



# Investigation of experimental and numerical methods, and analysis of stator clocking and instabilities in a high-speed multistage compressor

Johannes Schreiber

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**Johannes Schreiber**

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**Investigation of experimental and numerical  
methods, and analysis of stator clocking and  
instabilities in a high-speed multistage  
compressor**

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*Devant le jury composé de :*

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*To my beloved family and friends.*



# Preface

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# Abstract

The following experimental and numerical investigations aim at the deep understanding of the flow field in the 3.5 stages high-speed axial compressor CREATE, studied on a 2 MW test rig at the Laboratory of Fluid Mechanics and Acoustics (LMFA) in Lyon, France. This work focuses on three major objectives: Firstly, a global description of the flow field with an identification of limitations to the used exploration methods; Secondly, the characterization of the effect of stator-stator clocking in a high-speed compressor; Thirdly, the identification of instabilities arising at low mass flow rates for confirming studies on low-speed compressors and giving new insights.

This work demonstrates that a mis-interpretation of steady performance data occurs easily due to measurement constraints and correction coefficients are proposed. At certain locations in the compressor, the flow field exploration (experimental and numerical) methods are identified to be challenged. This identification will initiate further development of the methods. The main mis-predictions of the simulations concern the over-prediction of the blockage induced by the tip leakage flow and eventually an over-predicted pressure rise. Furthermore, the measurements provided by the pneumatic pressure probes over-estimate the static pressure upstream of the stators. This error is induced by the interaction between the stator potential field and the probe it-self. In addition, the laser Doppler anemometry method over-estimates the velocity downstream the stators. The transport of the rotor wakes through the stators might not be correctly captured with the seeding particles in this high-speed compressor.

The investigation of the stator clocking reveals only a small global effect within the measurement uncertainty band. Several contributions to the weak effect of clocking are identified by analysis of the flow structure transport, namely the time-mean mixing out of the stator wakes and the deformation of wakes along their flow path. The local effect of clocking depends on the span-height because of the variation of the circumferential position of the stator wakes and the stator blade shape over the span-height. Local possible positive and negative effects of clocking are identified and are shown to be almost in balance in this compressor. Furthermore, this work demonstrates that the unsteadiness in the flow field is not linked conclusively to the stator clocking.

In this compressor, the arising instabilities depend on the operating point and flow field exploration methods. At stable operating points and nominal compressor speed, the numerical results reveal a rotating disturbance in the rotors 2 and 3, whereas the measurements show a rotating disturbance only in the first rotor and only at part speed. In both cases the disturbance exhibits rotating instability like characteristics. An exhaustive numerical study allows to exclude the commonly assumed influence of rotor-stator interactions on the rotating disturbance and pinpoints its source. New insights into the stable behavior and periodicity of the measured rotating instability are derived contrary to the unstable behavior suggested by the naming and literature. This disturbance is shown to evolve into rotating stall cells when approaching the stability limit. At nominal compressor speed, a spike type surge inception is identified in the measured field. A precise description of the abrupt onset of the spike cell and its difference to a rotating stall cell are presented.

**Keywords:** high-speed multistage compressor, high-frequency experimental measurements, unsteady RANS simulations, clocking, instabilities, rotating disturbance, rotating instability, rotating stall



# Résumé

Les études expérimentales et numériques suivantes visent à la compréhension profonde de l'écoulement se développant dans le compresseur haute-vitesse axial de 3.5 étages CREATE, étudié sur un banc d'essai de 2 MW au Laboratoire de Mécanique des Fluides et Acoustique (LMFA) à Lyon, France. Ce travail a trois objectifs principaux : D'abord, une description globale de l'écoulement avec une identification des limites aux méthodes d'exploration utilisées ; Ensuite, la caractérisation de l'effet du clocking stator-stator dans un compresseur à haute-vitesse ; Troisièmement, l'identification des instabilités à faibles débits pour confirmer les études sur les compresseurs à basse-vitesse et contribuer à plus de compréhension.

Il est montré qu'une mauvaise interprétation des données de performance stationnaire se fait facilement en raison des contraintes de mesure et des coefficients de correction sont proposés. À certains endroits dans le compresseur, des limites aux méthodes d'exploration (expérimentales et numériques) de l'écoulement sont identifiées. Cette identification va permettre la poursuite du développement des méthodes. Les principales erreurs de prédiction des simulations concernent la surestimation du blocage induit par l'écoulement de jeu et l'augmentation de pression. En outre, les mesures fournies par les sondes de pression pneumatique surestiment la pression statique en amont des stators. Cette erreur est probablement provoquée par l'interaction entre le champ potentiel du stator et la sonde elle-même. De plus, l'anémométrie Doppler laser surestime la vitesse en aval des stators. Le transport des sillages du rotor à travers des stators n'est pas correctement capturé avec les particules d'ensemencement.

Le clocking a seulement un petit effet global dans la bande d'incertitude de mesure dans ce compresseur. Plusieurs contributions à ce faible effet de clocking sont identifiées par l'analyse du transport des structures d'écoulement : Le mélange circonférentiel du sillage de stator et la déformation des sillages le long de leur trajet dans l'écoulement. L'effet local du clocking dépend de la hauteur de veine en raison de la variation de la forme des aubages et du transport des sillages. Des effets positifs et négatifs sont présentés, qui globalement se compensent dans ce compresseur.

Les instabilités dans ce compresseur dépendent du point de fonctionnement et des méthodes d'exploration de l'écoulement. Aux points de fonctionnement stables et à la vitesse nominale du compresseur, les résultats numériques montrent une perturbation tournante dans les rotors 2 et 3, alors que les mesures montrent une perturbation tournante que dans le premier rotor et seulement à basse vitesse du compresseur. Dans les deux cas, les perturbations montrent des caractéristiques semblables. Une étude numérique permet d'exclure l'influence des interactions rotor-stator sur la perturbation tournante et met en évidence sa source. Des nouvelles connaissances sur le comportement stable et la périodicité du rotating instability (mesuré) sont dérivées contrairement au comportement instable suggéré par la dénomination et la littérature. Il est montré que cette perturbation évolue en cellule de décrochage tournante à l'approche de la limite de stabilité. A la vitesse nominale du compresseur, une entrée en instabilités de type spike est identifiée expérimentalement. Une description précise de l'apparition brutale du spike et sa différence par rapport à une cellule de décollement tournant sont présentées.

**Mots-clés:** compresseur axial haute-vitesse multi-étages, mesures expérimentales haute-fréquence, simulations RANS instationnaires, clocking, instabilités, perturbation tournante, rotating instability, décollement tournant





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# Nomenclature

## Acronyms and abbreviations

|       |                                                                                    |
|-------|------------------------------------------------------------------------------------|
| 25A   | Axial inter-row plane downstream of the IGV                                        |
| 260   | Axial inter-row plane upstream of the rotor 1                                      |
| 26A   | Axial inter-row plane downstream of the rotor 1                                    |
| 270   | Axial inter-row plane downstream of the stator 1                                   |
| 27A   | Axial inter-row plane downstream of the rotor 2                                    |
| 280   | Axial inter-row plane downstream of the stator 2                                   |
| 28A   | Axial inter-row plane downstream of the rotor 3                                    |
| 290   | Axial inter-row plane downstream of the stator 3                                   |
| 300   | Axial inter-row plane downstream of the compressor for the performance measurement |
| BPF   | Blade Passing Frequency                                                            |
| CCF   | Cross-correlation coefficients                                                     |
| CWT   | Continuous wavelet transform                                                       |
| DFT   | Discrete Fourier transforms                                                        |
| FFT   | Fast Fourier transforms                                                            |
| FM    | Frequency Modulation                                                               |
| LDA   | Laser Doppler Anemometry                                                           |
| LES   | Large Eddy Simulation                                                              |
| LPF   | Low Pass Filtered                                                                  |
| Norm. | Normalized                                                                         |
| OP    | Operating point                                                                    |
| PIV   | Particle Image Velocimetry                                                         |
| PNEUM | Pneumatic 3-hole probe measurements                                                |



---

|          |                                                        |
|----------|--------------------------------------------------------|
| POS      | A clocking position                                    |
| PS       | Pressure Side                                          |
| PSD      | Power spectral density                                 |
| R1,R2,R3 | Rotors 1, 2, and 3                                     |
| RANS     | Reynolds Averaged Navier-Stokes                        |
| ref      | A reference value                                      |
| rev.     | Revolution (of rotor)                                  |
| S1,S2,S3 | Stators 1, 2, and 3                                    |
| SS       | Suction Side                                           |
| TC       | Tip Clearance                                          |
| TCV      | Tip Clearance Vortex                                   |
| U(RANS)  | (Unsteady) Reynolds Averaged Navier-Stokes Simulations |
| ZDES     | Zonal Detached Eddy Simulation                         |

## Exponents and indices

|           |                                                              |
|-----------|--------------------------------------------------------------|
| $\ominus$ | Value in circumferential direction                           |
| '         | Blade angles                                                 |
| $f_x$     | Frequency in cross-correlation spectrum                      |
| 0         | Stagnation value                                             |
| 1,2,3     | Axial locations in the machine                               |
| 12        | Distance between two points in space or time                 |
| eq        | Equivalent                                                   |
| i         | Index for time step iterations                               |
| in        | Value at inlet of a row                                      |
| j         | Index for sub-iteration at a given time step for gear method |
| laser     | Value concerning laser measurement system                    |
| max       | A maximum value at a given point in space and/or time        |
| opt       | Optimum value                                                |
| out       | Value at outlet of a row                                     |
| pos       | Value concerning a position or positioning                   |
| PS        | Value at the pressure side of passage/blade                  |

|        |                                             |
|--------|---------------------------------------------|
| RI     | Rotating instability                        |
| rotor  | Value concerning the rotor                  |
| sc     | Measurement in settling chamber             |
| source | Value of a variable close to the source     |
| spike  | Value related to spike type stall inception |
| SS     | Value at the suction side of passage/blade  |
| st     | A statistical value                         |
| stator | Value concerning the stator                 |
| tip    | Value at the blade tip                      |

## Greek letters

|                          |                                                                 |
|--------------------------|-----------------------------------------------------------------|
| $\alpha$                 | Absolute flow angle                                             |
| $\alpha_{corr, struts}$  | Struts losses coefficient                                       |
| $\alpha_{drag}$          | Blade row drag coefficient                                      |
| $\alpha_{endwall}$       | Endwall losses coefficient                                      |
| $\beta$                  | Relative flow angle                                             |
| $\Delta$                 | Difference                                                      |
| $\delta$                 | Deviation                                                       |
| $\eta_t$                 | Non-dimensional time parameter for wavelet transform            |
| $\gamma$                 | Specific heat capacity ratio                                    |
| $\hat{q}(k)$             | Frequency spectrum from a Fourier transform                     |
| $\lambda$                | Throttle parameter for RANS outlet condition                    |
| $\lambda_{wave}$         | Fourier wavelength for wavelet                                  |
| $\mu_t$                  | Viscosity                                                       |
| $\omega_0$               | Non-dimensional Morlet frequency                                |
| $\omega_{m_\Theta, m_t}$ | Rotation speed of a rotor-stator interaction                    |
| $\Phi_{modal}$           | Phase coefficient matrix from double modal decomposition method |
| $\Psi^*$                 | Scaled and translated version of Morlet wavelet                 |
| $\Psi_0$                 | Morlet wavelet                                                  |
| $\sigma$                 | Blade solidity                                                  |
| $\tau$                   | Time delay between between arrival of a flow structure          |

$\Theta$  Circumferential coordinate

## Latin letters

|                     |                                                                                 |
|---------------------|---------------------------------------------------------------------------------|
| $A_{modal}$         | Amplitude matrix from double modal decomposition method                         |
| $c_{corr,Tt}$       | Correction factor for inlet total temperature                                   |
| $c_p$               | Total pressure loss coefficient                                                 |
| $Df$                | Lieblein diffusion factor                                                       |
| $dS$                | Passage size in space or time for blockage estimation                           |
| $ds$                | Fraction of a rotor passage (in space or time)                                  |
| $E_{RSI}$           | Energy of rotor-stator interactions                                             |
| $f$                 | Frequency                                                                       |
| $f_C$               | 1D value as result of fuzzy set for time period unsteady simulation convergence |
| $f_c$               | Center frequency for frequency modulation                                       |
| $f_m$               | Modulation frequency for frequency modulation                                   |
| $H_n$               | Helicity                                                                        |
| $it_{period}$       | Iterations per temporal period for numerical time discretization                |
| $L_{\Theta,min}$    | Minimal dimension of mesh in pitchwise direction                                |
| $m_{\Theta}$        | Spatial mode                                                                    |
| $m_t$               | Temporal mode                                                                   |
| $N_S$               | Number of stator blades in a compressor blade row                               |
| $no_{it}$           | Minimum number of iterations                                                    |
| $P_s$               | Static pressure                                                                 |
| $P_t$               | Total pressure                                                                  |
| $q_n$               | Discrete time series                                                            |
| $q_{std}$           | Measurement quantity after post-processing, before normalizing                  |
| $s_j$               | Set of scales for wavelet transforms                                            |
| $S_{Ps}$            | Static pressure signal                                                          |
| $Tt_{corr,inj}$     | Corrected inlet total temperature                                               |
| $Tt_{rise,nominal}$ | Nominal total temperature rise in the compressor                                |
| $V_{x,simpl}$       | Simple estimation of axial velocity gradient at the rotor tips                  |
| $X_{12}$            | Cross-correlation function                                                      |

---

|          |                                                      |
|----------|------------------------------------------------------|
| $x_{12}$ | Normalized cross-correlation                         |
| $c$      | Blade chord                                          |
| $E$      | Energy of a signal                                   |
| $f$      | A mathematical function                              |
| FB       | Flux balance                                         |
| H        | Non-linear second order scheme for gear method       |
| $h$      | Enthalpy                                             |
| I        | Turbulent Intensity                                  |
| $i$      | Incidence                                            |
| J        | Jacobian matrix of space operators and source terms  |
| Ku       | Kurtosis                                             |
| L        | Characteristic length for turbulence scale           |
| M        | Mach number                                          |
| N        | Number of points in discrete time series             |
| $n$      | Normal distance                                      |
| P        | Pressure                                             |
| $p$      | Mean order of accuracy for mesh discretization study |
| Pr       | Prandtl number                                       |
| Q        | Aerodynamic field for gear method                    |
| $r$      | Radius                                               |
| Re       | Reynolds number                                      |
| $s$      | Blade pitch                                          |
| Sk       | Skewness                                             |
| T        | Temporal period                                      |
| $t$      | Time                                                 |
| U        | Circumferential velocity                             |
| V        | Absolute velocity                                    |
| W        | Relative velocity                                    |
| $x$      | Axial machine coordinate                             |



# Introduction

Engineers are confronted with the demand for aircraft engines with higher efficiency and work output because airlines face the major challenge of growing air passenger and cargo transport by more than 5% every year, as reported by the IATA (International Air Transport Association), while the harmful impact of air traffic on the environment and climate needs to be reduced as clearly stated by the ICAO (International Civil Aviation Organization), which is a specialized agency of the United Nations. Among all engine components, the high-speed multistage compressor plays a crucial role for the overall performance of an aircraft engine.

The optimization of the compressor component is a complex challenge that requires deeper and deeper understanding of the flow physics because contentiously improvements have been achieved over the last decades already. The search for this understanding can be divided in low and high-speed axial compressor works. The low-speed compressor research work allows less costly insights in the flow field but needs to be confirmed in costly tests on high-speed compressors as used in rear stages of aircraft engines.

One of today's main goals pursued by the industrial research is to push further the limit of instabilities at low mass flow rates to operate the compressor at higher efficiency while being less sensitive to external disturbances. More compact engine designs are furthermore researched for reducing the specific fuel consumption. This might be achieved for example by smaller axial gaps between the compressor blade rows and less compression stages. A gain of 1% in compressor efficiency would decrease the specific fuel consumption of the aircraft engine by 0.5% to 0.8% following the principles of a Brayton-type engine (Cumpsty, 2003). The more compact design results in a higher blade loading and stronger blade row interactions, whose effects need a deep understanding to avoid damages to the compressor and efficiency penalties. Further possibilities for decreasing habitual losses in the compressor are researched without increasing the weight of the compression system. The blade row indexing (clocking) in the circumferential direction is for example a promising approach for rear stages in high-speed compressor.

Only a few realistic high-speed compressor test-rigs exist worldwide with an instrumentation fine enough for investigations of the flow field and flow conditions close to real engines conditions. They are rare because of the high cost of development and utilization. Computational Fluid Dynamic (CFD) methods might be a less costly approach depending on the applied method but still lack in capabilities when keeping reasonable simulation costs. Though, CFD is a useful method to obtain a supplementary view on the flow that allows the validation of the measurement methods and adds understanding of the flow field.

## Thesis structure

This thesis work investigates numerically and experimentally the flow field of the 3.5 stages high-speed axial research compressor CREATE, which has been designed by Safran Aircraft Engines and is tested at the LMFA (Laboratoire de Mécanique des Fluides et d'Acoustique) at the Ecole centrale de Lyon in France. The experimental data was acquired by a team of the turbomachinery group at the LMFA. The team profited from feedback and was guided thanks to this thesis work, which deals with the post-processing and post-treatment of the experimental data. Furthermore, steady and unsteady RANS simulations were conducted and post-treated in the scope of this thesis work. The simulations were achieved with the flow solver elsA from the French aerospace laboratory ONERA. The work on the numerical setup was presented at the ASME TurboExpo 2015 (Schreiber et al., 2015b). After a review of previous works in Chapter 1 and an introduction of the numerical and experimental methods in Chapter 2, the work presented in this thesis is divided into three parts:

The first part (Chapter 3) sets out to characterize the flow field in the compressor with a focus on the tip flow field and separated stator blade boundary layers. This study will be closely accompanied by an identification of challenges to the measurement and simulation methods for the advancement of these flow field exploration methods in the high-speed compressor environment. This work unfolds the trustworthiness of experimental and numerical data by deep analysis of the flow field, which goes further than only a classical uncertainty analysis.

In the second part (Chapter 4), the effect of stator clocking is investigated numerically and experimentally in the high-speed environment. This will show if the clocking is a valuable approach for reducing the losses in an aircraft engine compressor. The clocking effect is evaluated from a global perspective (compressor performance) to detailed analysis of local changes to the flow field. Major factors that impact clocking negatively are identified. This will help to decide about the use of clocking in aircraft engine compressors. A part of the work about clocking was presented at the ISAIF conference in 2015 (Schreiber et al., 2015a).

In the final part (Chapter 5), instabilities arising close to the stability limit of this compressor are identified in the numerical and experimental results and new insights are given. This analysis is divided into the study of instabilities occurring close to the stability limit but at a still stable operating point and instabilities that lead immediately to a flow breakdown. The experimental results will be investigated at nominal and part-speeds for the characterization of different types of rotating disturbances and their influence on the stall inception. These studies shall contribute to the understanding of the flow breakdown in high-speed multistage compressors. Moreover, this work will contribute to the understanding of the prediction capabilities of simulations for instabilities arising in multistage compressors. That will help aircraft engine engineers to understand the limits for the trustworthiness of these simulations. The study of the numerical results was presented at the ASME TurboExpo 2016 (Schreiber et al., 2016).

In Chapter 6, the conclusions drawn from this thesis work are summarized and perspectives for future works will be given.

# Chapter 1

## Review of previous works

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In this Chapter, a brief background for turbomachines and an introduction into general compressor aerodynamics will be given as foundation for this thesis work. Different types of blade row interactions will be then introduced to become acquainted with the multistage compressor environment and to give an overview of works about the stator clocking. Furthermore, the most recent knowledge about instabilities arising in compressors at low mass flow rates is discussed. Finally, the research objectives of this thesis work are derived from the review of previous works in this Chapter.



## 1.1 Background for a turbomachine

A jet engine describes any internal-combustion engine driving an aircraft by discharging a jet of energy-rich fluid rearwards. In principle a basic jet engine contains a compressor, a combustion chamber, a turbine and a nozzle as sketched in Figure 1.1. The ambient air enters the jet engine through the compressor which is adding energy to the flow and compressing it. Then the fluid enters the combustion chamber adding energy to the fluid at a constant pressure. In the following turbine, connected by a torque-shaft to the compressor, the fluid's thermal energy is converted to mechanical work. That generates the energy, which is required for the compressor's work. At the back of the jet engine, the ejection through the nozzle of the fluid with a surplus energy creates the thrust propelling the aircraft.

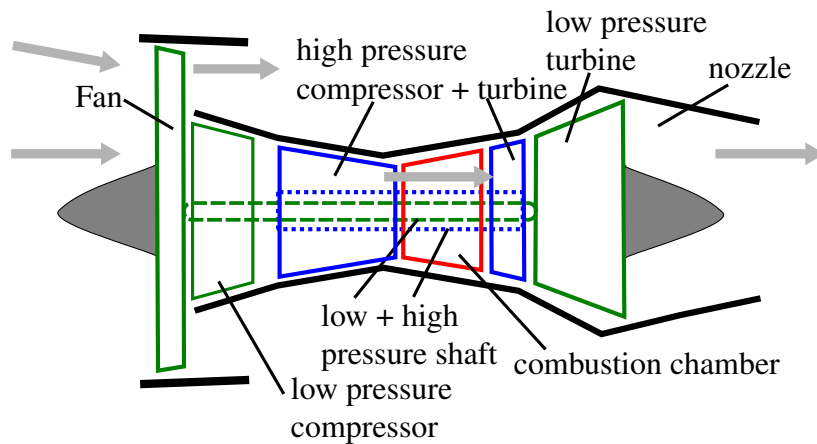


Figure 1.1 Sketch of a turbofan aircraft engine

In case of a turbofan, as a type of jet engine, a large low-pressure fan is added ahead of the compressor. A part of the low-pressure air bypasses the gas turbine and mixes with the fluid behind the turbine. The bypass air produces most of the thrust of a modern commercial aircraft turbofan engine by moving a large volume of air at low speed, rising consequently the efficiency of the jet engine. A low-pressure turbine can be added to the engine core in order to run the fan, and a possible low pressure compressor, through a slower turning shaft. The original core components become the high-pressure compressor, turbine and shaft (Dixon and Hall, 2010). The design of the research compressor, investigated in this thesis work, represents closely a part of the high-pressure compressor of a modern turbofan engine.

### 1.1.1 Principle of a compressor

The main function of the compressor in an aircraft engine is to increase the pressure of the incoming air before it enters the combustion chamber or the bypass. The pressure rise depends on the geometry of the compressor, the rotation speed of the rotor, and other secondary effects such as interactions between the stages of the compressor. In a typical aircraft engine, an axial compressor design is used. Compared to the radial compressor, the axial compressor allows transporting a bigger mass-flow with the same front surface.

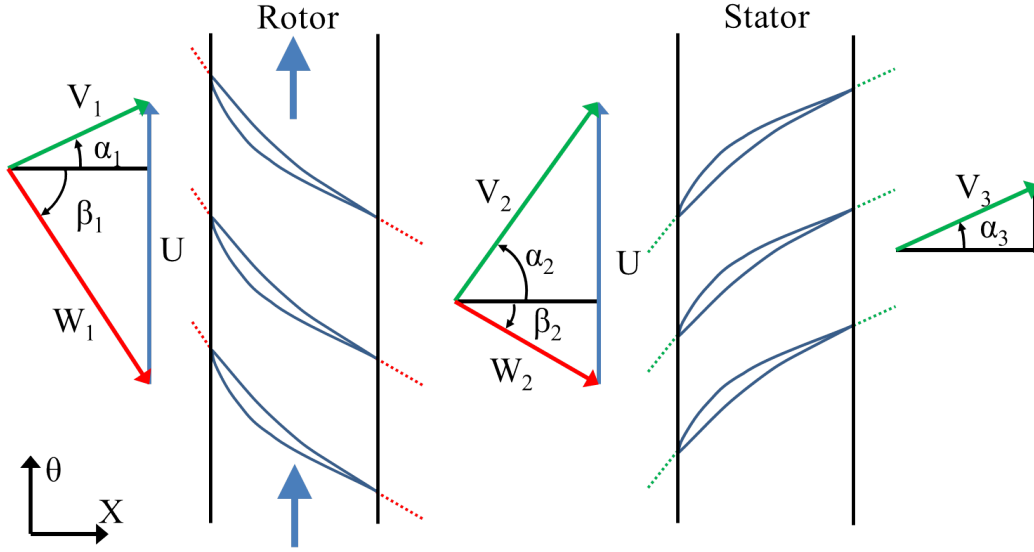


Figure 1.2 Sketch of ideal velocity triangles in a compressor stage

In other words, it produces the same amount of thrust with smaller drag. The design is characterized by a fluid entering and leaving in axial direction, hence axial-compressor. That allows connecting easily several stages in series, consisting of a pair of rotating (rotor) and stationary airfoils (stator). The relative motion of the rotating airfoils (blades) is adding energy to the fluid. Then the stationary blades convert the increased kinetic energy into a pressure rise. In order to obtain the needed pressure at the outlet of the compressor in relation to the pressure at the inlet (pressure ratio  $\pi$ ), a modern turbofan engine might consist of ten to twenty stages in series. A single stage has a total pressure ratio  $\pi$  of about 1.5, while a typical aircraft engine has a total pressure ratio  $\pi$  of about 30 to 40 (Cumpsty, 2003).

**Definition of flow description variables** Neglecting the small radial velocity component of the air passing through the compressor, the flow field can be represented in circumferential cuts at a constant radius (blade-to-blade cuts).

In a compressor stage with ideal flow conditions, as sketched in Figure 1.2, the air generally enters and leaves almost parallel to the rotor and stator blades in positive axial (x) direction. The absolute flow velocity ( $V$ ) can be found by the addition of the relative flow velocity ( $W$ ) and the rotor velocity ( $U$ ). When going through the rotor, the velocity component in the circumferential direction ( $V_\Theta$ ) is increased. The energy transferred by the rotor ( $\Delta h_{rotor}$ ) can therefore be calculated based on a difference between the circumferential velocity at the inlet and outlet of the rotor by the Euler equation as defined by Equation 1.1 for a constant radius (Cumpsty, 2003):

$$\Delta h_{rotor} = U \cdot (V_{\Theta out} - V_{\Theta in}) \quad (1.1)$$

The flow angles and blade angles are measured relative to the axial direction. In the relative frame, the flow angle is referred to as  $\beta$  (beta), and respectively in the absolute

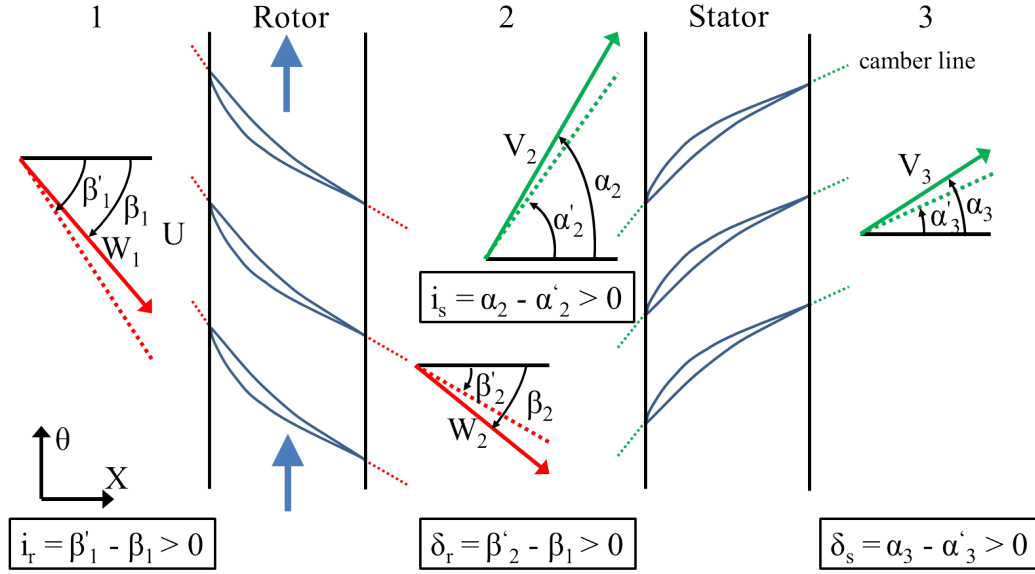


Figure 1.3 Definition of incidence and deviation angle, suction and pressure side, and camber line

frame as  $\alpha$  (alpha). Likewise, the blade angles are denoted by  $\beta'$  for a rotor, and by  $\alpha'$  for a stator, as illustrated in Figure 1.3. They are defined between the camber line and the machine axis. The actual angle between the entering air into a row and the blade angle is defined as incidence angle ( $i$ ). The corresponding angle between the air leaving the row and the blade outlet angle is defined as deviation ( $\delta$ ). Generally, the air does not turn as much as the blade due to losses. That results in a positive deviation angle. On the one hand, it is known that incidence angle changes do not necessarily lead to equally important deviation angle changes. On the other hand, a modification of the rotor speed or locally increased losses could change the deviation angle noticeably as demonstrated by Cumpsty (2003).

The accelerated flow is linked to a lower static pressure according to Bernoulli at the lower side of the blade profile, and hence, called suction side (SS). The decelerated flow at the upper side of the blade has consequently a higher static pressure, and is called pressure side (PS). The blade edge pointing against the main flow direction is called leading edge (LE), and respectively trailing edge (TE) in main flow direction.

### 1.1.2 Operating range of a compressor

A compressor is typically characterized by the pressure ratio and efficiency plotted against the mass flow rate at constant rotor speeds (Dixon and Hall, 2010). Figure 1.4 shows a compressor characteristic starting from the choke point (A), where the maximum mass flow rate is found. Lines of constant load are indicated that represent the compressor working at a constant outlet throttle valve position over variable compressor speed. The mass flow rate is limited by sonic blockage appearing in the compressor blade rows. The peak efficiency operating point is found at B. From point B to C, the efficiency is decreasing due to increasing

losses but the pressure ratio is rising because the work of the compressor rises more than the losses. That can be understood with the Euler equation (Equation 1.1). With decreasing mass flow rate, the incidence angle is rising more than the deviation angle. That results in a higher circumferential velocity difference through the rotor and hence higher work according to the Euler equation (Equation 1.1). From operating point C to D the losses increase dramatically, which prevent the pressure ratio to continue to increase and can lead to unstable operating conditions. A small disturbance can be enough to lead to compressor damaging instabilities, which will be introduced in Section 1.4.

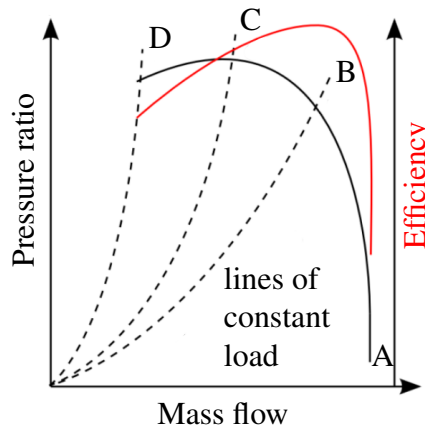


Figure 1.4 Compressor characteristic adapted from Dixon and Hall (2010)

## 1.2 Compressor aerodynamics

### 1.2.1 Wake and potential effects

In an axial compressor, wakes are emitted in downstream direction at the trailing edges of each blade, where the turbulent boundary layers of the blade's suction and pressure surfaces merge. Hence the wakes are mixing zones, which are characterized by a high turbulence, and total pressure and velocity losses, as shown in Figure 1.5. While moving downstream, wake and main stream are expected to become uniform due to viscous mixing at a certain distance downstream of the blade's trailing edge (Bullock and Johnsen, 1965).

#### Wake decay in multi-row compressor

Downstream of a single row, the major cause for wake decay would be viscous dissipation causing consequently mixing losses. By now, it is well known that the presence of a downstream row leads to a faster wake decay and can lead to a reduction in mixing losses (Poensgen and Gallus, 1991).

Smith (1966) proposed a theoretical mechanism for the wake transport and illustrated it by using a stream of dye representing a stator wake. The stream is chopped by a downstream rotor, reoriented, and stretched inside of the downstream rotor. The same would be valid for a rotor wake passing through a stator. According to the authors, the velocity deficit inside of

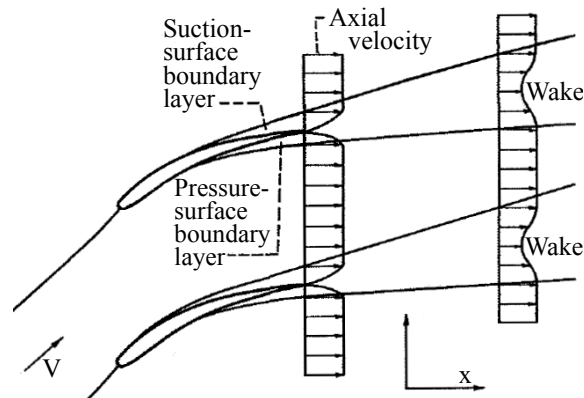


Figure 1.5 Boundary layers and blade wake in cascade flow (and compressor flow) adapted from a sketch made by Bullock and Johnsen (1965)

the wake is proportional to the width, and inverse proportional to the length of the wake. The effect can be summarized as an elongation of the wake while reducing the velocity deficit of its fluid. That effect is referred to as wake recovery due to inviscid stretching of the chopped wake segments, which reduces the losses due to viscous mixing.

Zante (1998) developed a model for the rotor wake decay that includes the effects of wake stretching and viscous dissipation. The model is based on an assumed analogy between the lengthening of a wake segment with constant mass in a converging channel and the wake stretching in a stator passage. The validated model reveals that wake stretching is the primary decay mechanism for rotor wakes in a stator. Hence, the losses due to viscous mixing were found to be strongly reduced in a high-speed axial compressor stage compared to the single rotor case.

### Negative jet effect

In a multi-row compressor, rotor and stator wakes are transported downstream through the following row. The negative jet effect occurs because of a lower velocity of a chopped wake segment in comparison with the free stream when traversing the downstream row, see Figure 1.6 (a). The absolute velocity is reduced within the wakes. The relative velocity is reduced as well, while the circumferential velocity is of same magnitude when comparing wake and free stream. That causes a slip velocity. The slip velocity leads to an accumulation of wake fluid at the pressure side of the blades. Free stream fluid replaces the chopped wake's fluid at the suction side. Thereby, the wakes get locally thinner towards the suction side of the blades (Mailach et al., 2008a; Sanders et al., 2002). Therefore the negative jet has also an influence on pressure and velocity distributions near the SS and PS side of the passage, see Figure 1.6 (b). Towards the SS, the velocity decreases at the upstream boundary of a wake segment, and thus a pressure peak can be registered. Towards the PS, the velocity increases at the upstream boundary, and therefore induces a pressure decrease. At the trailing edge of the downstream row, the accumulation of wake fluid thickens the emitted wake of the row itself.

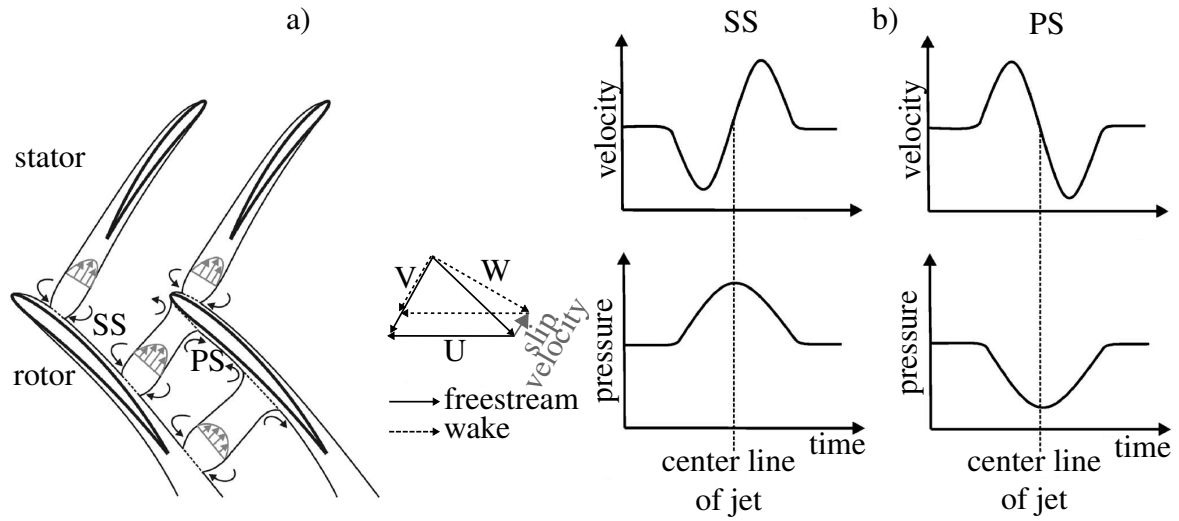


Figure 1.6 Sketch of negative jet effect (a), and velocity profiles in jet (b) adapted from the work of Mailach et al. (2008a)

### Potential effect

The potential field emitted by an object such as a blade, corresponds to the adaption of the fluid's streamlines to the object. The potential field change can propagate upstream as pressure waves if the main stream is subsonic. A simple geometry such as a cylinder in a flow illustrates well the potential effect. The flow decelerates and stagnates upstream of the cylinder, e.g. corresponding to the LE of a blade. This causes a zone of high static pressure ( $p_{source}$ ) according to Bernoulli. The pressure change information can travel as waves upstream with the speed of sound (minus the downstream flow velocity) in the case of subsonic flow. The axial damping of the potential effect can be estimated as function of the local Mach number, the distance to the source ( $x$ ) and the blade pitch ( $s$ ) as stated in Equation 1.2, adapted from Parker and Watson (1972). The larger the blade pitch, the lower the damping, the further the potential effect will propagate. In a high subsonic flow (downstream rotors) the potential effect can propagate further upstream from the source than in a low subsonic flow.

$$P(x) = P_{source} \cdot \exp\left(-2\pi\sqrt{1-M^2}\frac{x}{s}\right) \quad (1.2)$$

where:

$P_{source}$ : Circumferential static pressure modulation amplitude close to the source (blade in flow field)

$x$ : Axial distance to source

$s$ : Blade pitch

### 1.2.2 Tip leakage flow

A gap is realized between the tip of the rotor blades (rotating) and the casing (stationary). The radial tip gap size will be called tip clearance (TC) in the following Chapters. The pressure difference between PS and SS of a rotor blade drives the fluid to leak through the radial tip gap from the PS, see Figure 1.7. The maximum tip leakage flow (TLF) is found at the position of the maximum profile pressure difference between PS and SS. This position is understood as the point of origin of the tip clearance vortex (TCV). The main flow close to the LE is interacting with the tip leakage flow when it is entering the adjacent blade channel, and forming the tip clearance vortex core. The following tip leakage flow rolls around the core of the tip clearance vortex. Depending on the trajectory of the TCV, it can impact on the PS of the adjacent blade and mix with the boundary layer, or it can leave the inter-blade channel without direct interaction (Bullock and Johnsen, 1965).

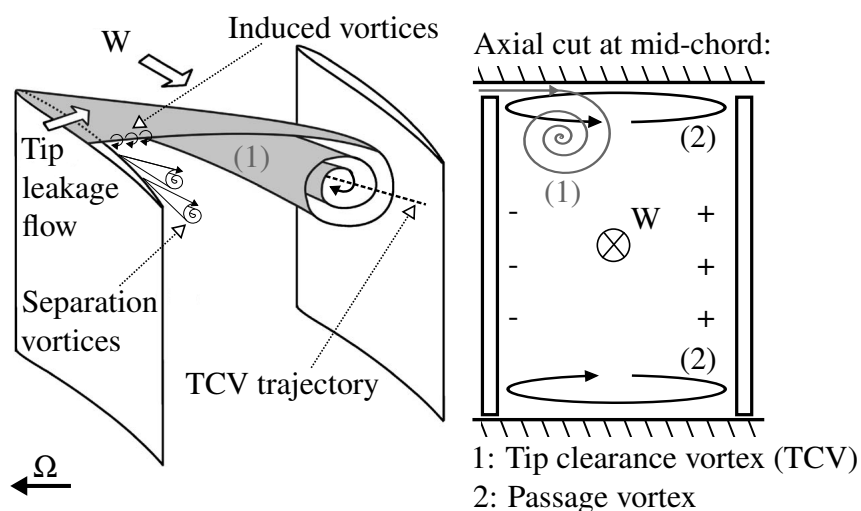


Figure 1.7 Tip clearance vortex adapted from You et al. (2007)

You et al. (2007) demonstrate, additionally to the TCV, the formation of separation vortices after mid-chord at the suction side blade tip on a cascade test case (with moving endwall) using Large-Eddy Simulations (LES). Smaller vortices are formed between the tip clearance vortex and the blade (marked as "induced vortices" in Figure 1.7). Furthermore, the authors observe a circumferential wandering of the tip clearance vortex trajectory with a frequency below the wake shedding frequency, both in their numerical study and experimental results. The wandering does not lead to an alignment of the TCV trajectory with the inlet plane. The authors considered the instability of the shear layer in the tip leakage jet flow as most probable cause. Boudet et al. (2015) also observed the wandering of the tip clearance vortex in an experimental and numerical work on an isolated rotor. A zonal RANS/LES approach was conducted by the authors with a full LES resolution for the tip flow region. The authors suggest that the wandering could be excited by turbulence from the incoming casing boundary layer and the adjacent tip leakage vortex. An interesting result is the classification

of the wandering as a precursor for rotation instabilities but not as the rotating instability itself. That will be further discussed in subsection 1.4.3.

An estimation of the velocity of the tip leakage jet is proposed by Storer and Cumpsty (1991) for cases of experimental work with limited measurement points. The estimation is based on the assumption that the momentum of the flow on the pressure side of the tip gap is conserved when being drawn into the tip gap. It is noted that the static pressure decreases towards the tip gap on the pressure side of the blade because of an acceleration of the fluid into the tip gap. Therefore, the authors propose to take the static pressure at mid-span outside of the potential field of the inflow. That leads to the proposition of Equation 1.3 for the estimation of the velocity ( $V_{tip}$ ) of the tip leakage jet passing through the tip gap.

$$V_{tip} = \sqrt{\frac{2(P_{tip} - P_{s_{mid-span}})}{\rho}} \quad (1.3)$$

where:

$P_{tip}$ : Total pressure close to blade tip (assumption of conservation of total pressure through tip gap from PS to SS)

$P_{s_{mid-span}}$ : Static pressure outside of the inflow to the tip gap (because of deceleration close to the tip gap), mid-span value proposed by Storer and Cumpsty (1991)

### 1.2.3 Flow separations

In a high-speed axial compressor flow, the Reynolds number is usually as high as  $10^6$ . With this Reynolds number, viscous effects are normally limited to a thin boundary layer, which is in contact to a blade, hub or casing surface. In the boundary layer, the flow velocity changes from the freestream velocity to the value zero imposed by the no-slip condition at the wall. In a two-dimensional flow, the boundary layer is constrained to remain obviously in a plane. In a three-dimensional flow, the boundary layer can develop a transverse component until a separation occurs. That component is mainly caused by the flow escaping from adverse pressure raise. The flow with a direction different from the freestream one is called secondary flow. Goodhand (2011) derive in more detail the flow escaping at the endwall. In a compressor the flow is being turned in the rows. That induces a pressure gradient from the pressure to the suction surface of a blade passage, see also Figure 1.8. The same pressure gradient is imposed on the endwall flow (2, red) as on the freestream (1, blue). The endwall flow needs to compensate the lower velocity with a locally increased streamline curvature (smaller Radius ( $r$ )) as derived from Goodhand (2011) with Equation 1.4. That causes a secondary flow - the passage vortex. Fluid consequently accumulates in the suction side corner, and plays an important role for corner separations as the passage vortex will intensify with loading of the blades.

$$\frac{\nabla p}{\delta n} = \rho \frac{V^2}{r} \quad (1.4)$$



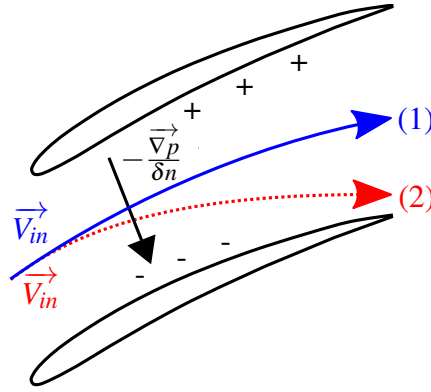


Figure 1.8 Secondary flow because of streamline adaption at endwall from Goodhand (2011)

**Diffusion factor** Lieblein et al. (1953) identified that a too important velocity diffusion on the blade surfaces leads to a thickening of the boundary layers and eventually detached boundary layer. Lieblein noted that the main part of the boundary layer growth occurs on the blade suction side surface. Consequently, the thickness of the emitted wake can be correlated to the growth of the boundary layer. The diffusion factor correlates the maximum velocity on the blade suction side to the outlet velocity (see Equation 1.5). The widely known Lieblein diffusion factor is an approximation because the determination of the maximum suction side velocity is difficult (see Equation 1.6). The average velocities  $V_{in}$  and  $V_{out}$  describe the flow in and out of a blade row in the reference frame of the blade row.  $Df$  factors in excess of 0.6 indicate a risk of blade stall. The Lieblein diffusion factor has been obtained for low Mach 2D flow in a cascade and is therefore not valid in high Mach flow ( $> 0.75$ ). Moreover, the factor is not valid close to the endwall and only valid for flow at the nominal blade operating point. Lieblein developed therefore an equivalent diffusion ratio for less restrictive conditions but still only for the flow in a 2D cascade, see Equation 1.7. A diffusion factors analysis can be consequently only conducted outside of the hub and casing boundary layers.

$$Df = \frac{V_{max} - V_{out}}{V_{in}} \quad (1.5)$$

$$Df = 1 - \frac{V_{out}}{V_{in}} + \left( \frac{|V_{\Theta out} - V_{\Theta in}|}{2\sigma V_{in}} \right) \leq 0.6 \quad (1.6)$$

$$Df_{eq} = \frac{\cos \beta_{out}}{\cos \beta_{in}} \cdot \left[ 1.12 + a |i - i^{opt}|^{1.43} + 0.61 \frac{\cos^2 \beta_{in}}{\sigma} \cdot |\tan \beta_{out} - \tan \beta_{in}| \right] \leq 2.0 \quad (1.7)$$

where:

$i, i^{opt}$ : Actual and optimum incidence angle

$a$ : A constant that depends on the blade profile (typically below 0.02)

$\beta$ : Relative flow angles;  $\sigma$ : Solidity

### Description technique for flow separations

The description of three-dimensional flow separation in the following shall be based on the work of Legendre (1956). Legendre proposed that through any non-singular point on a wall passes only one streamline. Flow separation was defined by Legendre consequently as convergence of the wall streamlines onto a wall streamline that originates from a singular point (saddle point).

Délery (2013) shows that the limiting (wall) streamlines are co-linear to friction lines close to a wall. Only one skin friction line goes through one point on the wall. Critical points (singular points) do not satisfy that rule. Here the shear stress vector vanishes. A set of skin friction lines or the shear stress vector field on the surface of an obstacle is called surface pattern.

The nature of a critical point can be identified based on the eigenvalues of the algebraic system formed by the shear stress vectors according to the work of Délery (2013), and is presented in Figure 1.9.

- At a critical point of type *node*, all the skin friction lines have a common tangent except for one of them. If the friction lines have their origin in the critical point, the node is an attachment node. Likewise, if the friction lines converge in a critical point, the node is a separation node. In a special case, an isotropic node, all friction lines go through the critical point.
- At a critical point of type *saddle point*, all skin friction lines avoid this critical point. Only two friction lines run through the saddle point.
- At a critical point of type *focus*, the friction lines spiral around it, and have hence no common tangent. At a *stable focus* the friction lines spiral into it. That can show the vortical surface as a result of flow separation. At an *unstable focus*, the friction lines spiral outward.

A special case, where the friction lines form closed curves encircling a limiting curve, is called a *center*.

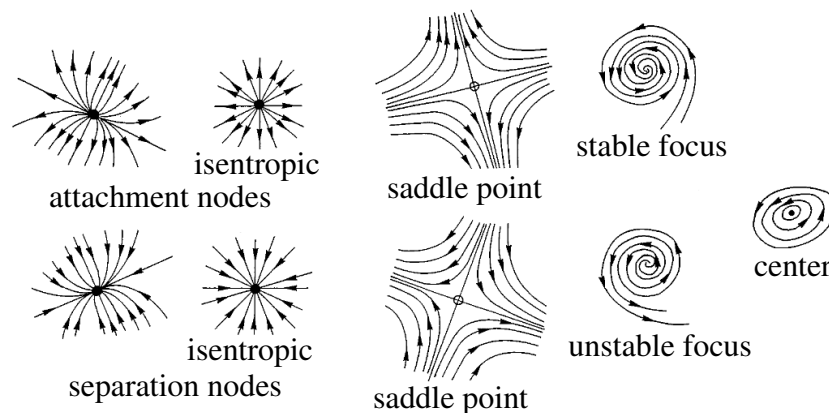


Figure 1.9 Critical point classification by Délery (2001)

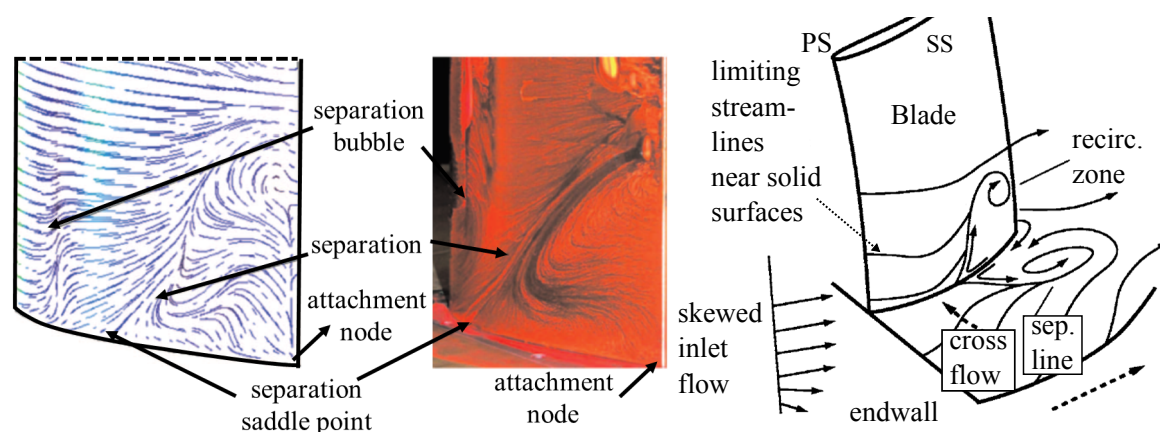
Délery (2013) defined rules for the interpretation of three-dimensional separations to obtain a topologically consistent surface pattern. Separation lines and critical points have to be identified using the surface pattern following these rules:

- The friction lines have to origin at one or several attachment nodes and terminate at a focus or detachment node. If the available flow field information is not fine enough, imaginary critical points might be necessary while following all the proposed rules hereinafter. Nodes and foci are equivalent in a topological sense because friction lines originate from them or end in them.
- A flow is detached if the surface pattern shows more than two nodes. In other words, the pattern contains at least one saddle point due to the line separating the friction lines of these nodes (separation line).
- The understanding of the flow field away from the surface is necessary to interpret the surface pattern correctly. Separation surfaces have to be associated to separation and attachment lines. Furthermore, the intersection of friction lines is wrong elsewhere than in a critical point.

### Corner separation

As introduced earlier, the secondary flow at the endwall causes an accumulation of low momentum fluid in the suction surface corners between endwall and blade. That causes the so-called corner separation which is present in every compressor, see Figure 1.10a. The endwall fluid and the blade suction surface flow form a separation line on the blade surface. Flow reversal is not necessary for such a small corner separation. Its importance for the flow field (in other words, the size of the corner separation) varies depending on the design of the blade and the operating point. (Gbadebo et al., 2005)

When a certain critical incidence is passed, a large open corner separation forms (see Figure 1.10b) and causes important losses and blockage, thus corner stall (Lei et al., 2008).



(a) Closed corner separation with CFD (left) and experimental method (right) from Cumpsty (2010) (b) Open corner separation sketched in the work of Lei et al. (2008)

Figure 1.10 Corner separations in compressors

## 1.3 Blade row interactions

The previously described flow structures such as wakes, potential fields, tip clearance vortices, separations can interact in compressors with one or more stages. These interactions shall be called rotor-stator-interactions (RSI). The literature review will show that the interactions can induce losses and noise. Using the understanding about the RSI, an ideal relative circumferential positioning of blade rows (either stators or rotors) relative to each other (clocking) could be found to reduce losses and noise.

### 1.3.1 Rotor stator interactions

Tyler and Sofrin (1962) suggested that the pressure field generated by the RSI consists of spinning spatial mode patterns that can be decomposed. Their mode order corresponds to linear combinations of the rotor and stator blade count. The authors furthermore identified that RSI can rotate above a certain critical Mach number, which allows the RSI to propagate far in the compressor.

Courtiade et al. (2012) have extended this modal decomposition to multistage applications. They could clearly identify multi-row interactions by using a newly extended double modal flow field decomposition method. They studied mainly casing pressure measurements in order to identify the RSIs in a previous (almost identical) version of CREATE. They showed that downstream of rotors, the RSI energy can be as strong as the fluctuations directly caused by the passage of the rotating blades (BPF harmonics). Moreover, the influence of the RSI on the global performance has been numerically investigated. Their overall share in production of entropy has been estimated to about 5% to 10%. Sanders et al. (2002) conducted Particle Image Velocimetry (PIV) measurements for the investigation of the flow field inside of a stator in a 1.5 stages axial compressor. The authors registered that the rotor wakes are still evident downstream of the stator row, which indicates possible multi-row interactions.

Three studies about rotor-stator interactions with relevant results for this work are presented hereinafter. Mailach et al. (2008a) conducted LDA measurements in a four-stage low-speed research compressor. The authors found that the incoming stator wakes have a dominant effect compared to the potential field on the periodical unsteady flow field of a rotor blade at mid-span. In the second part of their work Mailach et al. (2008b) showed that the periodic impingement of stator wakes on a rotor blade changes the pressure field around the blade. The periodically changing pressure field causes an oscillation of the trajectory of the tip clearance vortex. Furthermore, the authors visualized the interaction between the tip clearance vortex itself and the incoming stator wake, see Figure 1.11. The interaction is found to induce vortex pairs in the tip clearance vortex coming from the stator wake segment (due to the negative jet effect). Note that these results are obtained on a low-speed compressor.

Ernst et al. (2011) conducted measurements in a two stage axial compressor. The RSIs were in the focus of their investigations, which were carried out with LDA and pressure probe measurements. Ernst et al. (2011) could resolve the potential effect of a downstream row. It was observed that the pressure in a IGV wake rises due to an interaction with the potential field originating from the downstream rotor 1. At the same time, the trajectory of the rotor 1 TCV was found to be alternated due to an interaction with the IGV wakes. Downstream of

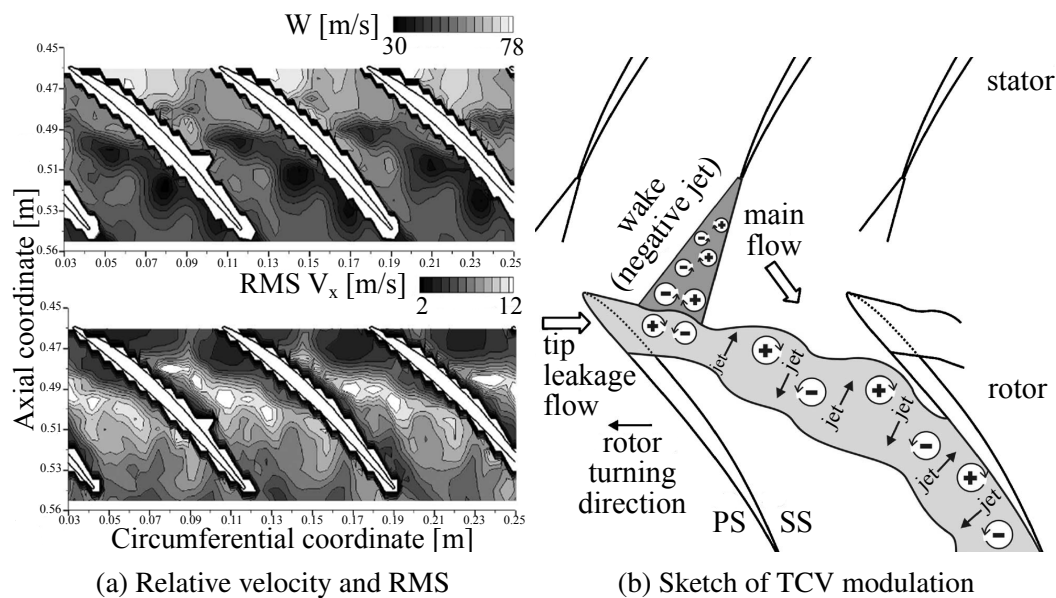


Figure 1.11 Interaction between wake and tip clearance vortex as seen by Mailach et al. (2008b) at 97.9% span-height in a rotor at design operating point

the first stator, the influence of the TCV originating from rotor 1 was proven to influence the flow field at about 20% of the span towards the casing. An interaction of the TCV with the stator wakes caused an increasing of the turbulence inside of the stator wakes. In general, the increasing of the RMS value was found to be connected with a common rotation sense of the vortices in each respective secondary flow structure. The authors could also identify multi-row interactions with a frequency analysis of the flow field.

Ottavy et al. (2012) conducted comparisons between numerical and experimental data for a previous version of the high-speed compressor CREATE. The authors found two well-fitting data sets in terms of main flow characteristics. It was established that the interaction between up- and downstream flows is under-estimated in the numerical data. It was assumed that this was solely caused by an over-dissipation of flow structures along the main flow axis. A more detailed identification of the over-dissipated structures was not given within the scope of the general comparison.

### 1.3.2 Stator stator clocking

Studies about the circumferential indexing of adjacent blade rows with similar blade count (clocking) date back to 1972. Walker and Oliver (1972) found early that clocking could have a beneficial - less noise - influence on the interaction of two successive rows in a low-speed 1.5 stages axial compressor.

One of the first experimental studies about the influence of clocking on the efficiency of a compressor was done by Barankiewicz and Hathaway (1997) on the 4.5 stages NASA Lewis low-speed axial research compressor in 1997. The overall change of efficiency was found to be 0.2% at maximum, and considered to lie in the band of the measurement uncertainties.

The optimal clocking position was found to vary depending on the operating point. With the potential of increasing the efficiency, reducing the noise generation and decreasing the unsteady blade forcing, clocking has been subject to works of many researchers. A general guideline for clocking especially in the multistage environment could not be identified yet. Experimental and numerical vane clocking studies have shown that stage efficiency changes can range from 0.1 to 2.7 points when applying clocking to low-speed axial compressors (Barankiewicz and Hathaway, 1997; He et al., 2002; Salontay et al., 2010b; Ståding et al., 2012).

The relation between clocking position and efficiency appears commonly sinusoidal. Most of the works identify the stator wake impingement on the downstream stator leading edge as beneficial for the efficiency at nominal operation point. The work of Ståding et al. (2012) presents an exception. They found that wake passing at mid-passage in the clocked row would be beneficial.

Mailach and Vogeler (2004) found that vane clocking influences the blade loading in the embedded rotor row in the low-speed four-stages compressor of Dresden. The overlap of upstream stator wakes and downstream potential effects are found to increase the pressure fluctuations. That is assumed to lead to important excitation of rotor blade vibrations.

A study on vane clocking in a multistage axial compressor was conducted numerically by He et al. (2002) on a 2.5 stages ALSTOM transonic case. They applied a nonlinear harmonic methodology to study the clocking. In sub- and transonic cases, the maximum change of efficiency was found below 0.1% for stator clocking. The best clocking position was identified to correspond to when the upstream stator wake impinges on the downstream stator leading edge. That was contributed to the fact that the largest velocity difference between the wake and the free stream fluid would be found when the wake was passing through mid-passage, and would cause hence higher mixing losses.

Fruth et al. (2013) investigated numerically the impact of clocking on the efficiency and aerodynamic forcing on the first 1.5 stages, including IGV/rotor/stator, of an industrial transonic compressor in cooperation with Siemens Industrial Turbomachinery AG. The maximum efficiency difference was found to be 0.125%. The authors found that the interaction of the low momentum fluid of an upstream stator wake with a given stator blade boundary layer would decrease in sum the losses downstream of the given stator. Here the authors noted the tip region as an exception, where stator wakes impinging on the downstream stator leading edge would cause a significant loss increase.

At the Purdue university, Key et al. (2009) investigated the effect of clocking on a 3.5 stages medium-speed axial research compressor (5000 rpm). The emitted stator 1 wakes are presented as mostly radial with slight skew towards the end-wall. The position with an impingement of the stator 1 wakes on the stator 2 leading edge gave the best efficiency with an increase of 0.27%. Also here the optimal clocking position was found to vary depending on the operating point. At a loaded operating point, the circumferential inclination of the stator wakes was observed to lead to a variation of the impact along the span. At this operating point, the optimum clocking position would be when upstream stator wakes impinge on the downstream stator leading edge in the hub region. In an accompanying work, Key et al. (2008) performed unsteady pressure measurements. The authors found that stator wake

impingement on a stator leading edge would cause a damping of the fluctuations induced by the upstream rotor in the stator blade boundary layer. That would lead to overall smaller losses. At the loaded operating point, the wake impingement was identified to cause a suction side boundary layer separation. That would lead to higher losses.

On the same configuration, Salontay et al. (2010b) performed unsteady RANS calculations using mixing planes up- and downstream of the experimentally investigated stator-rotor-stator rows. An agreement was found between experimental and numerical estimation of the performance increase only when extracting the numerical data at the limited number of measurement positions. An area average of the flow field showed a negligible overall change in efficiency. That was found to be caused by a bad prediction of the circumferential stator wake shape, and hence positive and negative clocking effects that cancel out over the span. The authors associated these discrepancies with the surface roughness and leakage flows that are not modeled. That shows the difficulties when trying to predict numerically the effect of clocking.

Saren et al. (1998) conducted a study of clocking in a high speed environment and showed that a change of one efficiency point could be globally possible in a 1.5 stages axial compressor. The authors noted that locally the efficiency could change as much as eight points in the endwall region.

### 1.3.3 Potential interactions

Mailach and Vogeler (2004) investigated the unsteady profile pressure distribution along mid-span of rotor and stator blades in a four-stage low-speed research compressor. The test case was introduced by Künzelmann et al. (2008). One major finding is that both the wakes from the upstream row and the potential field of a downstream row influence the unsteady pressure profile on a blade along mid-span. The profile pressure was observed to change instantaneously because of a propagation speed of sound (minus the fluid velocity in opposite direction) of the pressure waves associated with the potential fields of the stators. The changes were observed for every passing of a rotor blade by one of the up- or downstream stator blades. Furthermore it is established that the incoming wakes and the potential field have a comparable strong effect.

Mailach et al. (2004) developed a method to calculate the aerodynamic blade force based on recorded pressure measurements. Depending on the blade row and operating point, it was presented that the amplitude of the unsteady pressure could fluctuate by up to 15% to 35% of the mean values due to the interaction of a row with the upstream wakes and downstream potential field.

## 1.4 Rotating stall and instabilities, stall inception, and surge

The lowest mass flow rate end of the operating range is ultimately limited by the aerodynamic instability surge. Day (1994) found that in axial compressors surge is always preceded by a (short) period of the aerodynamic instability: rotating stall. Surge and rotating stall are to

be avoided because of the potential mechanical danger and for example unrecoverable flow conditions in a jet engine during flight.

Two main rotating stall inception types are identified in literature. The first is called modal inception which is a long length scale disturbance. Its modes are measured at the scale of circumferential lengths. The second stall inception type is a short length scale (a few blade pitches) disturbance which is commonly labeled as spike. The spike inception can occur with modal instabilities present in the flow field but not the other way around, as stated by Day (1993). Furthermore, rotating disturbances can be observed already at a mass flow rate greater than the stall inception point. These disturbances are most commonly called rotating instabilities in literature. The rotating instability does not induce a significant flow blockage in the concerned rotor contrary to the rotating stall.

The study of the preceding aerodynamic instabilities is the third major subject of this thesis in Chapter 5. The need for detailed measurements in high-speed axial compressors and their analysis was clearly stated by Day (2015) and derived from flow features that can only be found in high-speed compressor. That environment poses a challenge in terms of measurement cost and technique but here the measurements are necessary to understand the instabilities arising in a real aircraft engine compressor.

### 1.4.1 Rotating stall

Rotating stall is a local three-dimensional instability in compressors (in contrast to surge which is a global instability, described later on in Section 1.4.4). It can be characterized by one to several rotating cells of detached flow. Emmons et al. (1955) deduced from hot-wire measurements in a test compressor the stall propagation mechanism that is widely known today and illustrated in Figure 1.12.

The authors assumed that the presence of an initial (at the time unknown) disturbance causes one blade to stall. The flow ahead of the rotor is then diverted towards the adjacent rotor blade passages. The incidence angle is increased on the next blade in direction of the inlet velocity. At the same time the incidence is decreased on the preceding blade. The high incidence angle causes the next blade to stall, while the low incidence leads to a re-attachment of the flow. Consequently, the rotating stall cell is formed and propagates against the rotor

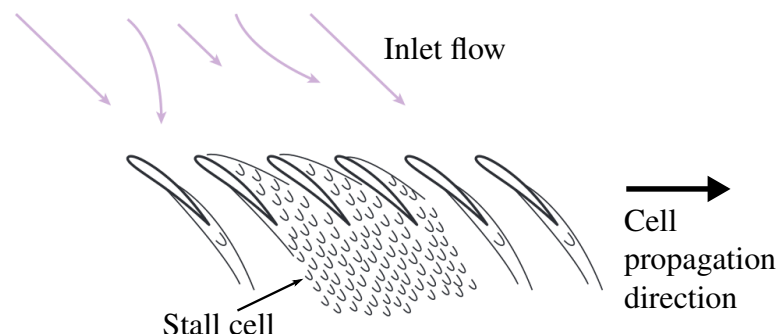


Figure 1.12 Stall cell propagation, Emmons et al. (1955), adapted from Tan et al. (2010)



turning direction in the rotor reference system. In the stator frame, the rotating stall cells are generally found to propagate at 30% to 80% rotor speed.

**Part-span and full-span rotating stall** It can be distinguished between part-span and full-span rotating stall. In part-span rotating stall, only a part of the span is stalled. Whereas in the full-span rotating stall, the flow is stalled over the whole span.

Part-span stall can occur in high-speed multistage compressor even though it is more likely to occur in low-speed compressors as shown among others by Giannissis et al. (1989) and Escuret and Garnier (1996), and is characterized by multiple cells (about up to 10) that can each cover one to several blade passages. The part-span stall of a rear stage is possible in a multistage high-speed compressor while the other stages maintain the compressor on quasi stable operation.

The part-span stall cells are found to rotate usually at 50% to 80% rotor speed. He (1997) could identify in an early numerical work that the smaller the rotating stall cell the higher the rotating speed. Day (1993) came to the same conclusion based on measurements in a low-speed multistage compressor. At the same time, the unstalled flow is diverted to the hub and accelerated while it stays stable with only a small reduction in the overall pressure output for the row.

Full-span rotating stall describes usually a single cell which can appear as a consequence of a spike type surge inception. The blockage induced by the cell leads to an important loss in pressure output of the compressor. An unloading of the blades (increasing of mass flow rate) is necessary to unstall the compressor. The single full span cell is found to rotate with between 30% and 50% rotor speed. Here the same is valid as for the part-span stall: The smaller the cell, the faster it rotates. Day et al. (1999) showed that the full-span rotating stall generally develops into surge within a very few rotations in high-speed compressors, as also observed in the compressor CREATE by Courtiade (2012).

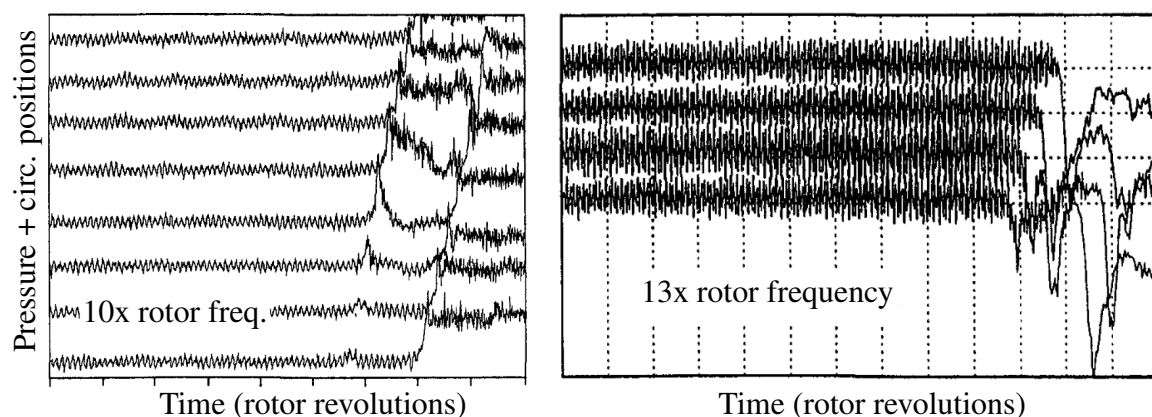


Figure 1.13 High-frequency part-span stall cell reported by Day et al. (1999) for high-speed axial compressors from Safran Aircraft Engines (left) and from the British Defence Research Agency (right)

Day et al. (1999) reported high-frequency rotating stall in several industrial high-speed axial compressors, including a four stages high pressure compressor from Safran Aircraft Engines. In this compressor, 14 stall cells are found to rotate with 82% rotor speed, inducing a passing frequency of about 10 times the rotor frequency in a casing pressure measurement as shown in Figure 1.13 on the left. The high-speed part-span stall cell is captured with eight pressure probes around the circumference at the casing. The "high-frequency" rotating stall appears as regular modulation of the signals prior to surge. A rotating stall onset within less than three rotations can be noted. Likewise in the Defence Research Agency compressor, a high-frequency rotating stall with 13 times the rotor frequency is detected (Figure 1.13 on the right).

Courtiade and Ottavy (2013b) and Courtiade (2012) observed similar high-frequency rotating pressure waves in a high-speed multistage compressor. The authors identified them as acoustic waves that would trigger rotating stall and surge due to acoustic resonance in the rear stages of the compressor. Courtiade and Ottavy (2013b) compared successfully the global characteristics of the acoustic resonance to a theoretical model.

### 1.4.2 Stall inception

There are two widely recognized stall inception types for low and high-speed axial compressors: modal and spike.

Camp and Day (1998) stated a criterion about which of the two stall inception types would occur given the flow and compressor properties. The criterion is still widely accepted today, as summed up by Day (2015). The criterion states that the spike stall inception occurs when a critical rotor tip incidence is exceeded before the peak of the overall total-to-static pressure rise characteristic is reached, see on the right in Figure 1.14. Furthermore, modal inception will occur if the peak of the overall pressure rise characteristic is reached before the critical rotor tip incidence is exceeded (on the left in the same Figure). At the peak, the

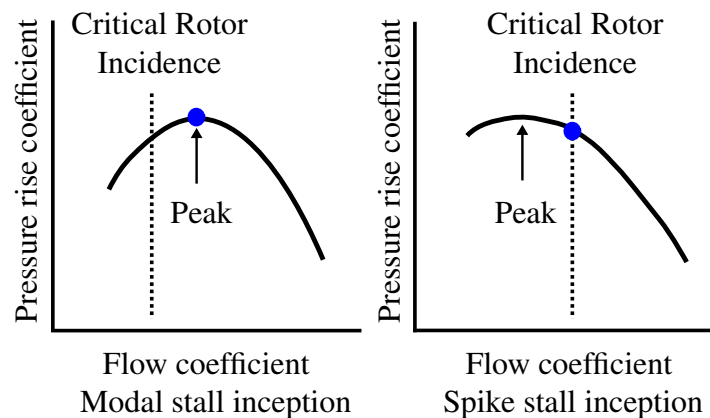


Figure 1.14 Left: modal stall inception criterion, right: spike inception criterion, adapted from the work of Camp and Day (1998)

slope of the total-to-static pressure ratio coefficient versus the mass flow rate is zero, and a small pressure perturbation can cause large variations of the mass flow rate. Thus the initially small perturbation can develop into a modal perturbation.

### Modal inception

The modal stall inception is characterized by a long-length scale disturbance (measured in circumference) that can be detected a long time before rotating stall inception, see Figure 1.15. That allows to use control systems for early detection of this stall inception type. The modal disturbance with a weak amplitude rotates below 50% of rotor speed. This type of stall inception is usually found in low-speed compressors, as stated by Day (1994). Attempts were made to trace the amplitude evolution of the modal waves prior to surge as an early stall warning, as for example by Tryfonidis et al. (1995) and Garnier et al. (1991). However, this stays limited to compressors who actually have a modal type stall inception.

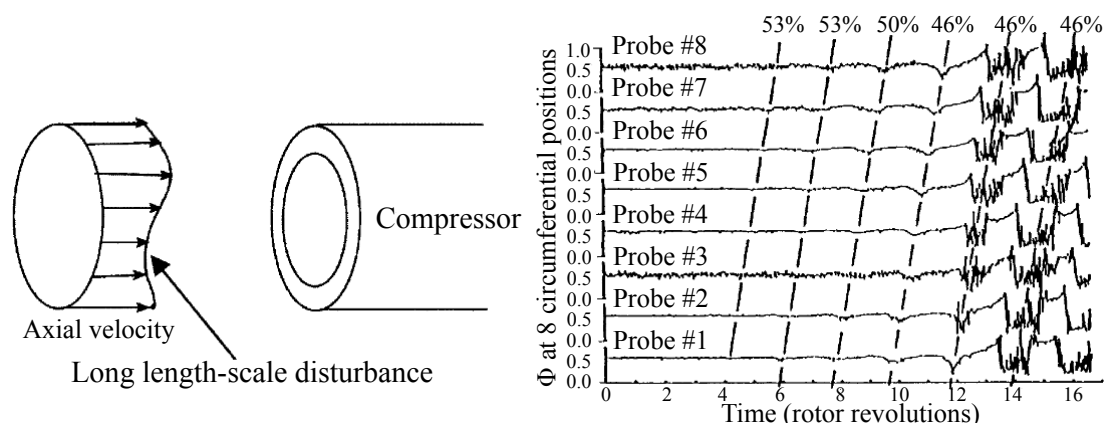


Figure 1.15 Modal (long length-scale disturbance) stall inception as measured by Day (1993), taken from the work of Vo (2001)

### Spike Inception

The spike stall inception can be characterized as a short length scale disturbance with a circumferential extension of one to two pitches when very first detected in the tip region of a rotor. The spike appears as sharp peak or valley (see Figure 1.16 as an example) in respectively a pressure or velocity measurement upstream of the one concerned rotor (Camp and Day, 1998; Day, 1993; Day et al., 1999; Escuret and Garnier, 1996), and propagates in circumferential direction with about 70% to 80% rotor speed. The spike disturbance grows quickly in circumferential extension with decreasing rotation speed and turns into a fully developed stall cell within less than five rotations. A clear trend cannot be identified in which of the rotors in a high-speed multistage compressor the spike occurs.

Cumpsty (1989) and Hoying et al. (1999) thought to have identified the importance of the tip clearance vortex for the stall inception in respectively a numerical and experimental work. Cumpsty (1989) noted that the upstream spillage ahead of the rotor would cause the initial blockage leading eventually to rotating stall. Hoying et al. (1999) noted an oscillation

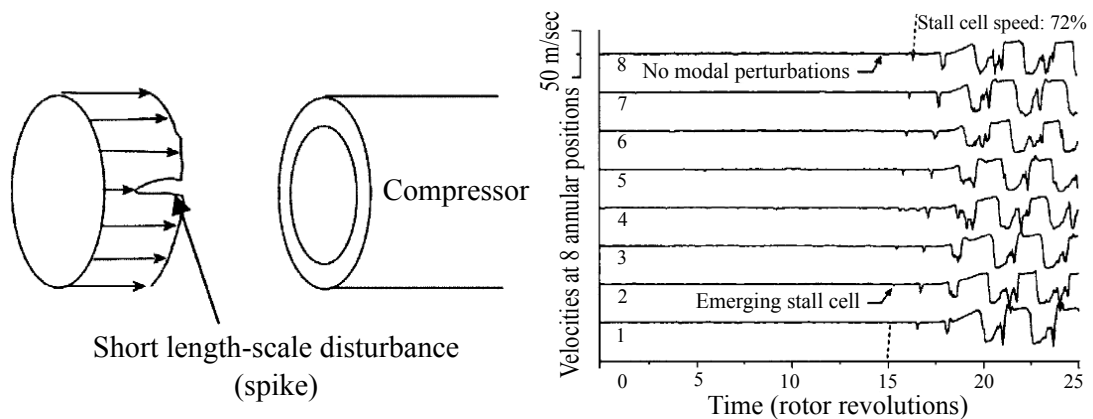


Figure 1.16 Spike (short length-scale disturbance) stall inception as measured by Day (1993), taken from the work of Vo (2001)

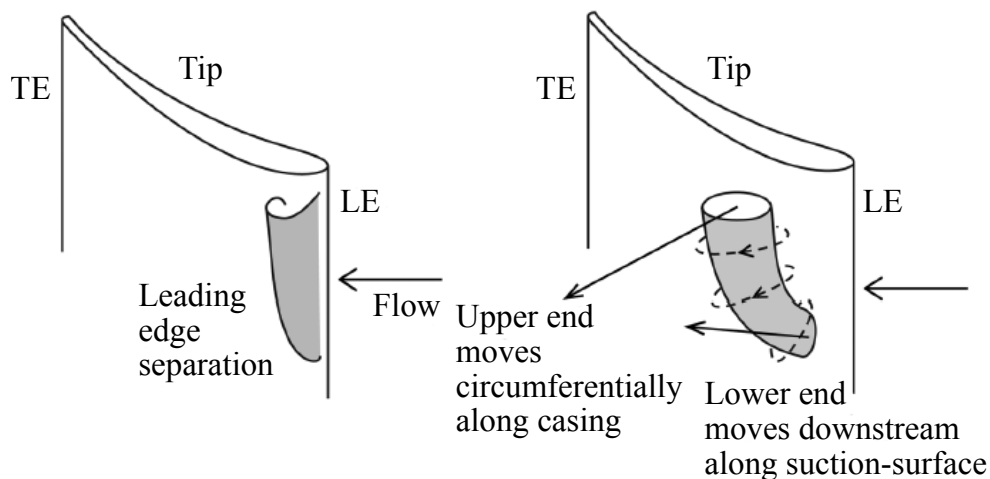
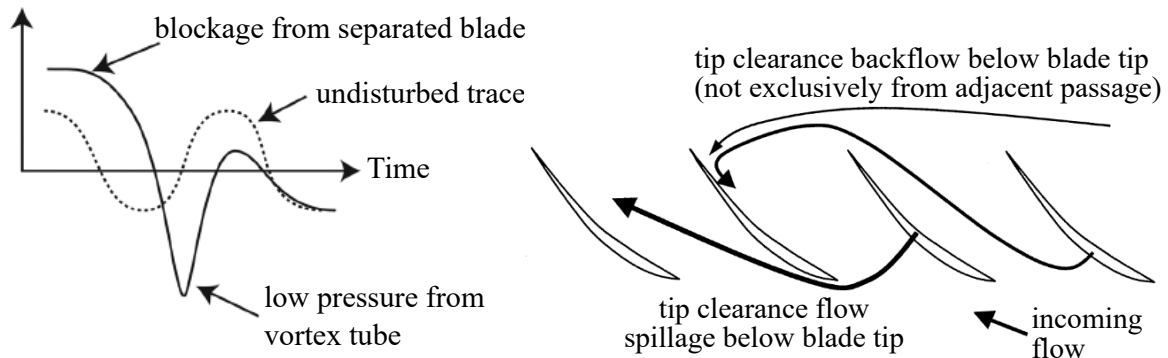


Figure 1.17 Leading edge separation causing tornado vortex sketched by Pullan et al. (2015)

of the tip clearance vortex and an important movement of it ahead of the rotor leading edge causing a stalling disturbance similar to a spike. The oscillation was believed to be caused by a breakdown of the balance between self-induced velocity of the vortex, which points in the up-stream direction, and the mean flow.

Inoue et al. (2000) identified the short length scale disturbance which induces the low pressure spots just upstream a rotor row for the first time. The authors proposed a tornado like separation vortex (sketched in Figure 1.17) extending from the blade suction side to the casing as the cause for the low pressure spot and increasing unsteadiness upstream of the rotor. The upper end of the separation vortex would move in circumferential direction, whereas the lower end would move downstream along the rotor blade suction surface.

The vortex induces a blockage of a blade passage that can be also detected upstream of the rotor. The high and low pressure zones close to each other result in the typical up-and-down signature of a spike which is depicted in Figure 1.18a.



(a) Typical spike trace in casing pressure measured by a stationary probe (Pullan et al., 2015) (b) Proposition of two criteria for spike type stall inception by Vo (2001)

Figure 1.18 Spike inception characteristics

Eventually, the disturbance is propagating in circumferential direction because the separation vortex will move to the pressure side of the adjacent blade and trigger another vortex there. The tornado like vortex could be the initial disturbance whose existence Emmons et al. (1955) has already assumed to be the initial cause for an important change in incidence to the blade row. This would consequently lead to rotating stall. Inoue et al. (2000) observed untypically the spike vortex propagation on a stabilized operating point with several reappearing of the vortex ("non-surviving" separation vortices), and called this state mild-stall. A new terminology should be avoided and the mild-stall could be understood as multiple part-span stall cells.

The works of Vo et al. (2008) and Vo (2010) propose two criteria that should be fulfilled for the occurrence of the spike stall inception (later results will show that they are not a necessity for cases without tip gap), see Figure 1.18b. These publications are mainly based on the results of the PhD thesis work from Vo (2001). The first criterion is saying that the interface between the tip leakage flow and the incoming flow should be parallel to the leading edge plane, causing spillage in the neighboring passage below the blade tip. The second criterion requires a tip leakage back flow at the trailing edge. The back flow impinges then on the pressure side of the adjacent blade and creates a blockage with partially inversed flow at the impinging location.

The numerical and experimental work of Pullan et al. (2015) supports the findings of Inoue et al. (2000) about the separation vortex forming on the rotor blade suction side but also shows that in a cascade and a shrouded rotor the spike leading edge separation vortex could form. Numerical results of their work are presented in Figure 1.19. Pullan et al. (2015) demonstrated that a corner separation would grow at the trailing edge of a rotor blade in the case without a tip gap (shrouded rotor). This occurs at the timepoint "t" in the passages with the "restaggered blades". A high incidence is consequently caused on the adjacent blade. Here a separation would form on the blade suction side leading edge (at timepoint "t+1.5") and propagate in circumferential direction. In the case with the tip gap, the increased

incidence was found to be caused by the blockage that is induced by the tip clearance vortex aligning with the rotor inlet plane. It grows no corner separation in this case. Furthermore, the authors observed that several spike type vortices could form and would not "survive" during the circumferential propagation. A final surviving vortex would cause then the rotating stall inception.

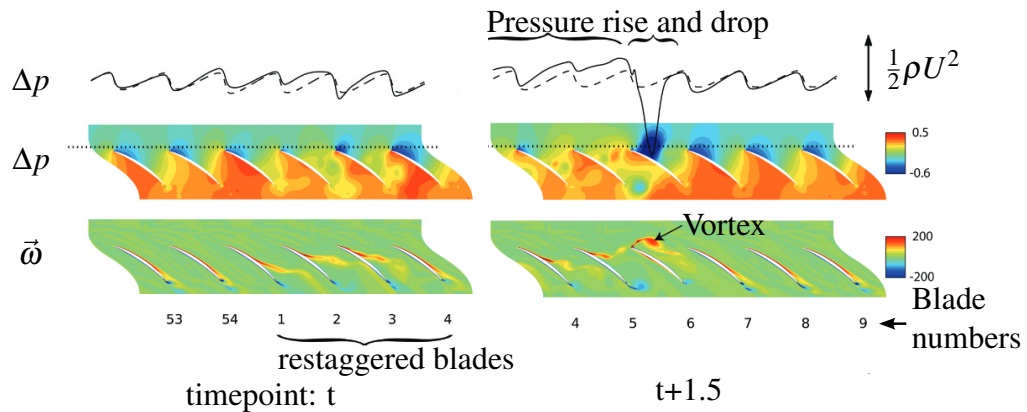


Figure 1.19 Spike caused by leading edge separation (restaggered blades) without tip clearance, adapted from Pullan et al. (2015)

Likewise, Weichert and Day (2013) found that the tip leakage spillage in the neighboring channel below the blade tip could be a consequence but not the cause for the spike inception. The authors did not observe a tip leakage vortex alignment with inlet plane prior to spike inception (one of Vo's criteria for the spike formation). They identified that the interface between the tip clearance vortex and incoming flow is disturbed prior to the initial formation of the leading edge separation. The perturbation of the interface (or passage flow) is located in the passage next to the suction side of the rotor blade with the highest pressure ratio through the tip gap. That has been identified to occur in the same blade passage in 90% of the investigated spike inceptions and related to a by 0.3% increased tip gap relative to the other tip gaps.

The work from Yamada et al. (2012) conducted very detailed pressure measurements and DES simulations on the already known research compressor from the studies of Inoue et al. (2000). The work comes to the same conclusions as the ones from Pullan et al. (2015) and Weichert and Day (2013) but was actually published earlier, and is illustrated in Figure 1.20. The tip leakage spillage below blade tip is understood to be caused by the separation vortex but not the other way around as initially thought by Vo (2001). The separation vortex was found to induce a blockage important enough to divert the tip leakage flow ahead of the rotor.

These results together suggest on the one hand likely formations of leading edge separation vortices (or spikes) while Vo's criteria being full-filled. On the other hand, Vo's criteria do not necessarily need to be full-filled for the formation of a spike separation vortex. The two criteria can be seen as possible conditions which give rise to high incidence. Another condition can be a corner separation causing blockage and therefore a high incidence as shown by Pullan et al. (2015).

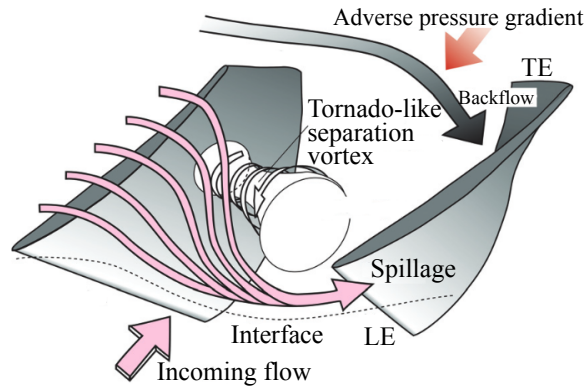


Figure 1.20 Tip leakage flow spillage because of spike separation vortex as proposed by Yamada et al. (2012)

### 1.4.3 Rotating instabilities

A certain rotating disturbance - today controversy called rotating instability (RI) - has been observed at stable operating points at loaded to near surge operating points, in contrast to the rotating stall which appears usually only very close to the stability limit at low mass flow rates. The rotating instability can be understood as a mild stall and induces a characteristic frequency bump in a pressure spectrum compared to a single peak for rotating stall, see Figure 1.21. The research on rotating instabilities has been mostly conducted with low-speed research compressor, and increased tip gaps so far. Among only a few others, Baumgartner et al. (1995) studied rotating instabilities in high-pressure compressors. That makes it a relevant subject to research in this work on a high-speed research compressor in Section 5.

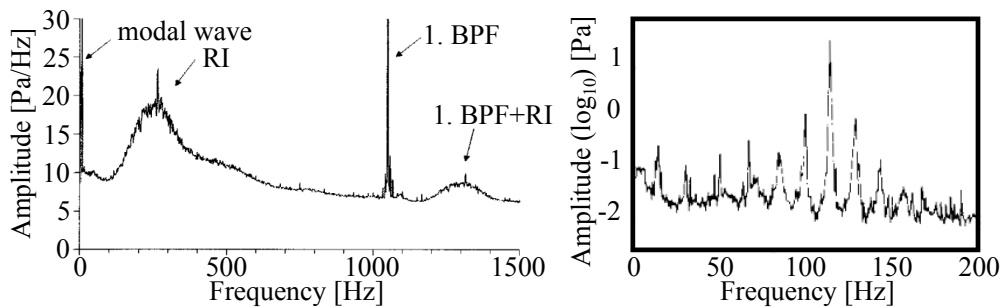


Figure 1.21 Frequency spectrum of casing pressure adapted from the works of Mailach et al. (2000) (left), and Mathioudakis and Breugelmans (1985) (right)

Mailach et al. (2000) observed an oscillation of the tip clearance vortex in the low-speed research compressor with increased tip gap based in Dresden, as depicted in Figure 1.22 on the left. These oscillations in one passage are found to influence the tip clearance vortex in the neighboring blade passage. The steep tip clearance vortex trajectory leads a lower tip leakage mass flow rate, thus flatter tip clearance vortex trajectory in the adjacent blade passage. That is repeated vice versa in the subsequent passage. The disturbance is therefore believed to propagate in circumferential direction, and to leave an alternating static pressure

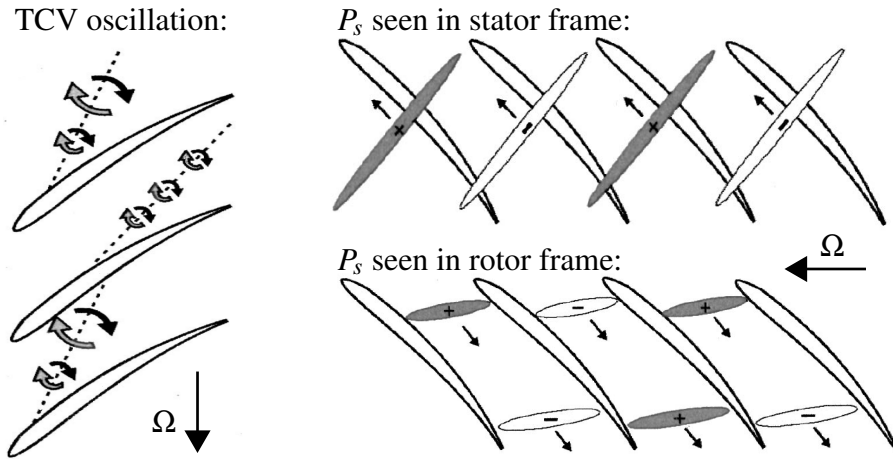


Figure 1.22 Tip clearance vortex oscillations (left) and propagation mechanism (right) of rotating instability seen by Mailach et al. (2000) in a rotor, in the stator frame (top) and rotor frame (bottom)

trace as sketched in Figure 1.22 on the right. The authors use the term "rotating instability" because the frequency bump suggests an unstable behavior regarding varying number of cells and rotation speed over time at a stable operating point. On the contrary, rotating stall with multiple cells can describe a stable condition over a few rotations (depending on the machine design and rotation speed), which would appear as one discrete peak in a frequency spectrum of casing pressure for example.

Despite of the newly introduced term, this type of rotating disturbance was already identified by Mathioudakis and Breugelmans (1985) as small rotating stall. They identified groups of rotating stall cells with varying cell numbers in a 1.5 stages research compressor.

Kameier and Neise (1997a) showed that the amplitude of the rotating instability would increase from its first appearance while closing the outlet valve in an experimental work on a low-speed high-pressure axial fan. Most interestingly the authors note that the frequency bump appears in signals obtained in the fixed and also rotating (rotor blade) frame. Furthermore, the authors noted a linear evolution of the phase angle of the cross-spectrum between the casing (or blade surface) pressure measurements of two probes. That indicates a constant (common) rotation speed and continuous existence for all peaks in the frequency bump. Only the amplitude of each peak appears to fluctuate over time. That allows to compute the rotating frequency ( $\omega_{RI,grad}$ ) from the gradient of the cross phase over the frequency bump, according to Equation 1.8. The linear cross phase evolution over the frequency bump is also supported by experimental work from Hermle (2015).

$$\omega_{RI,grad} = \frac{\Theta_{12}}{\frac{\Delta\Phi_{12}}{\Delta f_x}} \quad (1.8)$$

with:

$\Delta\Phi_{12}$ : Cross-correlation phase difference



$\omega_{RI,grad}$ : Rotating frequency of rotating instability calculated with gradient in cross phase spectrum

$\Delta f_x$ : Frequency range in cross spectrum

$\Theta_{12}$ : Circumferential distance between two pressure probes

Furthermore, Kameier and Neise (1997a) noted that the sum of the magnitudes of the rotation speeds of the rotating instability in the fixed ( $\Omega_{RI}^{fixed}$ ) and rotating frame ( $\Omega_{RI}^{rotating}$ ) is equal to the rotor rotation speed ( $\Omega$ ) as stated in Equation 1.9.

$$\Omega = \Omega_{RI}^{fixed} + \Omega_{RI}^{rotating} \quad (1.9)$$

where  $\Omega_{RI}^{fixed}$  and  $\Omega_{RI}^{rotating}$  are the rotating frequencies given relative to the shaft speed  $\Omega$ .

Moreover, Kameier and Neise (1997b) introduced the idea of a circumferential non-uniform fluctuating rotating source mechanism causing the multiple peaks in the frequency spectrum. The authors conducted a simplified experiment with a speaker, emitting a sinusoidal signal, mounted on a rotating disk, see Figure 1.23 on the left. In the rotating frame (RI frame) a single peak would appear in the pressure spectrum, whereas in the fixed frame multiple peaks would appear, see Figure 1.23 on the right. That experiment showed that the distance between the peaks in the frequency spectrum of the fixed frame corresponds to the rotation speed of the RI ( $\Omega_{RI}$ ). All peaks ( $\omega(m)_{RI}$ ) in the spectrum can be therefore related to a source frequency ( $\omega_{RI,source}$ ) with multiple circumferential modes ( $m$ ) according to Equation 1.10. Kameier and Neise (1997b) explained the multiple modes as characteristic for the Fourier components of a circumferential non-uniform source distribution. From their work, it is not clear how the speaker (a point source) is creating the circumferential non-uniformity.

$$\omega(m)_{RI} = \omega_{RI,source} + m \cdot \Omega_{RI} \quad (1.10)$$

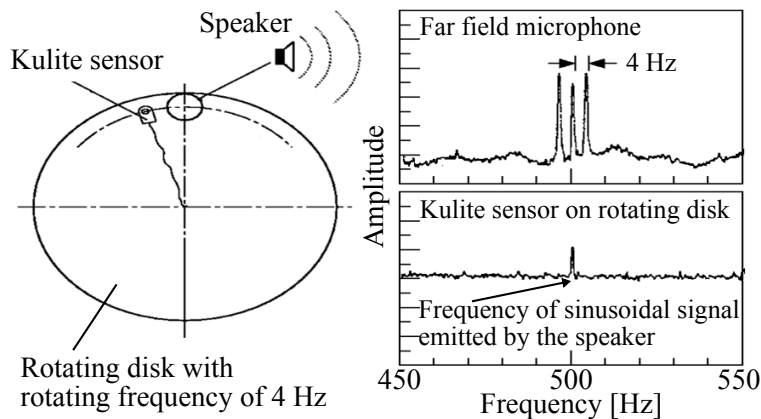


Figure 1.23 Rotating source (left), fixed and rotating frame spectra (right) as investigated by Kameier and Neise (1997a)

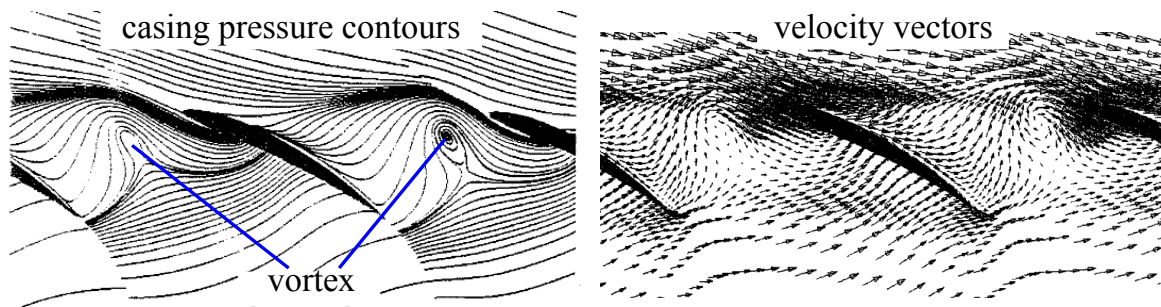


Figure 1.24 Interaction between reversed flow and tip leakage flow found by März et al. (2002) in a numerical work, casing pressure contours (left) and inst. velocity vectors (right)

März et al. (2002) investigated numerically and experimentally rotating instabilities in a single stage compressor with increased rotor tip gaps. The authors identified a radial vortex inside of the blade passage to be the cause for the TCV oscillations and hence the driving unstable mechanism for the rotating disturbance. The vortex appears as low pressure spot in the casing pressure measurements (see Figure 1.24 on the left). Axially reversed flow in the passage was found to interact with the tip leakage flow (see Figure 1.24 on the right), and assumed to cause the rotating instability vortex at about mid-passage close to the leading edge. The tip leakage flow is consequently diverted upstream of the blockage, induced by the separation vortex, and gets a trajectory almost parallel to the inlet plane. The vortex moves then from the suction to the pressure side of the passage at half the rotor speed. The downstream movement of the vortex was not described by the authors. Neither did the authors present the friction lines on the suction side of the rotor blades, where they could have missed that the separation vortex is actually extending from the blade suction surface to the casing.

Closely related are the results from Inoue et al. (2004) and Yamada et al. (2013) who also found a separation vortex forming at the leading edge of the rotor blade suction side near the tip. The authors performed experimental and numerical studies on the same 1.5 stages axial research compressor as already used for the spike stall inception studies (Inoue et al., 2000). The rotor tip gap was increased from 1% to 3% for this study. The separation vortex was found to be caused by the tip clearance vortex breakdown in the precedent rotor passage, see Figure 1.25. The breakdown induces a blockage and thus increased incident on the subsequent rotor blade (against rotor turning direction) and thus the separation vortex. A tip clearance vortex breakdown will follow in this passage, while the flow recovers in the preceding passage (separation vortex flows downstream and leaves passage). The disturbance thus rotates against the rotor turning direction. The description of the vortex formation is very similar to the one proposed for the spike like separation vortex. The separation vortex observed by Inoue et al. (2004) could correspond to the one already found by März et al. (2002) earlier.

The tip clearance vortex breakdown could not be observed by others, however the influence of one passage's tip clearance vortex on the one in the subsequent passage reminds of the propagation mechanism proposed by Mailach et al. (2000).

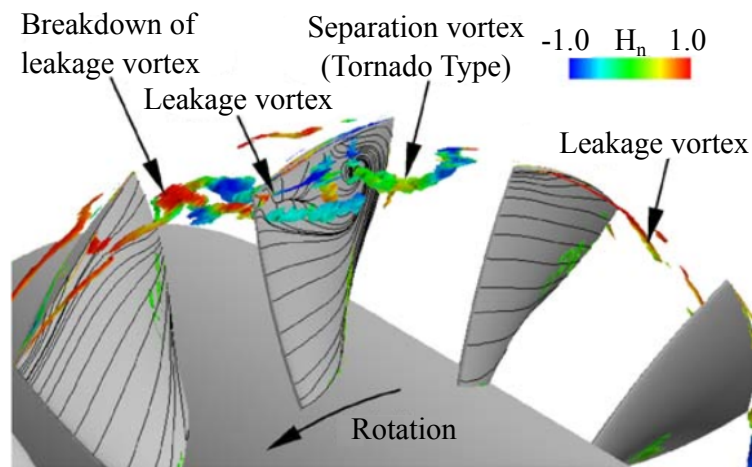


Figure 1.25 Rotating instability propagation mechanism as observed by Inoue et al. (2004), colored by helicity ( $H_n$ )

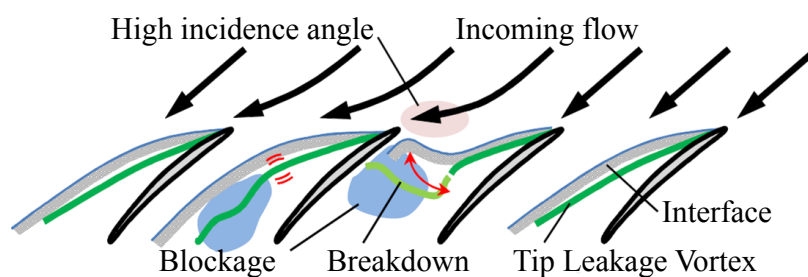


Figure 1.26 Rotating instability propagation mechanism as proposed by Yamada et al. (2013) and already seen by Inoue et al. (2004)

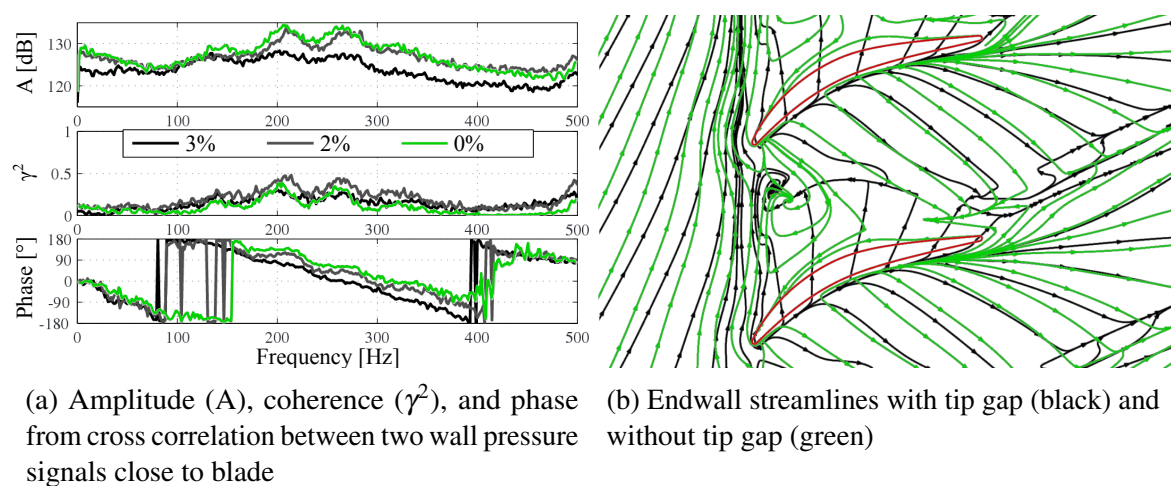


Figure 1.27 Characteristics of rotating instability with (black, grey) and without (green) tip gap adapted from Beselt et al. (2013)

Beselt et al. (2013) identified the typical rotating instability frequency bump in a numerical and experimental study on an annular cascade with and without tip (hub) gap, see Figure 1.27a. The coherence (similarity) spectrum between two signals obtained at different circumferential positions show a good fit for these frequencies. Characteristic is also the constant cross-spectrum phase evolution over the frequency bump, which has been found as well by Hermle (2015). The authors provoked the RI by increasing the incidence angle up to  $16^\circ$ . They therefore imply that another source mechanism than the tip clearance vortex has to be found. In the case with tip gap, their numerical study reproduced the results from März et al. (2002), Inoue et al. (2004), and Yamada et al. (2013) about a blade suction side separation vortex formation and the tip leakage vortex breakdown. In both gap cases, the authors noted a casing (hub) boundary layer separation spreading in the circumferential direction as depicted in Figure 1.27. A separation on the suction side of a blade induces a blockage and upstream spillage in the neighboring passages. Beselt and Peitsch (2012) demonstrated that the frequency of the rotating instability is proportional to the ratio of the blade chord length and inlet axial velocity to the concerned blade row ( $\frac{c}{V_x}$ ).

**In summary,** the source mechanism of the RI appears to be closely related to the spike formation at stall inception. For both instabilities, a flow separation has to occur, causing a probable leading edge separation vortex. With a large enough tip gap, the tip clearance vortex will induce the blockage, while no corner separation is found. A tip gap small enough or no tip gap and an incidence angle high enough will lead to the formation of an open corner separation, which could induce the necessary blockage for the formation of the vortex. The separation vortex appears to be weaker for RI than during the spike stall inception, as found during the comparison of two tip gap cases by Inoue et al. (2004). It is somewhat the mild-version, however letting a different trace in the spectra.

It has to be proven yet, what is eventually inducing the unstable behavior. Suggestions indicate that the tip clearance vortex breakdown causes the multiple frequencies as a consequence of the interaction between the tip clearance vortex and the separation vortex. That raises the question if there is likewise an important interaction between the corner separation and a leading edge separation or horseshoe vortex inducing the frequency bump in the case without tip gap. A sporadic formation of a leading edge vortex could also cause the unstable behavior, which is to be proven yet.

#### 1.4.4 Surge

Stall is an instability in the circumferential direction, whereas surge is a mono-directional instability in the axial direction. The whole compressor is affected and the mass flow rate is pulsing with a low frequency (1 to 100 Hz), which can lead to reversed flow if the pulsation is of large enough amplitude. Surge can be mainly of two types: classic and deep surge, see Figure 1.28. The classic surge is characterized by a low frequency oscillation of the mass flow rate without globally reversed flow. It occurs more often in compressors with a low pressure ratio or in compression systems with a small plenum volume ( $V_{pl}$ ). In these systems, the energy stored in the plenum is too small for a flow reversal. During a deep surge, the mass

flow rate oscillation is of large amplitude and causes periodically a global negative mass flow rate. That type of surge occurs in compressors with a high pressure ratio, and in systems with a large plenum volume (such as a combustion chamber in gas turbine applications), as stated by Day (1994).

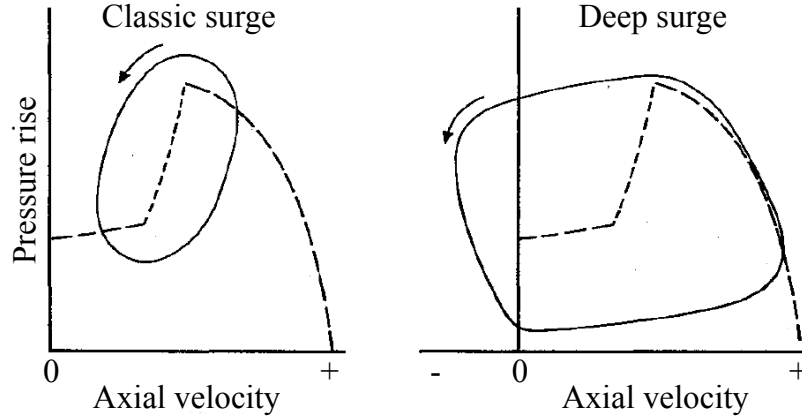


Figure 1.28 Classic and deep surge sketched on compressor characteristics by Day (1994)

Greitzer (1976b) understood the importance of the whole system (compressor, ducting, plenum volume, throttle) for the surge characteristics and proposed therefore a model taking into account the whole system. They defined the widely known Greitzer B parameter (Equation 1.11) from this model. It can indicate whether a certain compressor configuration will stall or surge. The system will experience surge if the value of B exceeds a critical one, which has to be found for every new compressor system (compressor, ducting, etc.). Greitzer (1976a) proposed  $B = 0.8$  as critical value. For smaller values, the system will only experience rotating stall around a time mean stable (possibly) lower mass flow rate and pressure ratio.

$$B = \frac{U}{2c} \sqrt{\frac{V_{pl}}{A \cdot L}} \Rightarrow \frac{\text{pressure force}}{\text{inertial force}} \quad (1.11)$$

with:

$U$ : Shaft speed

$V_{pl}$ : Volume of the plenum chamber downstream of the compressor

$L, A$ : Length and area of the compressor duct

$c$ : Speed of sound

Crevel (2013) estimated the B parameter to 0.87 with Equation 1.11 for a previous version of the compressor, which is the subject to study in this thesis work and will be introduced in Section 2.1. This was understood as an elevated value for B, and the compressor was shown to experience deep surge.

Courtiade and Ottavy (2013a) gave a very fine description of a surge cycle based on casing pressure measurements, and divided the cycle in four phases, as indicated in Figure 1.29:

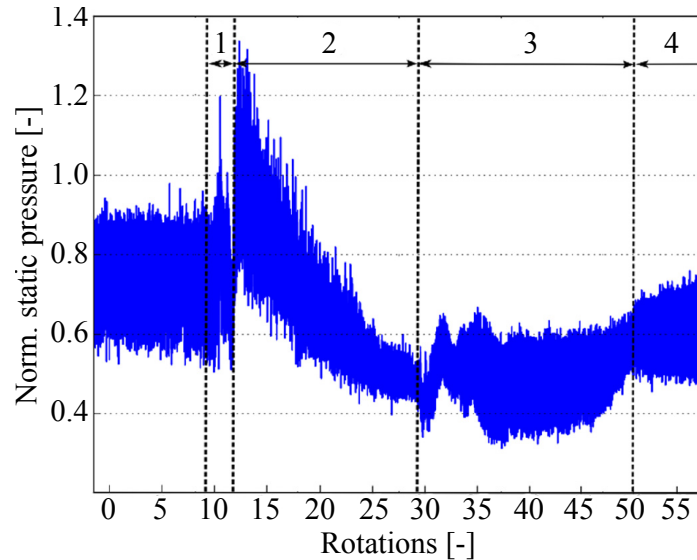


Figure 1.29 Surge cycle seen by casing pressure measurements of Courtiade and Ottavy (2013a)

- The phase 1 extends over only a few rotations, during which the stall inception occurs and the rotating stall cells grow very rapidly. These cells introduce a blockage. The flow reverses and impinges with non-adapted flow angles on the trailing edge of the stalled rotor blades.
- In the second phase, the flow is reversed and the static pressure is dropping while the compressor discharges. At this time point, no axial casing pressure gradient is found anymore.
- During the third phase the flow reattaches progressively while rotating stall cells are still present in the flow field.
- The compressor builds up slowly the pre-surge pressure ratio during the fourth phase. This phase can take as long as the three other phases together.

## 1.5 Research objectives

The literature review has shown that most of the studies are conducted on low-speed research compressors or single stage compressors, which represent a less challenging environment for the measurements than the one in high-speed multistage compressors. The results of these works need to be confirmed under realistic aircraft engine flow conditions. Furthermore,

the spike type stall inception and source of rotating instabilities is not fully understood yet. Therefore, this work will investigate experimentally and numerically the flow field in a high-speed multistage compressor at stable and unstable operating points, with a focus on three objectives:

1. The first objective of this work is to characterize the flow field of the high-speed multistage compressor CREATE at stable operating points, and to point out limitations to measurement and simulation methods. This limitation analysis in Chapter 3 goes beyond typical uncertainty estimations and requires deep knowledge of the flow field.
2. Clocking was presented as one possibility to decrease losses and thus increase the efficiency of a compressor. Studies about clocking in high-speed multistage compressors are still rare. The second objective of this work will be therefore the investigation of clocking in CREATE and will be presented in Chapter 4.
3. Open questions remain about the spike stall inception and rotating instabilities. Chapter 5 will describe several types of instabilities arising in CREATE at different operating speeds. The third objective of this thesis is the validation of the already known instability mechanisms and giving new insights into rotating instabilities found in measurements and numerical results.

# Chapter 2

## Flow field exploration and analysis methods

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Numerical and experimental data sets were acquired for the study of the flow field in a high-speed axial research compressor, which will be introduced in Section 2.1. The



measurements are challenging due to the high-speed environment that is characterized by high pressure and temperature. This environment causes strong forces on the measurement probes in the flow field with danger of probe rupture. At the same time, the probes need be of miniature size because the studied flow phenomena are at the scale of millimeters. The probes are furthermore required to have a high-frequency response for time-resolved measurements at high-speed. Numerical simulations have the advantage of giving access to the full flow field regardless of the challenging high-speed environment. Though, they have limited prediction capabilities when keeping reasonable simulation costs today.

In this thesis work, the flow field is studied experimentally with steady measurements of pressure and temperature, high-frequency casing pressure measurements, and laser Doppler anemometry measurements. The corresponding measurement methods and their uncertainties will be presented in Section 2.2. The numerical work consists of realizing and interpreting of steady and unsteady simulations that are introduced in Section 2.3. Furthermore, the signal treatment and analysis methods will be presented in Section 2.4. A summary of all conducted measurements and simulations can be found at the end of this Chapter in Section 2.5.

## 2.1 Presentation of the compressor and test rig

The 3.5 stages high-speed axial research compressor CREATE is the subject to research in this thesis work. CREATE stands in French for Compresseur de Recherche pour l'Etude des effets Aérodynamiques et TEchnologiques. The compressor was designed and built by Safran Aircraft Engines, and is tested on a 2 MW test rig in the laboratory for fluid mechanics and acoustics (LMFA) at Ecole centrale de Lyon, France. The geometry and rotation speed of the compressor are representative of high pressure compressor median rear blocks of a modern turbojet engine. The number of stages has been chosen so that the flow field contains a representative magnitude of secondary flow effects, while keeping a reasonable power requirement for the test-rig.

The compressor and the inter-row labeling are presented in Figure 2.1, and its characteristics are summarized in Table 2.1. The rotor shaft is driven at a design speed of 11543 rpm, and rotor 1 tip speed of  $313 \frac{m}{s}$ . The flow is slightly transonic in the first stage and fully subsonic in the last two stages. The circumferential periodicity of the 3.5 stages is reduced to  $\frac{2\pi}{16} = 22.5^\circ$  by choosing all number of blades as a multiple of 16 during the compressor design process as stated in Table 2.2. The number of stator blades is identical for the stators 1

Table 2.1 Standardized characteristics of CREATE at design point

|                               |                           |
|-------------------------------|---------------------------|
| Outer casing diameter         | 0.52m                     |
| Shaft speed                   | 11543 rpm                 |
| Rotor 1 tip speed             | $313 \frac{m}{s}$         |
| Rotor 1 tip inlet Mach number | 0.92                      |
| Nominal mass flow rate        | About $12.7 \frac{kg}{s}$ |

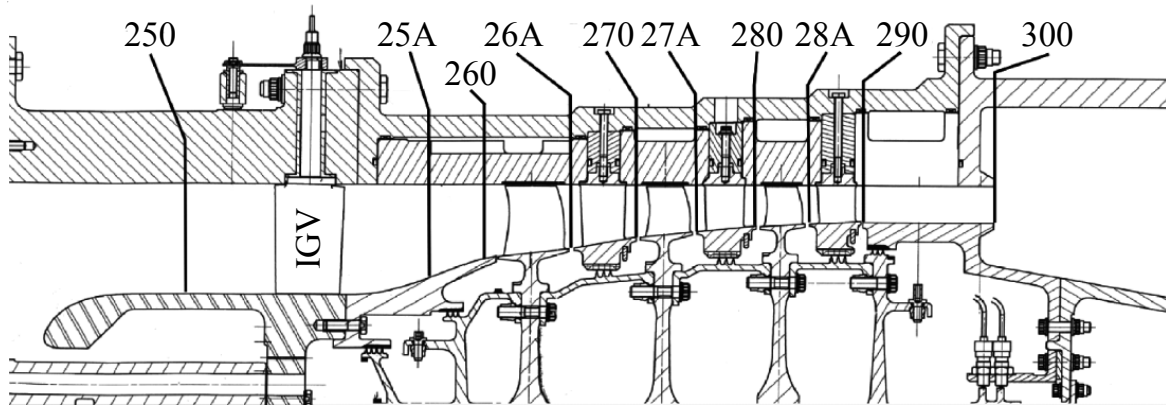


Figure 2.1 Meridian view on the 3.5 stages of the compressor CREATE

Table 2.2 Blade numbers of the compressor CREATE

|                                        | IGV | R1 | S1  | R2 | S2  | R3 | S3  |
|----------------------------------------|-----|----|-----|----|-----|----|-----|
| Number of blades $2\pi$                | 32  | 64 | 112 | 80 | 112 | 80 | 112 |
| Number of blades ( $\frac{2\pi}{16}$ ) | 2   | 4  | 7   | 5  | 7   | 5  | 7   |

to 3. Together with a movable stator 2 ring, this allows the study of clocking, which will be the subject of Chapter 4.

Eight struts with NACA 0024 airfoils are located far upstream of the IGV. The number of struts does not fit into one spatial period of the remaining compressor ( $\frac{2\pi}{8}$  vs.  $\frac{2\pi}{16}$ ). Their influence on the flow field has always been expected to be negligible because of their axial distance to the compressor but will be newly quantified and discussed for the estimation of the compressor performance in Section 3.1.2.

The inter-row planes are labeled from 25A to 300 as shown in Figure 2.1. All planes downstream of rotors are labeled with an "A", with exception of the plane "25A" downstream of the IGV. The planes downstream of the stators are labeled with a "0". The labeling will be used to identify the axial positions in the following Chapters. The shrouded stator design has to be regarded when studying the flow field close to the hub, and was already subject to research by Marty and Aupoix (2012). An increased inter-row distance allows probe traverses. The implementation of an outer casing moving-ring technology allows the moving of one or several probes in the circumferential direction to measure over the whole spatial periodicity of the compressor in the inter-row planes and at the rotor tips.

Numerous studies have been conducted on previous versions of CREATE. That gives a profound knowledge-base for numerical and experimental methods for the investigation of the flow field in the high-speed multistage compressor environment. The following list is not exhaustive but shall show the directions of research:

- Design and analysis of the flow field of the first version of CREATE, Touyeras and Villain (2004)

- Effect of turbulence model on flow field investigated with steady RANS simulations, Marty et al. (2008)
- Establishing CREATE test case for experimental and numerical methods, Ottavy et al. (2012)
- Proposition of extension for modal decomposition method for the analysis of rotor-stator interactions, Courtiade et al. (2012)
- Thesis work on the unsteady flow field and the onset of instabilities, Courtiade (2012)
- Interaction between the shrouded stator hub leakage flow and the main flow, Marty and Aupoix (2012)
- Acoustic resonance as surge precursor and fine description of surge, Courtiade and Ottavy (2013a)
- Simulation of surge inception and a surge cycle, Crevel (2013)
- Zonal Detached Eddy Simulations on interaction between the rotor 1 and the IGV, Marty et al. (2014)

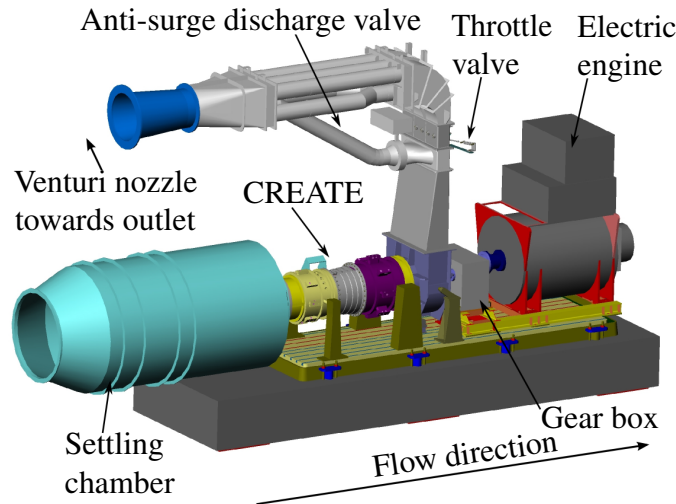


Figure 2.2 ECL-B2: 2 MW compressor test rig at the LMFA at Ecole centrale de Lyon, France

**The test rig** is realized as an open loop cycle (see Figure 2.2), which sucks the ambient air through a settling chamber into the compressor. The settling chamber includes a throttle which reduces the total pressure to about 0.74 times the ambient pressure at the inlet ( $P_{t_{inlet,mes}}$ ). This lowers the needed electrical power when performing measurements with a standardized compressor mass-flow rate, according to Equation 2.1. A 2 MW electric motor coupled with a gearbox drives the compressor shaft. The standardization helps to eliminate the influence of variations in the ambient measurement conditions as well. The secondary devices for lubrication and cooling demand 0.8 MW, while the real demand of the compressor is about 1.1 MW. Downstream of the compressor, a throttling butterfly-type valve is used to control the mass flow rate. Before the exhaust of the open loop cycle, a Venturi nozzle is installed to measure the mass-flow rate ( $\dot{m}_{mes}$ ), see Equation 2.1.

$$\dot{m}_{std} = \dot{m}_{mes} \cdot \sqrt{\frac{T_{inlet,mes}}{T_{ref}}} \cdot \frac{P_{ref}}{P_{inlet,mes}} \quad (2.1)$$

where the reference (ref) values correspond to ambient conditions,  $T_{ref} = 288.15 \text{ K}$  and  $P_{ref} = 101325 \text{ Pa}$ .

For eliminating the influence of the ambient measurement conditions, all the measurements are standardized according to Equations 2.2 to 2.5.

$$\Omega_{std} = \Omega_{mes} \cdot \sqrt{\frac{T_{ref}}{T_{inlet,mes}}} \quad (2.2)$$

$$P_{std} = P_{mes} \cdot \frac{P_{ref}}{P_{inlet,mes}} \quad (2.3)$$

$$T_{std} = T_{mes} \cdot \frac{T_{ref}}{T_{inlet,mes}} \quad (2.4)$$

$$V_{std} = V_{mes} \cdot \sqrt{\frac{T_{ref}}{T_{inlet,mes}}} \quad (2.5)$$

where *mes* always stands for a measured value after post-processing, and *std* for a standardized value. A specific heat capacity ratio of 1.4 is used for the post-processing of the measurement data. That corresponds closely to the mean specific heat capacity ratio between the inlet and outlet of the compressor.

**Machine oil in flow channel** The compressor shaft bearings are operated using oil for lubrication. In this compressor, an oil leakage into the flow channel was identified origin from upstream of the rotor 1. The leakage problem could not be repaired during the measurement campaign. The oil is centrifuged in the rotor 1 and stays close to the casing. This was identified with the help of laser Doppler anemometry measurements (see introduction of the measurement technique in Section 2.2.4). A contaminate fouling of the optical access for the laser Doppler anemometry was experienced during the measurements in the blade passages of the rotor 1 but not the rotors 2 and 3. The exact influence of the oil on the compressor flow field cannot be characterized but is considered small enough relative to the clean flow for allowing a trustworthy flow field analysis.

## 2.2 Experimental methods

This thesis work makes use of the existing knowledge about steady and high-frequency measurements in the high-speed compressor environment at the LMFA, see for example the work of Ottavy et al. (2012) and most recent thesis works of Bulot et al. (2010) and Courtiade (2012). Hereinafter the main characteristics of these measurements will be thus only briefly summarized.

### 2.2.1 Steady global performance measurements

The inlet total temperature is measured with ten probes in the settling chamber ( $T_{t_{in}}$ ). And the inlet total pressure ( $P_{t_{in}}$ ) is measured with a Kiel probe at plane 250 upstream of the IGV (see Figure 2.1). The outlet flow conditions ( $P_{t_{out}}$ ,  $T_{t_{out}}$ ) are measured with probe rakes at six circumferential positions and five span-heights at plane 300 (outlet of the compressor, upstream of the discharge collector). That gives a total of 30 probes for the flow conditions at the outlet of the compressor. As introduced before, the butterfly-type valve is used to adjust the operating point of the compressor. Where having multiple probe measurements, the data is arithmetically averaged. The isentropic efficiency is then calculated according to Equation 2.6.

$$\eta_{is} = \frac{\left(\frac{P_{t_{out}}}{P_{t_{in}}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{t_{out}}}{T_{t_{in}}} - 1} \quad (2.6)$$

The mass flow rate is estimated with a Venturi nozzle. The measurement uncertainties and the method for the mass flow rate estimation are detailed in the work of Courtiade and Ottavy (2013b). The relative experimental uncertainties for the compressor performance are given in Table 2.3. Apart from the measurement uncertainties, which only concern the data acquisition devices, errors can be committed while interpreting the performance data. These errors will be discussed in Section 3.1.2.

Table 2.3 Relative experimental uncertainties for the compressor performance measurements

|                       |              |
|-----------------------|--------------|
| Pressure ratio        | $\pm 0.07\%$ |
| Isentropic efficiency | $\pm 0.20\%$ |
| Mass flow rate        | $\pm 0.35\%$ |

### 2.2.2 Steady measurements

Steady measurements have been performed at all inter-row planes from 250 to 290 for the static and total pressure, total temperature, and the flow angle. These measurements have been conducted for a peak efficiency and a loaded operating point. The operating points will be defined in Section 3.1. At plane 250 only one circumferential position can be measured. At the remaining planes, the flow has been investigated over four stator pitches ( $\approx 12.857^\circ$ ) thanks to rotatable probe rigs, where seven would cover a complete spatial compressor period ( $22.5^\circ$ ). The limitation to four stator pitches comes from measurement project constraints. But numerical simulations demonstrated that the major physics of the flow is contained in four passages. The circumference is resolved with a measurement point every  $0.2^\circ$ . At casing, this corresponds to an arc of 0.9 mm, which is roughly the distance between two pressure taps of the probe. Seven three-hole probes and thermocouples are used for the steady measurements. The thermocouple is located 1 mm above the pressure taps on the probe. This is taken into account for the post-processing of the steady measurements. Multi-hole

probe calibrations and measurements are very common and not detailed any further here. More details about the alignment of the probes and the measurement uncertainty are stated in the report of Goguey et al. (2011) and listed as an example for the planes 25A and 290 in Table 2.4. The uncertainty estimation for multi-hole probe measurements is detailed for example in the work of Godard (2010).

During this thesis work, it has been identified that the multi-hole probe faces challenges in the multistage high-speed compressor environment that arise independently from the measurement uncertainty and are discussed in Section 3.2.1.

Table 2.4 Relative uncertainty of steady three-hole probe measurements in CREATE

|                   | 25A             | 290             |
|-------------------|-----------------|-----------------|
| Total pressure    | $\pm 0.2\%$     | $\pm 0.2\%$     |
| Static pressure   | $\pm 0.5\%$     | $\pm 0.3\%$     |
| Total temperature | $\pm 0.5^\circ$ | $\pm 0.6^\circ$ |
| Flow angle        | $\pm 0.5^\circ$ | $\pm 0.5^\circ$ |

### 2.2.3 High-frequency casing pressure measurements

The high-frequency casing pressure measurements allow studying non-intrusively the instantaneous tip flow field at stable and unstable operating points. At stable operating points, visualizations of flow structures, such as the tip clearance vortex, can be obtained with flow field reconstructions based on ensemble averages. They are necessary because of a physical limitation to synchronously recorded measurement points. This will be discussed in the following Section. The technique and skills for the high-frequency measurements have been developed at the LMFA for example during the thesis work of Bulot et al. (2010) and Courtiade (2012), and the acquisition system which has been used for this thesis work is detailed in the latter one.

The main high-frequency casing pressure measurement characteristics are summarized in Table 2.5. During the acquisition, an analog low-pass filter (4<sup>th</sup>-order Bessel filter with constant group delay) is used, and during the post-processing a perfect digital filter is applied to the signals for studying only the valid part of it. That follows the recommendation of the manufacturer and ensures having the resonance frequency of the sensors outside of the studied frequency band.

#### Localization of casing pressure probes

Measurements can be achieved simultaneously with 48 high-frequency pressure probes. Each inter-row plane can be equipped at maximum with six probes around the circumference. The probes are placed at 4, 14, 24, 40, 44, and 54 minutes (zero minutes corresponds to the physical top of the compressor) for allowing to capture circumferential rotating flow structures such as rotor-stator interactions or instabilities.

Table 2.5 Characteristics of high-frequency casing pressure measurements

|                       |                                                                                        |
|-----------------------|----------------------------------------------------------------------------------------|
| Pressure probes       | Kulite, XTE-190 family                                                                 |
| Sampling rate         | 500 kHz                                                                                |
| Resonance frequency   | between 290 and 440 kHz                                                                |
| Valid frequency range | up to 150 kHz                                                                          |
| Sensor surface        | 1 mm <sup>2</sup> , 2.8% axial chord (rotor 1)<br>0.22° of circumference at the casing |
| Pressure range        | 15, 25 and 50 PSI depending on axial sensor location                                   |

Above each rotor, twelve sensors can be installed in removable blocks, which are indicated in Figure 2.3. The probe locations in the blocks allows covering  $9^\circ$  in circumferential direction, and 12 different axial positions. The block can also be mounted reversed, giving additional 12 positions for all the rotors. For the rotors 2 and 3, an additional symmetrical block allows another 12 positions. Thus in total there are 24 axial positions for the rotor 1, and 36 for the rotors 2 and 3. All the casing pressure probes are fixed to rotating rings which allow a circumferential displacement of one spatial periodicity of  $22.5^\circ$ . The flow field can be reconstructed with ensemble averages using the whole spatial information. That results in the measurement grids shown for one example passage per rotor in Figure 2.3 on the right hand side. Missing axial positions are due to temporary probe defects during the measurements.

Transient phenomena can only be resolved with 12 sensors per rotor, where groups of three probes are located at an identical circumferential position. This makes it difficult to give spatially well resolved instantaneous views of the flow field.

The high-frequency casing pressure measurements were conducted with two different goals. Firstly, a global description is given for the unsteadiness in the compressor, which is obtained at 48 evenly distributed locations. In this way, the whole compressor operating range has been investigated from the choke to surge at different rotation speeds as listed in Table 2.6. The system has been brought to surge more than 26 times. Secondly, a fine spatial

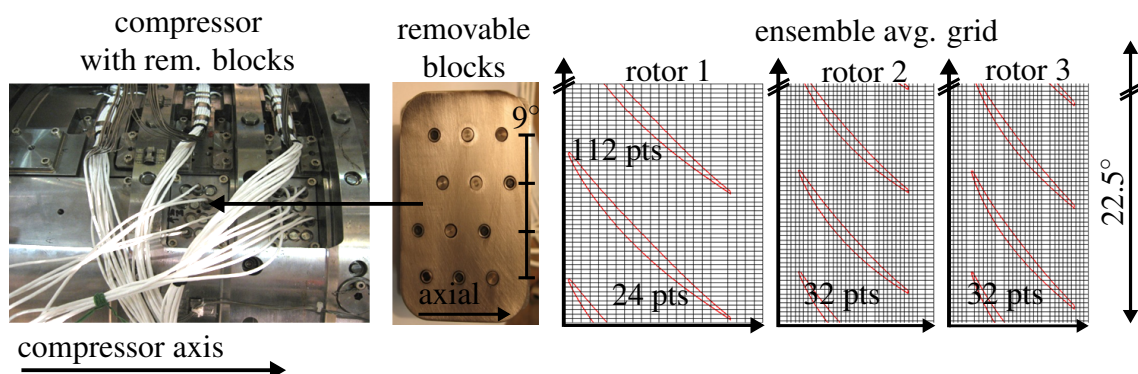


Figure 2.3 High-frequency casing pressure measurement mounted on removable blocks (left and middle) and zoom on the measurement grids per rotor (right)

description of the casing pressure flow field is conducted at the nominal compressor speed (100%  $\Omega$ ). These measurements were carried out for the peak efficiency, loaded and near surge operating point for one clocking position (see operating point definition in Section 3.1). The fine description is covering the full spatial period in all inter-row planes and above the rotors with a circumferential discretization of  $0.2^\circ$ . Thanks to the removable blocks, the compressor axis is resolved at 124 probe locations.

Table 2.6 Summary of high-frequency wall pressure measurements from choke to surge

| Objective | Rotor   | Shaft [°] | Clocking position [no.] | No. of OP pts. | Axial positions | Angular range [°] | No. of angular pts. |
|-----------|---------|-----------|-------------------------|----------------|-----------------|-------------------|---------------------|
| Inter-row |         | 1         | 1,2,3,4,5,6             | 15             | 6               | 360               | 6                   |
|           |         | 0.9       | 3                       |                | 6               | 360               | 6                   |
|           |         | 0.8       | 3,4,5,6                 |                | 6               | 360               | 6                   |
| Rotor tip | 1, 2, 3 | 1         | 3, 5                    | 15             | 12,12,12        | 9                 | 4 (12 probes)       |
|           | 1, 2, 3 | 0.9       | 3, 5                    |                | 12,12,12        | 9                 | 4 (12 probes)       |
|           | 1, 2, 3 | 0.8       | 3, 5                    |                | 12,12,12        | 9                 | 4 (12 probes)       |

### Uncertainty of high-frequency casing pressure measurements

The uncertainty of the high-frequency casing pressure measurement is composed of a spatial positioning and a calibration error. The details of the error calculation are given in the work of Courtiade (2012), who used the same measurement system and pressure probes, and applied here. Courtiade (2012) found that the statistical error is negligible when computing ensemble averages with more than 10000 points per temporal window. Also the digitization error is negligible with less than 5.3 Pa. The combined uncertainty on the steady and fluctuation parts of the static pressure signal are given in Table 2.7 for the three rotors. The uncertainties are calculated relative to the mean pressure at the given locations. The maximum uncertainty on the fluctuations is found at about 20% of blade chord related to the acceleration spot at the suction side of the blades.

Table 2.7 Combined uncertainty of high-frequency casing pressure measurements

|         | Steady value | Fluctuations, mean value | Fluctuations, max value |
|---------|--------------|--------------------------|-------------------------|
| Rotor 1 | 3.69%        | 1.33%                    | 2.63%                   |
| Rotor 2 | 3.73%        | 1.03%                    | 1.74%                   |
| Rotor 3 | 3.72%        | 0.76%                    | 1.45%                   |

### 2.2.4 Laser Doppler anemometry measurements

The laser Doppler anemometry (LDA) method allows non-intrusive measurements of the axial and circumferential velocity components in all inter-row planes from 25A to 290, and at



the rotor tips at several axial positions radially from 75% span-height to a span-height close to the casing. Seeding particles with an average size of  $1\ \mu\text{m}$  are added to the flow at the inlet of the test rig (upstream of the settling chamber). The particles are ideally supposed to follow perfectly the main flow in the compressor, while not modifying it (that will be discussed in the uncertainty evaluation in Section 3.2.2). The velocity components are each calculated from the scattered light by the seeding particles when passing through a network of interference fringes, which are produced by two laser beams coming from the same source with different ways. The LDA measurement setup for CREATE is not unique to this thesis work and is thus not presented in more detail here. Besides of being a widely known technique in literature, it is detailed in the work of Courtiade (2012).

The LDA method only allows a statistical description of the flow field because of irregular arrival times of the seeding particles and the low mean acquisition frequency compared to the BPF. Moreover, the deterministic velocity of the flow field is obtained from ensemble averages (see Section 2.4). This limits the LDA measurements to the study of stable operating points. In the following Section, all LDA measurement locations and operating points, and the measurement uncertainty are given.

### **LDA measurement locations and operating points**

The LDA measurements were performed in all inter-row planes, and in the rotor 2 and 3 tip regions. The LDA measurement was not possible in the rotor 1 tip region due to the oil leakage mentioned in Section 2.1. As for the high-frequency casing pressure measurements (Figure 2.3), a specific removable block is used to measure the velocities in the tip region of the rotor passages. Portholes in the blocks allow the optical access to the flow field at predefined axial positions. Hopper extensions are mounted on the portholes to avoid fast contaminate fouling of oil on the glass of the optical access holes.

The measurements were performed at peak efficiency (PE) and near surge (NS) operating points. That allows the study of two flow fields with the largest possible difference. Generally, the flow has been investigated over one stator pitch ( $\approx 3.214^\circ$ ) with an circumferential discretization of  $0.201^\circ$ . Downstream of the stators, the circumferential discretization has been adjusted locally to  $0.101^\circ$  for the fine description of the stator wakes. Also the span-wise discretization has been adjusted to the flow field with in-between 7 to 11 measurement points. Isolated measurements were conducted over the full spatial period ( $22.5^\circ$ ) for the investigation of rotor-stator interactions.

The number of acquired particles at a measurement point depends on the operating point and location due to measurement constraints. At a loaded operating point in the rear part of the compressor, the particle frequency might fall from usually 40 kHz at the inlet to 2 kHz in the hub and tip regions, or inside of wakes. Generally, 250,000 to 500,000 particles were acquired for one measurement point in space.

All measurement locations are listed in Table 2.8 but not necessarily all local discretization refinements are given for readability reasons. Two clocking positions have been investigated with the LDA measurements. Clocking was introduced in Section 1.3.2, and the results will be discussed in Chapter 4.

Table 2.8 Summary of LDA measurements

| Location    | OP    | Clocking position [no.] | No. of heights | Angular range [°] | Discretization [°] |
|-------------|-------|-------------------------|----------------|-------------------|--------------------|
| 25A         | PE    | 3                       | 9              | 11.5              | 0.201              |
| 26A         | PE/NS | 3                       | 9              | 11.5              | 0.402              |
| 270         | PE/NS | 3                       | 11             | 3.214             | 0.201              |
| 27A         | PE/NS | 3                       | 11             | 3.214             | 0.201              |
| 280         | PE/NS | 3,5                     | 7              | 3.214             | 0.201/0.101        |
| 28A         | PE/NS | 3                       | 11             | 3.214             | 0.201              |
| 290         | PE/NS | 3,5                     | 8              | 3.214             | 0.201/0.101        |
| R2,x1/2/3/4 | PE/NS | 3                       | 2              | 3.214             | 0.201              |
| R3,x1/2/3/4 | PE/NS | 3,5                     | 4              | 3.214             | 0.201              |

### LDA measurement uncertainty

The LDA measurement uncertainty calculation has to take into account several sources of error: the error induced by the measurement system ( $\sigma_{laser}$ ), the positioning error of the measurement volume ( $\sigma_{pos}$ ), and the statistical error ( $\sigma_{st}$ ). The details of the error calculation are given in the work of Courtiade (2012), who used the same measurement system. Following the law of the propagation of uncertainties, the LDA measurement uncertainty can be estimated with Equation 2.7. The results of the same uncertainty calculation applied to the flow field ( $V_i$ ) of the current version of CREATE are listed hereinafter.

$$\sigma(V_i) = \sqrt{\sigma_{laser}^2(V_i) + \sigma_{pos}^2(V_i) + \sigma_{st}^2(V_i)} \quad (2.7)$$

**Individual sources of error** The error of the Dantec LDA system ( $\sigma_{laser}$ ) has been estimated experimentally to be less than 0.3% of the axial and 0.1% of the circumferential velocity:  $\sigma_{laser} < 0.003 \cdot V_i$ .

The positioning of the measurement volume is achieved by a Kuka robot with an uncertainty of 0.1 mm in each of the three axis directions. Eventually the uncertainty related to the positioning error of the measurement volume ( $\sigma_{pos}$ ) highly depends on the gradient in the flow field as listed in Table 2.9.

Table 2.9 LDA positioning error at nominal operating point

|                          | Weak gradient     | Strong gradient   |
|--------------------------|-------------------|-------------------|
| Axial velocity           | $0.2 \frac{m}{s}$ | $3.0 \frac{m}{s}$ |
| Circumferential velocity | $0.2 \frac{m}{s}$ | $1.0 \frac{m}{s}$ |

The statistical error ( $\sigma_{st}$ ) is proportional to the RMS of the velocity over the root of the number of registered particles. This error increases where high RMS values are found: inside

of wakes or in the tip flow field. Here the particle passing frequency can decrease by a factor of five compared to the free-stream. At the same locations machine oil can be found (leakage of lubrication oil for the bearings), which can induce larger particles with an uncontrolled size and lower velocity than the actual flow. These particles are mainly centrifuged in the rotors, and consequently travel downstream inside of the rotor wakes at the rotor tip. These large particles might not follow the main flow correctly and therefore increase the statistical error, which is given in Table 2.10.

Table 2.10 LDA statistical error at nominal operating point

|                          | Normal seeded zone | Weakly seeded zone |
|--------------------------|--------------------|--------------------|
| Axial velocity           | $0.2 \frac{m}{s}$  | $2.0 \frac{m}{s}$  |
| Circumferential velocity | $0.3 \frac{m}{s}$  | $1.5 \frac{m}{s}$  |

The combined uncertainty of the LDA measurements can be calculated according to Equation 2.7 for each measurement point. The uncertainties are not given for every measurement point in the following Chapters for readability reasons but the averages and maximum values are nevertheless considered. They are given for all inter-row planes in Table 2.11, where the mean value is obtained with an arithmetic average.

Table 2.11 LDA combined error at nominal operating point

|     | $V_x$ , mean value<br>$\left[\frac{m}{s}\right]$ | $V_x$ , max value<br>$\left[\frac{m}{s}\right]$ | $V_\Theta$ , mean value<br>$\left[\frac{m}{s}\right]$ | $V_\Theta$ , max value<br>$\left[\frac{m}{s}\right]$ |
|-----|--------------------------------------------------|-------------------------------------------------|-------------------------------------------------------|------------------------------------------------------|
| 26A | 2.00                                             | 14.01                                           | 2.38                                                  | 7.09                                                 |
| 270 | 1.90                                             | 3.13                                            | 0.93                                                  | 3.46                                                 |
| 27A | 2.32                                             | 4.12                                            | 2.19                                                  | 4.02                                                 |
| 280 | 1.10                                             | 3.88                                            | 1.78                                                  | 4.39                                                 |
| 28A | 1.85                                             | 4.18                                            | 2.11                                                  | 4.72                                                 |
| 290 | 1.69                                             | 3.05                                            | 0.85                                                  | 3.34                                                 |

Radial profiles of the combined uncertainty for the axial velocity ( $V_x$ ) are given in Figure 2.4 as an example at the planes 270, 27A, and 280. The radial profiles are obtained with spatial weighted averages using  $V_x$  as weight because the density is not measured. They are given for the peak efficiency and loaded operating points (see definition in Section 3.1). The uncertainty fluctuations over the circumference are presented as colored envelopes. The mean uncertainty stays fairly constant over the whole span at all planes. High uncertainty fluctuations are induced generally by the wakes and the tip leakage flow. A noteworthy difference between weighted and arithmetic circumferential averages is not found with the given resolution (17 points for one stator pitch), and  $V_x$  as a weight. That can be seen when comparing the arithmetic averages from Table 2.11 with the values in the radial profiles (Figure 2.4).

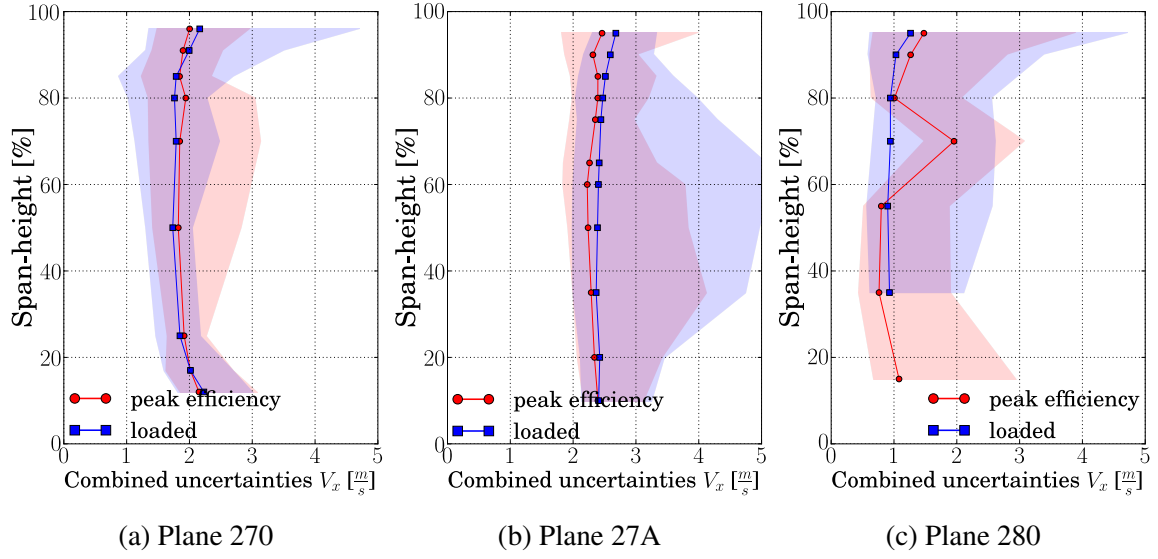


Figure 2.4 Radial profiles of combined axial velocity LDA measurement uncertainties

## 2.3 Numerical methods

This Section presents the numerical simulations that were conducted for this thesis work, and the parameters that were used for their realization. Post-treatment methods, specific to the numerical result fields, are also presented in this Section. All the numerical simulations were carried out using the flow solver elsA (Ensemble Logiciel pour la Simulation en Aérodynamique) developed by ONERA and CERFACS (Cambier et al., 2013). The code solves the Navier-Stokes equations for three-dimensional, viscous, and compressible flows with a cell centered approach on structured multi-block meshes. For this thesis work, only steady and unsteady Reynolds Averaged Navier-Stokes (RANS) simulations were conducted for the nominal rotation speed. Other methods such as Large Eddy Simulations (LES) could be conducted for obtaining a more precise prediction of the real flow field but this is technically and computation cost wise out of scope for this thesis work. A promising attempt with a spatially under-resolved LES on CREATE was presented by de Laborderie et al. (2016).

### 2.3.1 Simulation parameters

#### Reynolds averaged Navier-Stokes method

The Navier-Stokes equations for the description of a Newtonian viscous flow consist of time dependent continuity equations for the conservation of mass, conservation of momentum (three equations for three Cartesian directions), and the conservation of energy. The derivation and linking of the formulas are not unique to this thesis and not presented here. Certain important assumptions are made and simplifications applied - for linking and solving the Navier-stokes equations - that are also important for the comparison with the measurement results. These assumptions are summarized hereinafter. The fluid (air) in the

compressor is considered a perfect gas. Here the specific gas constant is taken as constant with  $R = 287.07 \frac{J}{kgK}$ , and the specific heat ratio is also considered constant with  $\gamma \approx 1.4002$ . The Prandtl number is fixed to  $Pr = 0.72$ , which is needed for the determination of the conductivity of air for the computation of the heat flux density vector - using the Fourier law.

A Direct Numerical Simulation (DNS) would solve the Navier-Stokes equations for all the spatial and temporal scales of the flow, which requires the spatial and temporal resolution to be fine enough. The mesh point number and size of the time step are closely related to the Reynolds number for a wall bounded flow, and increase with about  $Re^3$ , what leads to impossible computation costs for a multistage compressor simulation today. A less costly possibility is to solve the Reynolds Averaged Navier-Stokes (RANS) equations.

The RANS method consists of considering the instantaneous field as the sum of an ensemble averaged field and a fluctuation field. A classical time average (Reynolds average) is used for the density and the pressure. For the averaging of the velocity and energy, a Favre average is applied for taking into account density variations while simplifying the system that is to be solved. The averaging makes appear the Reynolds tensor, which relates the averaged and turbulent flow phenomena, and is equal to twice the turbulent kinetic energy. For the closure of the equation system a further hypothesis is therefore necessary. Here the most widely used one is the Boussinesq hypothesis, which is based on the analogy between the shear stress in a Newtonian fluid, and the effect of turbulence. The hypothesis implies isotropy in the Reynolds stresses. The Reynolds tensor is thus determined by relating the velocity field to a turbulent viscosity. That makes appear the two unknowns: the turbulent viscosity ( $\mu_t$ ) and the turbulent Prandtl number ( $Pr_t$ ), which is taken as constant with  $Pr_t = 0.9$ .

Summary of chosen constants:  $\gamma \approx 1.4002$ ,  $R = 287.07 \frac{J}{kgK}$ ,  $Pr = 0.72$ ,  $Pr_t = 0.9$

### Turbulence model

The flow is assumed to be fully turbulent because the mean Reynolds number based on the blade chord is about  $10^6$ . Only two equation turbulence models are used for the determination of the turbulent viscosity in this thesis work. A turbulent model with one equation would only link the turbulent viscosity to a mixing length. That requires less computation costs than a two equation turbulence model but the predictions of the real flow will be less accurate. In this thesis work, mainly the two equation  $k - \omega$  Wilcox (Goncalves and Houdeville, 2001) turbulence model has been applied, which expresses the turbulent viscosity as follows:

$$\mu_t = \rho \frac{k}{\omega} \quad (2.8)$$

where  $k$  is the turbulent kinetic energy, and  $\omega$  the specific rate of dissipation. There are advantages and disadvantages for every turbulence model, and in turbomachinery it is widely accepted that no model will give RANS the prediction capabilities of well resolved LES or DNS simulations today. A generally good flow field prediction could be obtained with the  $k - \omega$  turbulence model on a previous version of CREATE by Ottavy et al. (2012). In others works on CREATE, Marty et al. (2008) and Gerolymos and Vallet (2007) could show only minor improvements of the predictions with more advanced turbulence models

such as DRSM<sup>1</sup> (Speziale et al., 1991) or EARSM<sup>2</sup> (Wallin and Johansson, 2000). The magnitude of the influence on the flow field by a different (more advanced) turbulence model, such as the Wallin-Johansson EARSM, is discussed using steady RANS simulations in Section 3.1.1. Expecting only minor improvements, and due to an incompatibility of the EARSM models with the chosen unsteady rotor-stator interface information transport method in elsA, all simulations have been conducted with the original formulation of the  $k - \omega$  Wilcox turbulence model from 1998. The revisited (improved) version from 2006 is not available in the current version of elsA (Wilcox, 2008).

### Spatial scheme

The convective fluxes are computed with the third order upwind Roe scheme considering the minimal Harten entropic correction (Roe, 1981), which helps to respect the second principle of thermodynamics. The diffusive fluxes are calculated using a second order centered scheme.

### 2.3.2 Domain of simulation - space discretization

The simulation domain contains the compressor from the IGV to the stator 3. Although taking into account the IGV doubles the simulation domain in axial direction, numerical studies of the flow field in CREATE justify this approach. The studies have shown the importance of the wake and leakage vortices of the IGV for the correct simulation of the tip leakage flow field in the rotor 1 (Riéra et al., 2013). Certain simplifications had to be applied to the simulation domain to keep reasonable computation costs and due to feasibility reasons. They are summarized hereinafter, taking into consideration their impact on the flow field while performing a flow field analysis:

- To limit the calculation costs, only one natural periodic sector (i.e.  $\frac{2\pi}{16}$ ) is considered in the calculation domain. That imposes a spatial periodicity where instabilities cannot freely develop. Rotating stall cells could be only found as multiple of 16 for example.
- The eight struts upstream of the compressor would double the simulation domain axially (far upstream of IGV) and circumferentially due to the different spatial periodicity ( $\frac{2\pi}{8}$  vs.  $\frac{2\pi}{16}$ ). Their influence will be discussed in Section 3.1.2

<sup>1</sup>DRSM: The Differential Reynolds Stress Models are seven-equation turbulence models, which are not based on the Boussinesq hypothesis but solve a differential equation for each Reynolds stress component. That allows the model to account for anisotropy of turbulence and streamline curvature effects on turbulence. High computational costs and stability problems limit the use of this type of turbulence model (Troshin et al., 2014).

<sup>2</sup>EARSM: The Explicit Algebraic Reynolds Stress Model can be seen as an extended two-equation turbulence model, and a simplification of the DRSM. Assuming a local equilibrium of turbulence, an anisotropy tensor is added for the Reynolds stresses approximation, while neglecting advection and diffusion. The equations are fully Explicit and do not require an iterative process. The turbulent viscosity depends eventually of the velocity gradient (while taking into account the anisotropy of turbulence), whereas for a classical two-equation turbulence model, the turbulent viscosity is a constant value with the assumption of isotropic turbulence.

- Hot-air injection occurs at the hub just upstream of the rotor 1 (leakage of the air for counter balancing axial forces). The mass flow rate of this injection has been evaluated internally to be of negligible influence.
- The hub-leakage flow of the shrouded stators 1 to 3 is not taken into account. Their inclusion in the simulation domain is technically out of scope for this thesis work but will be discussed in Section 3.1.1.
- A floating IGV has been realized for meshing simplicity, whereas in the real machine the IGV can be rotated with a pivot, which blocks a part of the tip gap (see Figure 2.1). Furthermore, the hub-gap has been simplified. Equivalent hub and tip gaps have been estimated and are discussed in this Section.

## Mesh

Each blade passage is meshed with the software Autogrid (NUMECA, 2012) using a 0-H mesh topology with matching points at all block interfaces. The minimum cell size is equal to  $1\text{ }\mu\text{m}$  at the walls, which corresponds to a mean non-dimensional computed wall distance  $y^+$  of less than 1. The maximum wall distance around 5 can be found close to the leading edges of the rotor blades. The total number of grid points from IGV inlet to outlet boundary, considering a single passage for each row, is about 13.5 million. Its characteristics are summarized in Table 2.12. A spatial period of the compressor domain is discretized with about 66 million grid points for the URANS simulations.

Table 2.12 Number of nodes in the different blade passages of the simulation domain

| Direction        | IGV  | Rotor | Stator | Stator 3 |
|------------------|------|-------|--------|----------|
| Streamwise       | 243  | 155   | 155    | 183      |
| Pitchwise        | 73   | 57    | 49     | 49       |
| Spanwise         | 133  | 117   | 109    | 109      |
| Tip/Hub gap      | 25   | 25    | -      | -        |
| Total size (Mio) | 3.35 | 1.8   | 1.44   | 1.65     |

The mesh is built so that it captures accurately the RSI with the URANS simulations. The mesh convergence will be proven in the subsequent Section. The maximum numerical frequency, which can be captured with a mesh, is related to the minimal dimension of the mesh in pitchwise direction  $L_{\Theta,min}$  and the blade passing frequency (BPF), as advised by Gourdain and Leboeuf (2009) with Equation 2.9.

$$f_{max} = \frac{2\pi}{2 \cdot L_{\Theta,min} \cdot N_{Rotor}} \cdot BPF \quad (2.9)$$

The maximum numerical frequency ( $f_{max}$ ) which can be captured with this mesh is accordingly about 21 BPF, calculated based on the number of blades of the rotor 1. The fundamental frequency and its harmonics are therefore sufficiently taken into account. Gourdain

and Leboeuf (2009) make the link between a reasonable calculation time and the time step size based on the Shannon theorem in Equation 2.10.

$$no_{it} \geq \frac{2 \cdot f_{max}}{BPF} \quad (2.10)$$

where the minimum number of iterations ( $no_{it}$ ) per rotor passage should be higher than 42 in the present case. Note that this is only a minimum criterion.

### Mesh convergence

The eventually used mesh ("BASE" mesh) is mostly based on experiences that were obtained during simulations performed for previous versions of CREATE (Gourdain et al., 2012; Ottavy et al., 2012). During the creation process of the mesh, the "Grid Convergence Method" as proposed by Celik et al. (2008) was respected and is presented together with a view on differences in the flow field. The present grid convergence study is based on three cases and steady RANS simulations: The BASE mesh, a COARSE mesh with half the points in each mesh dimension, and a FINE mesh with double the points in radial direction. A refinement in one direction is accepted to limit the needed resources for the study with 3.5 compressor stages. The radial direction is found to have the most potential for space discretization improvement because of strong radial gradients related to the tip leakage flow.

Rotor 1 is taken as subject of the present study because it has the "poorest" space discretization regarding its relative blade height compared to the following blade rows. Grid refinement factors of 1.96 (COARSE/BASE) and 1.20 (BASE/FINE) are realized. As proposed by Celik et al. (2008), the mean order of accuracy is calculated based on the grid refinement factors and the ratio of the solutions (axial velocity) on the different grids. The corresponding formulas are detailed in the cited work. The mean order of accuracy ranges from 0.71 to 2.47 for different radial lines in the rotor 1 passage. That leads eventually to a discretization error for the axial velocity in the range from 0.09% to 1.25% with the BASE mesh, with its very local maximum in the tip region where the rotor flow field is strongly influenced by the tip clearance vortex. Figure 2.5 highlights the differences between the tip clearance vortex trajectories obtained with the BASE and FINE mesh. Two rotor passages are shown: In the left passage, high entropy contours in green (BASE) and red (red) shall highlight the zone mainly influenced by the tip clearance vortex. The passage on the left shows the tip clearance vortex trajectory (TCV) with the help of streamlines. Globally the tip clearance vortex trajectory is changed only marginally. That is found as red entropy zone extending slightly further in radial direction inwards (less than 1% span), and also with the help of the streamlines.

In the tip zone the mesh density is actually increased as shown in Figure 2.6. The radial grid point distance is given on the second horizontal axis. The axial velocity difference is given between the BASE and FINE mesh. The data is obtained with an extraction following a radial grid-line at the previously indicated axial position (Figure 2.6) in the rotor 1 passage. The grid density is constant over the main part of the span, with a densification close to the endwalls. A further increase of the grid points in the tip region, and coarsening of the mesh at mid-span might seem more ideal than applied here. The local differences between the



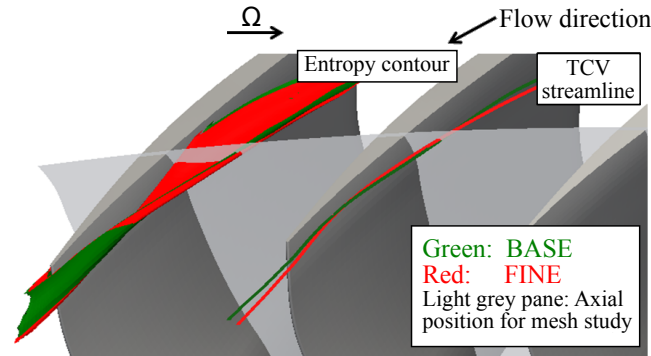


Figure 2.5 High entropy contour (left passage) and tip leakage vortex trajectory (right passage) in rotor 1 for BASE (green) and FINE (red) mesh

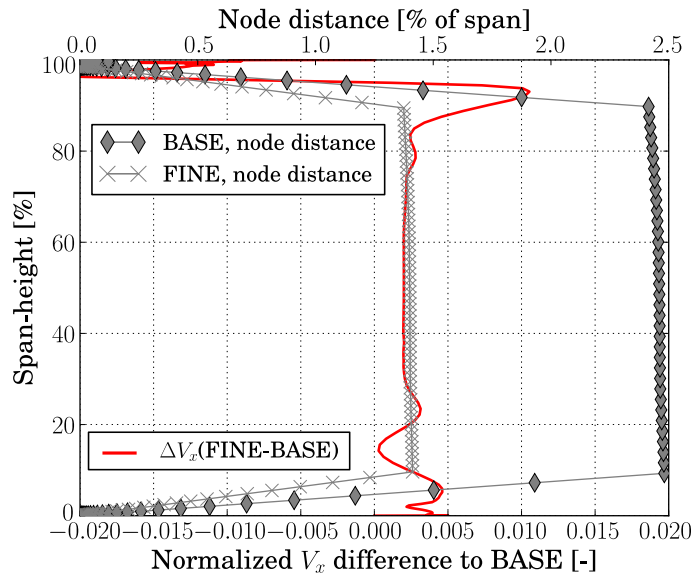


Figure 2.6 Axial velocity error as difference between BASE and FINE mesh, and radial node distance inside of rotor 1 passage

flow fields are smaller than 1% of the axial velocity. Here it is to be kept in mind, that this is the peak value found due to the small shift of the tip clearance vortex trajectory. This thesis work has also the goal to give a fine characterization of the flow field at mid-span because of possible separated blade boundary layers and for the study of stator clocking. This requires a sufficient discretization at mid-span as well, and explains the constant high discretization here.

The marginal tip clearance vortex trajectory change can be also found as local higher axial velocity close to the pressure side in Figure 2.7a. The rotor is turning from the left to the right. The axial velocity is given at the previously indicated axial position (see Figure 2.6) cutting through the tip clearance vortex at 95% span-height. The COARSE field finds an over-dissipated tip clearance vortex. That results in a rather linear axial velocity profile on the pressure side of the blade. This difference is considered more important than the one

found between the BASE and FINE mesh due the more important changing of the flow structure. Figure 2.7a shows the same axial velocity comparison for mid-span. In either mesh case, the rather undisturbed flow field in the rotor 1 at mid-span does not need the finest mesh discretization. Or in other words, the COARSE mesh is already fine enough for the good simulation of the velocity profiles at mid-span in the rotor 1 but not necessarily further downstream.

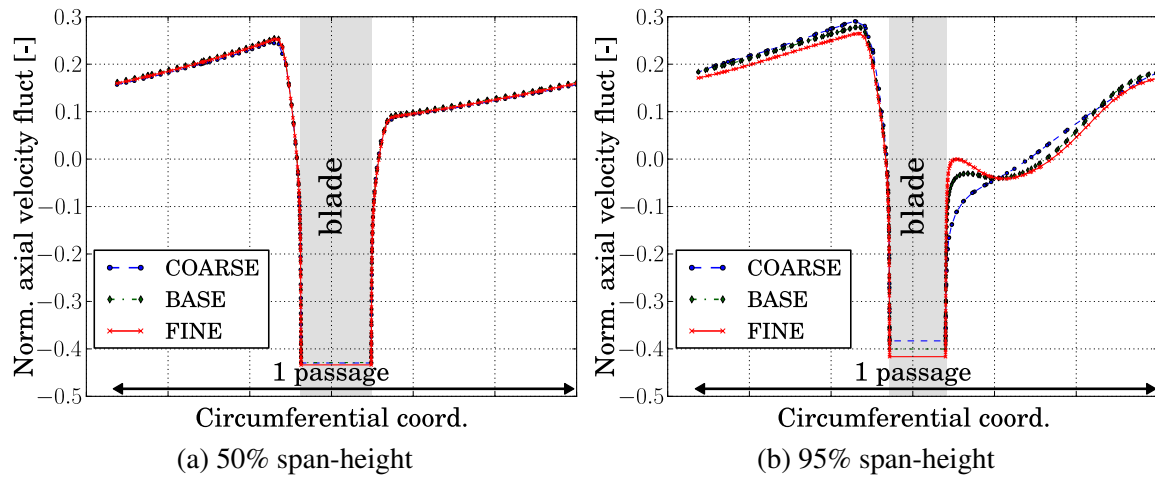


Figure 2.7 Mesh dependence based on axial velocity at about 66% blade chord of rotor 1, as indicated with a grey plane in Figure 2.5

The change in the overall compressor efficiency is equal for both the COARSE and FINE mesh with about 0.1 efficiency points as stated in Table 2.13. The changes in mass flow rate and pressure ratio are decreasing with a higher resolution of the mesh. That indicates the convergence of the performance values towards the finer meshes even though it is not perfect. Here it has to be considered that in the multistage environment, small errors accumulate in streamwise direction. That will be also shown for the time discretization study in Section 2.3.3. Note that the isentropic efficiency change from BASE to FINE in rotor 1 alone is smaller than 0.029 efficiency points for example.

Table 2.13 Mesh convergence in terms of overall performance, pressure ratio, efficiency

|        | Isentropic efficiency [%] | Pressure ratio [%] | Mass flow rate [%] |
|--------|---------------------------|--------------------|--------------------|
| COARSE | -0.09                     | -0.023             | -0.082             |
| BASE   | ref                       | ref                | ref                |
| FINE   | 0.10                      | 0.005              | 0.034              |

A further grid refinement in the tip and hub regions would significantly decrease the number of grid points at mid-span while keeping the same number of grid points along the span. That is not further considered. An increasing of the total number of grid points in radial direction cannot be realized because of the given size of the simulation domain and the related computation costs. Hence the following work makes use of the BASE mesh. The

changes in the TCV are considered as small enough in the multistage environment. High gradients in the tip region let differences appear large where only a negligible (small) shift of a flow structure occurs in space.

The time dependent transport of flow structures and interaction between blade rows can have an influence on the mesh convergence and needs to be evaluated with unsteady RANS simulations. The mesh independence of a rotating disturbance will be demonstrated in Section 5.2.1 for the BASE mesh and a finer mesh using unsteady RANS simulations.

### IGV hub and tip gaps

In the real compressor the IGV blades are mounted on a trunnion which allows a variation of the stagger angle. This trunnion is blocking a part (25%) of the tip gap. Note that the stagger angle is not changed for any data presented in this work. The hub gap geometry is simplified from a curved shape to a straight line between the leading and trailing edge as sketched (with exaggerated gaps) in Figure 2.8.

The influence of three versions of the simplified IGV geometries on the flow field is discussed here, see Table 2.14 for the gap characteristics.

- Original case: hub and tip gaps at leading and trailing edge as with real geometry
- Adapted case: Reduced hub and tip gaps for taking into account the gap surface reduction by the simplifying of the geometry. The hub and tip gap values at the leading and trailing edges are halved for this version, with the exception of the hub leading edge value because of having intersecting grid-lines otherwise. A decrease by 75% (reminder: 25% blockage) would not allow a good match between numerical and experimental values for the flow field downstream of the IGV.
- No gap case: Zero hub and tip gap as a third reference case.

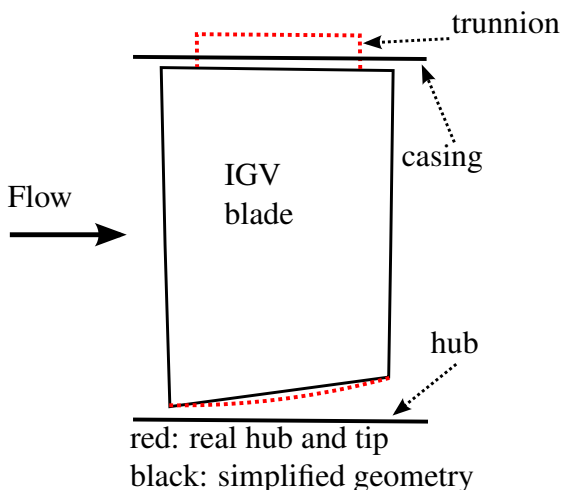


Table 2.14 Tested IGV hub and casing tip gaps (gaps used for this work: adapted gaps)

|          | hub  |      | casing |      |
|----------|------|------|--------|------|
|          | LE   | TE   | LE     | TE   |
|          | [mm] | [mm] | [mm]   | [mm] |
| original | 0.24 | 1.35 | 0.86   | 1.34 |
| adapted  | 0.24 | 0.67 | 0.43   | 0.67 |
| zero     | 0.00 | 0.00 | 0.00   | 0.00 |

Figure 2.8 Sketch of simplified (black) and real (red) IGV with exaggerated tip gaps

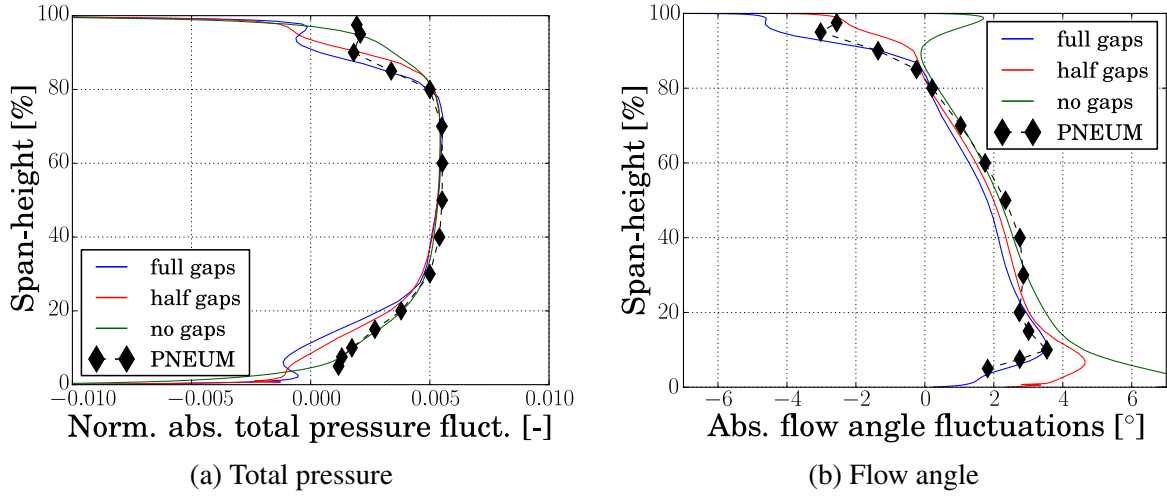


Figure 2.9 Radial flow profiles of normalized absolute fluctuations (rel. to average of full gaps case) for flow conditions at 25A (downstream of the IGV) with different IGV tip and hub gap sizes

The main goal for adapting the hub and tip is to obtain the best possible match between measurement and simulation data in terms of losses in the IGV, knowing the inlet (250) and outlet (25A) total pressure, temperature, velocity, and angle profiles from steady measurements at both planes, and the velocities from LDA measurements at plane 25A. Exemplary, the radial total pressure and flow angles profiles are shown for the three different tip/hub gap versions in Figure 2.9. The flow condition quantities are presented as normalized absolute fluctuations relative the overall mean of the "full gaps" case. The tip and hub leakage flow, and the hot-air injection cause thick boundary layers at the hub and casing. That is found as total pressure losses on 20% of span towards the endwalls. On the one hand, the full gaps version over-estimates the total pressure loss compared to the measurement and the no-gaps version. On the other hand, the flow is over-turned without any gaps due to classical but thick boundary layers (decrease of axial velocity) at the endwalls, see Figure 2.9b. The full and half gaps versions give an equally good fit with the measurement for both flow quantities. In this work, the half-gaps version is taken as compromise between a fit in total pressure losses and flow angle prediction.

### 2.3.3 Time discretization

Steady and unsteady RANS simulations are conducted for this thesis work. For the steady RANS simulations, a local time stepping method is used to "accelerate convergence". The pseudo time-marching is performed by means of an implicit time integration scheme, based on the Backward Euler scheme and a scalar lower-upper symmetric successive over-relaxation (LU-SSOR) method (Seokkwan Yoon and Jameson, 1987). The CFL (number of Courant Friedrichs Lewy) has been linearly rammed up during the first 2000 iterations of a steady simulation up to a CFL equal to 5.

The unsteady RANS simulations (URANS) have been conducted with the Gear method (Gear, 1971), which is based on Newton sub-iterations at each physical time step. The choice for this method is based on its usage for previous numerical work on CREATE (e.g., Ottavy et al. (2012)), and experience that has shown no difference in the flow field, while reducing the CPU costs, compared to one obtained with a Backward Euler scheme (Buffaz, 2012). The Gear method is briefly introduced hereinafter, followed by a discussion of the choice of the time step and the number of sub-iterations based on results from a published work during this thesis work (Schreiber et al., 2015b).

### Gear method

After the finite volume discretization in space, the URANS equations might be modeled as:

$$\frac{\delta Q}{\delta t} + FB(Q) = 0 \quad (2.11)$$

where  $Q$  represents the aerodynamic field and  $FB$  the flux balance. Discretizing the time derivative, the Gear method (Gear, 1971) is an iterative process giving access to the next time step as the root of the non-linear second order scheme:

$$H(Q) = \frac{3 \cdot Q - 4 \cdot Q^i + Q^{i-1}}{2 \cdot \Delta t^i} + FB(Q) \quad (2.12)$$

where  $i$  denotes the iteration process related to the physical progression in time. Searching the root  $W^{i+1}$  such that  $H(Q^{i+1}) = 0$ , sub-iterations are introduced based on the Newton method with the formulation

$$\left( \frac{\delta H}{\delta Q} \right)^{j,i+1} \Delta Q^{j,i+1} = -H(Q^{j,i+1}) \quad (2.13)$$

where  $j$  denotes the sub-iteration process of the Newton algorithm. That leads to the inner iteration

$$\left[ \frac{3}{2 \cdot \Delta t^i} \cdot I + J^{j,i+1} \right] \cdot \Delta W^{j,i+1} = -H^{j,i+1} \quad (2.14)$$

where  $J$  is the Jacobian matrix of space operators and source terms coming from a "steady like problem". Only the physical time step is used during the Newton sub-iterations, and has therefore a direct influence on the convergence process during the Newton sub-iterations. The sub-iteration residual  $H^{j,i+1}$  can be monitored as an average over the whole calculation domain in elsA.

### Setting time discretization by convergence study

The discretization in time is for one part fixed by the time step ( $\Delta t$ , see Equation 2.15), which is defined by a number of iterations per machine period ( $it_{period}$ ), respecting the mesh limitations that were introduced with Equation 2.10.

$$\Delta t = \frac{2 \cdot \pi}{16} \cdot \frac{1}{it_{period} \cdot \Omega} \quad (2.15)$$

The results of the study of the time discretization are presented in Schreiber et al. (2015b) and summarized hereinafter. The study has been conducted with  $it_{period}$  of 420, 840, and 1680. Here the biggest time step already fulfills the minimum criterion of Equation 2.10 with  $it_{period} \geq 42$ . A simulation with 210 iterations per period was found to be numerically unstable. The peak efficiency operating point is chosen for the following time discretization study. Note that it might lead to smaller differences than at a loaded operating point, where the expected unsteadiness is more important. It is a compromise between a reasonable calculation time of all the data points and an accurate convergence study.

A decrease of two order of magnitudes for the density residuals in the sub-iterations of the time stepping method is found to be a common practice in multistage turbomachinery (Gourdain et al., 2012; Salontay et al., 2010a). In cases with heavily increased grid size, a decrease of one order of magnitude is also accepted as a compromise between accuracy and calculation costs (Crevel et al., 2014; Riéra et al., 2013). The number of sub-iterations in elsA can be also fixed manually to have control over the simulation time.

Only a certain convergence level can be achieved with reasonable computations costs during the Newton sub-iterations of the Gear method. It is found that the sub-iteration convergence has an equally important influence as the decrease of the physical time step. A limitation of the number of sub-iterations to 12 does not degrade the sub-iteration convergence at subsequent physical iterations. Further tested numbers of sub-iterations are 20 and 32 sub-iterations.

A marginal change of the compressor performance is observed when going from 420 to 1680 iterations period in comparison to the increased calculation costs. Efficiency, pressure ratio and mass flow rate increase with finer time discretization at maximum about 0.2%, 0.36%, and 0.37% respectively. Significant differences are located in the tip region.

The tip leakage flow is sensitive to the change in time discretization. An overall normalized increase of the axial velocity is found with 2.3% at maximum in the tip region. That is shown to be connected to a decrease of the losses induced by the tip leakage flow with finer time discretization.

The velocity fluctuations are found to be far more sensitive to the time discretization, with an increase of 63% in the fluctuation energy with the finest time discretization. The decomposition of the flow field shows that the change in the rotor-stator interactions energy can locally be significant, but globally stays proportional to the overall fluctuations energy.

Eventually the physical time step has to be chosen as a function of calculation costs and motivation of the numerical work. The strong decrease of the time step size and increase of the number of sub-iterations are not in the interest of an industrial design process. If a study concerns effects of new technologies requiring accurate unsteady flow prediction, where the RSI could have a major contribution, the time discretization sensitivity has to be taken into account.

In conclusion for this thesis work, 840 iterations per period and 12 sub-iterations are found as best compromise between calculation cost and quality of the flow field prediction.

That combination does not limit the sub-iteration convergence at subsequent physical instants and leads to a trend of convergence of the most energetic RSI. An additional increase of the number of iterations per period or sub-iterations would not increase the relative importance of the energy of the RSI.

Summary of chosen parameters: 840 iterations per period, with 12 Newton sub-iterations

### **Time periodic convergence of the simulations**

Determining the state of convergence of an unsteady numerical simulation is important to ensure accurate and reliable results. It is not easy to judge the level of convergence of an unsteady RANS calculation involving a periodic flow compared to a simple iterative convergence in a steady RANS calculation. In the following, the time needed for traversing the whole domain with a defined speed is given as the ratio over the needed time for the rotation of a channel by one spatial period. The minimum number of calculated periods for covering the convection of flow field information from the inlet to the outlet is estimated at 13 periods. The number of periods for the upstream transport of flow information from the outlet of the machine to the inlet is estimated to 4.6 periods. The simulation should therefore calculate more than 17.6 periods. The convection speed and upstream transporting speed are estimated with the average axial velocity and the average speed of sound per row respectively. One convergence control method is the monitoring of the mass flow rates at the inlet and outlet of the calculation domain as for example done by Green et al. (2013) during a study of a single stage high pressure turbine. Convergence of the mass flow rate is reached when its difference between two consecutive periods would be below 0.02%. Salontay et al. (2010b) defined a change of 1% of mass flow rate over one rotor revolution as a convergence criterion.

Only the mass flow rate as one-dimensional value is not sufficient as a convergence criterion. Ahmed and Barber (2005) proposed therefore a FFT convergence criterion that relies on the evaluation of physical frequencies. Applying the criterion, a simulation is considered converged when the amplitudes of the main physical frequencies are stable between consecutive time intervals. Salontay et al. (2010b) conducted a computational investigation of vane clocking effects in a 3.5 stages research compressor at the Purdue University. The authors state that the periodic converged state was validated by means of Fourier Transform on the unsteady pressure field near the leading edge of the second stator. A periodicity of the rotor 2 wakes was supposed to be ensured by this method which is equal to the one proposed by Ahmed and Barber (2005). Among others Custer et al. (2012) check the periodic convergence in the same manner in a 1.5 stage turbine. A detailed value for the convergence level is not given in either of these cases.

Clark and Grover (2006) went further and introduced a single parameter for the estimation of the level of convergence by means of a multi-valued logic. The authors propose to define a series of fuzzy sets based on the time-mean level, DFT magnitudes and phase angles, cross-correlation coefficients (CCF), and fraction of overall signal power (PSD) for frequencies of interest. The overall convergence level is then an intersection of all fuzzy sets and defines itself a fuzzy set ( $f_C$ ). Here the authors define  $f_C \geq 0.95$  for two consecutive temporal periods as the criterion for reaching convergence of the periodic-unsteady flow field.

The convergence control for the CREATE simulations is for one part based on the Clark and Grover (2006) method. An example control trace is shown with only one harmonic ("h5") to ensure visibility in Figure 2.10. For another part, the mass flow rate is monitored at all inter-row planes to detect the maximum differences between two consecutive periods, and to look globally at the convergence process. Likewise, pressure and velocity signals of point probes are monitored at all inter-row planes.

The criterion of  $f_C \geq 0.95$  is applied because it appears possible to reach this level of convergence at all inter-row planes. It has to be considered that changes in small modulations of the flow field can lead to big variations of the amplitude ratios. Here it is advisable to look rather at an amplitude difference instead of a ratio. That was found at the inlet of the present calculation domain for example.

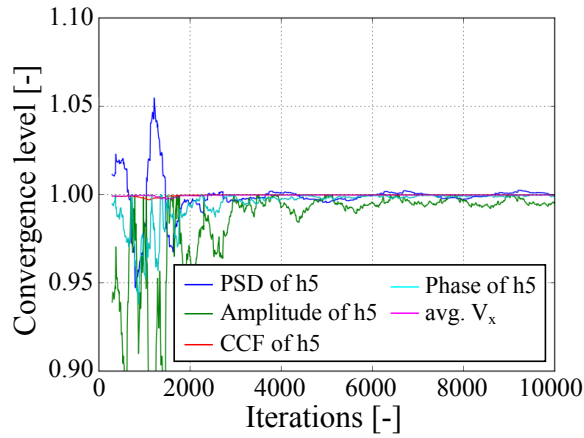


Figure 2.10 Example of Clark and Grover method applied to CREATE simulations indicating convergence after 10000 iterations ( $h_i$  = harmonic i)

### 2.3.4 Boundary conditions

Boundary conditions need to be set at the inlet and outlet of the simulation domain. The underlying flow conditions are derived from measurements at the inlet of the compressor. Furthermore, isothermal boundary conditions are applied on the hub and casing endwalls. The derivation of these boundary conditions is presented in this Section.

#### Inlet boundary conditions

The inlet conditions for the simulation domain are defined in the stator frame and constituted of the flow quantities enthalpy, total pressure, the direction of the velocity vector, and the turbulent quantities (turbulent kinetic energy and dissipation for the  $k - \omega$  turbulence model). Identical inlet conditions are used for the steady and unsteady RANS simulations because they are considered steady in both cases.

The flow quantities are deducted from measurements, which have been conducted as radial traverses with the pneumatic probes for one circumferential position about one chord



upstream of the IGV (plane 250). The circumferential position is located centered between the wakes of the struts. Furthermore, hot-wire measurements were conducted for the estimation of the turbulence intensity at the same location. Consequently, the losses induced by the struts are not taken into account for the inlet conditions but have to be considered for the comparison of the compressor performance (see in Section 3.1.2).

A delay in the start of the measurement campaign has forced the simulations to be ran with inlet conditions obtained from measurements of a previous version of CREATE ("old"). That has been done with the assumption of an unchanged inlet flow field for the two compressor versions because of an identical inlet of the test rig, ventilation system and IGV. The differences between the two flow fields at the inlet obtained afterwards ("new") can be seen for the total pressure and temperature in Figure 2.11a. They are smaller than the measurement uncertainty, with 0.1% and 0.5% respectively. Note the hot-air injection upstream of the IGV causing an total temperature peak towards the hub, and increased total pressure losses compared to the one found in the boundary layer towards the casing.

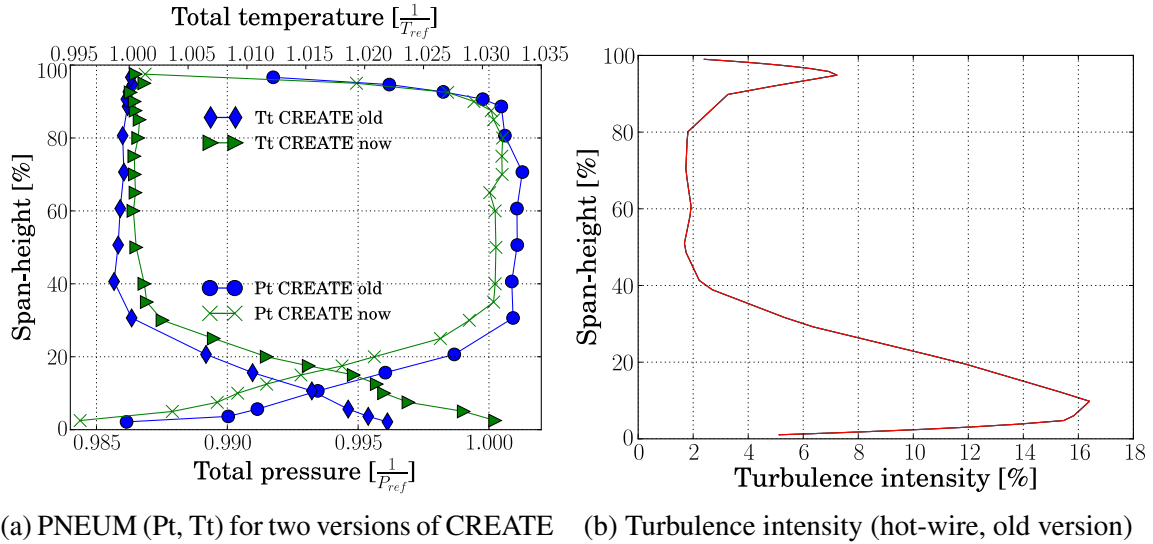


Figure 2.11 Measured inlet conditions for simulation domain

The radial profile of the turbulence intensity was measured only on the previous version of the compressor. Taking typical hot-wire measurement uncertainties  $\sigma_{\bar{V}}$  and  $\sigma_{V'}$  of 1% to 4% (Yavuzkurt (1984) or Ma (2012)) for the mean velocity ( $\bar{V}$ ) and the velocity fluctuations ( $V'$ ) respectively, the uncertainty of the turbulence intensity ( $I = \frac{V'}{\bar{V}}$ ) can be estimated with the propagation law of uncertainties according to Equation 2.16. This yields an absolute measurement uncertainty of the turbulence intensity smaller than 0.08% at any span-height at the inlet plane and is negligible for the (U)RANS simulations.

$$\sigma_I = \sqrt{\left(\frac{\partial I}{\partial V'} \cdot \sigma_{V'} \cdot 1.96\right)^2 + \left(\frac{\partial I}{\partial \bar{V}} \cdot \sigma_{\bar{V}} \cdot 1.96\right)^2} \quad (2.16)$$

The turbulent kinetic energy ( $k$ ) is derived from the measured turbulence intensity and the velocity vector according to Equation 2.17 for RANS simulations. The turbulent intensity is about 2% at mid-span, and has a peak due to the hot-air injection upstream of the considered inlet plane, see Figure 2.11b.

$$k = \frac{3}{2} \cdot I^2 \cdot V^2 \quad (2.17)$$

Eventually the turbulent kinetic dissipation rate ( $\omega$ ) is estimated with

$$\omega = \frac{\sqrt{k}}{0.09 \cdot L} \quad (2.18)$$

where the characteristic length ( $L$ ) is a scale for the largest turbulent structures at the inlet, and is determined empirically so that it leads to coherent predictions of the turbulence intensity when comparing with measured ones up- and downstream of the IGV. For this thesis work the characteristic length is set empirically to a value corresponding to about 1% of the pitch of the IGV at mid-span.

### Outlet boundary conditions

The pivot pressure  $P_{out}$  is calculated at each time step by the relation

$$P_{out}(t) = P_{ref} + \lambda \left( \frac{\dot{m}(t)}{\dot{m}_{ref}} \right)^2 \quad (2.19)$$

where the reference pressure  $P_{ref}$  and mass flow rate  $\dot{m}_{ref}$  are user defined constants to the outlet boundary condition.  $P_{ref}$  is the reference total pressure at the inlet of the compressor as introduced in Section 2.1. The choked mass flow rate is taken as  $\dot{m}_{ref}$ .  $\dot{m}(t)$  is the instantaneous mass flow rate calculated through the exit section. The characteristic of the compressor is described from choked to the stall inception point by increasing the value of the throttle parameter  $\lambda$ .

### Isothermal boundary conditions

A common feature of different flow solvers is the over-prediction of the total temperature ratio distribution at the end-wall, particularly at the casing in a rotor as determined in a NATO study using several different flow solvers (Dunham, 1998). At the NASA Glenn Research Center, studies about applying isothermal boundaries at the NASA Rotor 37 casing have shown a considerable minimization of the total temperature overshoot in about 20% of the passage height towards the casing (Bruna and Turner, 2013). With the isothermal boundary conditions, the surface (hub, blade, or casing) temperatures are held constant with an imposed value regardless of the heat flow through the boundary.

Empirically estimated non-linear temperature profiles (see Figure 2.12) are applied to the hub and casing boundaries in this work, using a standard function of the flow solver elsA. The normalized axial coordinates 0 and 1 correspond to the rotor 1 leading edge, and compressor

outlet plane 300 respectively. The influence of the isothermal boundary conditions on the flow field prediction will be discussed in Section 3.3.2.

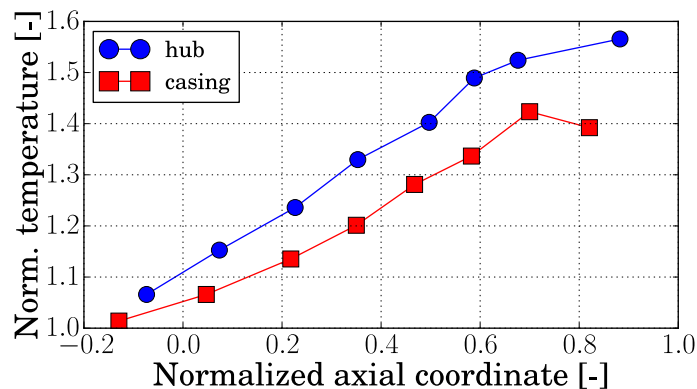


Figure 2.12 Empirical temperature profile for hub and casing provided by Safran Aircraft Engines

## Summary of the numerical parameters

The main numerical parameters are listed hereinafter in Table 2.15.

Table 2.15 Summary of main numerical parameters for the steady and unsteady RANS simulations

|                        | Steady RANS                                                                                            | Unsteady RANS              |
|------------------------|--------------------------------------------------------------------------------------------------------|----------------------------|
| Mesh points            | 13.5 million                                                                                           | 66 million                 |
| Spatial scheme         | Roe scheme (second order)                                                                              |                            |
| Temporal integration   | Backward Euler (first order)                                                                           | Gear method (second order) |
| Turbulence model       | $k - \omega$ Wilcox,<br>EARSM Wallin-Johannsson                                                        | $k - \omega$ Wilcox        |
| Wall boundaries        | Isothermal hub and casing wall boundaries<br>Adiabatic blade boundaries<br>Non-slip boundary condition |                            |
| Outlet boundary        | Throttle condition with radial equilibrium law                                                         |                            |
| Circum. boundary       | Spatial periodicity with matching nodes at border                                                      |                            |
| Rotor-stator interface | Mixing plane (Riemann average)                                                                         | Sliding mesh               |
| Rotation speed         | 11543 rpm                                                                                              |                            |
| Inlet conditions       | Radial measurement profiles                                                                            |                            |

## 2.4 Signal treatment methods

Many different signal treatment methods have been applied in the scope of this work and are introduced in this Section. The different averaging techniques and statistical analysis methods will be presented. Furthermore, the flow field analysis with the help of the frequency space will be introduced. Besides of a limited use of the statistical analysis on the RANS fields, all methods are applied on experimental and numerical data.

### 2.4.1 Arithmetic and weighted average

Several types of averages have been applied for this study, each adapted as best as possible to the physical problem working with the available numerical and experimental data.

**Arithmetic averages** are used for temporal averages and spatial averages of static pressure. The arithmetic averaging of the static pressure is only applied in case of a regular spaced grid in the averaging direction. In case of an irregular spaced grid, an area average is applied. That is the appropriate average as the static pressure is equal to the net force on an area.

**Weighted averages** are used for spatial averages of quantities such as total pressure, temperature, and velocities. Cumpsty and Horlock (2006) demonstrate that the mass (weighted) average of temperature is useful in a sense that one can consider a flux of enthalpy, where there is a linear relationship between enthalpy and temperature ( $H = c_p T$ ). There is no sense in considering a flux of total pressure as clearly stated by Cumpsty and Horlock (2006). But they also demonstrate that when the pressure variations are small (25%) compared to the absolute pressure, the ratio between a more adapted average technique (for which the measurements do not always have all data available) and the mass-average is smaller than  $\pm 0.004$ . Following this guideline, the pressure fluctuations are considered as small in CREATE and the mass average method is adapted for this work.

Depending on the measurement technique, there is no density information available. Rigorously all weighted averages are thus conducted as axial velocity weighted averages ( $\bar{q}$ ) for a quantity  $q$  according to Equation 2.20.

$$\bar{q} = \frac{1}{N \cdot \bar{V}_x} \sum_{i=0}^{N-1} q_i \cdot V_{x_i} \quad (2.20)$$

### 2.4.2 Statistics

#### Ensemble average

The ensemble average gives the ensemble of possible states - the deterministic time dependent description of the ensemble which is chosen. The averages are calculated phase-locked with a given periodicity - which can be the passing of blades or disturbance cells. Thus a signal is divided into sub-signals each describing a possible state of the chosen period. All the periods are then cut into an equal number of windows. All the data points that fall into a given window can be averaged for giving the ensemble averaged data point for this window.

That is repeated for all the windows. The data points in the windows can be used for further investigation of the statistics as described in the following paragraph.

- If not indicated otherwise, all the ensemble averages for pressure measurements are done with 200 windows on signals of generally more than  $800 \times 16$  temporal periods. That corresponds to about 2 million input data points at a sampling frequency of 500 kHz.
- All the ensemble averages for the LDA measurements are likewise computed with 200 windows, but on 300,000 to 500,000 registered LDA particles (data points). They are acquired over up to several thousands of rotations at sampling frequencies between 5 to 40 kHz (depending on the measurement position). This number of data points ensures ensemble averages, where a higher number of points would only lead to changes that are by far smaller than the measurement uncertainty.

### Standard deviation

The standard deviation can be computed on several different types of signals:

- It can be computed based on the deterministic flow signal, which has been prior obtained from an ensemble average or from the result of a URANS simulation. It does not give information about the turbulence level at a given spatial location, but a measure for the unsteadiness in the flow field.
- In the case of high-frequency measurements (LDA and wall pressure), all data points falling into one window of the ensemble average can be used as input for the calculation of the standard deviation. It allows the estimation of the dispersion of the measurements around the ensemble averaged value, caused by the turbulence in the flow, unsteadiness e.g. induced by the fluctuation of a wake, and the measurement uncertainties. That gives a temporal evolution of the standard deviation over one ensemble average period.
- Furthermore, it can be computed on a part of a time resolved signal, giving an indicator for the unsteadiness in the flow field. That can be useful to trace the unsteadiness evolution over time (several rotations).

### Skewness

As skewness ( $Sk$ ) is defined as the third order statistical momentum with Equation 2.21. The skewness quantifies the asymmetric deviation of the distribution from a normal distribution.

$$Sk = \frac{\frac{1}{N-1} \sum_{i=0}^{N-1} (q_i - \bar{q})^3}{\left( \sqrt{\frac{1}{N-1} \sum_{i=0}^{N-1} (q_i - \bar{q})^2} \right)^3} \quad (2.21)$$

The skewness describes the following phenomena in the flow field:

- $Sk = 0$ : Represents a normal (Gaussian) data distribution.

- $Sk < 0$ : The majority of the fluctuations of a quantity ( $q$ ) are above the mean value, while there are certain fluctuations with very low values of  $q$ .
- $Sk > 0$ : The majority of the fluctuations of a quantity ( $q$ ) are below the mean value, while there are certain fluctuations with very high values of  $q$ .

In shear layers, both positive and negative skewness are found, where also high values for standard deviation is present. Negative skewness can describe the exchange of high momentum fluid in low momentum fluid, where the same is valid for positive skewness the other way around. That was demonstrated among others e.g. by Beselt et al. (2014) for a corner separation in a compressor cascade.

### Kurtosis

Skewness is defined as the standardized fourth order statistical momentum, and calculated with Equation 2.22.

$$Ku = \frac{\frac{1}{N-1} \sum_{i=0}^{N-1} (q_i - \bar{q})^4}{\left( \sqrt{\frac{1}{N-1} \sum_{i=0}^{N-1} (q_i - \bar{q})^2} \right)^4} - 3 \quad (2.22)$$

Decarlo (1997) explain the meaning of kurtosis as summarized hereinafter.

- With the definition of Equation 2.22, a normal distribution has a kurtosis of 0.
- $Ku > 0$ : The distribution has heavier tails and a higher peak than the normal one, see also Figure 2.13 on the left. An elevated kurtosis reveals that infrequent deviation from the normal distribution have a major influence on the standard deviation.
- $Ku < 0$ : A negative Kurtosis has lighter tails and a flatter peak than the normal one.

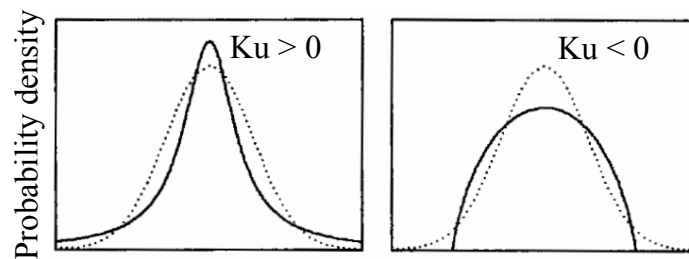


Figure 2.13 An illustration of positive (left) and negative (right) Kurtosis adapted from the work of Decarlo (1997)

### 2.4.3 Fourier transform

The Fourier transform allows the expression of a function of time  $q(t)$  regularly spaced with  $\Delta t$  over the period  $T$  of the length  $N$ , as a function of frequency  $\hat{q}(k)$  (frequency spectrum).  $\hat{q}(k)$  is obtained with Discrete Fourier Transforms (DFT) in this work by applying:

$$\hat{q}(k) = \sum_{n=0}^{N-1} q(n) \cdot e^{-2i\pi \frac{n \cdot k}{N}} \quad (2.23)$$

where  $k = 0, \dots, N-1$  and  $T = N \cdot \Delta t$ . The maximum frequency that can be calculated with the Fourier transform is equal to half the sampling frequency of the signal according to the Shannon theorem. The acquisition frequency has to be adapted accordingly. The Fourier transform is also used for obtaining the following information:

The amplitude spectrum shall be defined by:

$$|\hat{q}(k)| = \sqrt{\text{Re}(\hat{q}(k))^2 + \text{Im}(\hat{q}(k))^2} \quad (2.24)$$

The phase spectrum is obtained from:

$$\Phi(\hat{q}(k)) = \arctan\left(\frac{\text{Im}(\hat{q}(k))}{\text{Re}(\hat{q}(k))}\right) \quad (2.25)$$

The Parseval's theorem states that the total energy of a signal ( $q(t)$ ) can be equally estimated by summing the power per sample across time or the spectral power across frequency with Equation 2.26. The term "energy" in the following parts of this work will refer to this relation. It allows calculating the energy of a wake in a pressure field for example, when summing up the energy of the blade passing frequency and its harmonics.

$$E(\hat{q}(k)) = \sum_{k=0}^{N-1} |\hat{q}(k)|^2 \quad (2.26)$$

### 2.4.4 Double modal decomposition method

The influences of the rotor-stator interactions will be studied using a double modal decomposition method. The method gives the ability to decompose the flow field information into the influence of the rotors, stators and RSI. It is based on the assumption of Tyler and Sofrin (1962). It states that the flow field is a superposition of an infinite number of circumferential lobed structures, expressed in Equation 2.27. This approach has been extended by Courtiade et al. (2012) for multistage cases. A lobed structure can be the combination of any rotor and stator influence in a multistage compressor. A direct contribution of a row shall have not only the same number of lobes as its corresponding blade number, but also the same rotation speed as the row (rotor speed  $\Omega$  for a rotor and zero for a stator). The decomposition is achieved by spatial Fourier transforms of a signal for each time step (as illustrated in Figure 2.14) and then a temporal Fourier transform of each spatial mode signal. The maximum number of

detectable spatial and temporal modes is limited by the spatial and temporal resolution. The spectrum of the temporal Fourier transform allows identifying the set of temporal modes ( $m_t$ ) which constitute a spatial mode ( $m_\Theta$ ).

A rotation speed for every spatial and temporal mode combination can be obtained by applying Equation 2.28 and Equation 2.29. The application of Equation 2.28 requires deep knowledge of the flow field because several combinations could be possible.

$$f(x, R, \Theta, t) = \sum_{m_t=-\infty}^{\infty} \sum_{m_\Theta=0}^{\infty} A_{m_\Theta, m_t}(x, R) \cos[m_\Theta \cdot \Theta - m_t \cdot \Omega \cdot t + \Phi_{m_\Theta, m_t}(x, R)] \quad (2.27)$$

$$m_\Theta = m_t \pm k \cdot N_s \quad k \in \mathbb{Z} \quad (2.28)$$

$$\omega_{m_\Theta, m_t} = \frac{m_t}{m_\Theta} \Omega \quad (2.29)$$

$N_s$  corresponds to the possible numbers of stator blades (7 for a spatial period of CREATE, see Table 2.2). The temporal Fourier transforms allow the construction of a modal matrix  $A_{modal}$  and the phase coefficients ( $\Phi_{modal}$ ) based on Equation 2.27, see Equation 2.30.

$$A_{modal} = \begin{bmatrix} 0 & A_{i, M_t} & \dots & A_{M_\Theta, M_t} \\ \vdots & \vdots & A_{i, j} & \vdots \\ 0 & A_{1, 1} & \ddots & \vdots \\ A_{0, 0} & A_{1, 0} & \dots & A_{M_\Theta, 0} \\ 0 & A_{1, -1} & \ddots & \vdots \\ \vdots & \vdots & & \vdots \\ 0 & A_{1, -M_t} & \dots & A_{M_\Theta, M_t} \end{bmatrix} \quad \Phi_{modal} = \begin{bmatrix} 0 & \Phi_{i, M_t} & \dots & \Phi_{M_\Theta, M_t} \\ \vdots & \vdots & \Phi_{i, j} & \vdots \\ 0 & \Phi_{1, 1} & \ddots & \vdots \\ \Phi_{0, 0} & \Phi_{1, 0} & \dots & \Phi_{M_\Theta, 0} \\ 0 & \Phi_{1, -1} & \ddots & \vdots \\ \vdots & \vdots & & \vdots \\ 0 & \Phi_{1, -M_t} & \dots & \Phi_{M_\Theta, M_t} \end{bmatrix} \quad (2.30)$$

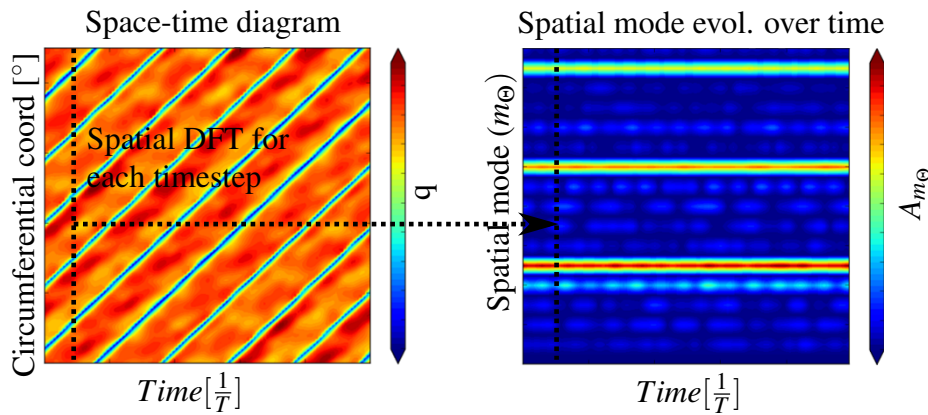


Figure 2.14 From space-time diagram (left) to spatial modes evolution over time (right)



The temporal modes  $m_t$  are direct contributions of the rotors (i.e. linear combinations of the rotor blade numbers). Courtiade et al. (2012) defined that the negative part of the temporal spectrum represents the temporal modes that propagate in the turning direction of the rotor. It is possible to filter a signal for the RSI with this method, by eliminating certain combinations of spatial and temporal modes from the modal matrix in Equation 2.30. The energy of selected (or all) RSI modes shall be defined as sum of the square amplitudes from the filtered (or unfiltered) modal matrix with Equation 2.31.

$$E_{RSI} = \sum_{i \in m_{\Theta}} \sum_{j \in m_t} |A_{modal, i, j}|^2 \quad (2.31)$$

An exhaustive decomposition of the flow field in a previous version of CREATE, and identification of the major rotor-stator interactions is available in the work of Courtiade (2012).

### 2.4.5 Wavelet transform

The wavelet transform allows analyzing amplitude variations over time at different frequencies. An admissible wavelet function ( $\Psi_{\eta}$ ) is required, where admissible means having a zero mean and being localized in the time and frequency space (Farge, 1992). The Morlet wavelet has proven to be useful for turbomachinery applications for example in the works of Inoue et al. (2002) and Lin et al. (2004), and will be used for all wavelet transforms in this work. The Morlet wavelet consists of a complex sinus wave modulated by a Gaussian according to Equation 2.32. It allows the identification of very narrow events in time such as spike type stall inception. Only the fluctuations in a signal are taken as input to the wavelet transform in this work, and they are normalized by the standard deviation of the signal. This treatment allows a judgment about the importance of structures in the wavelet power spectra.

$$\Psi_0(\eta_t) = \pi^{-\frac{1}{4}} e^{i\omega_0 \eta_t} e^{-\frac{\eta_t^2}{2}} \quad \omega_0 \in \mathbb{Z}, \quad \omega_0 > 6 \quad (2.32)$$

where  $\omega_0$  is the non-dimensional Morlet frequency, advised to be larger than 6 for the admissible condition. And  $\eta_t$  being a non-dimensional time parameter. It is known the smaller  $\omega_0$  the higher the time resolution of the wavelet spectrum, the larger  $\omega_0$  the better the frequency resolution.  $\omega_0$  has thus to be adapted for each studied phenomenon in the flow field. In this work values between 6 and 32 are used for  $\omega_0$ .

The continuous wavelet transform (CWT) of a discrete time series  $q_n$ , with a spacing of  $\delta t$ , and  $N$  points (with  $n = 0 \dots N - 1$ ) is given by:

$$CWT_n(s_j) = \sum_{n'=0}^{N-1} q_{n'} \cdot \Psi^* \left[ \frac{(n' - n)\delta t}{s_j} \right] \quad (2.33)$$

where  $\Psi^*$  is a scaled and translated version of  $\Psi_0(\eta_t)$ , see the work of Torrence and Compo (1998) for more information about the normalization of  $\Psi_0(\eta)$ . For each scale the convolution is conducted  $N$  times with Equation 2.33. The convolution can be done in Fourier space using a discrete Fourier transform.

In this work solely the wavelet functions made available by Torrence and Compo (1998) are used. These functions allow as well calculating the cone of influence for the beginning and end of the wavelet power spectrum. Here the Fourier transform expects periodic data, and the deviation from periodicity causes edge effects. These zones are blacked out in the wavelet spectra.

**Choice of wavelet transform scales** A set of scales ( $s_j$ ) is necessary for the wavelet transform. Estimations for the smallest and largest scales are used following the suggestions of Torrence and Compo (1998):

$$s_j = s_0 2^{j\delta j} \quad j = 0, 1, \dots, J \quad (2.34)$$

$$J = \delta j^{-1} \log_2 \left( \frac{N\delta t}{s_0} \right) \quad (2.35)$$

where  $\delta j$  defines the frequency resolution, a smaller value corresponds to a finer resolution.  $J$  defines the largest scale. The resolution has to be limited depending on the available computation resources and length  $N$  of the input time series. The Fourier wavelength ( $\lambda_{wave}$ ) for the Morlet wavelet is defined by:

$$\lambda_{wave}(s_j) = s_j \cdot \frac{4\pi}{\omega_0 + \sqrt{2 + \omega_0^2}} \quad (2.36)$$

Using  $\lambda_{wave}$ , the investigated frequency range can be adapted by setting the smallest scale - highest frequency ( $f_{max}$ ) - with Equation 2.37:

$$s_0 = \frac{1}{f_{max}} \cdot \frac{1}{\frac{4\pi}{\omega_0 + \sqrt{2 + \omega_0^2}}} \quad (2.37)$$

### 2.4.6 Data normalization

If not stated otherwise, all the results presented hereinafter are normalized by the following relations, where  $q_{std}$  represents a measured and standardized quantity after post-processing.

The total pressure and static pressure are normalized by the inlet total pressure and total pressure ratio at nominal operating point according to Equation 2.38, so as to obtain a total pressure around unity at the outlet of the compressor for the nominal operating point.

$$P = P_{std} \cdot \frac{1}{Pt_{std,inlet} \cdot \pi_{Nominal}} \quad (2.38)$$

The total temperature is normalized by the total temperature rise ( $Tt_{rise,nominal}$ ) in the compressor with Equation 2.39.

$$Tt = Tt_{std} \cdot \frac{1}{Tt_{rise,nominal}} \quad (2.39)$$

All the velocity measurements are normalized by using the same  $U_{ref}$  according to Equation 2.40.

$$V = \frac{V_{std}}{U_{ref}} \quad (2.40)$$

## 2.5 Summary of conducted measurements and simulations

Measurements have been conducted with the help of pneumatic probes and laser Doppler anemometry inside of the vane at all inter-row planes. High-frequency casing pressure probes allow the description of the flow field above all rotors and at the inter-row planes. Furthermore, the laser Doppler anemometry measurements allow the optical access to the rotor 2 and rotor 3 tip flow fields.

Overall the uncertainties of each measurement show the high quality of the measurements but also challenges in the high-speed multistage compressor. The measurement uncertainties have to be kept in mind during the interpretation of the results, and are summarized hereinafter:

- Steady measurements: Relative uncertainty of about  $\pm 0.2\%$  on the pressure measurements and  $\pm 0.5^\circ$  on the flow angle and total temperature measurements.
- Laser velocimetry measurements: Globally the uncertainty of the mean axial and circumferential velocity components lies between  $\pm 0.93 \frac{m}{s}$  and  $\pm 2.38 \frac{m}{s}$ . High fluctuations of the uncertainties are observed due to wakes, separations, and flow structures close to the endwall. This has to be taken into account when studying the flow field at these locations.
- High-frequency casing measurements: The uncertainty of the fluctuations is low with  $0.6\%$  compared to a fairly high uncertainty on the mean value with  $3\%$ . The high uncertainty comes from the non-compensation of the ambient (flow field) temperature for the probes.

Numerical result fields are available from steady and unsteady RANS simulations for the compressor domain from the IGV to the stator 3. Unsteady RANS fields are available for all five clocking positions at the peak efficiency operating point, and two clocking positions at the loaded operating point, and one clocking position at the near surge operating point. Several additional studies have been conducted with steady RANS simulations (turbulence model, technologies, boundary conditions) and unsteady RANS simulations for the investigation of instabilities arising from the loaded operating point on.

# Chapter 3

## Discussion of the compressor flow field and its exploration methods

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The ultimate goal of conducting the measurements and simulations of the CREATE flow field is to understand deeply its flow physics. The correct interpretation of the flow field data requires also the knowledge of the limitations to its exploration methods. That knowledge needs to go beyond typical estimations of measurement uncertainties. Here the high-speed environment is particular challenging. In this Chapter, the results are all based on the clocking position 3 because the stator indexing (clocking) will be discussed in more detail in the next Chapter. All results presented here are nevertheless assured to be valid for the other stator clocking positions.

This Chapter will go from a global perspective on the flow field to a more detailed analysis. Firstly, the performance of the compressor will be presented together with the corrections that need to be taken into account for overcoming interpretation errors of the data. Only those corrections allow the good interpretation of the performance offset between numerical and experimental data. Secondly, a global description of the flow field will be given, and major differences between the experimental and numerical flow field will be identified. That is followed by detailed discussions of the rotor and stator flow fields. Each with a focus on where the respective flow field exploration method reaches its limit and gives false views of the flow field. Interestingly the methods are not necessarily challenged at identical locations in the compressor.

### 3.1 Performance of the compressor

The numerical predictions of the compressor performance can be validated with measurements. This comparison is more complicated than thought at first sight because of simplifications to the simulation domain and corrections that need to be applied to the measurement data for their correct interpretation. Those corrections have never been applied before for this comparison. Numerical performance data is obtained from full integrals of flow field data at the inlet and outlet of the simulation domain. The total pressure and temperature are measured in a single point at the inlet of the compressor and with 30 probes at the outlet of the compressor (close to the outlet of the simulation domain). Why an extraction of the numerical data at the limited measurement positions is not useful will be discussed in Section 3.1.2.

In a first step, the uncorrected performance data with a common normalization is presented in Figure 3.1. The performance data is normalized by the measured peak efficiency and

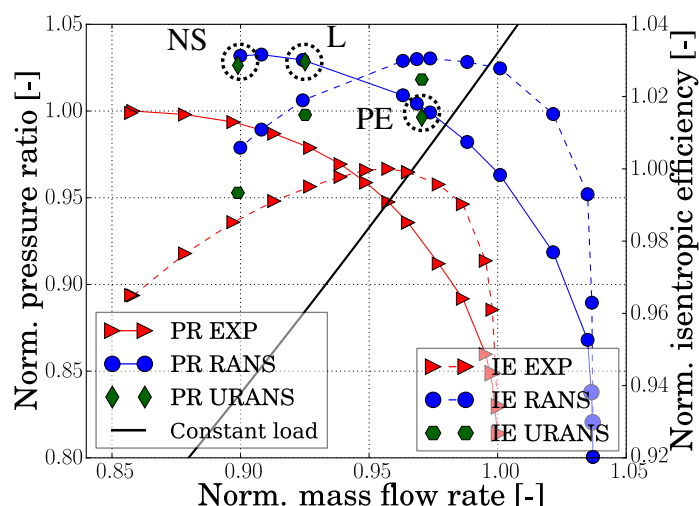


Figure 3.1 Common normalization of RANS (blue) and experimental (EXP, red) performance curves for the total pressure ratio ("PR", left vertical axis) and the isentropic efficiency ("IE", right vertical axis)

pressure ratio, and mass flow rate at choked conditions. The measured (EXP, red) and predicted (RANS, blue) total pressure ratios are presented on the left vertical axis, and the isentropic efficiencies on the right vertical axis. An almost constant offset of +3.8% mass flow rate, +2.6 efficiency points, and +0.1 total pressure ratio points is noted for the RANS results relative to the measurements. At first sight with this scale, the numerical results appear completely off but there is more to it.

Also in Figure 3.1, URANS performance predictions are shown with diamonds (total pressure ratio) and hexagons (isentropic efficiency) at nominal (PE), loaded (L) and near surge (NS) operating points. A small offset between URANS and RANS is noticeable. The analysis of local differences between those flow fields will be discussed with this thesis work for example in Section 3.4.1. Even if not very different in a sense of global performance, the local flow field can be different. Though, RANS counterbalances mis-predictions with redistribution of the flow along the span, which is found to neutralize the mis-prediction in a global sense in CREATE. It shall be noted that the peak efficiency operating points in the experimental and numerical performance data do not necessarily lie on line of constant load (following a quadratic mass flow rate law as stated in Equation 2.19), which is indicated with the black solid line in Figure 3.1. This could not be known at the time point of the realization of the peak efficiency URANS simulations because this was prior to the measurements. Though the offset is acceptable small and the flow field characteristics do not change significantly.

For overcoming the offset between the numerical and experimental performance curves, the mass flow rate, isentropic efficiency, and pressure ratio are normalized by their respective maximum values in Figure 3.2. With the individual normalization, a good match is found globally between the evolutions in the two data sets. The pressure ratio evolutions show an almost perfect match but RANS under-estimates the surge limit by about 1% of normalized

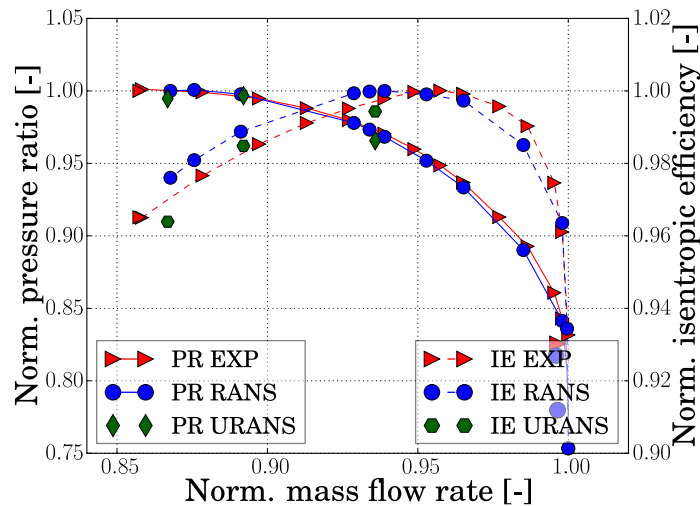


Figure 3.2 Individually normalized RANS (blue) and experimental (EXP, red) performance curves for the total pressure ratio ("PR", left vertical axis) and the isentropic efficiency ("IE", right vertical axis)

mass flow rate, as reported already for a previous version of CREATE by Courtiade (2012) and Ottavy et al. (2012). The simulation over-estimates slightly the isentropic efficiency towards surge, and under-estimates the efficiency between peak efficiency and choked mass flow rate. The small offset between the peak efficiency points is found again.

The influence of the simplifications that are used for the simulation domain on the performance data will be discussed in Section 3.1.1. This is followed by a definition of the correction factors that are to be applied to the steady performance measurement data in Section 3.1.2.

### 3.1.1 Discussion of simplifications to the numerical domain

It is known from previous studies on CREATE, that simplifications in the simulations, such as non-taking into account the gaps between the rotor-stator hub platforms would lead to an additional offset between the predicted efficiency and measured performance (Marty and Aupoix, 2012; Ottavy et al., 2012). An experimental characterization is not available for the hub leakage mass flow rate underneath the shrouded stators. The numerical work of Marty and Aupoix (2012) takes into account the gap below the first stator. The authors show that the leakage flow can modify the global performance but also leads to possible over-predictions of hub-corner separations in the stators. They note an isentropic efficiency decrease of about 1 point, and a mass flow rate decrease of about 5%. These exaggerated effects might all be the result of an over-predicted hub leakage mass flow rate as stated by Marty and Aupoix (2012). Because of technical limitations for the setup of the simulation domain, and for not adding another uncertainty to the simulation domain, the hub leakage is not taken into account for any stator in this thesis work. The adjusting of the correct simulation of the hub leakage flow (especially without further measurement data) is out of scope for this thesis work.

The influence of the eight struts are neither taken into account with the measurements at the inlet because of the measurement positions, nor in the simulations because of the spatial periodicity of  $\frac{2\pi}{16}$ . A quantification of the influence of the wakes of the eight struts on the unsteadiness in the flow field, and local changes of losses would require extensive additional simulations. Their impact is judged of minor impact on the main flow field at the peak efficiency point. It does not appear as significant disturbance ( $\frac{2\pi}{8}$ ) to the main spatial period ( $\frac{2\pi}{16}$ ) of the compressor in the measurements. This is always verified with an additional measurement in circumferential direction closing the spatial period.

The choice of the turbulence model is always a difficult one because of all the widely known troubles with the RANS turbulence models in predicting "simple" flows in cascades and especially "complex" flows in multistage compressors. It is out of scope for this thesis work to test for several turbulence models and their influence on the compressor performance. From unpublished (confidential) thesis work (partly presented in (Sharma et al., 2007) and (Marty et al., 2008)) it is known that no matter the turbulence model, the differences to the measurement data are not substantially reduced in terms of flow aerodynamics. Furthermore, it is known from these works that the isentropic efficiency can be influenced by about 1% between the results of the  $k - \omega$  Wilcox and the k-1 (Smith, 1994) turbulence model. Here the latter shows globally a lower efficiency for CREATE.

### 3.1.2 Corrections applied to measured performance

So far, differences between the global experimental and numerical performance data of CREATE have been attributed to mis-predictions by the RANS method (e.g., in the works of Ottavy et al. (2012) and Gourdain et al. (2012)). In the complex multistage environment it is possible to make errors in the interpretation of the steady performance measurement data though. Those errors are not the same as classical measurement uncertainties, and can be corrected to a certain extent. This Section shall make aware of the challenges for the steady performance measurements in the high-speed multistage environment, and can be used as guide for correcting the CREATE performance data and as general reminder for possible interpretation errors in multistage compressors. The concerned measurements are listed hereinafter:

- The total temperature is measured in the settling chamber upstream of the compressor. This temperature corresponds to the one found at mid-span upstream of the IGV at the plane 250. A needed correction to this measurement is proposed in this Section.
- The total pressure is measured between two wakes of the struts at mid-span at plane 250. The total pressure loss induced by the struts is evaluated in this Section.
- The influence of the limited measurement points at the compressor outlet will also be discussed.

#### Losses induced by struts

Eight support struts are located upstream of the compressor by about one compressor length. Their influence is not taken into account in the numerical setup, and neither with the Kiel probe measurement at the inlet at plane 250 between two strut wakes. Obviously the related losses are taken into account by the flow condition measurements at the outlet of the compressor. The blade profile of the struts corresponds closely to the one of NACA 0024 blades. Their respective drag is stated in the extensive work about NACA blades of Abbott et al. (1945) with  $C_d$  for the given flow conditions. A blade row loss coefficient  $\alpha_{drag}$  can be defined according to Equation 3.1 which is taken from the work of Dixon and Hall (2010).

$$\alpha_{drag} = C_d \cdot \frac{c}{s} = 0.015 \cdot \frac{c}{s} = 0.01422 \quad (3.1)$$

where  $c$  is the chord length and  $s$  the blade pitch. The endwall losses caused by the presence of the struts is defined by Equation 3.2 according to the work of Howell (1945).

$$\alpha_{endwall} = 0.02 \cdot \frac{s}{H} = 0.0245 \quad (3.2)$$

where  $H$  is the blade height. The overall coefficient for taking into account the pressure losses induced by the struts can be defined with Equation 3.3.

$$\alpha_{corr,struts} = \alpha_{drag} + \alpha_{endwall} = 0.03872 \quad (3.3)$$

This loss coefficient translates to a pressure loss according to Equation 3.4, where  $P_{s_{sc}}$  is measured in the settling chamber and  $P_{t_{in}}$  with the previously described Kiel probe.  $P_{loss,struts}$



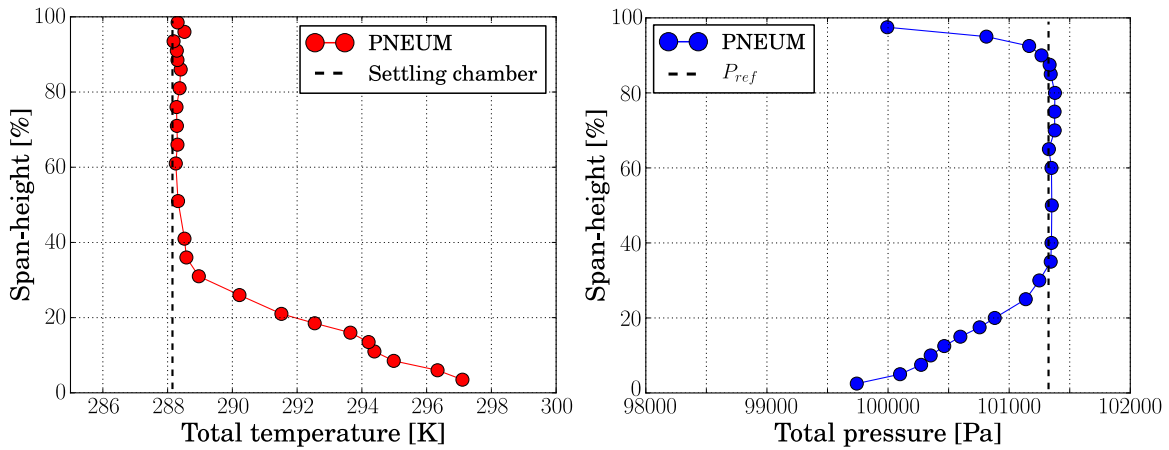
needs to be combined with the total pressure losses induced by the hot air injection upstream of the IGV using Equation 3.7.

$$P_{loss, struts} = \alpha_{corr, struts} \cdot (P_{t_{in}} - P_{s_{sc}}) \quad (3.4)$$

### Hot air injection upstream of the IGV

The inlet total temperature is estimated as average of measurements obtained with ten probes in the settling ( $T_{t_{in}}$ ) chamber far upstream of the compressor. It corresponds about to the total temperature at mid-span at plane 250 upstream of the IGV.

Upstream but close to the IGV a hot air injection occurs at the hub because of air recirculation in the test rig, which are necessary for counterbalancing the axial force of the compressor. The injection mass flow rate is as small as 0.05% of the nominal compressor mass flow rate. The injection causes a peak in the total temperature of up to 9K compared to the mid-span flow as shown in Figure 3.3a based on PNEUM measurements.



(a) Total temperature and standardized settling chamber temperature (b) Total pressure and reference total pressure

Figure 3.3 Inlet conditions at plane 250 (inlet plane to the URANS simulations) based on PNEUM measurements

Those measurements are conducted at one fixed circumferential position. More positions are not available due to measurement constraints. The wakes of the struts are not expected to have an influence on the total temperature at 250 as the struts are not delivering work to the fluid. The settling chamber temperature is indicated with the vertical black dashed line. Taking the full plane information or just the settling chamber temperature obviously changes consequently the isentropic efficiency.

Taking the  $T_{ref}$  as inlet total temperature for the numerical and experimental performance calculation is a possibility because the actual total temperature profile at plane 250 is imposed as boundary conditions in the (U)RANS simulations. Though this procedure would not give the actual compressor performance, and also does not allow to take into account small

differences between the imposed (previous version of CREATE, see Section 2.3.4) and the actual measured total temperature profile at plane 250.

The total temperature profile at plane 250 is not measured during every operating of the compressor contrary to the settling chamber temperature. Thus a correction for the settling chamber temperature is proposed in order to take into account the hot air injection for the experimental estimation of the isentropic efficiency. The numerical isentropic efficiency will be therefore estimated with a total temperature average over the full plane at 250.

The weighted average of the total temperature at 250 over the whole span corresponds to 289.69K based on the PNEUM data. The correction factor will be thus defined with Equation 3.5.

$$T_{t_{corr, inj}} = T_{t_{in}} \cdot c_{corr, Tt} = T_{t_{in}} \cdot \frac{289.69K}{T_{ref}} = T \cdot 1.0054 \quad (3.5)$$

The correction factor of  $c_{corr, Tt} = 1.0054$  decreases the measured temperature ratio between inlet and outlet of the compressor. Consequently the isentropic efficiency rises by 1.5 efficiency points at the peak efficiency operating point. Small differences between the total temperature profiles of changing compressor versions exist and should be carefully taken into account. These changes can have a large impact on the isentropic efficiency. Thus each compressor version will need a specific correction factor. A previous version of the compressor (studied by Courtiade (2012)) needs a  $c_{corr, Tt}$  of 1.0029 for example.

The same type of correction is necessary for the total pressure measurement ( $P_{t_{in}}$ ), which is conducted with a Kiel probe at mid-span at plane 250 and almost identical to the reference total pressure ( $P_{ref}$ ) as indicated in Figure 3.3b. The hot-air injection causes mixing losses thus pressure losses in the hub region. The correction is noted in Equation 3.6. With the correction, the isentropic efficiency will increase by 0.13 efficiency points at the peak efficiency operating point. A  $c_{corr, Pt}$  of 0.9986 is to be applied for the previous version of the compressor. Again, the numerically estimated efficiency is calculated with a full flow field average of the total pressure at the plane 250.

$$P_{t_{corr, inj}} = P_{t_{in}} \cdot c_{corr, Pt} = P_{t_{in}} \cdot 0.9978 \quad (3.6)$$

### Performance data statistics

Historically the steady performance data is averaged over one second and stored at every second of compressor rotation. For a given stabilized operating point, only one data point was taken as information until now. Reducing the statistical uncertainty is possible by taking more than one data point. With the given operation of the compressor, 30 seconds (points) are available for estimating the compressor performance thanks to averages over all these data points. This helps to eliminate mass flow rate variations of  $\pm 0.2 \frac{kg}{s}$  and efficiency variations of  $\pm 0.1\%$ , and is advisable for any steady performance measurement.

This treatment is applied for all performance data in this work if not indicated otherwise.

### Constant or variable specific heat capacity ratio

The RANS simulations are conducted with a constant specific heat capacity ratio ( $\gamma \approx 1.4002$ ). Historically the experimental constant specific heat capacity ratio is estimated based on an interpolation between the non-standardized inlet and outlet total temperatures at a given measurement day (called "variable  $\gamma$ " hereinafter). This takes into account the reduction of the specific heat capacity ratio with increasing total temperature (for example during measurements in the summer). The influence of a constant versus a variable specific heat capacity ratio is illustrated in Figure 3.4. The same total pressure and total temperature measurement values are taken as input for the two ways of the efficiency estimation. A constant  $\gamma$  is applied for the round (blue) data points, and a variable one for the diamond (green) data points. All these data points are obtained from measurements without any instrumentation (e.g., PNEUM probes) in the flow field but not necessarily stabilized operating points and are not averaged over 30 seconds (see Section 3.1.2). This might cause fluctuations but does not change the general trend for this comparison. The inlet total temperature on the horizontal axis varies between a cold day (282 K) and a hot day (307 K).

The variable  $\gamma$  allows to obtain an efficiency estimation with variations in the range of 0.5 efficiency points. That is close to the measurement uncertainty of  $\pm 0.2$  efficiency points plus the eliminated statistical uncertainty of  $\pm 0.1$  efficiency points (Section 3.1.2). A constant  $\gamma$  raises the efficiency between 0.5 to 0.8 points. The larger range with the constant  $\gamma$  compared to the variable  $\gamma$  comes from the eventual dependency of the efficiency on the measurement day (inlet total temperature). This behavior is not wanted but cannot be avoided when having measurement campaigns with a duration of over a year. The dependency is nevertheless smaller than the effect between constant and variable  $\gamma$  (0.3 vs. 0.8 efficiency points). Furthermore, the compressor performance characterization is done during a few consecutive days during which the inlet temperature will not vary by much. A constant  $\gamma$  is therefore used for the comparison between numerical and experimental results, more specifically the  $\gamma$  used for the numerical simulations (see Section 2.3.1).

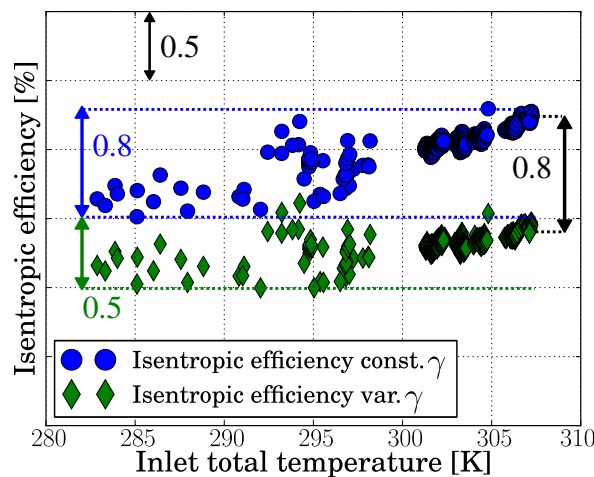


Figure 3.4 Influence of variable and constant heat capacity ratio ( $\gamma$ ) on the isentropic efficiency estimation depending on measurement day (inlet total temperature)

### Influence of limited measurements points

The exit flow conditions are measured with total pressure and temperature probe rakes, which are located at six circumferential positions and each holding probes at 5 different span-heights evenly spaced between 10% and 90% span-height. That gives in total 30 measurement points of total pressure and temperature at the outlet of the compressor at plane 300. The numerical domain ends one stator chord upstream of this plane, where the numerical outlet also serves for the extraction of the exit flow conditions.

The two different axial positions for the outlet flow conditions influence the mixing out of flow structures but are not expected to change significantly the radial distribution of the flow conditions. Thus the numerical results allow the quantification of the influence of the limited number of measurement points compared to the full plane average on the isentropic efficiency. The extraction is done with a probe grid resolving the actual probe size. It also allows rotating numerically the 30 probes for quantifying possible fluctuations of the efficiency depending on the rake positions, which are not rotatable on the actual test rig. These two points are quantified in Figure 3.5a. Instead of six rakes, only three rakes are taken into account because of the imposed spatial periodicity in the numerical work. All three virtual rakes are rotated simultaneously.

In average, the "3-rake efficiency" lies 0.17 points below the one calculated with the full plane average. This is surprising because 10% of the span towards the hub and casing are not taken into account with 3-rake extraction. In the 10% of span towards the endwalls, the total temperature increases and total pressure decreases compared to the mid-span flow at the outlet of CREATE. This is therefore expected to result in an over-estimation of the efficiency. But the lower efficiency is actually the result of the limited measurement positions over the

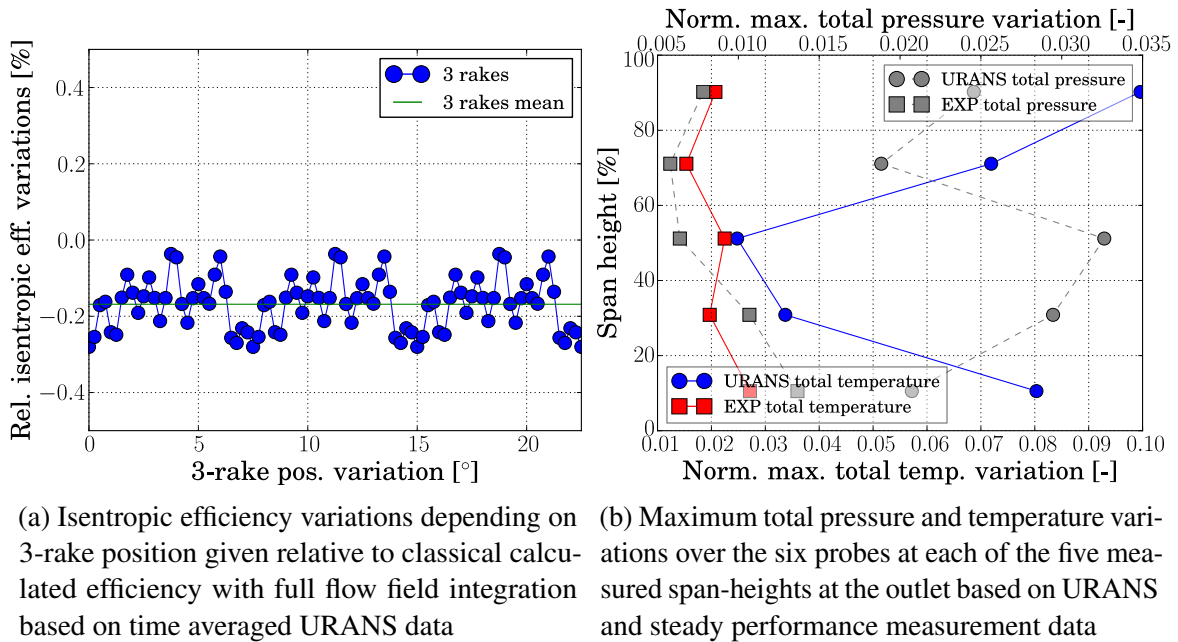


Figure 3.5 Study of limited flow condition measurement points at the outlet of the compressor

circumference. Additionally, fluctuations of  $\pm 0.13$  efficiency points are registered depending on the circumferential position of the "3 numerical rakes". These fluctuations might be of smaller amplitude at the actual measurement plane 300 because of a more mixed out flow than in the numerical results.

The fluctuations of the total temperature and pressure over the six probes of each of the five span-heights are presented in Figure 3.5b. The total temperature variations are given in color, and the total pressure variations in grey, where the URANS data is given with circles and the measurement data with squares. The numerical data is extracted for one arbitrary position of the 3 rakes. Respective to the variations in the URANS data, the measurement data shows little variations over the six probes at each span-height. At 10%, 70%, and 90%, four times higher temperature variations are found in the numerical results than in the measurements. Likewise the total pressure variations are overall three to four times higher. That suggests a probable over-estimation of the effect of the limited number of measurement points on the overall efficiency estimation. Thus a correction factor cannot be conclusively defined here for the experimental data. A general underestimation of up to 0.17 efficiency points might occur due to the limited number of circumferential measurement points.

Therefore a full plane average is advisable for the estimation of the outlet flow conditions based on the URANS results, with the given extension of the simulation domain. That minimizes the effect of the different states of mixing out of the flow structures compared to the measurement data. In this thesis work, all presented numerical performance data is based on full plane averages (of the time averaged field in case of URANS simulations).

### **Influence of operating with depression at inlet**

The test rig is operated with an inlet total pressure below the ambient total pressure (see section 2.1). Then all measured values are standardized. The difference between measured and standardized values corresponds also to a change in the Reynolds number. The fully turbulent RANS setup cannot capture the effect of the changing Reynolds number.

This effect could be quantified with experimental work but that is out of scope for this thesis work. Heidelberg and Ball (1972) [Figure 11] evaluated the effect of a Reynolds number change on the overall performance of a six stages axial compressor. 0.74 times the ambient pressure (CREATE) would lead, according to their work, to a performance decrease of up to 0.7 efficiency points. This could therefore explain a part of the offset between the experimental and numerical performance found for CREATE. However, Heidelberg and Ball (1972) worked with inlet pressures corresponding to Reynolds numbers in the range of  $10^4$  to  $10^5$ , whereas  $10^6$  is found in CREATE and therefore the performance decrease might be over-estimated. A correction factor adapted to CREATE would need to be experimentally defined in a future work.

### Summary of correction factors for measured compressor performance

The numerical isentropic efficiency is estimated with full plane averages at the plane 250 and the outlet of the simulation domain (close to plane 300). The steady performance measurements need to be corrected as listed hereinafter (attention, the coefficients might depend on the compressor version) but give a globally valid idea of the magnitude of the correction. As a reminder, all flow condition values are standardized as introduced with Equations 2.1 to 2.5:

- The inlet total temperature is to be corrected according to Equation 3.5 with the coefficient  $c_{corr,T} = 1.0054$
- The inlet total pressure measurement is to be corrected according to Equation 3.6 and with the coefficient  $\alpha_{corr,struts}$  defined in Equation 3.3 derived from the drag force equation. That leads to the equation:

$$Pt_{corr} = Pt_{in} \cdot c_{corr,inj} - \alpha_{corr,struts} \cdot (Pt_{in} \cdot c_{corr,inj} - Ps_{sc}) \quad (3.7)$$

where the static pressure ( $Ps_{sc}$ ) is measured in the settling chamber, and  $\alpha_{corr,struts} = 0.03872$ , and  $c_{corr,inj} = 0.9978$ . That yields in average (over the whole operating range) a  $Pt_{corr}$  which is 0.3% lower than  $Pt_{in}$ .

- The correction of the measured inlet total pressure and temperature results in a modification of the standardized mass flow rate because of the nature of Equation 2.1, which depends inversely proportional on the inlet total pressure and proportional on the square root of the inlet total temperature. The pressure correction increases in average the standardized mass flow rate by 0.3%, and the temperature correction increases the standardized mass flow rate by 0.27%.
- The total temperature correction suggests that the standardized shaft speed is overestimated during the measurements (see Equation 2.2). That offset is marginal with 32 rpm, and the effect on the compressor performance is negligible. An extrapolation of the total pressure rise, using data obtained at three different shaft speeds, yields a 0.36% pressure ratio point and +0.33% standardized mass flow rate increase with +32 rpm shaft speed.
- An identical specific heat capacity ratio (from the numerical setup) shall be taken for the estimation of the efficiency based on either database (RANS/EXP).
- The Reynolds number effect should be taken into account but a correction factor needs to be determined in a future experimental work.

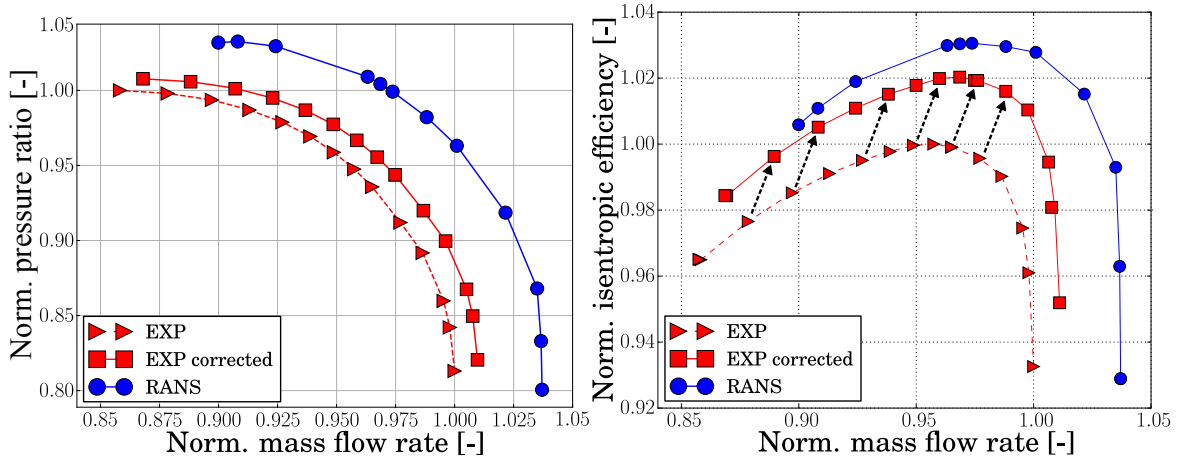


Figure 3.6 Applied corrections to the experimental steady performance measurements of the total pressure ratio (left) and isentropic efficiency (right)

All corrections together lead for the steady performance data to an increase of 1.8 efficiency points, 0.72% pressure ratio points, and 1.35% of the nominal standardized mass flow rate. The remaining mass flow rate correction (additionally to the 0.3%, 0.27%, and 0.33%) is conducted on the assumption that any displacement of the performance curve should be on a line of constant load. Consequently, the mass flow rate increases due to taking an identical specific heat capacity ratio for RANS and EXP. The correction of the data is presented for the total pressure ratio (left) and the isentropic efficiency (right) in Figure 3.6. The uncorrected and corrected steady performance data are presented with red triangles and squares respectively. The RANS data is presented with circles (blue).

The remaining offset between the numerical and experimental performance (0.9 efficiency points, 3.24% pressure ratio, and 2.47% mass flow rate) might be explained by the previously discussed simplifications to the simulation domain and consequently or independently by mis-predictions of the flow field. A faulty measurement of several flow condition measurements (temperature, pressure, and mass flow rate) is considered unlikely. That would be necessary to obtain the rather constant shift between the RANS and EXP performance data. The following parts of this thesis work will highlight the most important mis-predictions by the (U)RANS method.

## 3.2 Global description of the flow field

This Section characterizes the flow field in CREATE and points out global challenges to the flow field exploration methods. The flow field characterization is conducted with radial distributions of total pressure and temperature, flow angle, and entropy at the peak efficiency operating point. These aerodynamic quantities give information about:

- The total pressure is either decreased due to losses in a row, or increased by the exchange of work in a row.

- Total temperature is an indicator for the work exchanged between the fluid and the blades rows. Zones of high total temperature can indicate zones of high exchange of work. That has to be carefully interpreted with the help of the flow angle because high total temperature can also indicate zones, where losses are produced by viscous mixing.
- The entropy distributions indicate losses, where zones of high entropy reveal zones of high losses. Globally the entropy increases in flow direction through CREATE.
- The flow angle allows also a first identification of flow separations, which cause generally a decrease of the absolute velocity and hence an increase of the flow angle.

The aerodynamic quantities are presented each in one respective Figure (3.7, 3.8, 3.9, and 3.10) for all the inter-row planes from 25A to 290.

The URANS (blue) results for the peak efficiency point are averaged in space and time for the radial views. The PNEUM data has been chosen for the EXP (red) data. The total pressure, total temperature and entropy are presented as absolute fluctuations relative to the average obtained with the EXP data downstream the respective rotors of the investigated stage (or IGV). These fluctuations are then normalized by the overall increase of each respective aerodynamic quantity through CREATE at the peak efficiency operating point (following the definition in Section 2.4.6). The flow angle is presented as difference between URANS and EXP (URANS minus EXP). Planes downstream a stator or rotor are plotted with dashed and solid lines respectively. This representation method gives a very good overview of the flow field evolution through the compressor and allows the identification of zones with major time mean differences between results obtained with the numerical and experimental methods.

The steady RANS method is known to show globally very similar results compared to the URANS and are not presented here. A more detailed analysis of specific zones of interest results will be presented in the subsequent Sections using also all the other available databases.

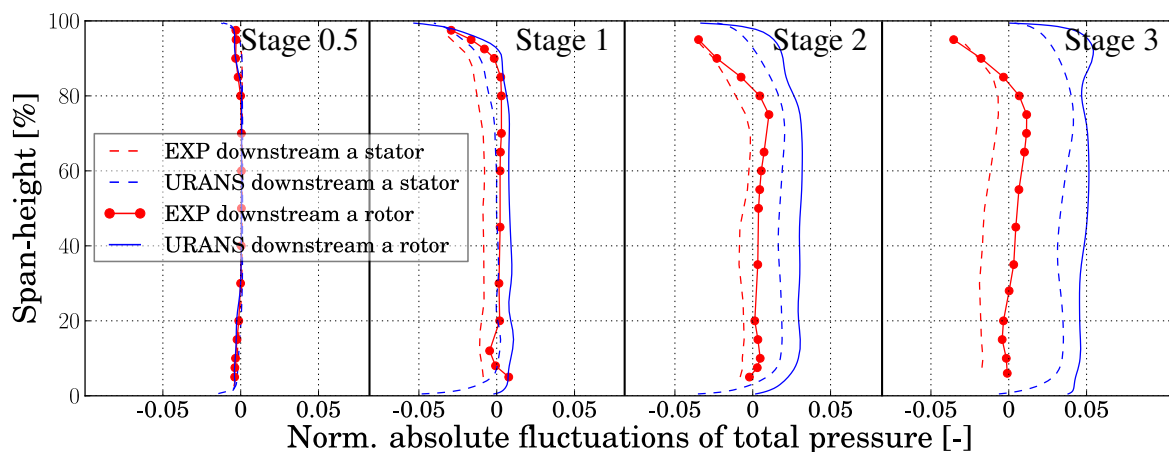


Figure 3.7 Overview for total pressure fluctuations evolution through 3.5 stages with EXP and URANS, relative to average EXP from downstream of each rotor (or IGV)



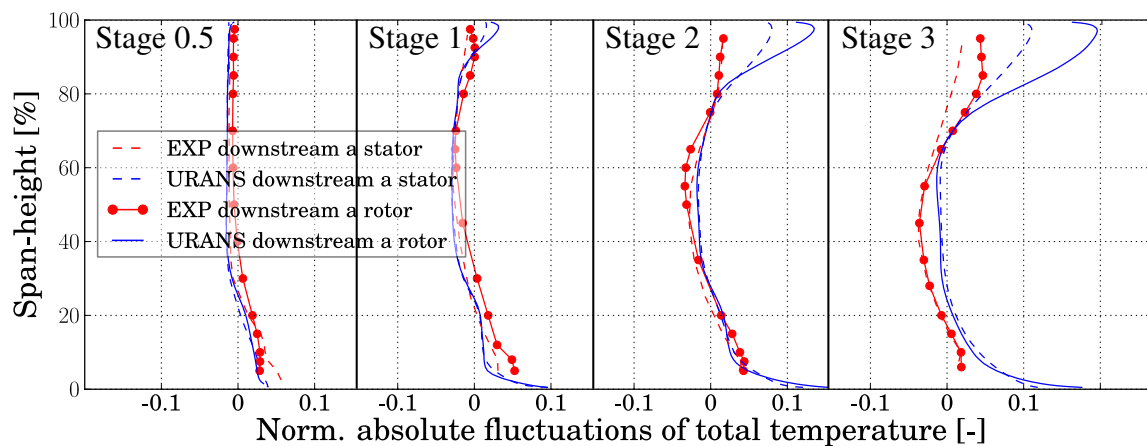


Figure 3.8 Overview for total temperature fluctuations evolution through 3.5 stages with EXP and URANS, relative to average EXP from downstream of each rotor (or IGV)

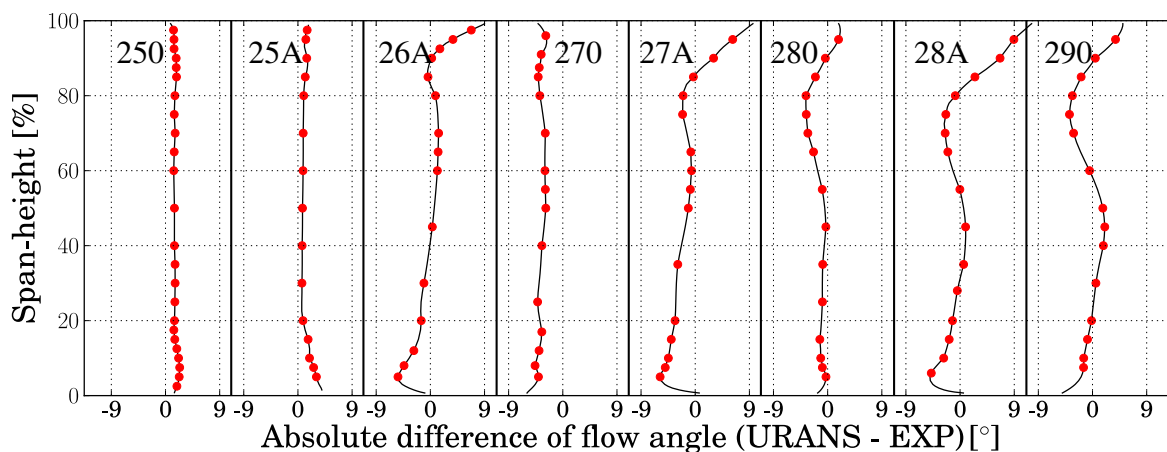


Figure 3.9 Overview for flow angle evolution through 3.5 stages as difference between EXP and URANS

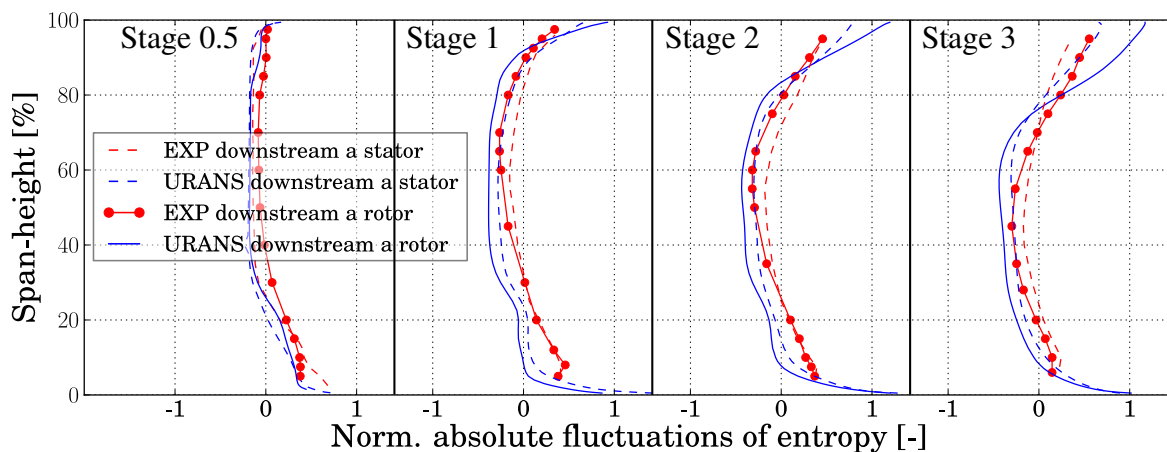


Figure 3.10 Overview for entropy fluctuations evolution through 3.5 stages with EXP and URANS, relative to average EXP from downstream of each rotor (or IGV)

### Crossing the IGV

- Losses induced by the hub and tip leakage flow appear as small total pressure deficit, flow angle increase, and as entropy rise at 10 and 90% span-height respectively.
- An offset of  $1.5^\circ$  in flow angle is found at the inlet of the compressor between the two databases. Downstream the IGV an offset of  $1.0^\circ$  is found. Such a small variation in the inlet flow angle of a blade row is not expected to lead to an almost equal change of the outlet flow angle of the respective blade row. This offset is thus related to the measurement uncertainty and not to wrongfully imposed boundary conditions. It suggests a constant mis-alignment of the probes in the acquisition system. The same offset is found downstream the stator 1 but not downstream any other stator.

### Crossing the rotors

- A globally increasing offset in total pressure but not total temperature rise is found from rotor to rotor. The global comparison of the entropy indicates an under-prediction of losses in the rotors. The entropy rise is higher in all the rotors in EXP than RANS, except in the rotor tip region. That means with the same amount of work, the RANS predicts a higher total pressure rise.
- The work is roughly evenly distributed from the hub towards 85% span-height in the rotor 1, 80% span-height in the rotor 2 and 60% span-height in the rotor 3. The zone above these span-heights shall be labeled boundary layer in a sense of that it shows flow physics very different from the main stream one. That boundary layer thickens in axial flow direction. The boundary layer is characterized by a very significant increase of the total temperature, where it decreases only very close to the casing again. This total temperature increase is largely over-predicted in the RANS. The increase can be attributed to two effects. Firstly, the tangential velocity of the blades increases naturally with bigger radius, and additionally their stagger angle increases. That can lead to an increase of the work exchanged between the blade and the fluid according to the Euler equation (Equation 1.1), if the deflection of the flow is roughly the same. Secondly, the tip leakage flow causes a decrease of the axial velocity in the tip region. That is found as an over-deflection of about  $9^\circ$  of the flow (see radial distributions of flow angle downstream rotors). The over-deflection increases the tangential velocity component and therefore, according to the Euler equation, has a positive effect on the work of the rotors. That leads eventually to a notable over-prediction of the total pressure rise and entropy production in the (U)RANS results.

Only very close to the casing, the velocity of the fluid decreases significantly due to the casing boundary layer. That causes a decrease of the work output, hence total temperature and pressure decrease.

That does not necessarily lead to an increase of the incidence angle in the subsequent rows. The tip mis-prediction will be discussed later on in Section 3.3.

- A small zone of increased losses are found in an entropy rise up to 20% span-height in the IGV and the rotor 1 induced by the hub clearance vortex emitted from the IGV and transported far downstream (difficult to see at this scale).

### Crossing the stators

- The re-circulations underneath the shrouded stators are not taken into account in the simulations, which could be an explanation for increased losses in the hub region in the experimental data (see entropy distributions).
- A small closed corner separations is detected in the stator 1. The related losses can be seen as weak valley at 85% span-height in the total pressure distributions for EXP and URANS. This separation will be discussed in more detail in Section 3.4.1.
- Likewise, a separation is found in the stator 3 at mid-span. It causes higher total pressure losses in the (U)RANS than EXP (a deeper valley in the radial distribution). A focus is put on this separation in Section 3.4.2.
- In the tip region (from about 80% span-height to casing) the total temperature, pressure and nonphysical entropy decreases reflect the radially inwards diving of the tip clearance vortex.

### 3.2.1 Interaction between pneumatic probe and downstream stator

The inter-row planes downstream the rotors 2 and 3 have been identified to cause systematically difficulties for the steady three-hole PNEUM measurements. These problems are not captured with a classical measurement uncertainty analysis. In the case of the PNEUM measurements, the velocity is calculated based on the measured static and total pressure, and total temperature. The total pressure and temperature measurements are considered trustworthy judged by extensive analysis of the measurement data. The difficulty lies in the measurement of the static pressure as demonstrated in Figure 3.11. Here the absolute flow angle  $\alpha$  is indicated in the legend. The velocity triangles are given based on URANS, PNEUM, and LDA data for the flow field at mid-span at the peak efficiency operating point.

- A good fit is found in the rotor 1, the flow angle variation is smaller than  $1.4^\circ$  - well measured and predicted, and the axial velocity variations are smaller than the measurement uncertainty (not readable because of the scale of the Figure and the normalization).
- The absolute velocity ( $V$ ) is under-estimated with the PNEUM measurements by 8.5% downstream the rotor 2 and by 8.0% downstream the rotor 3. At the same time the absolute flow angle is considered well measured with variations smaller than  $2^\circ$  between the databases. Likewise, Coldrick et al. (2004) demonstrate in an experimental work that a multi hole probe gives good estimations of the flow angle in the highly unsteady flow field downstream a rotor. The rotor wake passing in front of a multi-hole

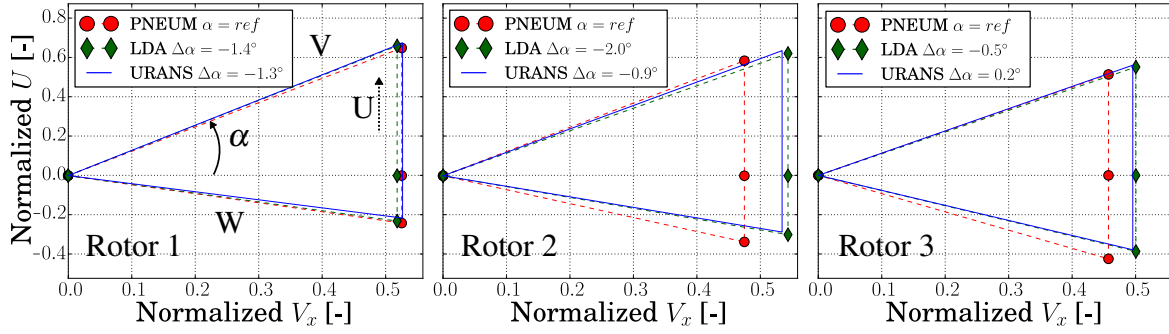


Figure 3.11 Velocity triangles for mid-span downstream the rotors 1, 2, and 3 based on URANS, PNEUM, and LDA data, flow angle relative to PNEUM indicated in legend, ratio of  $U/V_x$  deformed because of confidentiality

probe would lead to an equal time mean modulation of the static pressure on all holes of the probe, and eventually to no change in the flow angle.

- Note that the same comparison of the velocity triangles downstream the stators would not show an under-estimation of the velocity.

Having a good fit in the rotor 1 between all databases, and still a good fit between the (U)RANS and LDA results downstream the rotors 2 and 3, the PNEUM velocity (actually static pressure) measurement is considered falsified downstream the rotors 2 and 3. This wrong measurement of the static pressure can also be highlighted by a deficit in the mass flow rate calculated (integration) with the PNEUM measurement (not shown here). The error in the static pressure measurement shall be quantified with the help of the LDA measurements and the trustworthy PNEUM total pressure and temperature measurements. The static temperature can be computed from the first law of thermodynamics, applying the steady flow energy equation (Dixon and Hall, 2010) while assuming a constant specific heat capacity at constant pressure according to Equation 3.8.

$$T_{s,LDA} = T_{t,PNEUM} - \frac{V_{LDA}^2}{2c_p} \quad (3.8)$$

For compressible flows the static pressure can be computed from the ratio of the total temperature ( $T_{t,PNEUM}$ ) to the static temperature ( $T_{s,LDA}$ ) with Equation 3.9.

$$P_{s,LDA} = P_{t,PNEUM} \cdot \left( \frac{T_{s,LDA}}{T_{t,PNEUM}} \right)^{\frac{\gamma}{\gamma-1}} \quad (3.9)$$

where  $\gamma = 1.4$  is taken. The static pressure is presented for PNEUM, URANS, LDA in Figure 3.12. A clear over-estimation of the static pressure can be seen with the PNEUM measurement. The PNEUM static pressure is higher than the predicted one and also higher than the one estimated by the LDA data with Equation 3.9. When treating the last measurement point close to the casing as an exception (a static pressure drop is not expected here), the

PNEUM data over-estimates the static pressure also in comparison to the one found at the casing (labeled as casing  $P_s$ ).

The difference between the static pressure estimated with PNEUM and LDA can be used as a correction value for the PNEUM measurement. The resulting static pressure distribution is the red dotted line in Figure 3.12. A linear fit is applied to this distribution, while ignoring the last point close to the casing. The difference to the URANS data is not decreasing but the linear fit (based on the corrected values) gives a better fit to the casing pressure measurement than the original PNEUM data.

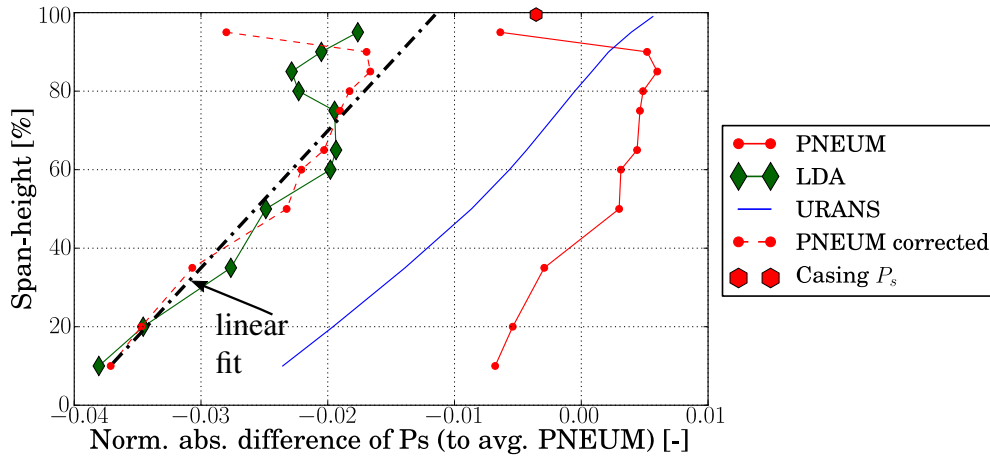


Figure 3.12 Absolute difference of static pressure for PNEUM, URANS, LDA, and the corrected values for PNEUM (red dashed line) as an example at plane 27A. Additionally the casing pressure measurement is indicated. Static pressure is given relative to average PNEUM

The static pressure is systematically over-estimated by the PNEUM probes downstream the rotor 3 as well. The differences between the static pressure estimated with PNEUM and LDA are presented in Figure 3.13 for the planes downstream the rotor 2 (27A) and 3 (28A). The difference is normalized by the reference pressure ( $P_{ref}$ ). On the vertical axis, the blockage induced by the PNEUM probe (and shaft) is indicated as ratio over the area of one stator channel. The blockage is presented for span-heights from 20 to 95% span-height. The shaft and the probe are simplified as a rectangular with a 4 mm width. The blockage is increasing when descending with the probe (and shaft) into the compressor vane. The data for the plane 27A is presented in red, and for 28A in blue. Linear fits are computed for the two data-sets without taking into account the last measurement point close to the casing. A possible interaction between the probe and the casing flow field does not allow establishing a valid link between blockage and static pressure over-estimation. The linear fits are labeled "correction function". An almost constant offset is found at the plane 27A, which corresponds to about 7.8% of the reference pressure. The offset at plane 28A corresponds to between 5.9% and 9.9% of the reference pressure, where the smaller offset is found towards the casing. Here several data points differ from a possible offset. Thus they have a notable influence on the linear fit. In consequence, a common correction function for the PNEUM

data in the two planes cannot be defined. Though, a constant offset of 7.8% would be a good approximation for the correction.

A constant offset suggest that this over-estimation of the static pressure does not origin in the channel-blockage induced by the probe in the flow field. An increased effect would be otherwise expected at the measurement points towards the hub, where the largest blockage is found (deepest immersion of probe). The correction factor cannot be compared between different operating points because PNEUM and LDA are not available at another common operating point.

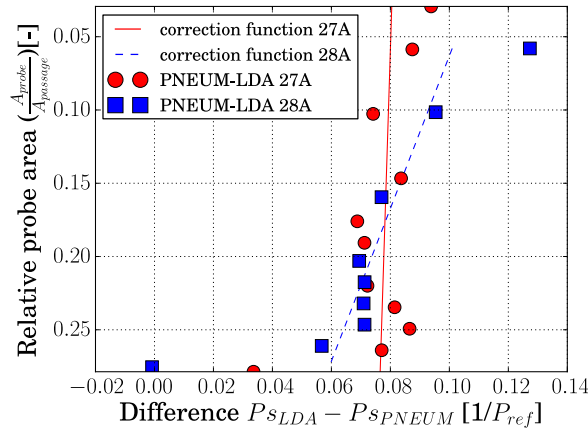


Figure 3.13 Probe induced blockage area over difference between  $P_{s,LDA}$  and  $P_{s,PNEUM}$  from 20% to 95% span-height

With the available data, the static pressure measurement problem cannot be attributed to a specific influence by the downstream stator on the flow field. The difference in static pressure between the  $P_{s,PNEUM}$  and  $P_{s,LDA}$  is presented in Figure 3.14. A weak modulation is found in the difference map but the  $P_{s,PNEUM}$  is over-estimated at all measurement points (note the scale of the color map). The same is valid for the plane 28A. The weak circumferential modulation is a sign for an interaction between the probe and the downstream stator. This interaction might falsify the measurement to higher static pressure measurements independently of the circumferential position. An isolated study would be necessary for the understanding of the mechanism of the interaction.

Hoenen et al. (2012) found a local acceleration around the probe head due to a nozzle effect created between a blade trailing edge and the probe head. That would cause the static pressure to decrease but is not the case with the PNEUM measurements in CREATE. One of very few applicable studies of interactions between a pneumatic probe and a downstream stator in a compressor was conducted by Coldrick et al. (2003). The authors conducted a numerical study on a high-speed axial compressor about placing a multi-hole probe in proximity of a stator leading edge. In their study, the mass flow rate was not kept constant between the case with and without the probe. Furthermore, the probe was placed directly at the inlet plane of the stator. That does not correspond to the probe placement in CREATE and thus the results of the work are not transferable.

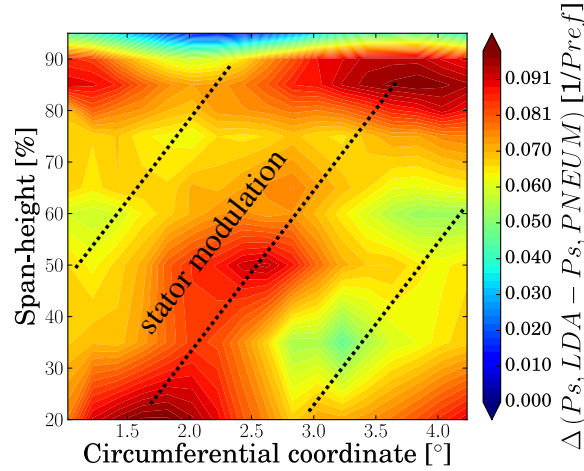


Figure 3.14 Difference of static pressure between  $P_{s,LDA}$  and  $P_{s,PNEUM}$  at the plane 27A

In the scope of the present thesis work, no further experimental and numerical studies could be conducted for understanding the mechanism behind the interaction between the pneumatic probe and the downstream stator. Possible studies could be:

- Experimental: A simplified study with the pneumatic probe upstream of two bars that represent the stator leading edge. A possible change in the measured static pressure could be investigated. Additional PIV (Particle Image Velocimetry) measurements could give time resolved information about the velocity field around the head of the probe.
- Numerical: A numerical study would allow a deeper insight in the flow field around the probe head. The setup could be fairly simple with an isolated stator in a first test case. The difficulty would be the meshing of the probe in simulation domain, which is expected to be difficult with a structured mesh. A second test case should include an upstream rotor because its influence cannot be neglected. The numerical study has to be realized without a shift in the operating point between the configuration with and without a probe in the domain. Consequently the circumferential extension of the simulation domain should be large enough for minimizing the global effect of the probe.

In summary, a correction of the measured static pressure is necessary during the post-processing of the PNEUM static pressure data at the planes 27A and 28A. In a first approximation, a constant decrease of the measured static pressure is advisable in the order of 7.8% of the reference pressure. Another option would be to rely solely on the casing pressure measurements for the static pressure. The increase of the static pressure from hub to casing should be considered though. The PNEUM data has not been corrected in this work because mostly velocity fluctuations are presented instead of absolute values.

### 3.2.2 Challenge for LDA measurement downstream a stator

The LDA measurement technique is challenged by the measurements downstream the stators. The axial and tangential velocities are given as relative difference between the LDA measurements and the PNEUM and URANS data in Figure 3.15. The respective PNEUM and URANS velocity component is taken as reference at each span-height.

A systematic over-estimation of the axial velocity (solid lines) by the LDA technique is identified as positive values of up to 16%. The difference tends to increase from the hub to the casing between LDA and PNEUM. Only downstream the stator 1, the difference increases from mid-span to the hub again. Downstream the stator 3, the axial velocity comparison between LDA and PNEUM reaches negative values below mid-span. This is attributed to difficulties for the PNEUM measurement in capturing the good time mean axial velocity in the presence of a separated blade boundary layer, which will be discussed in Section 3.4.2.

The tangential velocity (dashed line) is locally under or overestimated but a systematic error is not identifiable here. The tangential velocities of LDA and PNEUM fit fairly well downstream the stators 2 and 3, with a difference close to the measurement uncertainty for the most part of the span. The tangential velocity differences between LDA and URANS are understood as actual mis-predictions. For example, the rotor tip leakage flow is transported through the stators, and this specific mis-prediction is found as positive peaks at 80% downstream the stators 2 and 3 (blue dashed line).

The axial velocity over-estimation goes beyond the measurement uncertainty range ( $\pm 2\%$  to  $\pm 4\%$ ). The following points were verified and justify the analysis in this Section:

- A problem with the PNEUM measurement data is excluded here as explanation because the comparison between LDA and URANS shows the same tendency as the one between LDA and PNEUM. Moreover the URANS is expected to rather over-predict the velocity due to the mass flow rate offset between the simulations and the measurements (shown earlier in Figure 3.1)

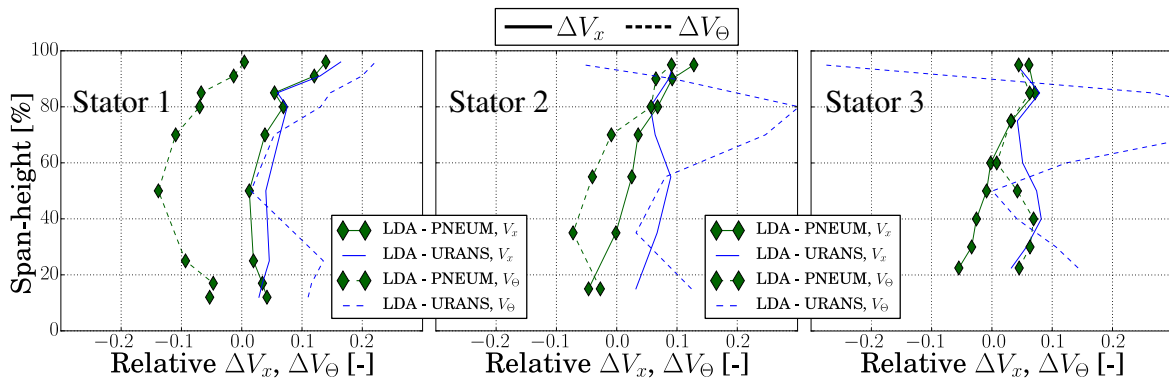


Figure 3.15 Axial velocity (solid lines) and tangential velocity (dashed lines) given as relative differences between the LDA measurements and the PNEUM and URANS as reference data for the planes downstream the stators 1 to 3



- The number of ensemble average windows is not found to have an influence (not detailed here). A lower number of windows decreases the temporal ensemble average resolution, while having more particles in each window for the ensemble averaging. This was verified with the goal of decreasing the statistical uncertainty but no difference could be found in the radial axial velocity profiles.
- Moreover it has been verified that there is no link between the over-estimation of the axial velocity downstream of the stators, and the LDA rotor wake measurements downstream of the rotors. The rotor wake depth is well captured by the LDA downstream of the rotors (not shown here).

Figure 3.16a shows the time averaged axial velocity based on LDA, PNEUM, and URANS data for 70% span-height downstream the stator 2 . The LDA data is only available over one stator pitch. PNEUM and URANS data are given for the same matched stator pitch. The variation compared to the other stator pitches is given with the colored bands, which trace the minimum and maximum time averaged axial velocities over all 7 (URANS) or 4 (PNEUM) stator pitches for each given relative position of a stator passage.

LDA over-estimates the time mean axial velocity at all circumferential positions besides of at the stator wakes suction side (right hand side). The time mean stator-stator interaction influence on the PNEUM data (red band) is of smaller amplitude than the actual difference between LDA and PNEUM. Thus the over-estimation is not expected to origin from simply a too limited view of the flow field. It is to note though that the stator-interactions are of greater importance in the URANS results than found by the PNEUM. The prediction capabilities of the stator-stator interactions by the URANS is discussed in Section 4.3.

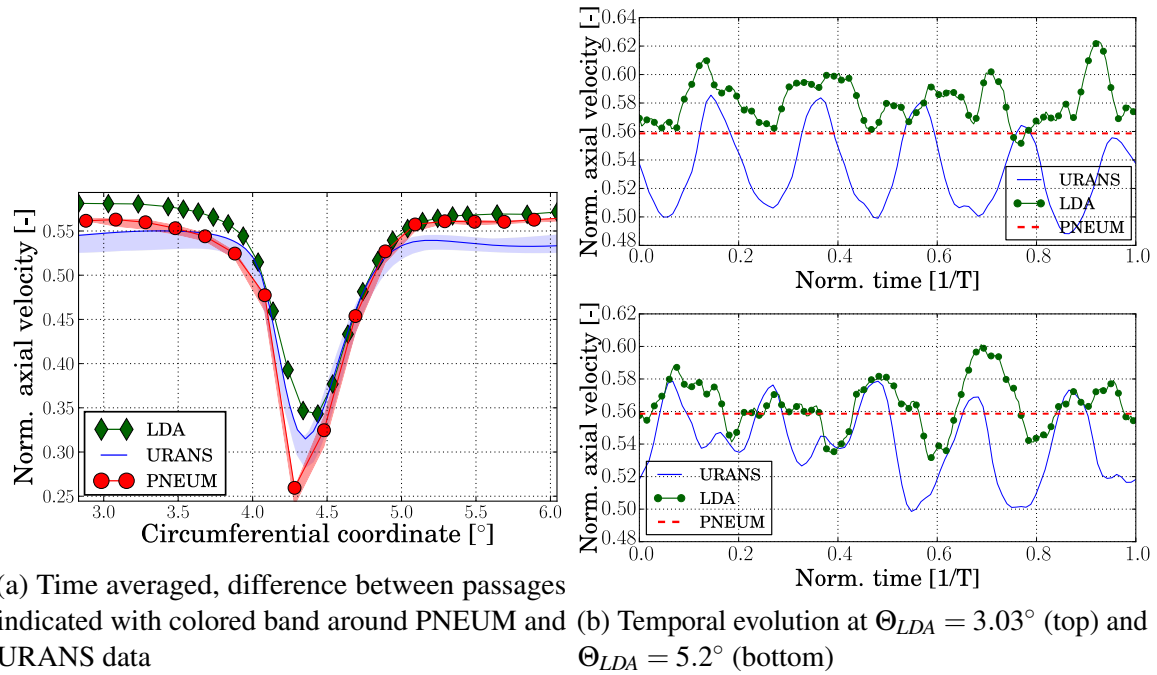


Figure 3.16 Axial velocity based on LDA and URANS data at 280 at 70% span-height

In Figure 3.16b, the axial velocity is presented over one temporal period at two circumferential positions based on URANS and LDA data. Additionally, the PNEUM value is indicated with a red dashed line. The circumferential position  $\Theta = 3.0^\circ$  is chosen to present the flow field evolution over time at about mid-passage. A matching spatial position is chosen for the URANS extraction, where the temporal match is an approximation because of the limited spatial information (one stator pitch) with the LDA measurements. Fluctuations of higher amplitude are found in the URANS results compared to the one measured by LDA.

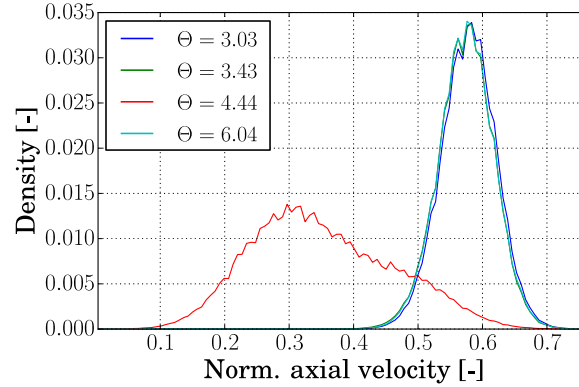


Figure 3.17 Time averaged histograms calculated based on LDA particles ordered in their respective ensemble average window for four circumferential positions at plane 270 at 70% span-height

The valleys are mainly caused by the rotor 2 wake passing. They induce a temporal mode 5, corresponding to 5 blade passings over one temporal period ( $1/16^{th}$  of a rotation). The mode appears mixed out in the measurements. Moreover the LDA measurement gives a higher axial velocity than PNEUM at almost all ensemble average time points.

Probability density distributions of the registered LDA particles are a good first measure for the measurement quality. They can be computed on the particles per ensemble average window and "time" averaged in the next step. A Gaussian normal distribution would indicate that there are no outliers in the measurement, which can originate from small variations in the particle size or local too little seeding. A sufficient high number of the registered particles has been verified per ensemble average window. Variations can occur due to the rotor passing but they are small and not considered to influence the measurement result (particle number is not detailed here). Figure 3.17 shows the time mean probability density functions for four circumferential positions:  $3.03^\circ$ ,  $3.403^\circ$ ,  $4.44^\circ$ , and  $6.04^\circ$ .

The third position ( $4.44^\circ$ ) represents a measurement point in the stator wake. The histogram has a Gaussian peak at a low axial velocity but a long arm towards higher velocities which are physical and mainly caused by the flutter of the wake in circumferential direction due to RSI. Outside of the stator wake, Gaussian like distributions are found which are a sign of a good measurement quality. Furthermore, the probability density distributions have been verified window by window (not shown here), and they do not indicate any measurement problem.

**In conclusion**, the LDA measurement over-estimates the time mean axial velocity of the downstream the stators. The measurement quality does not indicate any reason for the over-estimation. Downstream the rotors, the rotor wake depth and mean velocity is accurately captured and is therefore not the origin of the problem. A measurement inside of the stator passage would be necessary to identify the onset of the velocity over-estimation with the LDA technique. A correction factor could be applied for the time mean axial velocity values but would be specific to an operating point and inter-row plane.

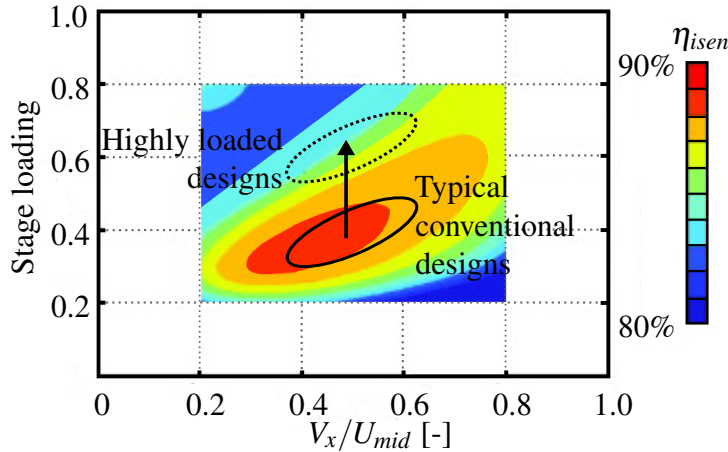


Figure 3.18 Compressor Smith chart showing stage loading over the normalized axial velocity with contours of efficiency ( $\eta_{isen}$ ) and indication of conventional and loaded stage design adapted from Dickens and Day (2010)

### 3.3 Characterization of the flow field in the rotors

For decreasing the fuel consumption of an aircraft engine, one way is to lower its weight. This can be achieved by reducing the number of compression stages while increasing or keeping the overall pressure ratio. That requires a higher pressure rise per stage, which can be achieved by increasing the stage loading coefficient (enthalpy rise in the rotors). Dickens and Day (2010) show in their work that the stage efficiency of a high loaded design stage is lower than the one from a conventionally loaded stage. The classification of conventional and high loading is shown in Figure 3.18. The horizontal axis gives with  $\frac{V_x}{U_{mid}}$  a dimensionless mass flow rate equivalent and the vertical axis shows the stage loading.

Dickens and Day (2010) find in a numerical and experimental work that increasing the stage loading from 0.45 to 0.55 leads to a minimum stage efficiency penalty of 0.9%. From 0.55 to 0.65 stage loading (high loaded), a more important additional efficiency loss of 2.4% can occur. Furthermore their work shows that the degree of reaction (enthalpy rise in the rotor relative to the one in the overall stage) needs to be increased together with the stage loading to obtain an optimal efficiency for a given stage. Otherwise the downstream stator risks large separations.

The stage loading coefficient for CREATE is presented in Figure 3.19 based on the PNEUM data in red, LDA in green, and URANS in blue for the peak efficiency point. Addi-

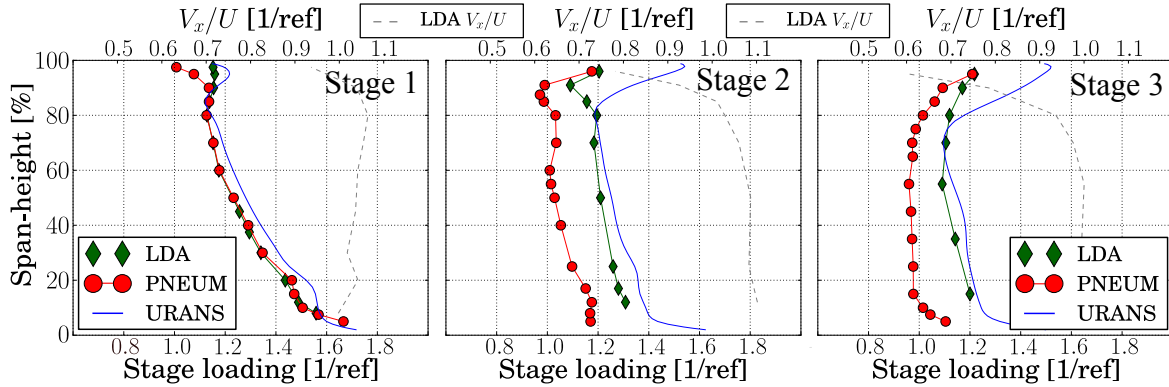


Figure 3.19 Stage loading for the stages 1 (left), 2 (middle), and 3 (right) based on PNEUM (red), LDA (green), URANS (blue), data normalized by one common set of reference values for all three stages defining a conventional design because of confidentiality

tionally the axial velocity is traced relative to the circumferential velocity at each respective span-height on the second horizontal axis for comparing the data points with Figure 3.18. The data for the stages is presented relative to one common set of reference values defining a conventional design. A normalized loading above 1 corresponds to an unconventional loading. Here the stage loading coefficient is calculated according to Equation 3.10.

$$\Phi = \frac{\Delta h_{rotor}}{U^2} = \frac{V_{\Theta, rotor-out} - V_{\Theta, rotor-in}}{U} \quad (3.10)$$

where  $U$  is the circumferential velocity of the blade at the given span-height, and the  $\Delta h$  is calculated with the help of the Euler Equation 1.1.  $V_{\Theta}$  is the absolute tangential velocity of the fluid at the inlet and outlet of a rotor respectively.

For the most part of the span, the stage loading corresponds well between LDA and URANS results. The latter has a tendency of higher loading and work output as already found in the radial total pressure and entropy distributions in Figure 3.7 and Figure 3.10.

The stage 1 loading with a low normalized  $\frac{V_x}{U}$  of about 0.75 lies close to the conventional design at mid-span. The hub loading with higher values tends towards unconventional high loading with a possible penalty in efficiency (see Figure 3.18). The loading of the stages 2 and 3 is rather conventional from the hub up to 80% span-height. The PNEUM and LDA capture the effect of the special rotor 2 blade design, which allows an unloading of the rotor tip. The URANS is not able to capture this effect; the mis-prediction of the tip flow field appears as a large over-loading of the rotor 2 and 3 tips. That was shown to result into an over-prediction of the pressure rise in Figure 3.7. The constant offset between the PNEUM stage loading and the ones of the other databases lies in local problems with the measurement technique in the multistage high-speed environment and will be discussed in Section 3.2.1.

The degree of reaction is calculated with Equation 3.11 taken from Dixon and Hall (2010), where 1, 2, and 3 label respectively the rotor inlet, outlet, and the stator outlet. An uncertainty in the calculation arises due to the distance of the inter-row planes to the blade rows, especially at plane 25A as inlet-plane to the rotor 1. Figure 3.20 presents the degree of reaction based on the URANS result and the LDA data. The degree of reaction is normalized

by a conventional value, whereby a value higher than 1 corresponds to a degree of reaction considered as higher than conventional.

$$Reaction = \frac{0.5(W_3^2 - W_2^2) + 0.5(U_2^2 - U_3^2)}{U_2 V_{\Theta,2} - U_1 V_{\Theta,1}} \quad (3.11)$$

Globally a good prediction of the degree of reaction is achieved with the URANS compared to the measured value. The degree of reaction is high with values above the conventional one (used for the normalization), besides of in the rotor 1 below 40% span-height. Especially in this region, a high degree of reaction would be beneficial to avoid a hub-corner separation in the stator 1 according to the work of Dickens and Day (2010). It will be shown that no hub-corner but a casing corner separation is found in the stator 1 in Section 3.4.1. Here the degree of reaction might actually not be high enough between 80 and 90% of span-height for the stage 1. In the tip regions of the rotors 2 and 3 a higher degree of reaction is found with the measurements than with the URANS. That is related to the tip flow field mis-predictions. The under-prediction of the degree of reaction in the stage 3 at mid-span might promote the over-prediction of a possible separation in the stator 3, which is discussed in the Section 3.4.2.

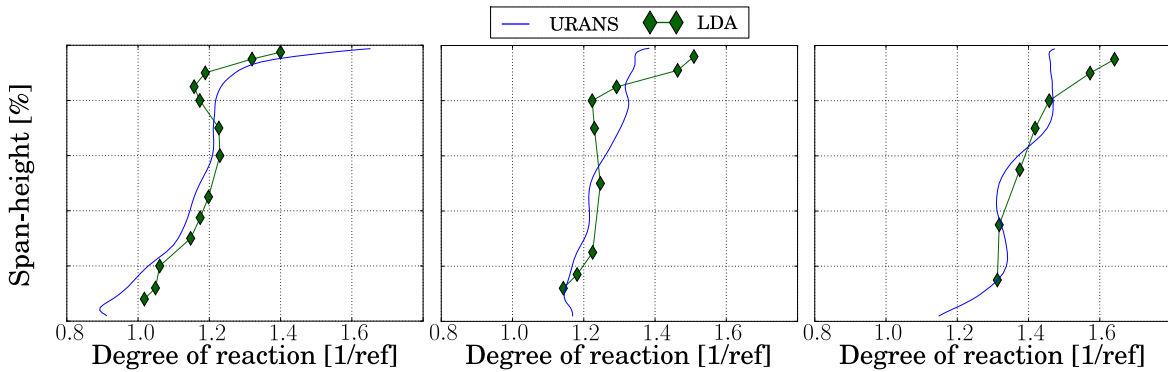


Figure 3.20 Degree of reaction of the stages 1, 2, and 3 based on LDA (green), CFD (blue), normalized by a conventional degree of reaction because of confidentiality

### 3.3.1 Description of the rotor tip flow field

The characteristics of the rotor tip flow field shall be discussed with a focus on why and to what extent it is not correctly predicted by the RANS. The mis-match between the measured and simulated time averaged tip flow field data is larger at the peak efficiency than at the near surge operating point. The time dependent (but not time mean) differences at the near surge operating point, and their important influence on the onset of instabilities are presented later in section 5.1. For the peak efficiency operating point the following flow field data is available:

- LDA measurements in the rotor 2 and 3 tip regions
- High frequency casing pressure measurement above all the rotors
- LDA and PNEUM measurements downstream the rotors at the inter-row planes
- (U)RANS simulation data



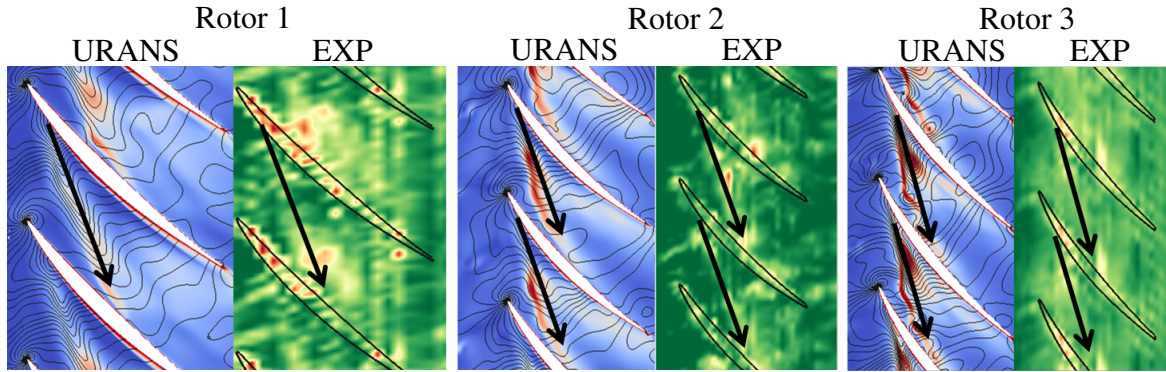


Figure 3.21 Rotors 1, 2, and 3 tip flow field at nominal OP for each rotor: standard deviation on casing pressure measurement (left), and numerical static pressure contour in black, and numerical relative velocity gradient in red/blue (right); Red shows high values

The rotor tip flow field will be discussed using unsteady numerical results and high-frequency casing pressure measurements. That shall give an idea about the tip clearance vortex (TCV) trajectory in the blade-to-blade cut plane. The relative velocity gradient is used to visualize the edges of the TCV induced blockage zone in the tip region of the rotors in the case of numerical results. The blockage is characterized by a lower axial velocity compared to the one in the free-stream. Furthermore, static pressure contours give additional information about the TCV trajectories, and help to detect any spillage or backflow from one passage to another. The instantaneous flow field maps allow the detection of important differences between the passages.

Instantaneous screenshots of the whole spatial period of the tip flow field are not measurable with the available measurement techniques for CREATE. The high-frequency static pressure measurements, recorded over the spatial period of CREATE, allow only the calculation of an ensemble average phase locked to the temporal period with 100 temporal windows at each measurement grid point. Using 2 seconds of measurement data with a sampling frequency of 500 kHz at a rotating frequency of 192 Hz, each window contains about 10000 measurement points. The study of the standard deviation is necessary for detecting phenomena that are not determinist, such as non-synchronous rotor tip clearance vortex oscillations. In a next step the standard deviation can be averaged over the 100 temporal windows in the frame of the rotor. That averages out fluctuations from passage to passage which are caused by RSI. The interaction between the TCV and the incoming stator wakes is expected to induce an oscillation of the TCV trajectory as depicted by Marty et al. (2014), and therefore give rise locally to the standard deviation.

The numerical results for the tip flow field just below the blade tip are presented for the rotors 1, 2, and 3 in Figure 3.21. The standard deviation based on the measurements is shown in the same Figure on the right hand side for each rotor. In the numerical and experimental representation, simplified TCV trajectories are indicated with black arrows. They have an identical inclination in both representations to have a common visual reference. It has to be noted that the standard deviation maps let rather identify the upstream blockage zone edges induced by the TCVs (similar to the relative velocity gradient maps) than their trajectory.

The standard deviations maps are not built with an identical scale from rotor to rotor. A common scale would not allow the trace of the tip flow field features due to an increasing unsteadiness in the main flow direction through the compressor.

The numerical results show clearly the modulation of the TCV by the interaction with the incoming stator wakes. The modulation does not induce any significant changes to the TCV trajectory, and is mainly seen in the wave-like edge of the blockage zone in the velocity gradient map. The modulation of the TCV in the rotor 1 by the incoming IGV wakes is weak compared to the modulations found in the rotors 2 and 3 by the incoming stator 1 and 2 wakes respectively. In the experimental results an increased standard deviation is found especially at the rotor blade tips. That reflects the high unsteadiness of the tip leakage flow. The trajectory of the highly unsteady tip clearance vortex can be traced on a diagonal in each passage. It marks a frontier between low and high standard deviation. The trajectories are very similar in all passages, pointing towards about mid-chord with the very simplified representation of the trajectory as a straight line (black arrows).

At mid-chord, the point of the interaction between the TCV and the tip gap flow of the adjacent blade is marked by a spot of higher standard deviation than anywhere else in the passage. That is the easiest detectable in the rotor 1 because of the general lower unsteadiness of the rotor 1 tip flow field compared to the ones in the rotors 2 and 3 (first stage vs. rear stages). The spot is found to be modulated by RSI (not visible here with time averages).

The comparison between the numerical and experimental TCV trajectories shows a fairly good match in all rotors at the nominal operating point. The numerical prediction of the important blockage zone edge upstream of the TCV in the rotor 3 is reflected in an only slight increase of the standard deviation in the experimental results. Judged by the weak increase of standard deviation, the blockage zone appears over-predicted in the rotors 2 and 3 already at the nominal operating point. The standard deviation maps do not allow a judgment about the blockage zone downstream the TCV trajectories. Here the relative velocity gradient maps indicate an important blockage zone spanning over the whole passage width in the rotors 2 and 3, which will be discussed in this Section.

Globally, the comparison between URANS and the high-frequency casing pressure measurement shows a good match of the tip clearance vortex trajectory in the blade-to-blade cut at the peak efficiency point.

### Downstream the rotors

Technical measurement constraints do not allow LDA measurements inside the rotor 1 passages at the time being. Therefore, a 2D view from downstream on the rotor shall give an idea about the tip leakage flow at the peak efficiency operating point, and is presented in the rotor frame for the plane 26A in the Figure 3.22a.

The time mean (rotor frame) axial velocity flow field is presented with a grey color map for the URANS results at the peak efficiency operating point, and superimposed is the time mean standard deviation of the velocity (computed based on the LDA particle velocities ordered in the ensemble average windows) with blue-to-red contours. The standard deviation helps to identify the non-deterministic unsteadiness linked to the wakes and the tip leakage flow. One rotor 1 and IGV period ( $\frac{2\pi}{32}$ ) is presented here.

The rotor 1 wakes can be seen as almost vertical stripes of low axial velocity and an increased standard deviation. The rotor is turning in positive circumferential direction. The focus shall be put on the wake in the middle of the Figure. At the hub a widening of the wake is found at the suction side, and further increased standard deviation. This is a sign of a separated blade boundary layer, which will be discussed in Section 3.3.3. The tip leakage flow is "accumulating" in the casing corner on the pressure side of the rotor wake. This zone is also marked by an increased unsteadiness. The contour lines of the standard deviation follow well the axial velocity distribution in this casing corner. In other words, the general shape of the rotor 1 tip flow field is well predicted.

Figure 3.22b shows the axial velocity distributions for 90% and 95% span-height. For this representation, the axial velocity signal is averaged in time, and then ensemble averaged over the passages to obtain one representative passage. The velocity is given relative to the maximum velocity at the given span-height. The previously mentioned tip leakage flow "accumulation" is found in both databases mainly between  $1.5$  to  $3^\circ$  at 95% span-height. URANS predicts a wake which is too deep. Furthermore, the casing boundary layer is too thick in the URANS results. That causes the axial velocity to drop between  $3.1^\circ$  and  $5.5^\circ$  at 95% span-height. The interaction between the tip leakage flow and the rotor blade boundary layer results in slightly wider and significantly deeper wakes in the URANS results than captured by LDA at 90% span-height. A measurement problem is excluded for the measurement of the wake depth because of two reasons: Firstly, probability

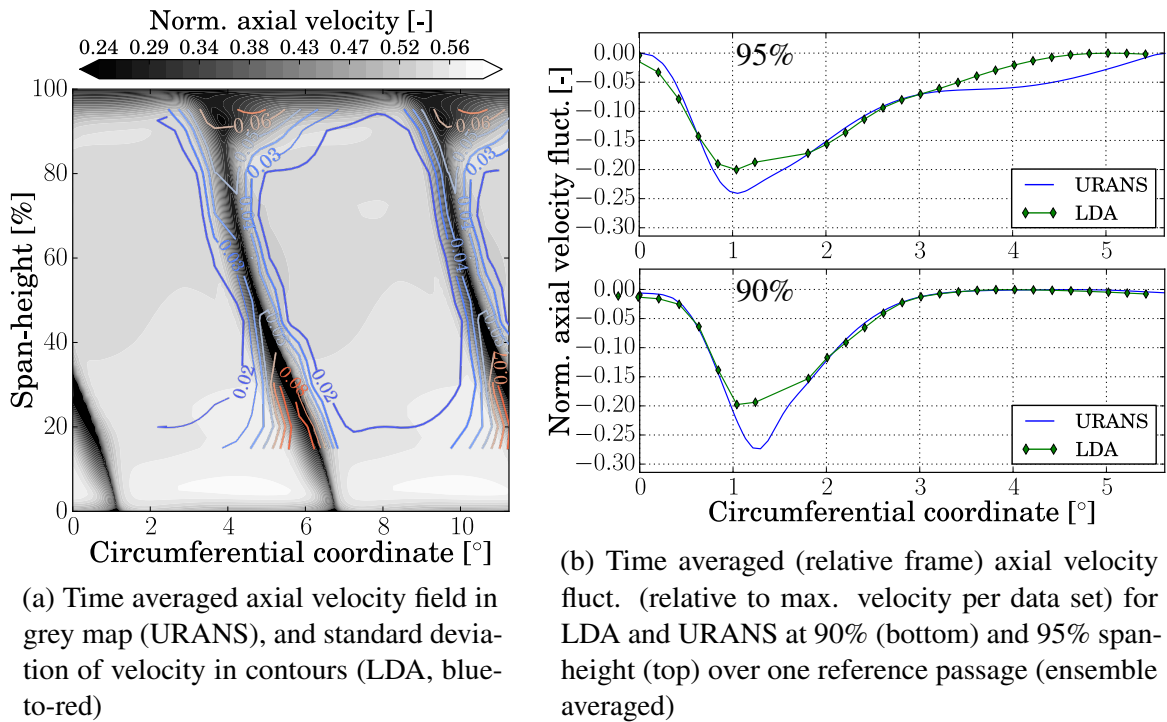


Figure 3.22 Characterization of the rotor 1 peak efficiency flow field at plane 26A in the rotor frame



density distributions on the particles, which are falling into the temporal ensemble average windows representing the rotor wakes, resemble closely normal distributions (not shown here). Secondly, the number of recorded LDA particles inside of the rotor wakes has been verified to be not significantly lower than in the passage flow.

**The blockage** induced by the tip leakage flow, wakes, and secondary flow structures shall be quantified with an integration of the related axial velocity deficit by using Equation 3.12. The importance of an axial velocity deficit is taken into account relative to the circumferentially averaged axial velocity. This eliminates the influence of differences in the mass flow rate when comparing multiple databases.

$$\text{blockage}(\text{span-height}) = \int_{\Theta, \min}^{\Theta, \max} \frac{V_{x, \max} - V_{x, \Theta}}{V_{x, \text{mean}}} \cdot \frac{d\Theta}{s} \quad (3.12)$$

where the blockage per span-height is defined relative to the blade pitch ( $s$ ), and all velocity and angle values are looked at per span-height. The blockage is shown for the URANS (blue) and LDA (green) at the plane 26A in Figure 3.23. The blockage increases towards the hub due to the hub corner separation, where URANS over-predicts the blockage by up to 4%. In the tip region from 80% span-height to the casing, the blockage over-prediction increases to up to 10%. This quantifies the results about the differences in the wake width and casing boundary thickness.

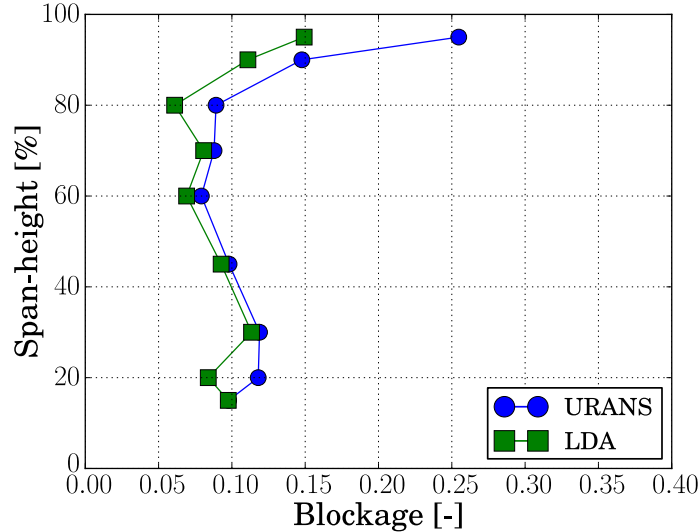


Figure 3.23 Blockage relative to the rotor 1 pitch based on LDA (green), URANS (blue), for the peak efficiency operating point

### Inside of the rotor passages

Further discussion of the blockage induced by the tip leakage flow requires the LDA measurements inside of the rotor 2 and 3 passages. They were conducted at the axial positions indicated in Figure 3.24.

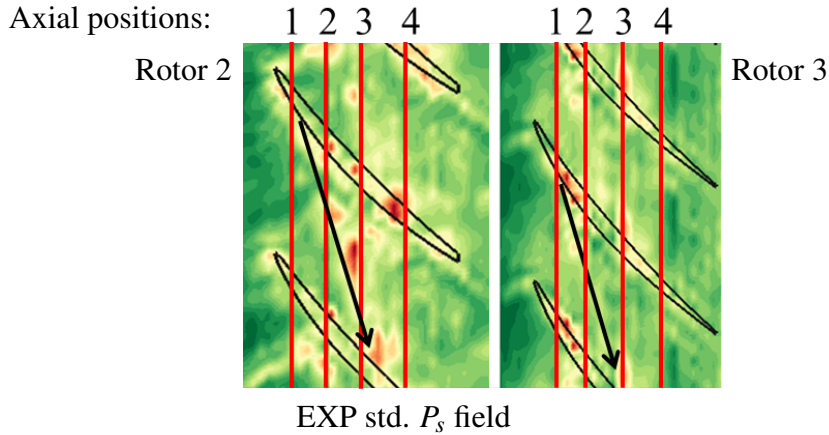


Figure 3.24 Axial measurement positions (labeled 1 to 4) in the rotors 2 (left) and 3 (right)

The tip leakage flow characteristics in the rotor 2 are discussed with the help of the axial velocity fluctuations at the axial measurement position 1, 2 and 3 in Figure 3.25. The URANS data is arithmetically averaged in time in the rotor frame. Only rotor clocking could induce any difference from passage to passage in this case. Thus, the URANS data is ensemble averaged over the five passages to obtain one reference passage signal. This treatment filters the time mean rotor clocking influence. The LDA data is given ensemble averaged over time because of a less good spatial discretization. Also here, a reference passage signal is calculated. Having the blade passing as reference, a good space-time match is possible. The blade passing is found as holes in the velocity distributions. In this comparison between signals over space and time, an uncertainty arises from the stator influence on the signal that is presented over time. There is a time mean influence, which is avoided by comparing only fluctuations in the following. Furthermore, there is a time dependent influence, which cannot be eliminated for this comparison but it has been estimated to be by far smaller than any differences presented in Figure 3.25.

Firstly, the validity of the space time comparison is proven without analysis of the tip flow field characteristics. In the top right Figure, the influence of the stator on the LDA time evolution is given as grey envelope. The pressure and suction sides are respectively located on the right and left hand side of each passage. One blade is indicated with a grey vertical bar. The stator influence is estimated as maximum difference between the time evolutions of 17 signals obtained from circumferential measurement locations covering one stator pitch. The difference is calculated in the rotating frame. The slim envelope represents the only weak modulation of the tip flow field by the surrounding stators. Therefore, the characteristic differences between URANS and LDA are considered valid for any circumferential position of the LDA measurements at the tip of the rotors.

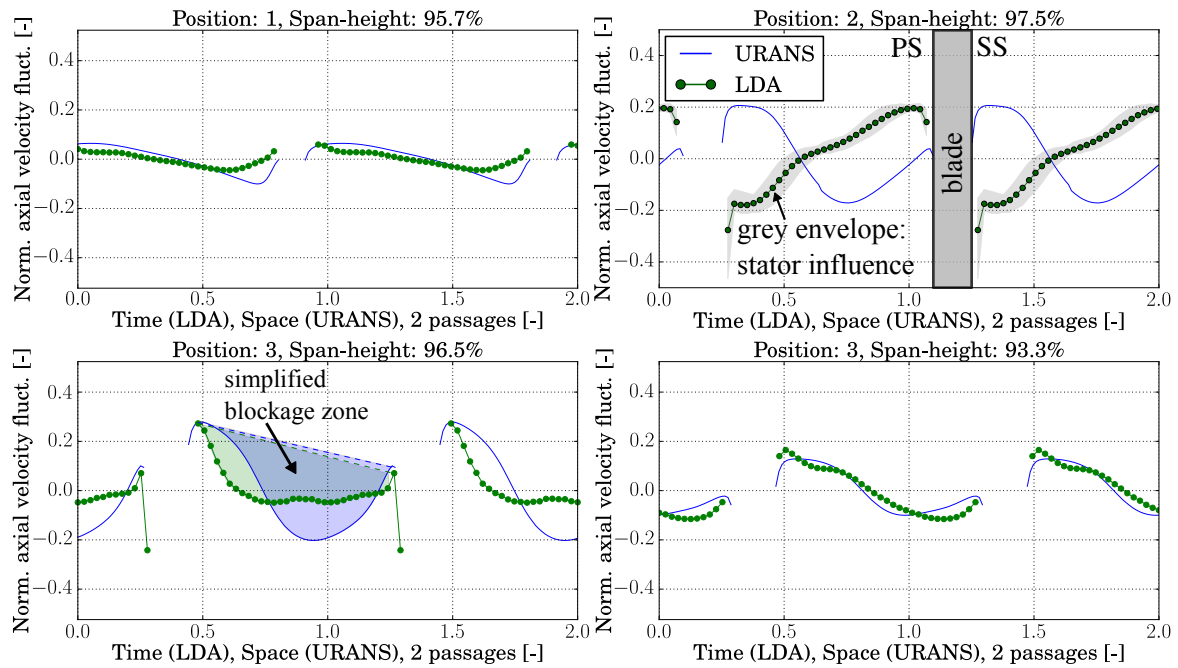


Figure 3.25 Axial velocity fluctuations at axial measurement position 1, 2, and 3 (two span-heights) in the rotor 2 at the peak efficiency operating point, simplified blockage zone marked in colors for position 3

Secondly, the main characteristic differences between the URANS and LDA rotor 2 tip flow field are summarized hereinafter.

- The axial position 1 does not show any trace of the tip clearance vortex. That corresponds well to earlier presented results in the blade-to-blade view (Figure 3.21).
- At the axial position 2, an axial velocity drop is found surprisingly close to the suction side (left side of passage) in the LDA measurements. It corresponds to the axial position of the spilling over of tip leakage flow through the tip gap (see Figure 3.21 and Figure 3.24).

In the URANS the tip leakage flow causes a blockage of the incoming flow over almost the whole passage width at this axial position. This was seen as increased velocity gradient upstream of the indicated TCV trajectory in Figure 3.21. There must be a small shift of the tip clearance vortex trajectory in the URANS results compared to the measurements. In the URANS data, this axial position is almost aligned with the tip clearance vortex trajectory, which explains the circumferential extend of the blockage zone.

- At the axial position 3 at 93.3% span-height, the radially inwards edge of the TCV is found, and shows a good match between URANS and LDA. The velocity gradient from the suction to the pressure side fits well up to mid-passage. Note that LDA shows a larger velocity drop than URANS close to the pressure side of the passage. That is

found because of the less circumferential dissipated TCV in the URANS, which shows a clear maximum velocity deficit at the horizontal position around 0.9.

- The main difference is found when cutting through the radial center of the tip clearance vortex, which is illustrated with the position 3 at 96.5% span-height. The LDA finds an axial velocity deficit, induced by the tip leakage flow, over almost the whole passage width. This can be deducted from the comparison between the velocity gradient in the passage between position 1 and 3.

Most importantly, the URANS predicts the influence of the tip leakage flow more concentrated in pitch-wise direction than measured. That results in a time mean axial velocity deficit, which is significantly over-predicted inside of the tip leakage flow and will be quantified in the following paragraph.

A quantification of the **blockage** induced by the tip leakage flow will be given for the two surfaces that are colored in Figure 3.25 for the position 3 at 96.5% span-height. A simplified velocity profile ( $V_{x,simpl}$ ) is drawn with dashed lines for the URANS (blue) and LDA (green) results. The simplified velocity profile and blockage are defined as listed hereinafter:

- If the axial velocity close to the suction side is lower than the one at the pressure side, an axial velocity gradient is imposed for capturing the blockage on the suction side. The velocity gradient at the inlet of the rotor at 90% span-height, without influence of the tip leakage flow, serves as a reference.
- The simplified velocity profile is drawn between the maximum axial velocity close to the pressure and suction side.
- Blockage shall be defined as axial velocities below the one of the simplified velocity profile.
- A negative blockage (acceleration of the fluid) is not taken into account.

The tip blockage is estimated per span-height as defined by Equation 3.13.

$$blockage_{tip} = \int_0^1 \frac{V_{x,simpl} - V_x}{V_{x,simpl}} \cdot \frac{ds}{dS} \quad (3.13)$$

where  $ds$  represents a fraction (distance between two data points) of a rotor passage, and  $dS$  the passage size (in time or space). At the position 3 at 96.5% span-height, the URANS results give an over-prediction of the blockage by 0.11 fractions of a rotor passage.

Following this guideline, the blockage in the tip region of the rotor 2 is presented for four investigated axial positions in Figure 3.26. Noise in the blockage distribution estimated with the LDA data can be explained by the measurement resolution close to the blade passing, and the simplifications that are necessary for feasibility reasons of this blockage estimation.

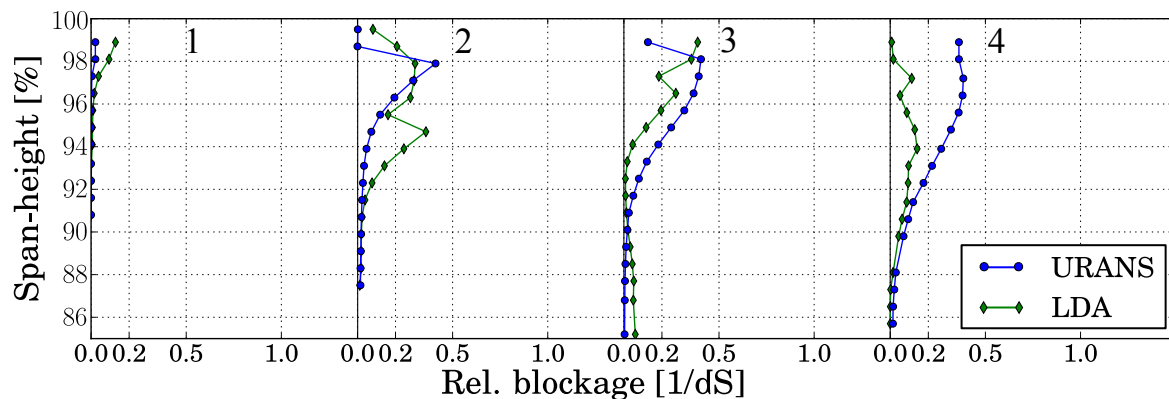


Figure 3.26 Blockage in tip region of the rotor 2 at four axial positions estimated with Equation 3.13

At the peak efficiency operating point, the axial position 1 shows the blockage induced by the TCV in upstream direction above 97% span-height. The URANS results allow capturing how the radial extend of the tip leakage flow influence zone is growing from one axial position to another in downstream direction.

- A blockage peak is found at the position 2 at about 98% span-height in both databases. LDA captures a higher blockage below this peak. That means the measured tip leakage flow dives axially sooner into the vane than predicted but the radial extend reaches the same level from position 3 on.
- Globally at the positions 3 and 4, the URANS over-predicts the blockage by up to 20%. Here the mixing-out of the tip clearance vortex decreases the maximum blockage in the LDA results. The blockage drops from values above 50% to below 20%. This effect is weaker in the URANS predictions, the maximum blockage decreases by only 5% from position 3 to 4, which signals a problem in the prediction of the mixing-out in the RANS simulations.

**In the rotor 3**, the focus of the analysis is put on the difference in the axial velocity deficit induced by the tip leakage flow. As already shown for the rotor 2 and thus not shown here: Outside of the zone influenced by the tip leakage flow, the URANS results capture well the velocity gradient in the rotor passage and compare well to the LDA measurement.

The axial velocity fluctuations for the two span-heights at the axial position 2: 91.8%, and 96.9% are presented in Figure 3.27. The LDA results are presented over time and the URANS results over space, as already explained for Figure 3.24. Additionally, all registered particles are plotted with grey dots. They appear as vertical bars because of the distribution over a certain velocity range in each ensemble average window.

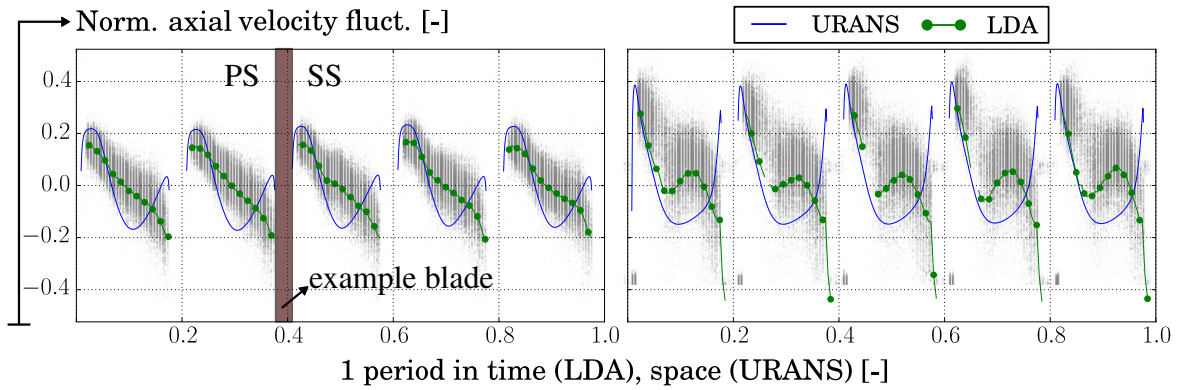


Figure 3.27 Axial velocity fluctuations 91.8% (middle) and 96.9% span-height at axial measurement position 2 in the rotor 3 at the peak efficiency operating point

- 91.8%: The measurements only show a weak axial velocity deficit linked to the tip leakage flow. It is found as a small step close to mid-passage. The URANS shows a tip leakage flow that is inducing a large concentrated axial velocity deficit, which is found as substantial negative peak below the LDA measurement points. This resembles the mis-predictions already presented for the rotor 2. For one part this is caused by a circumferential shift of the tip clearance vortex trajectory towards mid-passage, and for another part by an over-prediction of the blockage induced by the tip leakage flow.
- 96.9%: The complexity and high unsteadiness of the flow field is found as the large band of grey measurement points. The URANS predictions are outside of the main part of the grey measurement points. And URANS predicts therefore no possible mean state of the measured flow field. The axial velocity deficit close to the suction side is well matched. This was also found as good match of the TCV trajectories in Figure 3.21. Though, URANS over-predicts the blockage over the remaining passage width. That is found to lead to an over-turning of the flow and consequently to an over-prediction of the circumferential velocity, and hence over-prediction of the work in the rotors 2 and 3. This explains the over-prediction of the total pressure rise from 80% span-height to the casing in the rotors 2 and 3 as shown earlier in Figure 3.7.

The (U)RANS simulations are considered to have the same trouble in predicting the tip flow field of the rotor 2 and also rotor 3. An over-prediction of the blockage is noted. Furthermore the mixing out of the tip leakage flow might be badly predicted. In consequence an over-prediction of the tangential velocity component is found (higher work output). In the following Section, possible sources for this mis-prediction are discussed. Unfortunately an accurate tip gap leakage mass flow rate comparison is not possible with the available measurement data.

### 3.3.2 Discussion of the mis-prediction of the tip flow field

The tip gap sizes have been verified and are stated in an internal report. The average measured tip gap size in rotation corresponds well to the realized tip gap size in the simulation domain. There are of course variations from blade to blade within the manufacturing precision. That is valid for all three rotor blade rows. Having a fitting mean tip gap size, it is not considered to have an influence on the comparison between the experimentally obtained and simulated flow field at the peak efficiency operating point.

A comparison of the measured and predicted incidence angle to the rotors is not simple because of possible interactions between the pneumatic probes and the flow field close to the endwalls, or a challenge of measuring the mean velocity component for the LDA downstream the stators (discussed in Section 3.2.2). A comparison between PNEUM and URANS (not shown in detail, but the flow angle was presented in Figure 3.9) reveals possible differences of the incidence angle in the order of  $2^\circ$  to the rotor 2 and 3 at 95% span-height (highest measured span-height) at the peak efficiency operating point. This is not believed to be the driving factor for the mis-predictions of the tip flow field. Of course, the incidence angle difference leads to an additional source of error that can worsen the mis-prediction.

Two promising ways of improving the tip flow field prediction are the use of a more advanced turbulence model and the application of isothermal boundary conditions. Their actual impact on the flow field is discussed in the two Sections hereinafter respectively.

#### Changing the turbulence model for over-coming tip flow field mis-prediction

Steady RANS simulations were conducted for a comparison of the flow field obtained with the Wallin-Johansson EARSIM (advanced, introduced in Section 2.3.1) and the  $k - \omega$  Wilcox (standard) turbulence model. The radial profiles of the total pressure downstream the rotor 2 (left) and 3 (right) are presented in Figure 3.28 for the two RANS fields and the PNEUM measurements. The total pressure is presented as normalized absolute difference to the mean total pressure obtained with the "RANS  $k - \omega$ ". The global view allows a good judgment about the local minor differences that are found in the two RANS flow fields. Minor differences are found for example in the size of separations or the tip leakage vortex trajectories. Their description is of no gain for this thesis work.

The radial total pressure profiles show for the two rotors a clear over-prediction of the total pressure (rise) relative to the PNEUM measurement. Apart from the previously reported general offset, the most disturbing difference lies in the 20% of span towards the casing and appears as a somewhat step in the radial distributions. This over-prediction is minimally corrected with the EARSIM turbulence model but that correction is by far smaller than the difference to the PNEUM results.

Clearly the EARSIM turbulence model is not a way for substantially improving the flow field prediction with RANS simulations in CREATE. Further (costly) tests would require unsteady RANS simulations, which are not possible in the combination of the sliding mesh and the EARSIM turbulence model in the current version of elsA.

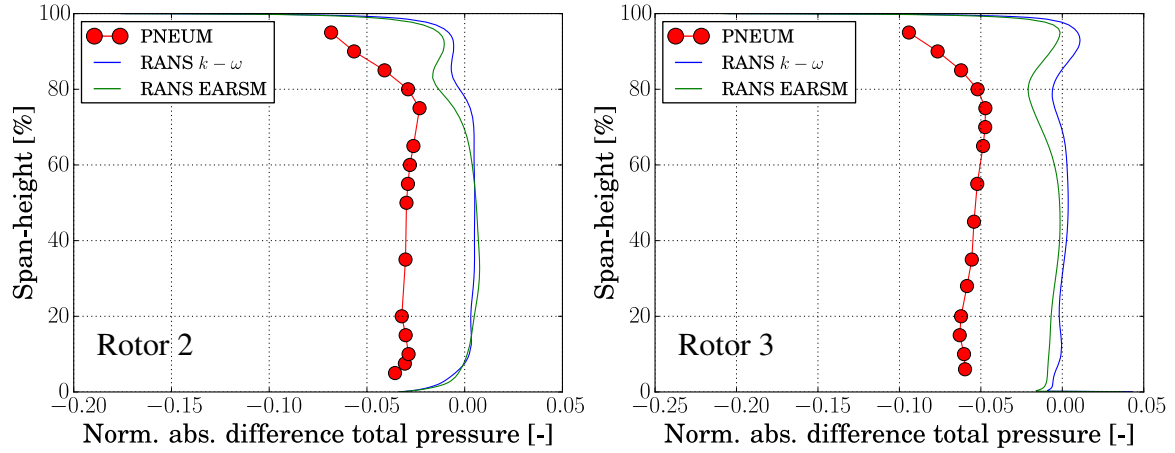


Figure 3.28 Influence of EARSM turbulence model visualized by radial total pressure distributions downstream the rotors 2 (left) and 3 (right) for the RANS results and PNEUM data, given as normalized absolute difference to average "RANS  $k-\omega$ " value

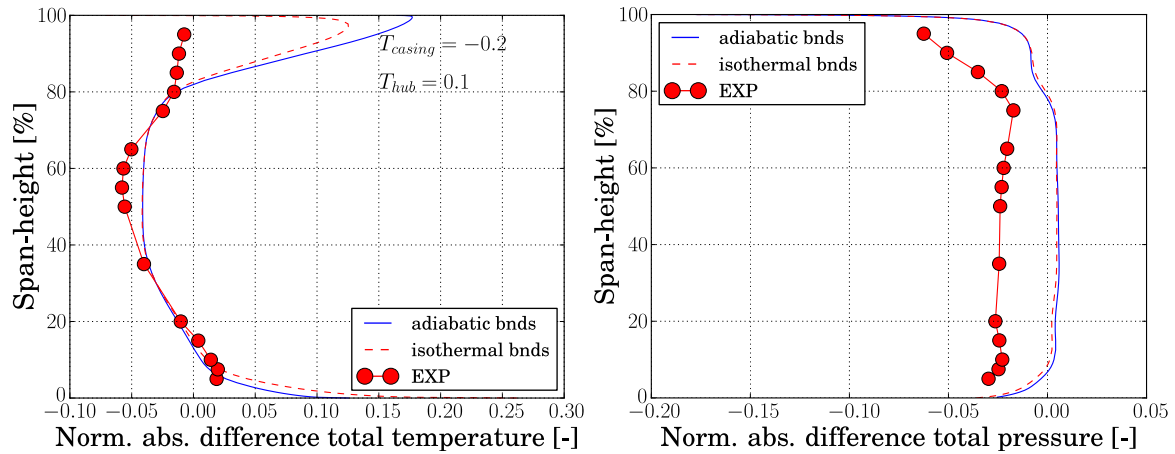


Figure 3.29 Influence of isothermal boundary condition on total temperature (left) and pressure (right) at 27A, in comparison to PNEUM data, given as normalized absolute difference to average "adiabatic bnds" value

### Effect of isothermal boundary conditions on the rotor tip flow field

Empirically estimated temperature profiles are imposed at the hub and casing (as introduced in Section 2.3.4) for all the (U)RANS simulations in this work. Their influence shall be discussed with the total temperature and pressure profiles downstream of the rotor 2 in Figure 3.29. The imposed temperature are stated as well. The total temperature is presented as normalized absolute difference to the mean value of the data for "adiabatic bnds".

The isothermal boundary conditions allow a decrease of the total temperature overshoot in the 20% of span towards the casing. The overshoot decreases by 0.05 normalized value (relative to the overall total temperature rise in the compressor) at maximum which



corresponds to 40% of the maximum difference between PNEUM and RANS adiabatic. At the hub the influence is smaller because of an imposed hub temperature close to the one in the flow field. The isothermal boundary conditions are not found to influence the flow field at mid-span at all. The total pressure distributions with adiabatic and isothermal boundary conditions almost overlap. At this scale the influence of the isothermal boundary conditions is negligible on the flow field.

Despite the small influence on the actual flow field characteristic, the application of the isothermal boundary conditions is not costly and gives a better estimation of the total temperature rise in the compressor. Therefore this type of boundary condition is applied for all the (U)RANS simulations in this thesis work.

### 3.3.3 Hub-corner separation in the first rotor

The first rotor is hub-loaded, which can cause a hub-corner separation consequently. In the current version of the compressor, the hub-endwall and blading was therefore adapted to handle this issue. More details on the design are confidential. The surface friction lines for the old (conventional) and current design of the hub-endwall are presented in Figure 3.30 for the rotor 1. The results are based on steady RANS simulations, where the relative Mach number ( $Ma_{rel}$ ) is shown for a blade-to-blade cut at 5% span-height.

Both designs give a closed corner separation on the blade suction surface at both operating points. The corner separation is evidently growing from nominal to loaded valve operating point. The old design causes a region with reversed flow on the blade surface. According to the extensive work of Taylor and Miller (2015), this type of closed corner separation with partly reversed flow can abruptly change its topology to an open separation when increasing the incidence sufficiently. The open corner separation is considered as blade failure with important losses.

The new design causes a stronger transverse surface pressure gradient, similar to what is shown by Taylor and Miller (2015) in a comparison between two different lean angle designs of a stator blade (see Figure 3.31). The inner part of the low momentum boundary layer flow is over-turned towards the mid-span because of the increased transverse pressure gradient. A possible open-corner separation is avoided with this design but higher wake mixing out losses are possible at mid-span because of the migration of the low momentum fluid. Recent work by Choi et al. (2008) shows that a hub-corner separation in a rotor can interact with the tip clearance vortex, and provoke the rotating stall inception. Therefore it is important to ensure that the separation in the rotor 1 in CREATE does not grow too importantly in size from the peak efficiency to the near surge operating point. However, sometimes the corner separation can induce a blockage that helps the tip flow field to keep stable. In the following paragraph the validity of the simulations is discussed with experimental results.

The radial profiles of the entropy fluctuations around the mean value are obtained for each database (RANS, URANS, PNEUM) at the plane 26A downstream the rotor 1 and presented in Figure 3.32a for the peak efficiency operating point. On the second horizontal axis, the arithmetically time and space averaged standard deviation on the absolute velocity (LDA) is traced. The standard deviation is computed on each ensemble average window,

which are containing between 100 and 3000 particles each (wakes are less resolved than the free stream).

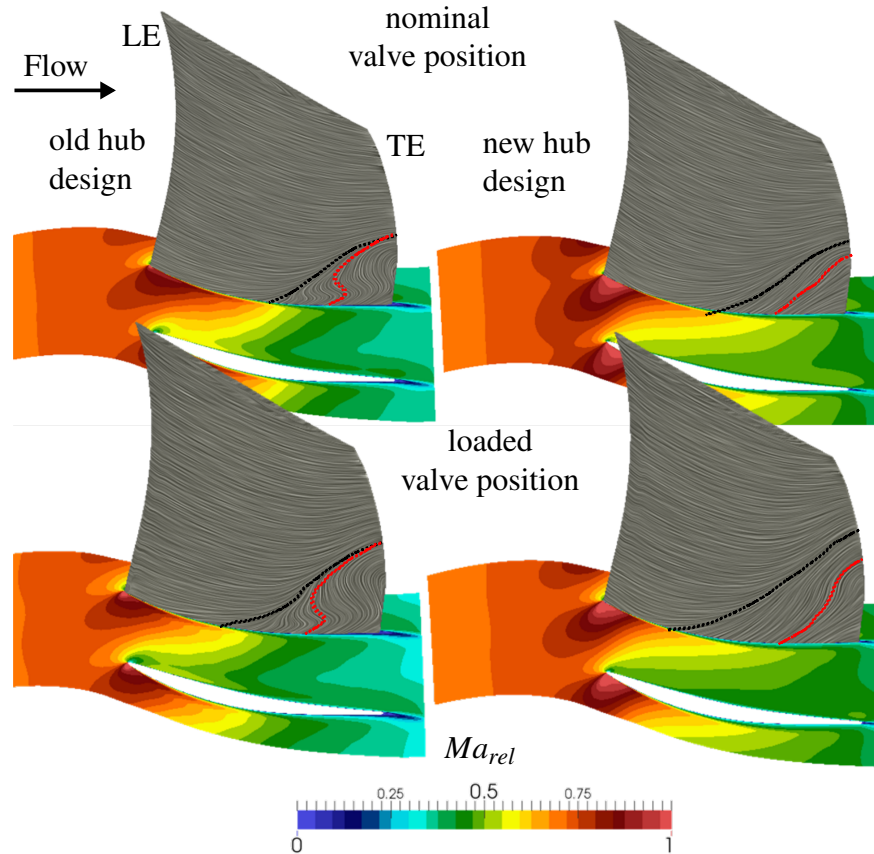


Figure 3.30 Surface friction lines on the rotor 1 blade for the nominal (top) and loaded (bottom) operating point, for the old hub design (left) and the new hub design (right); Relative Mach number at 5% span-height

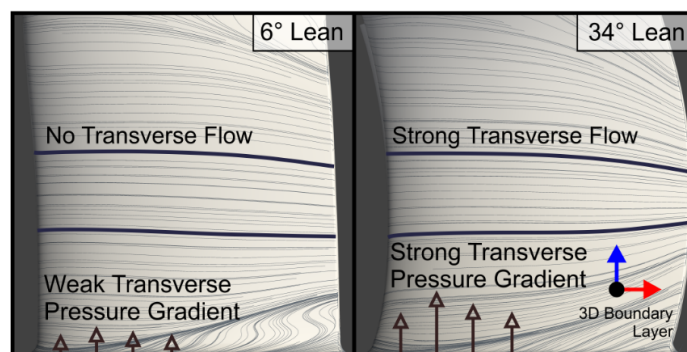


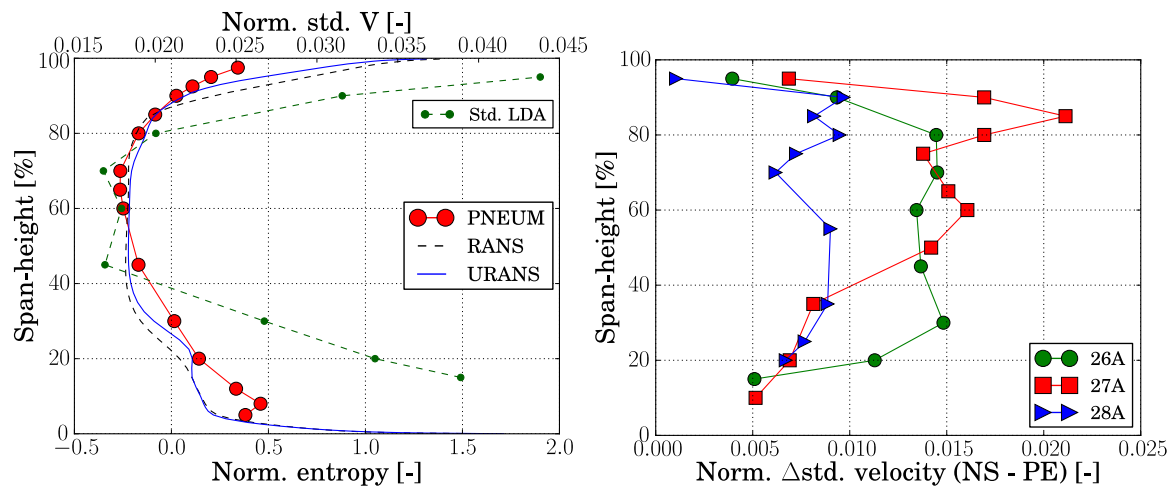
Figure 3.31 Influence of transverse pressure gradient on streamline curvature from Taylor and Miller (2015)

As previously shown, the losses (entropy rise) are over-estimated in the tip region. An elevated unsteadiness (increased standard deviation) is found here as well. Below 60% span-height, the entropy rise is under-predicted by RANS and URANS. This part of the span can be divided in a zone 1 between 20% and 60% span-height and a zone 2 below 20% span-height to the hub. Over both zones, the standard deviation is increased compared to the one at mid-span.

- The difference in the zone 1 originates from the incorrect simulation of the mixing out of the hot-air injection upstream of the IGV. The radial outwards migration of the highly turbulent injection is not well captured with the simulations.
- In the zone 2, the losses induced by the hub-corner separation in the rotor 1 are under-estimated in the simulations.
- The difference between RANS and URANS at 20% span-height can be explained by the presence of a mixing plane upstream of the rotor 1 with the RANS simulations.

The difference between the standard deviation of the velocity (LDA) at near surge and peak efficiency operating points highlights the zones of important increase of unsteadiness in the rotors, presented in Figure 3.32b. An increasing standard deviation is expected from peak efficiency to near surge operating point because of an increasing unsteadiness related to the tip leakage flows.

- The maximum increase of unsteadiness is found downstream of the rotor 2, where the tip leakage grows noticeably more than in the other rotors between the two operating



(a) Entropy fluctuations around respective mean value of the plane for PNEUM, RANS, and URANS at 26A downstream the rotor 1 at peak efficiency operating point, and standard deviation of velocity (V, LDA)

(b) Standard deviation of velocity (V) relative to respective overall mean axial velocity, given as difference between near surge and peak efficiency operating point, based on LDA

Figure 3.32 Characterization of unsteadiness in the rotor 1

points. The increase of the standard deviation is three times higher at the tip than at the hub here.

- The standard deviation increases globally the least in the rotor 3.
- The rotors 1 and 3 show a fairly evenly distributed increase of the standard deviation over the whole span, with decreasing difference between the operating points towards the endwalls. Thus the unsteadiness induced by the tip leakage flow does not increase more than the unsteadiness related to the rotor wakes (and all other unsteady flow structures). Likewise the unsteadiness induced by the separation in the hub-corner of the rotor 1 does not increase more than the one found at mid-span.
- The flow field is expected to be more unsteady in the rear stages than the front stages because of the RSI. The major contribution to the unsteadiness though comes from the direct contributions of the row just upstream, compared to the mean flow. The endwall design in the rotor 1, which causes the over-turning of the low momentum boundary layer flow of the hub as previously discussed with Figure 3.30 and the high loading, may explain the important velocity deficit in the rotor wakes that generates high temporal fluctuations.

### 3.4 Identification of separations in the stators

This Section shall show the capability of the numerical and experimental methods to capture possible separations on the blade suction surfaces of the stators 1 to 3 in CREATE. That will give an idea about the reliability of the RANS simulation during the design of the stator blades.

The Lieblein diffusion factor (see Equation 1.6) is calculated for the identification of high loaded zones in the stators. According to Lieblein, blade boundary layers are likely to have separations with a diffusion factor greater than 0.6. The diffusion factor should be calculated with flow field data at the nominal operating point of a blade with a 2D flow but here only the flow field data at nominal compressor operating point is available. Consequently the diffusion factor in the hub and casing boundary layer flow is not trustworthy. The equivalent diffusion factor (see Equation 1.7) is found to give the same tendencies for the present configuration, and is thus not presented here. The Lieblein diffusion factor is presented in Figure 3.33 for the stators 1, 2, and 3 based on LDA and URANS data. The factor is given relative to the mean URANS value for each respective stator because of confidentiality. The PNEUM data is not considered here because of the problem with the static pressure measurement reported in Section 3.2.1. The LDA data globally under-estimates the diffusion factor because of the time mean over-estimation of the axial velocity downstream the stators (Section 3.2.2).

Given the nature of Equation 1.6, a separated suction side blade boundary layer increases the diffusion factor. More specifically, an increased  $V_{\Theta,out}$  (separation) causes a high diffusion factor. In other words, an increased diffusion factor can also be the consequence of the

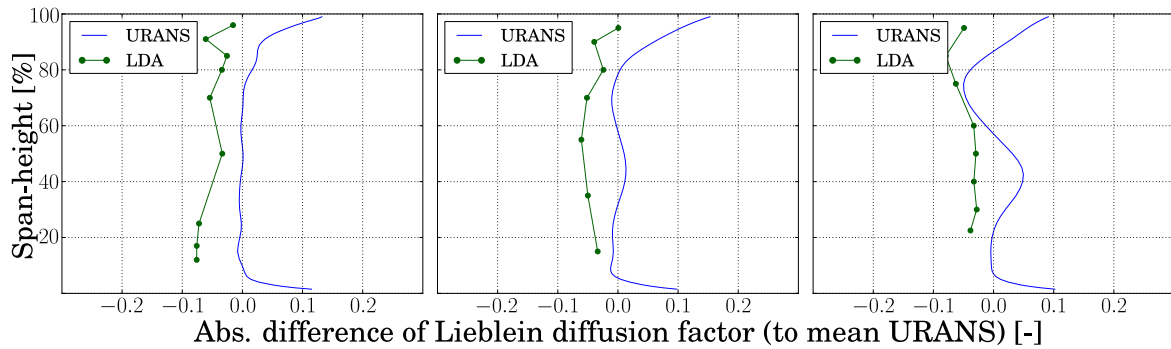


Figure 3.33 Lieblein diffusion factor for the stators 1, 2, and 3 based on LDA (green), and URANS (blue). Given as absolute difference to mean URANS per stator (Globally under-estimation noted by LDA because of axial velocity measurement problem downstream the stators)

incoming flow field to a stator, which might not be adapted to the given incidence angle resulting of the multistage flow field.

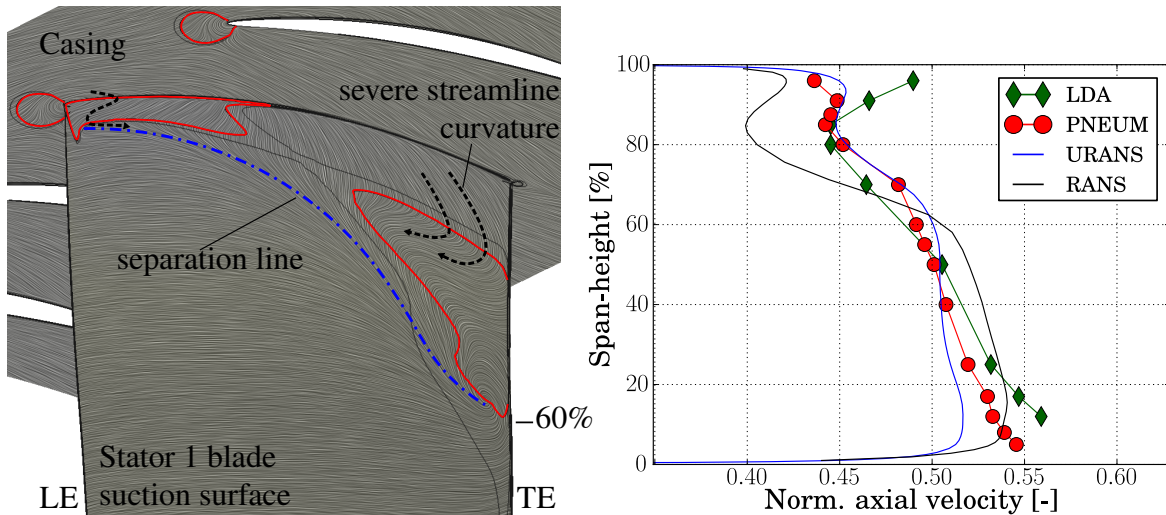
The information obtained from the Lieblein diffusion factor evolutions can be summarized as follows:

- In this high-speed compressor, every absolute difference of the Lieblein diffusion factor above 0 indicates zones of likely separated blade boundary layers. The span-wise extend of these zones increases in streamwise direction.
- In the stator 1, a small bump is found in the URANS results between 80% and 90% span-height. Here a small corner separation is present. Its representation in the different databases will be discussed in the following Section.
- In the stator 2, the diffusion factor is equally high over the whole span, with a small bump at mid-span in the URANS results.  
This stator is known to separate progressively on the blade suction surface over the whole span at the loaded and near surge operating point. This type of separation is obtained thanks to the lean angle of the blade. The lean design can help to modify the low momentum flow accumulation away from the corner between the blade and the endwalls towards the main span, as already discussed with the results of Taylor and Miller (2015). The progressive separation is well captured with the URANS and LDA data and not further discussed in this thesis work.
- In the stator 3, the diffusion factor is increasing significantly at mid-span in the URANS and LDA results. Here, the URANS predicts a stronger increase than LDA relative to the mean URANS value. In this zone, a separated suction blade boundary layer is present and will be discussed in Section 3.4.2.

### 3.4.1 Casing corner separation in the stator 1

A closed casing corner separation is identified on the stator 1 blade suction surface at peak efficiency, loaded, and near surge operating points in the experimental and numerical results. First, the 3D structure will be presented with the help of surface friction lines. Then, the size and characteristics will be compared between the databases going from 1D to 2D views.

The time mean surface friction lines (see introduction in Section 1.2.3) are presented for one example stator 1 blade at the near surge operating point in Figure 3.34a. This operating point is chosen to show the most severe (simulated) separation. The top of the Figure represents the casing, and one stator 1 blade is shown down to about 30% span-height with a view on the suction surface. The separation line is indicated with a manually drawn blue dashed line marking the separated zone on the blade suction surface. Additionally,  $V_x = 0 \frac{m}{s}$  and  $V_x = 0.5 \frac{m}{s}$  are given respectively as red and grey contours on the blade and casing surface.



(a) Time averaged surface friction lines on the stator 1 blade and casing at the near surge operating point based on URANS results, red:  $V_x = 0 \frac{m}{s}$  (b) Radial profiles of axial velocity for LDA, PNEUM, URANS, and RANS at the plane 270 at the near surge / loaded (PNEUM) operating point

Figure 3.34 Characteristics of the separation in the stator 1

The zone of an axial velocity close to zero or smaller than zero extends diagonally over the blade suction surface. It is lifted away from the trailing edge blade-casing corner thanks to the blade design. The low momentum fluid from the casing boundary layer is over-turned from the casing towards lower span-heights. The severe streamline curvature is highlighted with black dashed arrows in the separated zones. Fully axial reversed boundary layer flow is found between about 70% and 90% span-height at the trailing edge at the near surge operating point.

Radial profiles of the axial velocity are given in Figure 3.34b for LDA, PNEUM, URANS and RANS results from plane 270 at the near surge operating point. Note that the PNEUM measurements are conducted at the loaded and not near surge operating point. The radial



profiles can be directly compared to the previously introduced surface friction lines. All databases show an axial velocity deficit of at least 10% at 85% span-height compared to the mid-span. That matches the zone with fully reversed surface friction lines.

RANS is over-estimating the axial velocity deficit by a factor of about 2 compared to the other databases. The circumferential mixing out of the rotor influence (wake and tip leakage flow) at the mixing plane interface has a deteriorating effect on the separation in the stator. Having a matching effect in all databases means that the PNEUM technique tends to under-estimate the effect of the corner separation.

The last two LDA measurement points close to the casing show a re-acceleration of the flow to a mid-span level, which is not found in the other databases. That might be related to the previously discussed challenge for the LDA to measure the good time mean velocity downstream a stator (see Section 3.2.2).

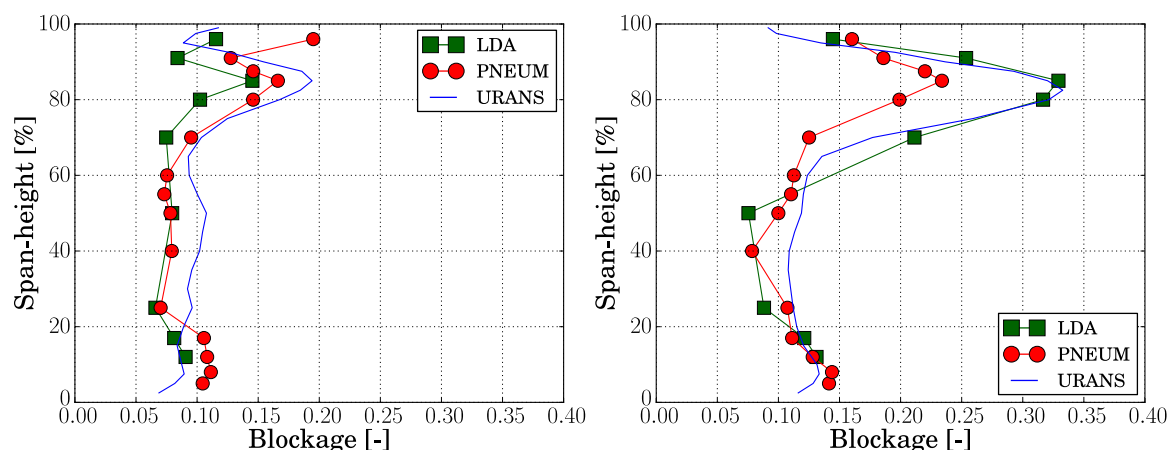


Figure 3.35 Blockage relative to stator 1 pitch based on EXP (red), LDA (green), URANS (blue), for the peak efficiency (left), and for the near surge (PNEUM: loaded) OP (right)

The blockage induced by the corner separation shall be quantified with an integration of the related axial velocity deficit using the previously introduced Equation 3.12. For simplicity reasons the equation takes also into account the wakes and other velocity deficit inducing flow phenomena. The blockage is presented for the peak efficiency (left) and the near surge operating point (right) in Figure 3.35 for LDA, PNEUM and URANS data. Note that PNEUM presents data from the loaded operating point. The URANS blockage is computed with the same spatial resolution as the experimental one.

The wakes induce a blockage of 6% to 10% and 9% to 14% at peak efficiency and near surge operating point respectively below the zone influenced by the corner separation. Globally the blockage increases with decreasing mass flow rate. The corner separation is found as peak at about 85% span-height at both operating points. The separation almost doubles in maximum blockage with decreasing mass flow rate and increases its span-wise extension by about 20% span. At maximum, a blockage of about 34% of the stator pitch is found at the near surge operating point. Here, PNEUM shows a lower blockage because it is calculated with data of the loaded and not near surge operating point (not available).

### 2D shape of the stator 1 corner separation

The 2D shape of the corner separation is presented with the help of cuts at a constant axial position corresponding to the plane 270 (downstream the stator 1) in Figure 3.36. The time mean axial velocity flow field is presented with grey color maps for the URANS results, and superimposed is the time mean standard deviation (computed based on the particles in the ensemble average windows) with blue-to-red contours. The standard deviation helps to identify the unsteadiness linked to the corner separation and can highlight the shear layer between the separation and the free stream flow. Here the color scales for the standard deviation are matched for both operating points. Only 3.5 stator wakes (half a spatial period) are presented here because they give all the necessary information for this comparison.

The stator wakes appear as dark vertical stripes due to the induced axial velocity deficit. Differences between the passages appear mainly due to an interaction between the IGV and the stator 1. Globally the standard deviation (unsteadiness) induced by the wakes is lower than the one linked to the casing corner separation. A small zone of unsteadiness is found in the hub corner at the suction side of the wakes. This small separation is not growing by much with decreasing mass flow rate. Measurement constraints do not allow a measurement closer to the hub which would give access to a more detailed analysis of a possible small hub corner separation.

At the peak efficiency operating point, the casing corner separation is hardly visible in the URANS results. Here the blockage was shown to differ by only 5 to 7% between mid-span and separated zone. The LDA highlights well the shear layer between the separation and the main flow, which is found as high gradient of axial velocity in the URANS map. The highest measured span-height does not indicate a decreasing standard deviation close to the

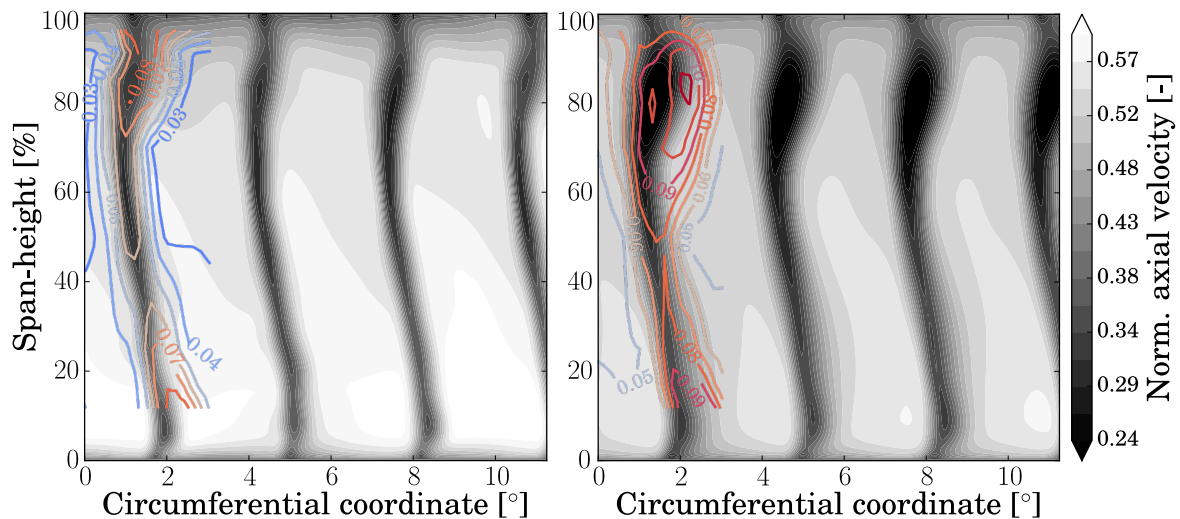


Figure 3.36 Axial velocity field in color map (URANS), and standard deviation of velocity in contours (LDA, blue-to-red) at the plane 270 (downstream the stator 1) for the peak efficiency (left) and near surge (right) operating point



casing, signifying that the corner separation is not lifted of (or less than 5% of span) the corner between the blade and casing.

At the near surge operating point, the lower edge of the highly unsteady shear layer is found almost at mid-passage in the LDA results. A radially and circumferentially larger separation is found by the LDA than predicted (agrees with the small difference of 3% in blockage at 85% span-height seen in Figure 3.35). A second spot of increased standard deviation appears in-between the blade suction surface and the shear layer. That is typical for the highly intermittent behavior of a corner separation. The standard deviation contours capture well the structure of the corner separation, which does not extend up to the casing downstream the stator as also previously shown in Figure 3.34a.

### Modulation of the separation by rotor wake passing

The stator 1 flow field is used as an example for highlighting the importance of the flow field modulation by an upstream rotor. The Figure 3.37a presents an instantaneous blade-to-blade cut at 85% span-height showing the entropy flow field from the stator 1 to the rotor 2 based on URANS results at the near surge operating point. The rotor turning direction is from the left to the right. It is the span-height with the maximal size of the corner separation. The color red highlights high entropy production. The incoming rotor wakes are seen as diagonal stripes of increased entropy (losses) in this representation. Those wake segments modulate the stator wake in the numerical results, but not significantly the size of the corner separation. Here the corner separation causes the most part of the entropy production. The modulation pushes a part of the stator wake in rotor turning direction because of the negative jet effect (see Section 1.2.1).

In a next step the modulation shall be quantified and is estimated at each of the circumferential evenly spaced LDA measurement points at a given span-height. The amplitude of

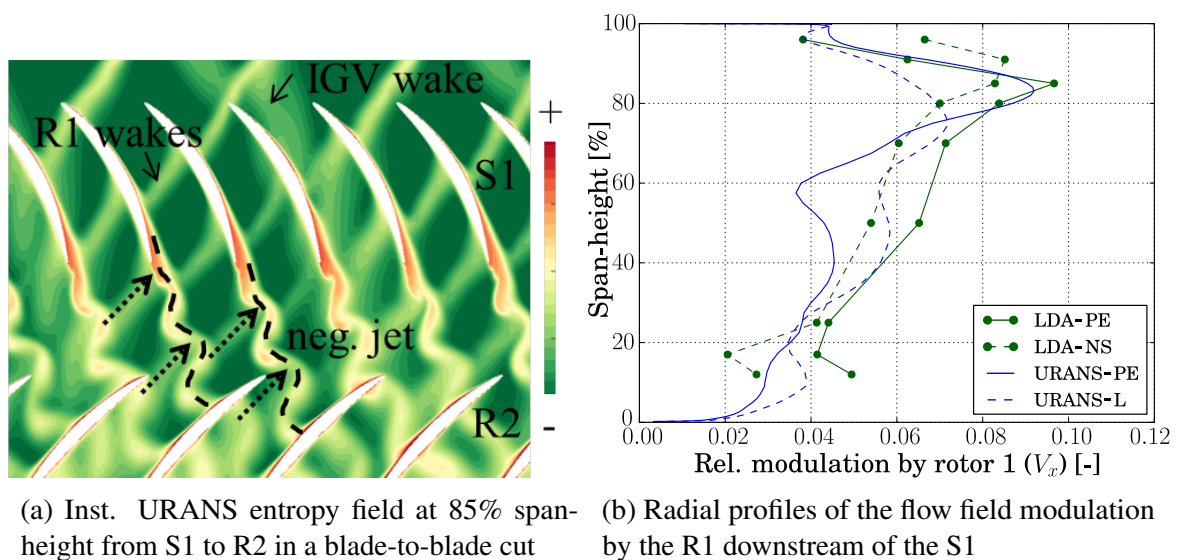


Figure 3.37 Stator 1 (S1) flow field modulation by the rotor 1 (R1)

the modulation is computed with temporal Fourier transforms and then averaged over all measurement points of a span-height for the peak efficiency and near surge operating points. Here the first three harmonics of the rotor 1 influence (4 blades) are taken into account. The Figure 3.37b gives radial profiles of the stator 1 flow field modulation by the rotor 1 as an amplitude relative to the mean axial velocity of each database at the given operating point.

Globally the modulation is the strongest at the location of the corner separation, which superposes also with radial span-heights strongly influenced by the rotor 1 tip leakage flow. The stator wake is subject to the strongest oscillations here. A good fit is found between the databases at the peak efficiency operating point.

The maximum relative amplitude decreases from peak efficiency to near surge operating point. In the case of the LDA, the two operating points give two well-fitting evolutions of the relative amplitude (taking into account the measurement uncertainty). Thus the relative amplitude does not change with the operating points. No sudden rise of importance of the separation oscillation is found close to surge, which is positive for the operation of the compressor. An open corner separation is not forming at any instant of the URANS simulation (not shown in detail here).

URANS finds a stronger modulation below 70% span-height at the near surge than peak efficiency operating point. The point of the maximum amplitude decreases by 10% span-height, which corresponds well to the descending of the separation but also to the diving of the tip leakage flow between the two investigated operating points.

Figure 3.38 shows the time mean axial velocity based on LDA, URANS, and PNEUM results at 85% downstream of the stator 1 (270) for the peak efficiency (left) and the near surge operating point (right). Additionally, the bottom of the Figure shows the rotor 1 flow field modulation at each measurement point of the presented span-height. The modulation amplitude is estimated with temporal Fourier transforms as root square of the energy of the first three harmonics of the rotor 1 (4 blades). The suction side is found on the right hand side of the wakes.

It is evident that even though the mean amplitude of the modulation (Figure 3.37b) and the blockage (Figure 3.35) were shown to fit well (differences between 1 to 2%), the wake characteristics appear different between the databases in Figure 3.38.

- At the peak efficiency point, URANS predicts a deeper but slimmer wake than the LDA. That can be explained by the rotor modulation:  
LDA captures a more important oscillation of the wake at the suction side than at the pressure side in comparison to URANS. Even though in average the modulation equals to the predictions of the URANS (see also Figure 3.37b), it results in a wider but less deep time mean stator wake than predicted.
- Trusting the LDA measurements, the PNEUM measurement shows a questionable fit with the URANS predictions at the peak efficiency point. The pneumatic probe appears to have trouble capturing the good (LDA) time mean velocity inside of the stator wakes because of the oscillation induced by the rotor 1.
- The modulations are of lower amplitude at the near surge than the peak efficiency operating point. The corner separation grows in size and the local influence of the rotor

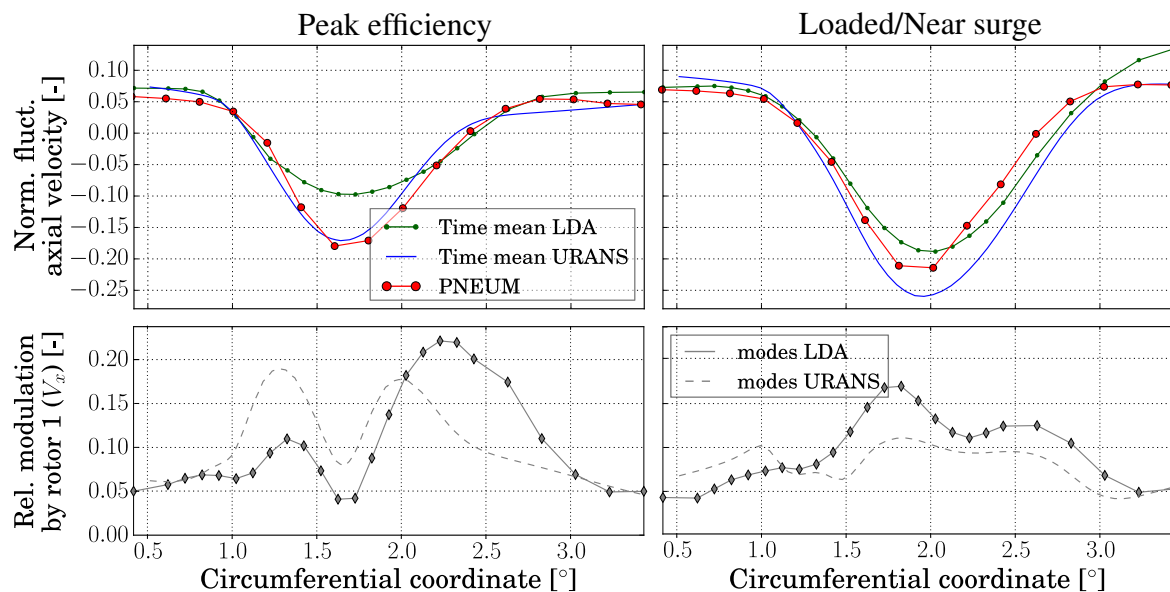


Figure 3.38 Time mean axial velocity at 85% span-height downstream at 270 based on URANS, LDA, and PNEUM for the peak efficiency (left) and near surge (PNEUM: loaded) operating point. Rel. (to mean velocity) amplitude of rotor 1 harmonic on 2nd vertical axis

modulation decreases. That corresponds to results of the work by Schulz et al. (1989), which shows that the incoming wakes do not penetrate into the separated region.

URANS and LDA do not show a good fit in wake width or in the wake depth. Again this can be explained by the under-predicted wake oscillations.

### 3.4.2 Separation at mid-span in the stator 3

A separated suction side stator 3 blade boundary layer is found at all operating points in all databases. This Section will discuss the over-prediction of the size of this separation by (U)RANS method at the peak efficiency operating point.

The surface friction lines for one example stator 3 blade are shown in Figure 3.39a. The lean angle of the blade deviates the low momentum flow from the hub and casing corner towards mid-span. Here a separation of the suction side boundary layer is found. It extends radially from 20% to 60% span-height at the peak efficiency operating point. Stable foci are found at the lower and upper end towards the trailing edge (see also Figure 1.9), with friction lines spiraling into it.

The inlet flow field conditions shall be clarified with the exemplary velocity triangle for the peak efficiency flow field at 60% span-height at 28A in Figure 3.39b. PNEUM and the other databases do not show a match because of the PNEUM problems noted in Section 3.2.1. The incidence angle differences between URANS and LDA are in the order of 0.7 to 1.2° at the separated span-heights. These differences are considered non-significant.

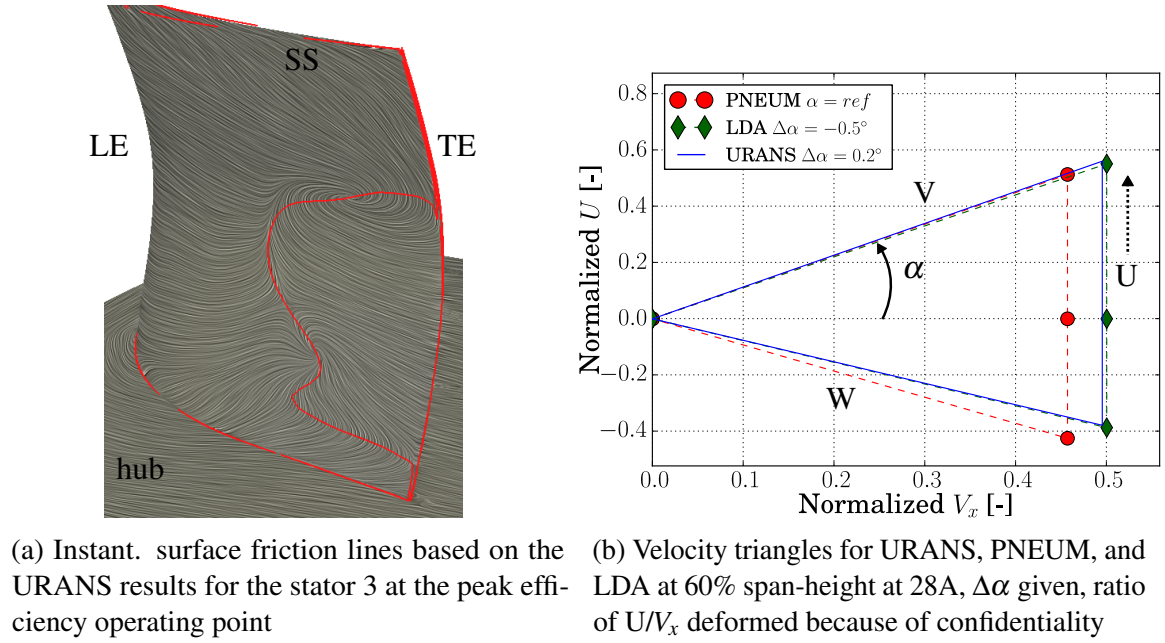


Figure 3.39 Characteristics of the separation in the stator 3

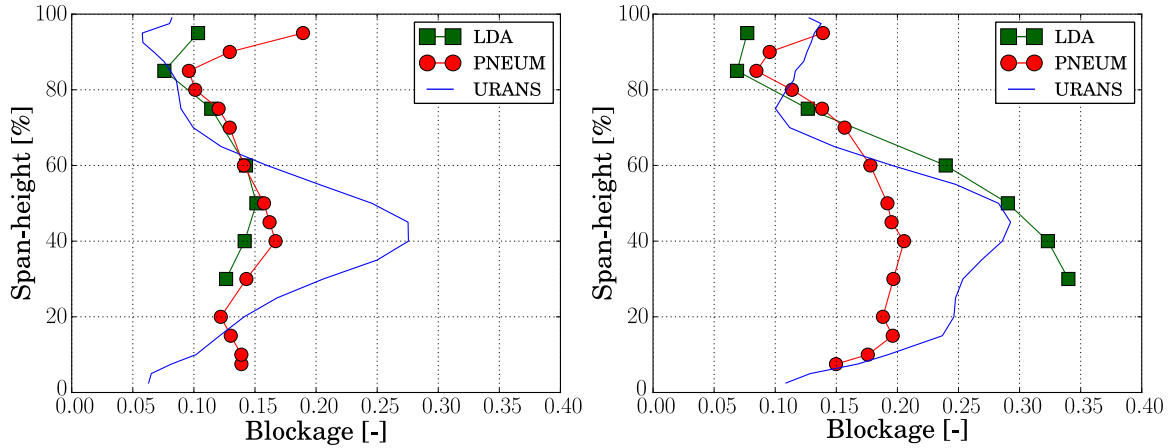


Figure 3.40 Blockage relative to the stator 3 pitch based on EXP (red), LDA (green), URANS (blue), for the peak efficiency (left), and for the near surge operating point (right), PNEUM: loaded instead of near surge operating point

The size of the separation is quantified with the blockage factor defined earlier with Equation 3.12 and presented in Figure 3.40 for the peak efficiency (left) and near surge operating point (right). Globally, the URANS is over-predicting the separation by at maximum 10% of pitch at the peak efficiency operating point but gives a good prediction at the near surge operating point. PNEUM might mainly under-estimate the separation at the near surge operating point because actually the loaded operating point was measured. Though, the PNEUM technique might also have trouble to capture well the time mean velocity due to the

oscillation of the separated boundary layer. The pneumatic tubes connected to the sensors average the pressure fluctuations induced by the modulation of the separation and minimize peak like fluctuations. Nevertheless, PNEUM also detects a growth of the separation with decreasing mass flow rate. At maximum the separation takes 29% (URANS) to 37% (LDA) of the pitch at the near surge operating point.

Judged by the little difference in inlet flow angle, the over-prediction of the separation is believed to be caused by troubles of the RANS method related to the turbulence model. It is known from literature that RANS tends to over-estimate the size of corner separation but not necessarily the losses related to it, as demonstrated by Zambonini and Ottavy (2015). The total pressure deficit across the stator 3 is calculated relative to the inlet total pressure for each respective database (PNEUM, URANS) and presented in Figure 3.41 as difference to the average total pressure losses determined with PNEUM for confidentiality. The inlet total pressure is chosen as reference and not the dynamic inlet pressure because of the troubles of the correct estimation of the static pressure upstream a stator by the PNEUM probes (Section 3.2.1). Globally the total pressure loss is under-predicted by the URANS. This is not a systematic error, which can be seen in the earlier presented Figure 3.7. The evolutions above 80% do not represent the real losses between inlet and outlet because of radial inward gradients in the flow field due to the tip leakage flow. The separated boundary layer causes an increased total pressure loss with a maximum at about 45% span-height in the URANS results. Though the predicted total pressure loss is never higher than the measured one. That relates to the literature findings about RANS simulations.

Observing the separation already at the peak efficiency operating point in all databases, it points towards a blade design which is not well adapted to the incoming flow. The lean angle deviates the low momentum flow from the hub and casing towards mid-span but cannot suppress the consequent separation at mid-span.

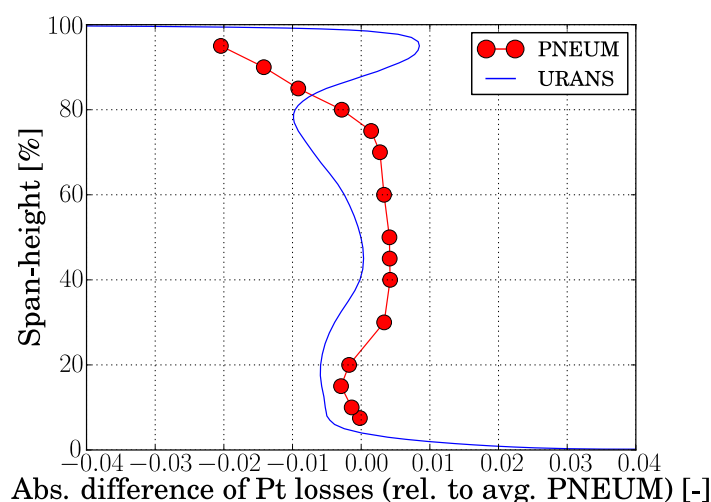


Figure 3.41 Radial profiles of total pressure deficit (relative to inlet total pressure) through the stator 3 based on PNEUM and URANS results, given as normalized absolute differences to the mean losses estimated with PNEUM

### 3.5 Summary of challenges in the flow field exploration

The identified major challenges for the different flow field measurement and simulation methods are summarized in the list hereinafter:

- Mis-interpretations of steady performance data is possible due to measurement constraints beforehand. Correction coefficients were proposed for the flow field conditions at the inlet of the compressor. These corrections need to be considered during the standardization of the measurement values of pressure, temperature and shaft speed. With the defined corrections, the influence on the standardized shaft speed is negligible and does not shift the compressor performance curve noticeably.
- An almost constant shift was found between the predicted and measured compressor efficiency and pressure ratio despite the applied correction coefficients: 0.9 efficiency points, 3.24% pressure ratio, 2.8% of nominal mass flow rate. The source of the differences is expected in faulty predictions of the flow field or simplifications to the simulation domain.
- Two causes were identified for the global shift of the performance curves between the (U)RANS and EXP: Firstly, (U)RANS over-predicts the work in the tip region of the rotors 2 and 3 in CREATE. This is caused by an important over-prediction of the blockage induced by the tip leakage flow. Secondly, (U)RANS predicts globally a higher total pressure rise per rotor with less work than measured. But a systematic mis-prediction of the losses does not occur in the stators.
- The pneumatic probes are not adapted for the static pressure measurement upstream of a stator due to a probable interaction with the latter. This causes an over-estimation of the static pressure, and thus under-estimation of the velocity. Further fundamental studies were proposed for understanding the interaction mechanism.
- The LDA measurement technique is challenged by the given flow field to measure accurately the time mean axial velocity component downstream the stators. The measurement quality was verified and measurements inside of a stator passage would be necessary for an identification of the source of this challenge.
- The pneumatic probes do not capture well the time mean velocity of an oscillating wake, where there is an important oscillation due to a separated boundary layer.
- RANS over-predicts a separation (in the stator 1), where rotor-stator interactions have an impact. Here, the URANS gives a closer fit to the measurement data because of resolving the rotor-stator interactions. However, (U)RANS over-predicts a separation in size but not losses in the stator 3 at the peak efficiency but shows a good fit to the measurements at the near surge operating point. The turbulence model might lead once to a fault and once a good prediction.



# Chapter 4

## Clocking

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The previous Chapter introduced the main characteristics of the flow field in CREATE and discussed the limitations of the flow field exploration methods. Keeping these limitations in mind, in this Chapter the focus is put on the effect of stator clocking in CREATE. Clocking was introduced (Section 1.3.2) to have a possible positive effect on the compressor performance by reducing losses in the stators, and also being able to reduce the unsteady rotor blade loading. Most of the presented work was conducted with low-speed research compressors. The originality of this work is to study the effect of clocking in a multistage high-speed compressor.

This Chapter focuses on three global points which are important for the clocking. The first part of the Chapter will discuss the global effect of clocking on the compressor performance. In the second part, the stator wake characteristics are presented because they obviously play a



key role for the clocking effect. The dissipation and transport of the stator wakes are therefore described in detail. In the third part, the stator-stator wake interaction is characterized. Possible optimal and critical clocking positions are explained based on their influence on the flow field.

## 4.1 Definition of clocking in CREATE

Clocking in CREATE shall be defined as changing of the circumferential position (POS) of the stator 2 relative to all the other stators rows against the shaft turning direction, as sketched in Figure 4.1. The test rig allows the study of 5 clocking positions (POS 1 to POS 5), which cover one stator pitch with an open spatial period. An additional POS 6 can be studied, which is closing the spatial period because it is a repetition of POS 1 but one stator pitch further. Therefore POS 1 and POS 6 can be used to validate the repeatability of the performance measurements for a theoretically identical clocking position.

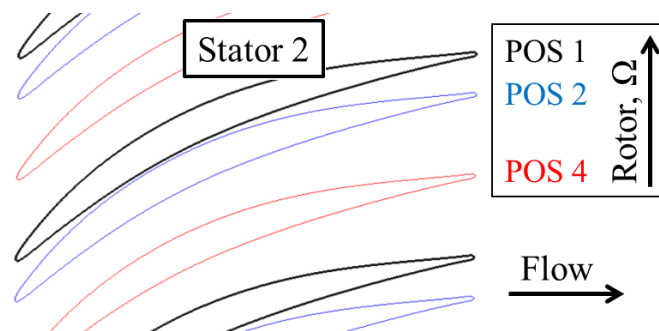


Figure 4.1 Sketch of clocking definition for CREATE

## 4.2 Influence of clocking on the global steady performance

The global performance data can be obtained from three different databases for the study of the global effect of clocking:

- Steady global performance data is labeled EXP (obtained according to Section 3.1). The isentropic efficiency is computed with a variable heat capacity ratio, which depends on the actual inlet and outlet total temperature at the measurement day. That avoids a bias of the estimation of the clocking effect by the measurement conditions. This point was discussed in Section 3.1.2. Here the absolute offset between URANS and measurement is of no importance, thus a constant heat capacity ratio is not applied.
- PNEUM measurement data is used as weighted averaged measurements at the inlet plane (250) and at the outlet plane (290).
- URANS performance predictions are obtained likewise from performing weighted averages on the flow field conditions at the inlet plane (250) and outlet plane (290).

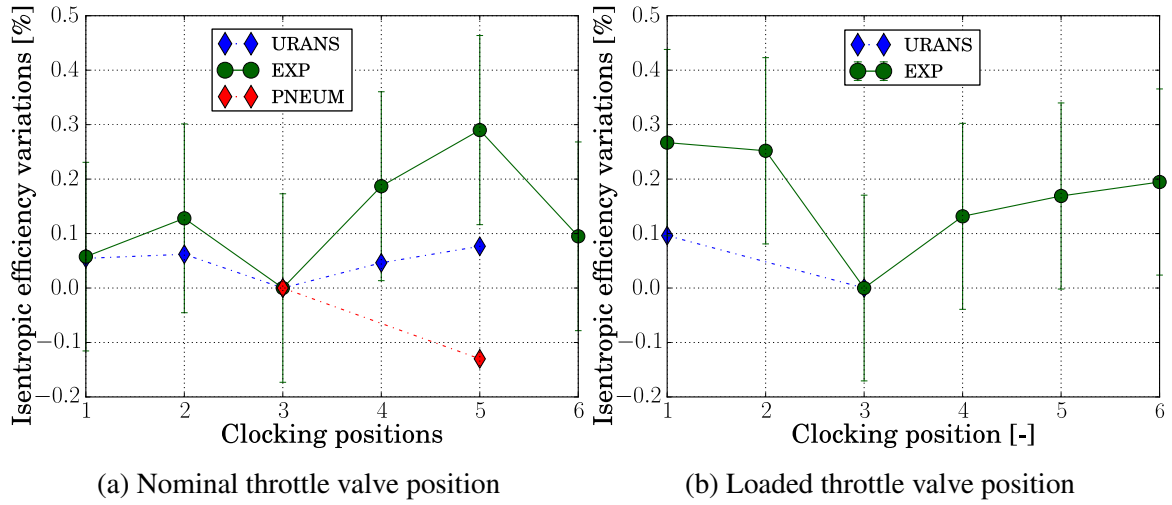


Figure 4.2 Influence of clocking on the global efficiency of CREATE

The **isentropic efficiency** is given for the three databases (EXP, URANS, PNEUM) for several clocking positions relative to the respective minimum efficiency of each database in Figure 4.2a for the nominal throttle valve position, and in Figure 4.2b for the loaded throttle valve position.

The PNEUM performance values are shown for the two available POS 3 and POS 5 in red for the nominal throttle valve position. The global steady performance measurement data is shown in green (EXP). The relative measurement uncertainty for the isentropic efficiency ( $\pm 0.2\%$ , see Table 2.3) is indicated with green errorbars. Here the relative uncertainty is taken as absolute one due to confidentiality where there is no significant difference between the two values. Furthermore the available numerical URANS performance estimations are shown in blue.

The comparison between POS 1 and 6 shows differences that are smaller than the measurement uncertainty. The difference is  $0.006\%$  at the nominal throttle valve position and  $0.08\%$  at the loaded throttle valve position. In both cases, POS 6 shows a lower isentropic efficiency than POS 1. Those differences fall within the band of measurement uncertainties, which on the one hand confirms the repeatability of the measurements. On the other hand, a difference of  $0.08\%$  compared to  $0.2\%$  measurement uncertainty clearly shows that the measurement uncertainty should not be neglected when judging the global effect of clocking.

At the nominal throttle valve position, the maximum effect of clocking is  $0.3\%$  for the global measurements (green),  $0.08\%$  for the URANS predictions (blue), and  $0.13\%$  for the PNEUM measurement (red). At the loaded throttle valve position, the maximum effect of clocking is estimated to  $0.27\%$  for the global measurements,  $0.1\%$  for the URANS predictions, and PNEUM measurement data is not available here.

Ignoring the measurement uncertainty, a trend is visible in the global performance measurement data with a critical clocking position 3 (POS 3) and an optimal clocking position 5 (POS 5) at the nominal throttle valve position. The evolution of the clocking effect could be described as sinusoidal as also often reported in literature (see Section 1.3.2), with

one peak and one valley over the full clocking period. POS 1 and 5 do not fit perfectly the sinusoidal trend, repeated performance measurements could maybe give a better fit. The sinusoidal trend is confirmed by the URANS results but only with a weak amplitude level. The PNEUM measurement data reveals POS 5 as critical clocking position opposing to the steady performance measurements and URANS. This clearly raises doubts about the trend of optimal and critical clocking position.

At the loaded throttle valve position, POS 1 appears as optimal and POS 3 again as critical clocking position. POS 5 shows again a higher efficiency than POS 3, and could be therefore seen as an optimal clocking position over the operating range from nominal to loaded valve position. The performance evolution could be carefully labeled sinusoidal as also at the nominal valve position. A sinusoidal trend means in theory that the clocking effect is similar for a wake of an upstream stator passing on the pressure or suction side of the downstream stator passage. Though, the 3D profiles of the stator wakes and blades make this more complicated as it will be shown in this Chapter.

Overall, URANS gives a good estimation of the global effect of clocking on the isentropic efficiency. All the data points lie either within or very close to the measurement uncertainty band at both investigated valve positions.

Figure 4.3a shows the **total pressure ratio** relative to the one of the presumably optimal clocking POS 3 for the nominal throttle valve position and Figure 4.3b for the loaded throttle valve position.

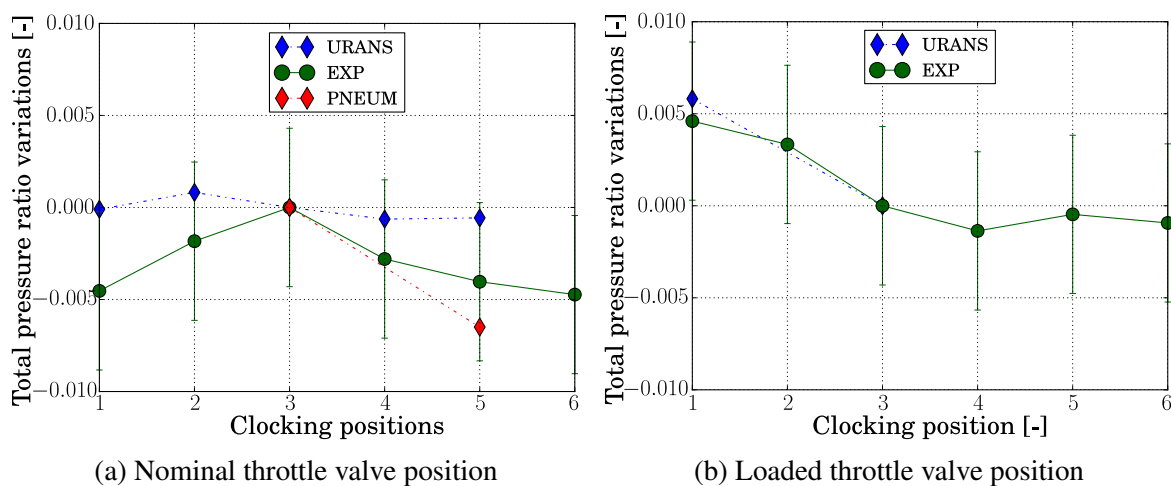


Figure 4.3 Influence of clocking on the total pressure ratio of CREATE

The measured (EXP) total pressure ratio variations are in opposing phase to the efficiency variations at the nominal throttle valve position. At the presumably critical clocking position (POS 3) the highest total pressure ratio is obtained. POS 6 reproduces well the pressure ratio of POS 1. Whatever the clocking position, the variation of the total pressure ratio stays within the measurement uncertainty band. At this scale of variations, URANS predicts almost no (five times smaller) clocking effect compared to the measured one. The "highest" total pressure rise is found with POS 2. The PNEUM finds a higher pressure ratio with POS 3

than POS 5, corresponding well to the isentropic efficiency trend. From an optimal clocking position, it would be expected to obtain the lowest losses in the stators and highest overall pressure ratio while having little influence on the rotor work. This was also found for the embedded stage 2 performance in an experimental work on a three stages compressor by Key et al. (2009). Thus, especially the EXP results are surprising and might indicate the relatively low effect of clocking compared to the influence of the measurement uncertainty.

At the loaded valve position, the total pressure ratio variation is not sinusoidal. And the difference between POS 1 and POS 6, with an amplitude of the maximum clocking effect, questions the validity of POS 1 as optimal clocking position. The URANS results predict a good match of the total pressure ratio variation with the POS 1 but it stays within the measurement uncertainty band. Only two clocking positions are studied with URANS simulations at the loaded valve position because of the little change of the amplitude of the global effect of clocking from nominal to loaded valve position.

The **total temperature** is fixed with the inlet boundary conditions for all conducted simulations. Also in the measurements the standardized inlet total temperature is kept constant. Therefore, the influence of the clocking on the compressor work can be found in the outlet total temperature variations, for the nominal throttle valve position in Figure 4.4a and for the loaded throttle valve position in Figure 4.4b. The repeatability of the outlet total temperature measurement is good with at maximum a variation of  $0.1^\circ$  (see POS 6, POS 1), compared to the measurement uncertainty of  $0.5^\circ$  (Table 2.3). Globally, all temperature variations stay within the measurement uncertainty band.

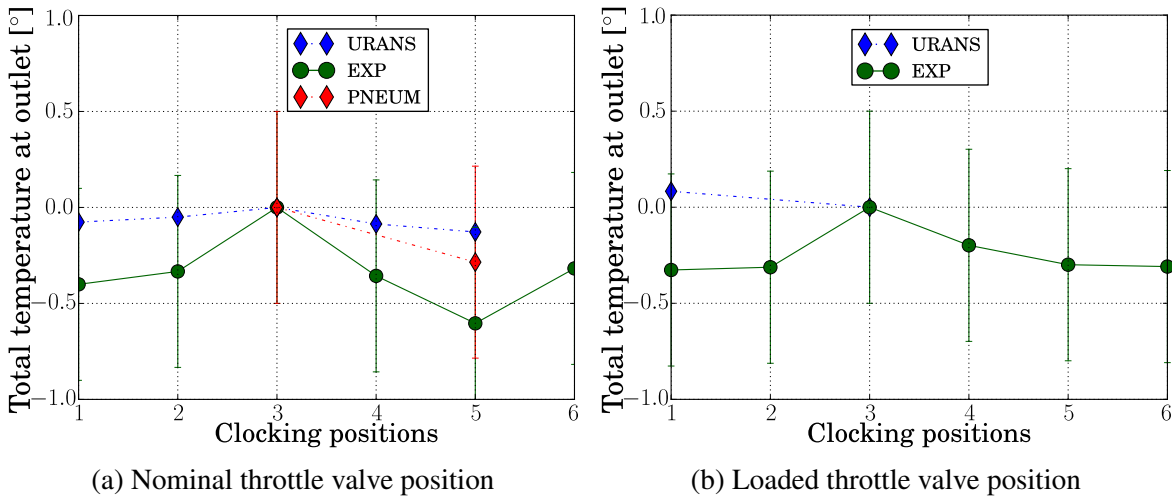


Figure 4.4 Influence of clocking on the outlet total temperature of CREATE (rel. to POS 3)

The critical POS 3 shows the highest outlet total temperature for all databases at both valve positions. URANS data shows the only exception, with the highest total temperature for the POS 1 at the loaded throttle valve position. The optimum POS 5 gives the maximum decrease of  $0.60^\circ$  with the global steady measurements,  $0.28^\circ$  with PNEUM, and the smallest effect with URANS of  $0.13^\circ$ . At the loaded throttle valve position, the clocking effect on

the outlet total temperature lets appear the POS 3 as an outlier with an offset of about  $+0.3^\circ$  compared to the other clocking positions.

The results for the global clocking effect are summarized hereinafter:

- The global effect of clocking is small (0.3 efficiency points), and correlates with results in literature for low-speed research compressors as introduced in Section 1.3.2.
- With all investigated outlet throttle valve positions (nominal and loaded), the clocking effect stays within the measurement uncertainty range. The repeatability test (POS 1 vs. POS 6) illustrates the importance of keeping in mind the measurement uncertainty.
- POS 3 could be seen as a negative outlier, with an over-estimation of the measured outlet total temperature ( $+0.3\text{K}$ ).
- URANS gives a good estimation of the magnitude of the clocking effect with 0.08 efficiency points.
- Ignoring the measurement uncertainty, POS 3 and POS 5 could be carefully considered respectively optimal and critical clocking position over the nominal and loaded operating point. These clocking positions are thus worth a deeper flow field analysis, as they correspond to two particular relative positions of upstream wakes and blades.

The actual differences in the flow field between the POS 3 and POS 5 are presented in the following part of this Chapter.

### 4.2.1 Influence of clocking on the surge margin

Surge inception measurements were conducted for all clocking positions using an identical valve closing method for each position. Starting from the last stable operating point, the outlet throttle valve was closed with such a low speed that it allows quasi stationary flow conditions. The absolute valve position is of no importance here, only the differences between the investigated cases are of interest. The flow conditions (mass flow rate, outlet pressure) measurements are limited to an acquisition frequency of 1 Hz. Besides, synchronous high frequency pressure measurements can be used to estimate very precisely the onset of stall (pressure acquisition frequency of 500 kHz, compared to about 192 Hz shaft speed).

Generally, the randomness and uncertainty of the time point of the stall (and eventually surge) inception could be explained by a small perturbation triggering the onset of stall. Here the randomness can be estimated by comparing the valve positions and mass flow rates for surge inceptions between identical clocking positions, see Table 4.1. The differences in throttle valve closing between the two POS 3 and POS 5 are 0.15 mm and 0.17 mm at the surge inception time point respectively, which corresponds to 4.5 s and 5.1 s delay respectively. The delay is not linearly related to the mass flow rate. The 0.17 mm valve closing difference, and 1.0% mass flow rate difference shall be understood as repeatability uncertainty for the surge inception measurements at a given clocking position. The measurement uncertainty

for the mass flow rate is lower ( $\pm 0.35\%$ , Table 2.3) than the range of mass flow rate within which the surge inception occurs. This signifies that the surge inception does not occur at a fixed mass flow rate but over a mass flow rate range. A disturbance from upstream is likely to cause the very rapid surge inception in this compressor, and the surge inception is not linked to a fixed incidence angle to the blade rows for example.

Table 4.1 Valve position and mass flow rate at stall inception

| Clocking | Valve position<br>mm | Mass flow rate<br>relative to nominal |
|----------|----------------------|---------------------------------------|
| POS 3    | 172.45               | 85.9%                                 |
| POS 3    | 172.30               | 84.9%                                 |
| POS 5    | 171.70               | 86.1%                                 |
| POS 5    | 171.87               | 85.6%                                 |

The instantaneous data points for the total pressure ratio during the surge inception measurement (with slow continuous closing of the throttle valve, 1 Hz acquisition frequency) are presented for the five clocking positions in Figure 4.5. A zoom on the measurement data is seen (less than 10% of the operating range) in the Figure with a close focus on the surge inception that causes the abrupt drop of the pressure ratio. The largest difference is found between POS 1 and POS 4, where the data for POS 1 suggests an extension of the operating range by +0.11% nominal mass flow rate respectively. This extension is below the previously identified repeatability uncertainty but would suggest an extension of about 0.7% of the operating range. Based on the given data, a clocking effect on the surge margin cannot be stated with certainty.

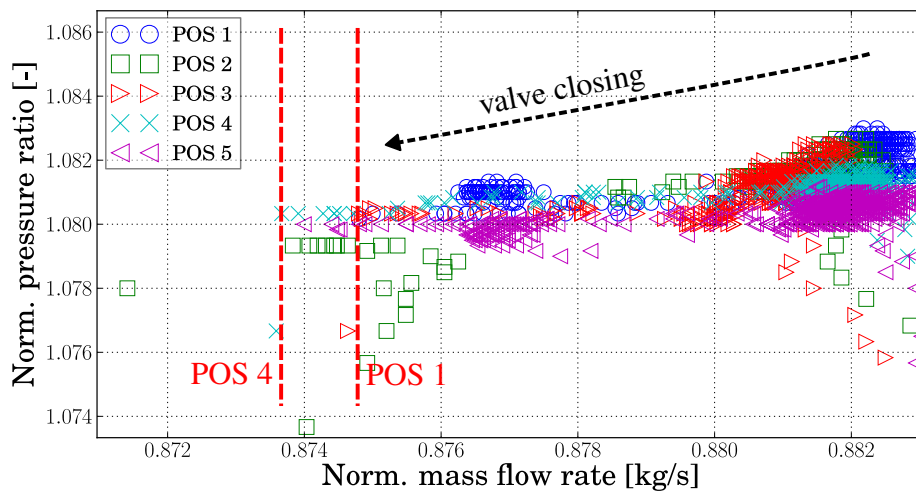


Figure 4.5 Influence of clocking on the surge inception illustrated with the total pressure ratio of CREATE

### 4.2.2 Bias of clocking effect estimation by a limited measurement grid

The flow field conditions at the outlet are obtained with a limited number of measurement points (30, see Section 3.1.2). This was shown numerically to have a possible impact on the performance estimation ( $\pm 0.13$  efficiency points) depending on the common circumferential position of those 30 probes ("numerical turnable probe rake"). The method of the numerical turnable probe rake is applied to several clocking positions, and the isentropic efficiency variations relative to the common mean efficiency are presented in Figure 4.6a for three clocking positions. Depending on the rake position, the prediction of the clocking effect can reach up to 0.25 efficiency points. Also the optimal clocking position is found to depend on the circumferential rake position. Exemplarily, POS 4 can be the optimal position at  $6^\circ$  or the critical one at  $0^\circ$  rake turning angle. The seven peaks indicate the influence of the stator flow field (seven blades) on the efficiency estimation.

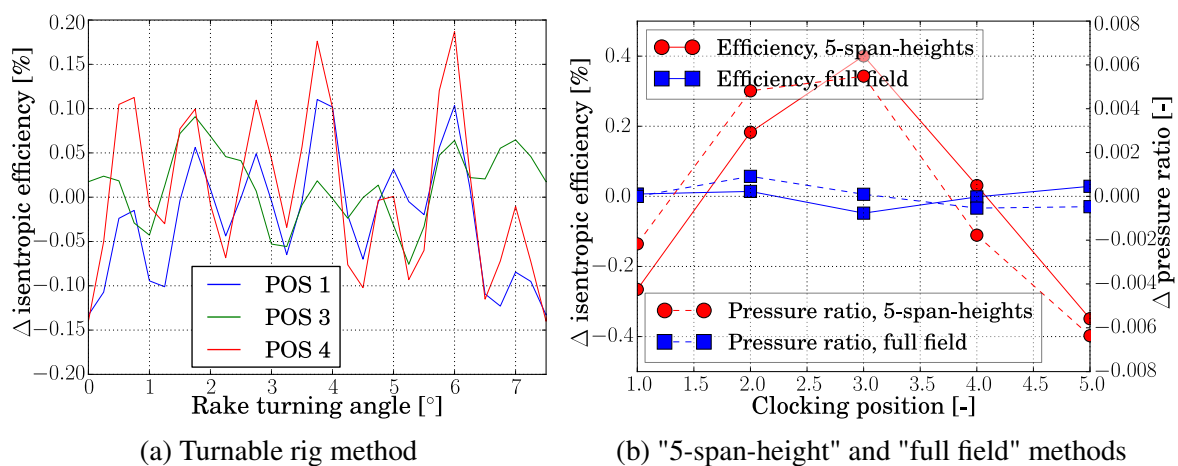


Figure 4.6 Isentropic efficiency estimation with different numerical extraction methods given as difference to the respective mean values over all rake turning angles or clocking positions

Another possibility for an experimental-like estimation of the total pressure and temperature at the compressor outlet is to take the whole circumferential numerical information from the five measured span-heights. Instead of an arithmetic average (done with the measurement data) a weighted average is applied and is presented in Figure 4.6b. That method reproduces the limited taking into account of the boundary layers at the outlet by the experimental measurements. This method is labeled "5-span-height" and is compared to the performance estimation based on the "full field". The efficiency and pressure ratio data is given relative to the respective overall mean value of each method.

The "5-span height" method leads to a clear over-estimation of the clocking effect with more than 0.7 efficiency points (red) and more than 0.006 pressure ratio points compared to the full field method. This over-estimation occurs because the low radial resolution does not allow to take accurately into account possible positive and negative effects of the clocking in the radial direction. The radial variations of the clocking effect will be discussed in Section 4.4.

In conclusion, it is very important to have a fine measurement grid for an accurate estimation of the clocking effect. Otherwise the distinction between an optimal and critical



clocking position is not trustworthy because it would depend on the extraction method from the numerical results or the measurement method.

### 4.3 Stator wake characteristics

This Section presents the dissipation of stator wakes along its flow path from one stator to another, and the characteristics of this wake transport.

#### 4.3.1 Dissipation of stator wakes

The trace of the actual stator wake trajectory from stator 1 through rotor 2 until arriving at the stator 2 passage is not obvious. An explanation for this is the axial time mean evolution of the total pressure deficit induced mainly by the stator 1 and stator 2 in Figure 4.7. It is obtained from spatial discrete Fourier transforms on the time mean flow field at several span-heights for several axial positions (not inside of blade rows). In a signal processing sense, the square root of the energy of the spatial mode 7 (7 stator blades over the simulated sector) and its harmonics (14 and 21) is taken as amplitude for the total pressure deficit (see introduction of Equation 2.26). This amplitude is averaged for every axial position over the span. The data is given based on the URANS (blue) and PNEUM (red) results for the POS 3 and POS 5.

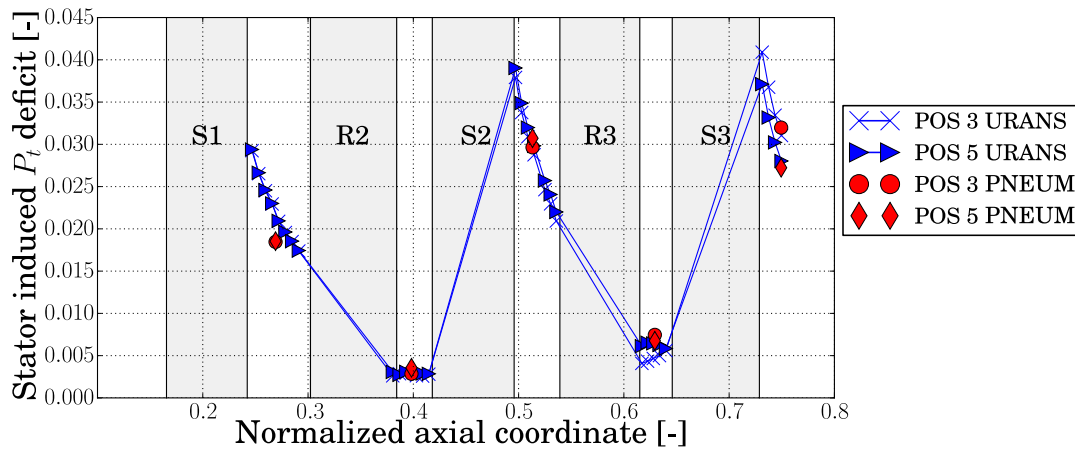


Figure 4.7 Time mean dissipation of stator wakes expressed as total pressure deficit averaged over span at axial position from the stator 1 to the stator 3

Globally, the experimental and numerical data show a good fit for the time mean total pressure deficit. A fast mixing out of the total pressure deficit induced by the stator wakes, hence decreasing amplitude, is found upstream of the rotor 2 already (idem upstream of the rotor 3). More than two third of the wake deficit is mixed out up to the arrival at the rotor 2 leading edge. Downstream of the rotor 2, there is almost no trace of the stator 1 influence anymore. The amplitude decreases to less than 8% of the one found just downstream of the trailing edge of the stator 1. The small amplitude suggests a small influence of the stator 1 wakes on the flow field downstream of the rotor 2 and hence also a small effect of clocking.



At this scale, almost no difference is found between the data of the two clocking positions up to the rotor 3 leading edge. Downstream, it increases in the axial flow direction but stays smaller than 0.5% of normalized total pressure. Numerical and experimental data show the same tendency here between the two clocking positions downstream of the stator 3. This is related to the effect of clocking on the separated suction side boundary layer of the stator 3 at mid-span (separation identified in Section 3.4.2). This interaction will be discussed later on in Section 4.4.

### Comparison of wake decay to a model

Raj and Lakshminarayana (1973) conducted experimental investigations of the near and far wake ( $\frac{x}{c} > 1$ ) characteristics in a cascade of NASA-65 type airfoils. The authors measured the mean velocity, turbulence intensity and Reynolds stresses downstream the airfoils. From these results, they developed a theoretical formulation for the wake decay in streamwise direction while taking into account the shape of the airfoil and its loading. The wake structure is described by the wake center velocity with Equation 4.1. The influence of cascade parameters such as solidity and incidence on the wake decay is also illustrated. This allows the careful application of theoretical formulation to wakes behind the stator blades of an axial compressor. Of course it is known that the downstream blade rows have an impact on the decay of the stator 1 wakes (such as by the negative jet effect, or wake recovery by stretching). This estimation shall give an idea about the importance of the wake decay on the clocking effect.

$$\frac{V_c}{V_e} = 1 - \frac{1.25 \cdot C_d^{\frac{1}{2}}}{\left(\frac{x}{c} + \frac{x_0}{c}\right)^{\frac{1}{2}(1-m)}} \quad (4.1)$$

where  $V_c$  is the wake center velocity,  $V_e$  is the wake edge velocity,  $C_d$  is the coefficient of drag of the blade of interest, and  $m$  is a constant influenced by the pressure gradient.  $m < 0$  for favorable pressure gradient and  $m > 0$  for adverse pressure gradient. The authors found  $m$  to be equal to 0.16, 0.08, and 0.03 for incidences of  $-6^\circ$ ,  $0^\circ$ , and  $2^\circ$  respectively.  $\frac{x_0}{c}$  is the virtual origin and found most commonly between 0.02 and 0.03 for cascades according to the authors.  $x$  is the axial coordinate, and  $c$  the chord length of the blade.

The model of Raj and Lakshminarayana (1973) is applied for the wake emitted by the stator 1, and its dissipation up to the stator 2 leading edge ( $\frac{x}{c} = 1.7$ ). Equation 4.1 is applied with the value of the chord of the stator 1 at mid-span. The incidence angle to the stator 1 is estimated to about  $6^\circ$  where Raj and Lakshminarayana (1973) give values for the variable  $m$  up to incidence angles of  $2^\circ$ , here the maximum value is taken with  $m = 0.03$ . The blade solidity is estimated at mid-span.

The modulated and measured wake velocity are shown from the stator 1 trailing edge to the stator 2 leading edge in Figure 4.8. The URANS data is shown with the blue dotted line, where  $\frac{x}{c} = 1.49$  corresponds to the inter-row plane 27A. The coefficient  $m$  is evaluated with two values, and it does not change the modeled decay of the stator 1 wake at the chosen scale for the Figure.

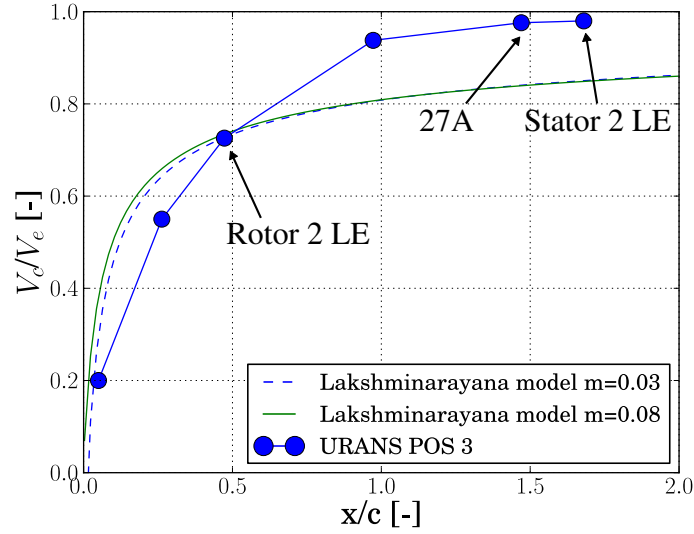


Figure 4.8 Comparison of variation of the stator 1 wake velocity downstream stator 1 between URANS prediction and model

Half a chord downstream the stator 1 trailing edge (at the rotor 2 leading edge), a good fit is found between the modeled and measured wake velocity deficit, where already 72% of the wake velocity deficit is mixed out. Further downstream, the model is off by about 14% because it does not take into account the rotor 2 blade row. Though, the model gives a good idea about 84% of the wake velocity deficit being mixed out along the flow path from one stator to another. The URANS results help to estimate that the multistage environment leaves only 2% of the stator 1 wake deficit non-mixed-out at the stator 2 leading edge. That corresponds well to the total pressure wake deficit results presented in Figure 4.7 and explains mainly why the clocking effect is not significant.

### 4.3.2 Unsteady transport of stator wake segments

Important characteristics of the stator wake transport will be shown with the help of an instantaneous entropy flow field blade-to-blade view in Figure 4.9. That shall contribute to the understanding of the low time mean total pressure deficit induced by the stator 1 as seen in Figure 4.7 and the additional mixing-out losses compared to the model in Figure 4.8 previously.

Downstream of the stator 1 trailing edge, a wavelike modulation of the wakes of stator 1 is seen. This modulation is caused by the chopped segments of the rotor 1 wakes and by the potential field of rotor 2. That could be quantified when analyzing discrete Fourier transforms over the time at points in the stator 1 wakes (not shown here). In the rotor 2 passage, the chopped stator 1 wake segments experience a deformation because of the negative jet effect, which causes an accumulation of the low momentum fluid at the pressure side of the blades. A further contribution to the deformation is the inter-channel pressure gradient together with

the rotor-rotor interactions. That causes the stator 1 wake segments to have a hook-like shape, as illustrated with the black dotted lines in Figure 4.9.

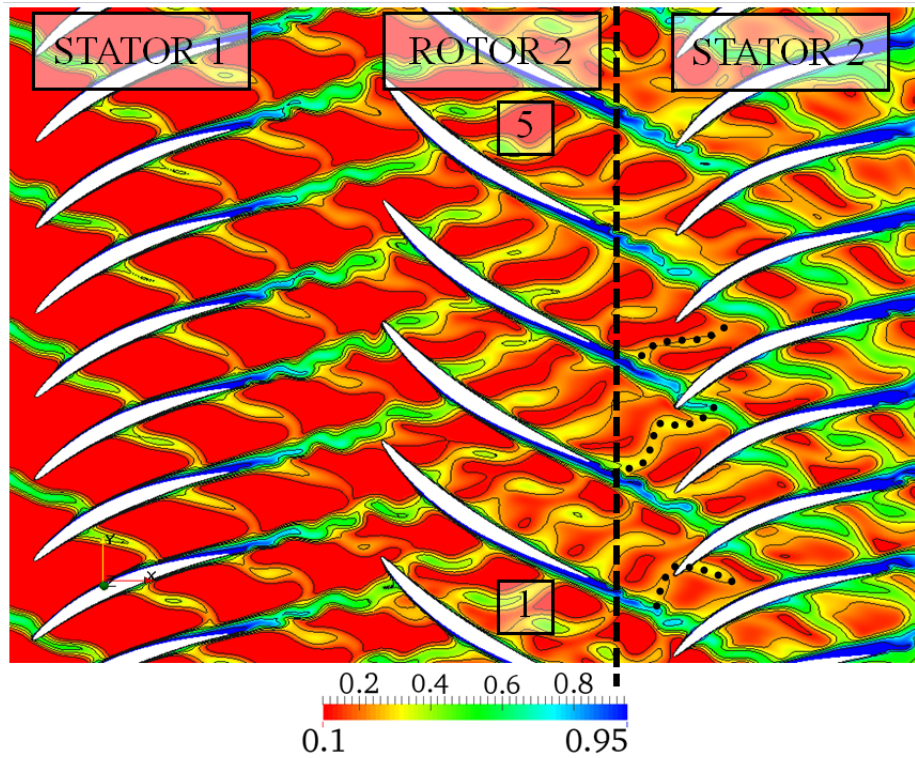


Figure 4.9 Normalized entropy distribution on blade-to-blade view at 60% span-height from the stator 1 to the stator 2, for clocking POS 3, at an arbitrary time point, with passage labels 1 and 5 for Figure 4.10

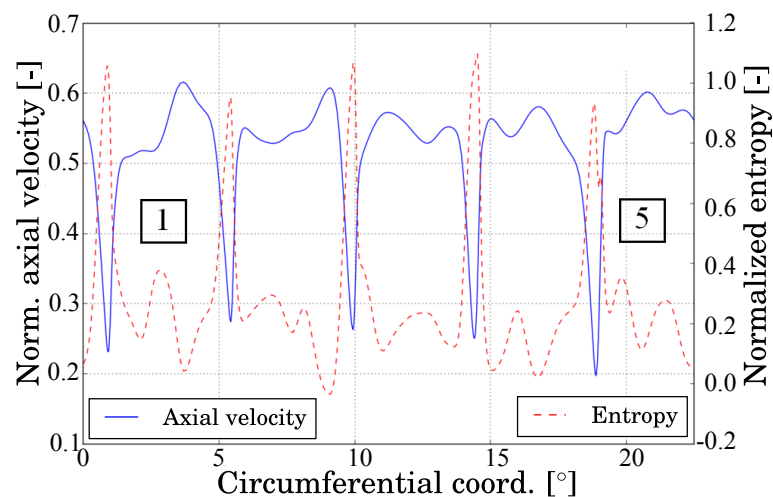


Figure 4.10 Instantaneous entropy and axial velocity distributions over circumference at 60% span-height at plane 27A for POS 3 as indicated in Figure 4.9 at the same arbitrary time point

For the same arbitrary time point, evidence of the deformation is also found in the circumferential distribution of entropy and axial velocity at mid-span in Figure 4.10. At the axial position downstream the rotor 2 (plane 27A), five entropy peaks associated with deep and narrow axial velocity valleys are induced by the rotor 2 wakes. Here, the stator 1 wake segments are expected to introduce seven small entropy peaks but they actually introduce more peaks because of the deformation they experience in the rotor 2 passage. That is found as non-equally circumferentially spaced small entropy peaks where it is difficult to identify all of them because of the overlap with the rotor 2 wake peaks. As already presented with Figure 4.7, the stator 1 wake induced axial velocity deficit is found as weak valleys between 1% and 2% of deficit.

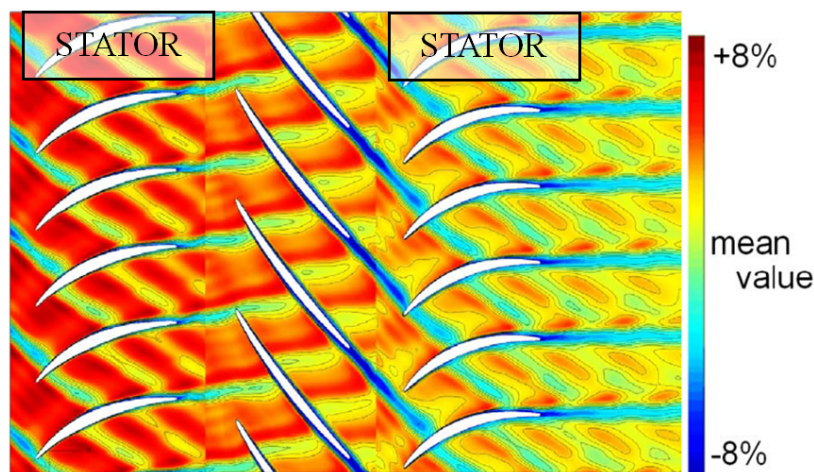


Figure 4.11 Standardized entropy distribution on blade-to-blade view at a constant span-height from a stator 1 to a stator 2 with NLH method from Pesteil et al. (2012)

For further emphasizing the importance of the wake transportation for the clocking effect, a comparison is performed between the flow fields obtained with the URANS method and the Nonlinear Harmonic Method (NLH) which needs less CPU resources. This comparison shall also point out the importance of the numerical method for the study of clocking. The NLH results have been obtained on a comparable compressor, as presented in the work of Pesteil et al. (2012). The NLH estimates the time dependent field using a Fourier decomposition of the periodic fluctuations while only simulating one blade passage per row (Vilmin et al., 2013). The time mean flow is then influenced by the periodic fluctuations. The presented method takes also into account the clocking harmonics. It is known that the quality of the wake transport prediction can be influenced by the number of harmonics that are taken into account as shown by Guédeney et al. (2013).

Pesteil et al. (2012) found a global clocking effect of 0.2 efficiency points using NLH. This is a larger effect than the 0.08 efficiency points reported for CREATE. An analysis of the entropy distribution in the blade-to-blade view obtained with the NLH (Figure 4.11) contributes to the understanding of the difference in the global clocking effect. Small discontinuities are found at the interfaces. That is an indicator for a too little number of harmonics that was taken into account according to the work of Guédeney et al. (2013). As a

consequence, the wavelike modulation of the stator 1 wakes by the upstream and downstream rotors appears very weak. Furthermore, almost no deformation of the chopped stator wake segments is visible in the rotor passages. The rotor-rotor clocking is clearly not taken into account with this method because no difference in stator wake modulation is found between the different rotor passages. Only the accumulation of low momentum fluid towards the pressure side due to the negative jet effect is visible. With the given method and taken unknown number of harmonics, the stator 1 wakes pass almost straight without deformation through the rotor passages. They can either impinge on the stator 2 leading edge or pass at the middle of the stator 2 inter-blade channel.

In summary, the weaker interactions between the rows lead to less deformation of the stator 1 wakes and hence probably to a weaker mixing out of the total pressure deficits in the NLH than in the URANS result field. The difference in wake transport between the NLH and sliding mesh method might be a major part of the explanation of the prediction of a more important effect of clocking with the NLH. This is not necessarily a problem of the NLH method but possibly a non-adapted number of harmonics which was taken into account.

## 4.4 Stator-stator interactions

This Section gives a fine description of the characteristics and the effect of the stator-stator interactions with the help of the data for the clocking positions 3 and 5.

### 4.4.1 Identification of relative stator wake positions

Considering the given stator blade design of CREATE, and more particularly the azimuthal location of the trailing edge along the span, it is expected that the circumferential position of the emitted stator wakes is not constant along the span. The circumferential position of the stator 1 and 2 wakes relative to the respective downstream stator blades are discussed in this Section.

Generally downstream of a stator, its wakes are found as strongest perturbation in signals of total pressure or axial velocity. The perturbation by the wakes from a possible upstream stator is a lot weaker, as shown previously in Figure 4.7. Two fictitious total pressure signals are sketched for an axial position downstream of the stator 2 (plane 280) in Figure 4.12 with solid lines in red for the POS 3 and blue for the POS 5. The direct wakes of the stator 2 appear as deep valleys, whereas the wakes from the upstream stator 1 is seen either as weak valley (POS 3) or mixed with the direct wake (POS 5). In the actual very unsteady flow field, the weak valleys would not be identifiable as easy as here. A difference between the two signals (POS 5 - POS 3) does not give any information about the upstream stator 1 wake position. The direct contribution of the stator 2 shall be therefore filtered by a comparison of the flow fields between two clocking positions in the frame of the stator 2. This method will be called "grid-by-grid" comparison, and requires a spatial shift (see Figure 4.12) of one of the signals so to cancel out the direct contribution of a stator. The identification of the stator 1 wake position is then possible, where POS 3 shows as a positive peak and POS 5 as a negative peak.

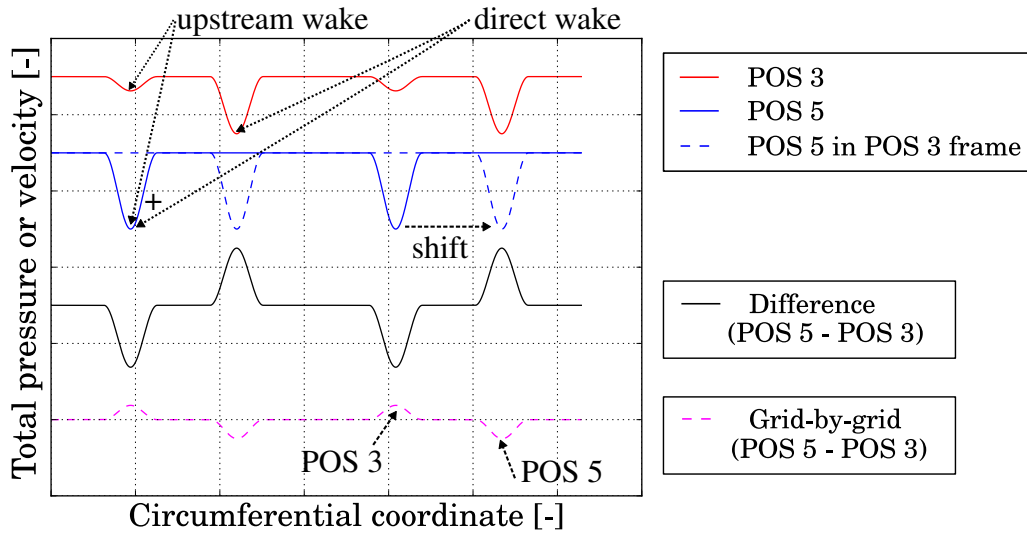


Figure 4.12 Sketch of grid-by-grid comparison method for the identification of the upstream stator wake position

**Analysis in plane 280** Due to the weak effect of clocking on the stator 2 flow field, a grid-by-grid comparison based on measurement data would yield differences within the measurement uncertainty. This does not allow a clear identification of a stator wake structure. URANS results are taken for the demonstration here because they allow a good identification of the upstream stator wake structure and presenting the wake characteristics. Figure 4.13 shows the "grid-by-grid" comparison based on the URANS time averaged total pressure field at plane 280 downstream of the stator 2 for the identification of the relative stator 1 position. Here, the time averaged field is used for filtering the rotor wake influence. Eventually, the comparison is given as normalized total pressure, where red and green indicate the stator 1 wake position for the POS 3 and POS 5 respectively. In addition, a black dashed line indicates the stator wake center of the stator just upstream of the plane. The wake center is identified using the minimum axial velocity per span-height as a reference. The pressure side and suction side are indicated as well for a shaft rotation in positive circumferential direction.

The weak stator 1 wakes (previously established) appear as circumferential inclined structures over almost the whole span-height. The stator 2 wake (black dashed line) shape is different compared to the one of the upstream stator 1 wake, it shows clearly the effect of the blade lean angle, which makes the wake shape look like a hook. Thus, a possible interaction is found between the stator 1 wake (black dashed line) and the stator 2 wake only between 40% and 70% span-height with POS 5. This suggests a radial dependency of the clocking effect.

Results of the near surge or loaded operating point are not presented here because they demonstrate the same clocking characteristics. Note that with a shift in mass flow rate, the circumferential position of the upstream stator wakes changes relative to the downstream stator wake. This can modify the local effect of clocking.



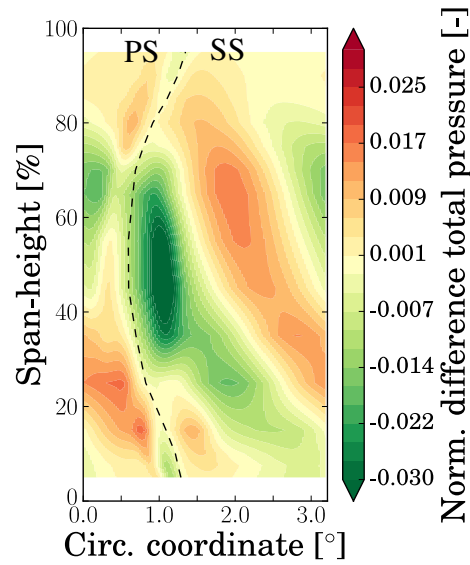


Figure 4.13 Normalized difference between flow field downstream the stator 2 in the reference frame of the stator 2 for URANS at the peak efficiency operating point; Stator 2 trailing edge indicated with valley in axial velocity signal; Red: POS 3, Green: POS 5

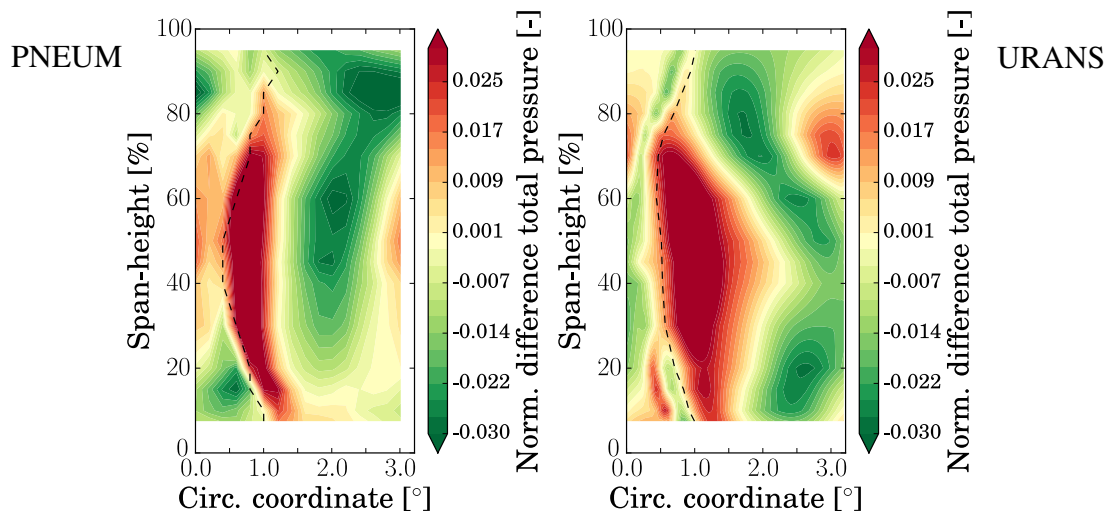


Figure 4.14 Normalized difference between flow field (POS 5 - POS 3) downstream the stator 3 in the reference frame of the stator 3 for PNEUM (left), and URANS (right) at the peak efficiency operating point; Stator 3 trailing edge indicated with valley in axial velocity signal; Red: POS 3, Green: POS 5

**Analysis in plane 290** Figure 4.14 shows the grid-by-grid comparison between POS 5 and POS 3 for the total pressure field with PNEUM and URANS at plane 290 downstream of the stator 3. The differences induced by the clocking are higher at this location than downstream the stator 2 and allow an identification of a wake structure with the PNEUM measurement. The evolution of the differences from the stator 2 downstream to the stator 3 corresponds well to the trace of the time mean dissipation of the stator wakes in the compressor as presented earlier in Figure 4.7.

Both databases reveal a stator 2 wake passing on the suction side and inside of the stator 3 wake from the hub to about 80% span-height for the POS 3. Here, an interaction occurs between the incoming stator 2 blades and the separated (Section 3.4.2) stator 3 blade boundary layer. The influence of the clocking on the losses related to this separation will be discussed in the following Section. Possible interactions between the stator 2 wakes and the hub and casing boundary layers are found above 80% span-height as green spots overlapping with the black dashed lines in all the databases. At POS 5 (green), the radial shape of the stator 2 wake is different between the two databases. The separation deviates the incoming stator 2 wake, which is more pronounced in the URANS results than the PNEUM measurement. This might be caused by the over-estimation of the separation size by the URANS simulation, as established earlier in Section 3.4.2.

#### 4.4.2 Influence of the relative stator wake position on stator losses

A link shall be made between the changes in the losses in the stators and the previously established variations of the circumferential positions of the incoming stator wakes relative to the downstream stator blades. The total pressure losses are calculated with Equation 4.2.

$$c_p = \frac{Pt_{out} - Pt_{in}}{Pt_{in} - Ps_{in}} \quad (4.2)$$

where "in" and "out" label the inlet and outlet of the investigated stator. The total pressure losses across the stator 2 are presented in Figure 4.15 as a difference between POS 5 minus POS 3. Globally, the total pressure loss variations are below 0.5% of the dynamic inlet pressure ( $Pt_{in} - Ps_{in}$ ). PNEUM would show exaggerated differences in losses due to the measurement uncertainty and the systematic over-estimation of the static pressure upstream a stator (Section 3.2.1) and is therefore not presented here.

With the URANS results, an interaction between the stator 1 wake and the stator 2 blade boundary layer at POS 3 was established at 35% span-height in Figure 4.13 earlier. Likewise an interaction was identified for 60% span-height with POS 5. In either of the cases the total pressure losses decrease but only by less than 0.2% with an interaction between the incoming wake and the stator blade boundary layer. The total pressure losses will be presented in detail for these two span-heights hereinafter.

The total pressure distributions and total pressure losses are presented for the plane 280 (downstream the stator 2) at 35% and 60% span-height in Figure 4.18. For this representation, the total pressure field is averaged in time and the circumferential direction at the inlet of the stator. Then the total pressure loss is obtained according to Equation 4.2. This data is ensemble averaged over the seven stator passages for obtaining one representative passage.



The total pressure losses are given as difference between POS 5 and POS 3 at bottom of the Figure. The total pressure is presented as absolute difference to the overall mean obtained with the PNEUM method at each respective span-height.

Downstream of the stator 2 at 35% span-height (Figure 4.16a), the stator 1 wake is located on the stator 2 blade (wake) suction side with POS 5, and rather at the pressure side with POS 3 (see also Figure 4.13). The interaction between the stator 1 wake and the blade suction side boundary layer (POS 5) causes higher total pressure losses than the wake passing at the pressure side. This can be seen between the circumferential positions  $0.6^\circ$  and  $2.2^\circ$ . However,

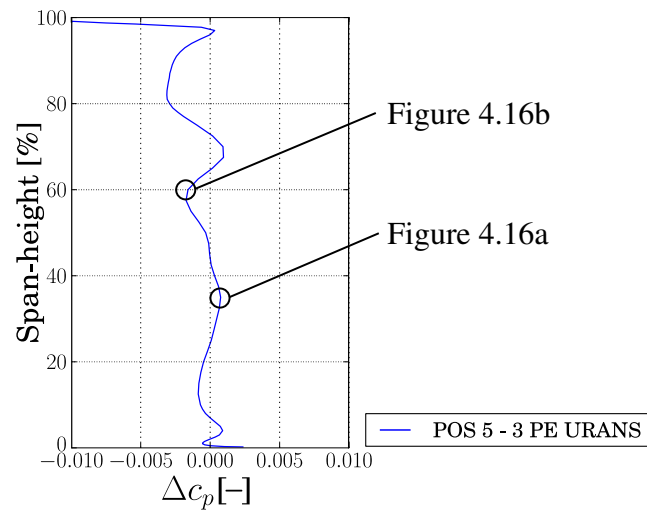


Figure 4.15 Time averaged total pressure loss across the stator 2 based on URANS given as difference (POS 5 minus POS 3) for the peak efficiency operating point

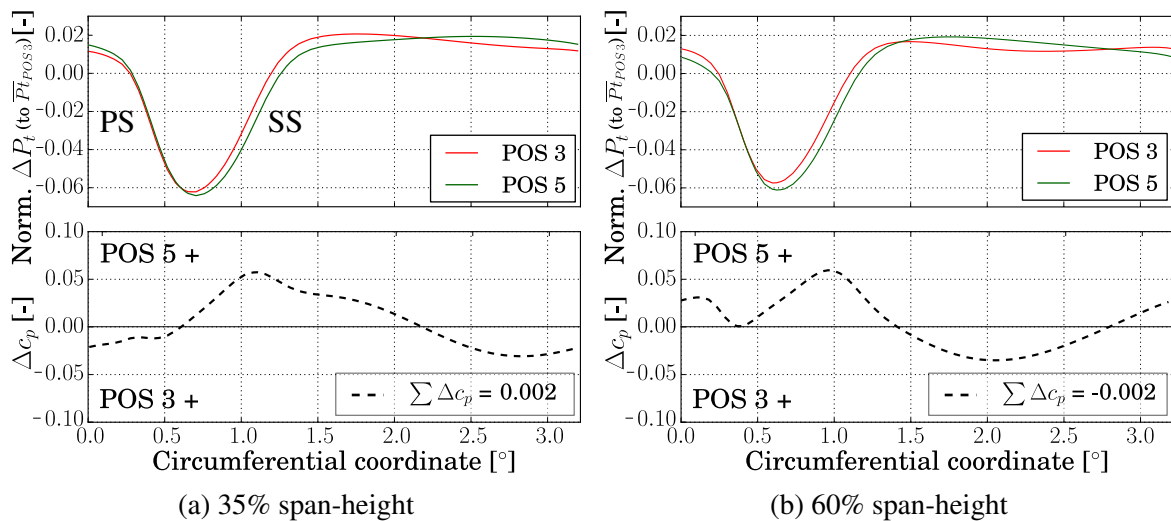


Figure 4.16 Total pressure distributions at the plane downstream the stator 2 and total pressure loss across the stator 2 based on time averaged URANS (POS 3+ and POS 5+ indicate higher losses for the respective clocking position) for the peak efficiency operating point

the difference in the sum of the losses over the circumference is low with only 0.2%, as indicated in the legend.

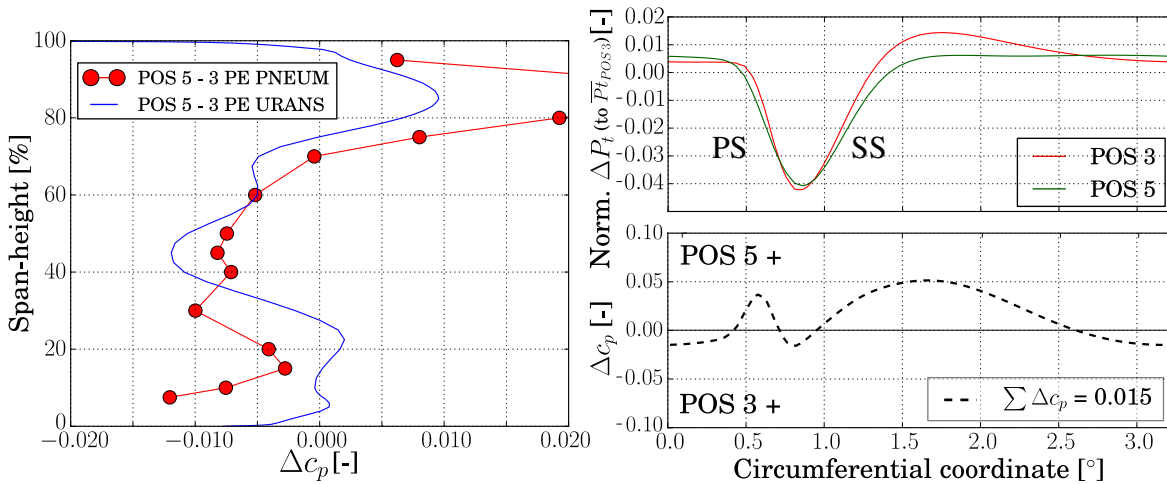
At 60% span-height, the passing of the stator 1 wakes appears as total pressure deficit at mid-passage for the POS 3 compared to the POS 5. That supports the previously stated observations about the circumferential stator wake positions. The stator 1 wake increases the local losses in the stator 2 with either clocking position. Clearly the losses are lower where an interaction occurs between the incoming wake and the stator 2 blade suction and pressure side boundary layers ("wake-wake" interaction). This is the case for the POS 5 at this span-height (green). At the POS 3, the stator 1 wakes causes higher mixing-out losses at mid-passage of the stator 2.

### Effect of clocking on a separated blade boundary layer

The total pressure distribution at the plane downstream the stator 3 and the total pressure losses across the stator 3 - calculated according to Equation 4.2 - are presented as difference (POS 5 minus POS 3) in Figure 4.17a. Two major effects of clocking are found in the stator 3:

- Above 70% span-height, a peak is found with higher losses for the POS 5 than POS 3.
- Between 30% and 70% span-height, a wake-wake interaction causes higher losses with the POS 3 than POS 5.

As explanation for the spike above 70% span-height, the total pressure distribution and total pressure losses are presented for 85% span-height (inside of the losses "spike") in Figure 4.17b. The total pressure signal of the POS 5 reveals a wider stator 3 wake mainly on the suction side between the circumferential positions  $1^\circ$  and  $2.6^\circ$ . In the same zone, the total pressure losses are higher with this clocking position. In comparison, the stator 2 wake increases only very little the losses at the mid-passage flow ( $2.6^\circ$  to  $3.0^\circ$ ) with the POS 3.



(a) Total pressure loss in the stator 3 given as difference (POS 5 minus POS 3)

(b) Total pressure and losses based on URANS at 85% span-height

Figure 4.17 Overview of total pressure and total pressure losses across the stator 3 at the peak efficiency operating point

This negative effect of the wake-wake interaction might be found because of a redistribution of the flow along the span-height between the two clocking positions. The radial total pressure loss profiles (Figure 4.17a) show two peaks towards the hub and casing in favor of lower losses with the POS 3, and at mid-span a peak indicating higher losses with the POS 5. The wake-wake interaction at mid-span might alter the stator 3 blade boundary layer towards the hub and casing. Though, a hub or casing corner separation does not form at either of the clocking positions.

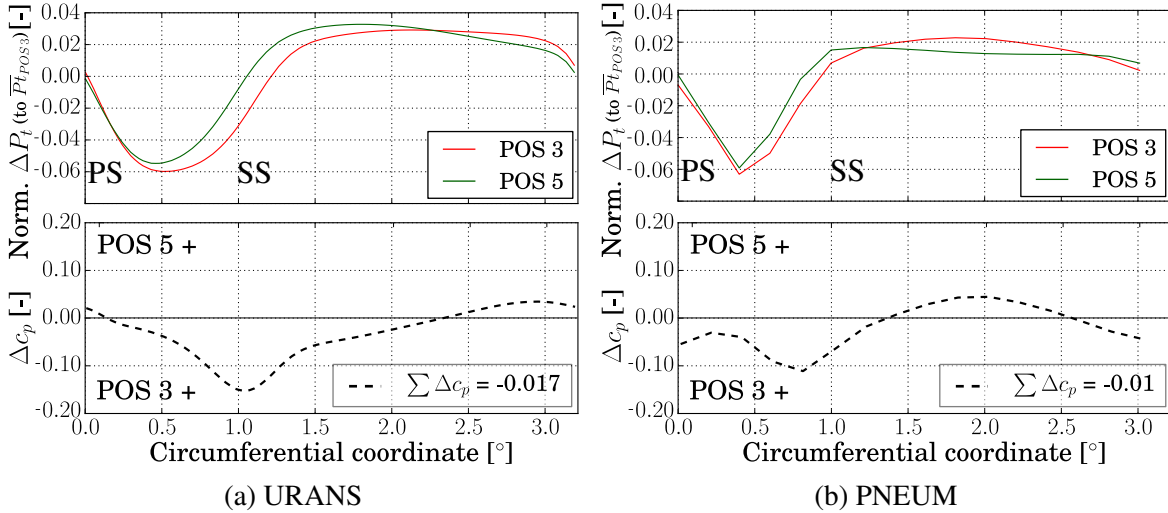


Figure 4.18 Total pressure distribution downstream the stator 3 (top) and total pressure losses across the stator 3 (bottom, POS 5 minus POS 3) at 45% span-height

At mid-span, the effect of the wake-wake interaction is clearer than towards the hub and casing. A description of the flow topology of the separated boundary layer at mid-span is found in Section 3.4.2. This suction side boundary layer is found to be sensitive to the clocking position. Figure 4.18 shows the total pressure distributions downstream the stator 3 and total pressure losses across the stator 3 for 45% span-height with the URANS (left) and PNEUM (right). Both Figures show one ensemble averaged stator passage, which takes into account the available 7 (URANS) and 4 (PNEUM) stator pitches.

The wake-wake interaction causes a growth of the separation with the POS 3. Here the total pressure losses are locally higher for this clocking position. The stator 2 wake passing induces a locally increased incidence angle of up to  $1^\circ$  (not shown here), to an already separated blade boundary layer of the stator 3, and causes therefore the negative clocking effect. Towards the pressure side of the passage, the stator 2 wake passing is seen as total pressure deficit with POS 5. Though, these additional losses are lower than the one related to the grown suction side separation. The PNEUM measurement supports the URANS results. This shows that the URANS simulations are able to predict trustworthy clocking effects. At high loading conditions, Key (2007) found likewise that a wake-wake interaction can trigger a boundary layer separation and result in the highest losses. The local negative effect of clocking identified in CREATE is sketched in Figure 4.19, where the stator 2 wake passing

influences the incoming flow direction and eventually the separated boundary layer size. The higher incidence angle leads to a growth of the already present separation.

In summary, it shall be noted that the wake-wake interaction does not always lead to lower losses in the stators. Higher losses are noted when the interaction occurs only at the suction side, or when there is a separated boundary layer present already. The wake-wake interaction might also trigger a separation, which could lead to higher losses.

A radial redistribution of the flow has been noted as a consequence of the changing size of a separation with different clocking positions. This radial redistribution due to clocking might alter the interpretation of the local effect of clocking, which is no longer mainly an effect following the streamlines (mainly axial direction) but equally a radial effect.

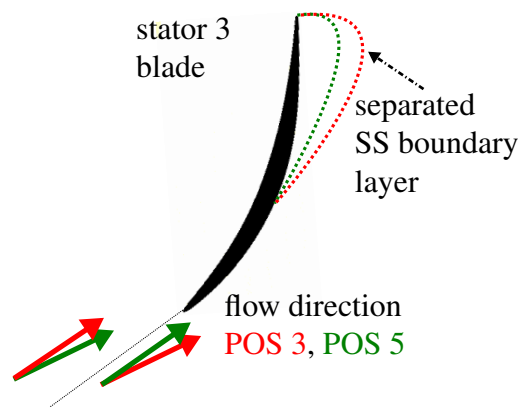


Figure 4.19 Sketch of the clocking effect in case of a separated blade boundary layer, stator 2 wake causes a local increase of the incidence angle for POS 3 (red) and POS 5 (green)

#### 4.4.3 Radial differences in stage efficiencies

The presentation of the change in the stage 1 efficiency between two clocking positions will be used to demonstrate the effect of the measurement uncertainty. An effect of clocking on the stage 1 efficiency is not expected because it is located upstream of the rotatable stator 2. The "uncorrected" isentropic efficiency is presented for the stage 1 as difference (POS 5 minus POS 3) in Figure 4.20 on the left side. The URANS results are presented with the blue solid line, and the PNEUM results with the red circles and solid line. The flow conditions are obtained from the planes upstream of the rotor 1 (25A) and downstream the stator 1 (270). The measurement uncertainty is indicated with the horizontal bars which show a combined uncertainty of  $\pm 1$  efficiency points. This has been obtained from the uncertainties for the total pressure and total temperature stated in Section 2.2.2. Variations of up to  $\pm 1.7$  efficiency points are seen in the PNEUM data, whereas the URANS data shows only small bumps towards hub and casing. The largest variations are outside of the measurement uncertainty band.

The rotor 1 work and the total pressure losses across the stator 1 are not expected to change with the clocking of the downstream stator 2. The small bumps towards the hub and casing can occur because of differences in radial gradients between the clocking positions

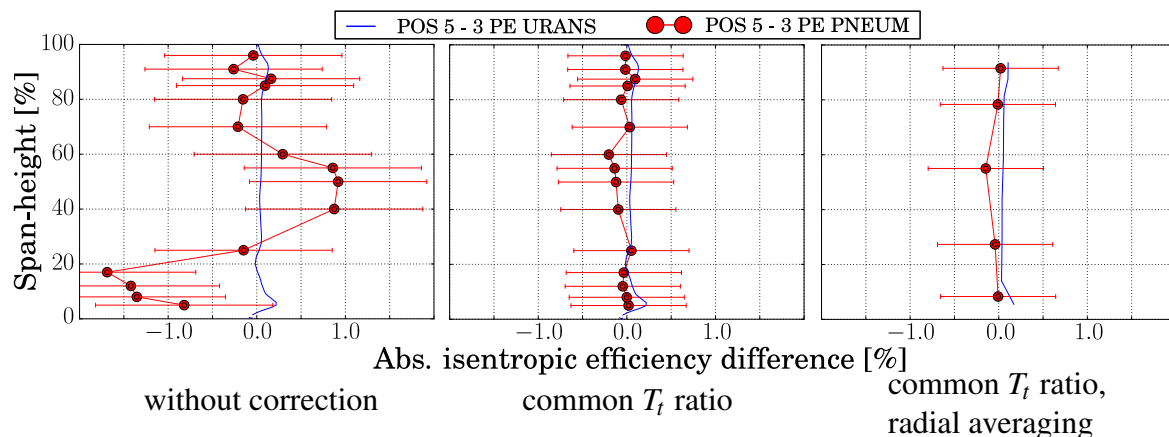


Figure 4.20 Isentropic efficiency of the stage 1 without correction (left), a common total temperature ratio for the two clocking positions (center, right), and radial averaged (right)

in the flow between the inlet and outlet of the stage. At these locations, the hub and tip gap leakage vortex of the IGV is located and might shift with small changes of the operating point with clocking. Differences in the measured total temperature ratio of the stage 1 are nevertheless found (not presented here) and result in the large efficiency variations in the radial profiles. These variations are attributed to the measurement uncertainty but not to the influence of clocking. The changes in the total temperature ratio for the rotors 2 and 3 are of the same order of magnitude as in the rotor 1 (not shown here).

A possible "correction" of the data is proposed: The total temperature ratios are averaged between the clocking positions ("common  $T_t$  ratio") for the calculation of the radial evolution of the isentropic stage efficiency differences. This correction is applied to the remaining PNEUM plots (center, right) in Figure 4.20. That reduces theoretically the high uncertainty from  $\pm 1.0$  efficiency points to  $\pm 0.65$ . With the total temperature ratio correction, the efficiency profiles show consequently mainly changes due to differences in total pressure loss across the stators.

The temperature corrected profiles show variations of only up to  $\pm 0.15$  efficiency points. These variations stay within the measurement uncertainty band, and are close to the expected zero effect of clocking on the stage 1 performance. The streamlines between the inlet and outlet of a stage do not follow perfectly a constant span-height. Considering this, the flow condition measurement data is averaged over three measured span-heights at the inlet and outlet. This gives flow tubes of up to about 20% span-height based on which the isentropic efficiency is presented on the right hand side ("radial averaging") in Figure 4.20 (with a common total temperature ratio as well).

In conclusion, the stage 1 performance is as expected not affected by the stator 2 clocking and any variations are induced by the measurement uncertainty or radial velocity components in the flow. The presented data treatment considers carefully what effect of clocking is measurable with the given measurement uncertainty while knowing what effect of clocking to expect based on the URANS results. The efficiency for the stage 1 is presented here for making aware about mis-judgements about local effects of clocking because of the measurement uncertainty.

### Radial evolution of efficiency for stage 2

The differences between the isentropic efficiency of stage 2 at POS 5 and POS 3 are presented in the Figure 4.21 for the uncorrected URANS and PNEUM data (left) and the total temperature corrected PNEUM data (center). Furthermore, the grid-by-grid comparison for the plane downstream the stator 2 (plane 280) is shown on the right hand side for the discussion of the link between the relative circumferential position of the stator 1 wakes and the stage efficiency. Horizontal black dashed lines are presented for a discussion later on in this Section. Without the total temperature correction, PNEUM shows variations of up to  $\pm 1.2$  efficiency points compared to about  $\pm 0.5$  efficiency points found in the numerical results.

Assuming little effect of the clocking on the rotor 2 work (just about as little as on the rotor 1 work), the total temperature ratio correction is also useful for the stage 2. Using the correction (center of Figure 4.21), the variations between the URANS and corrected PNEUM data show a good match in terms of amplitude of the variations. All differences between the clocking positions stay within the measurement uncertainty band (horizontal error bars). Therefore no ideal clocking position shall be identified for this stage. The URANS results are nevertheless worth to identify locally a link between the relative circumferential wake positions (see grid-by-grid comparison on the right) and the isentropic efficiency.

The following conclusive results can be noted for the URANS results:

- At 35%: With POS 3, the interaction between the blade's BL and the incoming stator 1 wake results in lower losses (see Figure 4.16a) and eventually a higher isentropic efficiency.
- At 60%: With POS 5, the wakes interact with the stator 2 blade boundary layer, and a 0.16 points higher efficiency is found for POS 5 than POS 3 in the numerical results.

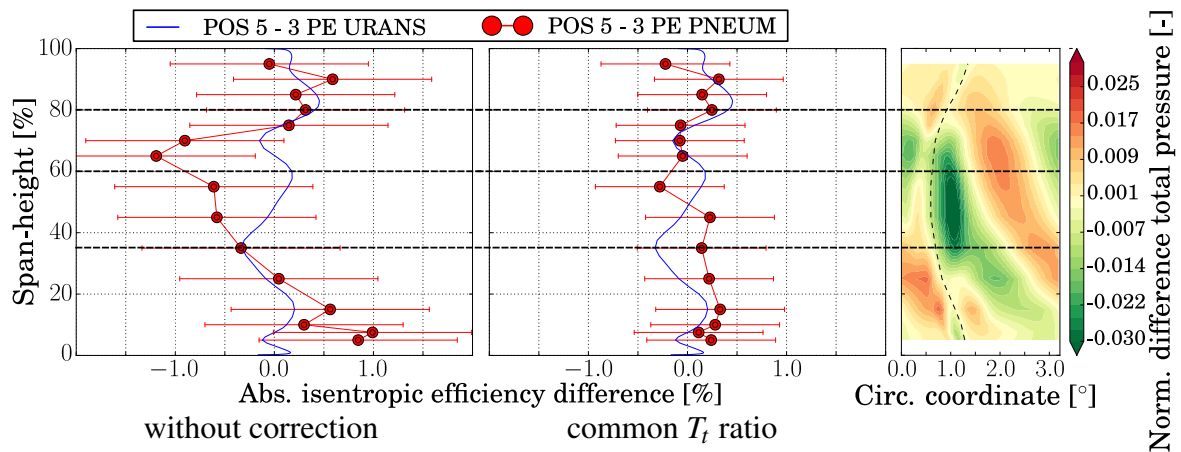


Figure 4.21 Isentropic efficiency of the stage 2 with full information (left), averaged total temperature ratio (center), and URANS axial velocity difference (right, red: POS 3, green: POS 5)



The radial measurement points do not resolve this span-height but suggest likewise a small clocking effect, judged by the surrounding measurement points.

- Above 80% span-height in the casing BL (20% span-height towards casing), POS 3 interacts with the stator 2 BL in the numerical results. This shows one example (also found at the hub) for a not beneficial interaction between an incoming stator wake and the stator blade BL. The isentropic efficiency difference is almost 0.5 points here.

In conclusion, the inclination of the incoming stator 1 wake relative to the stator 2 blade shape is reflected in the radial efficiency profiles of the URANS results as also earlier presented for the radial profiles of the total pressure losses across the stator 2. In the stage 2, the local effect of clocking reaches up to 0.5 efficiency points but there are positive and negative effects. These effects are in equilibrium and this results in an identical overall stage 2 efficiency for POS 3 and POS 5.

### Radial evolution of efficiency for stage 3

Clocking of the stator 2 can influence the rotor 3 work, even if the differences might be small (in the order of the measurement uncertainty as in the rotor 1). The isentropic efficiency differences (POS 5 minus POS 3) are presented on the left in the Figure 4.22 for the uncorrected URANS and PNEUM data. Furthermore the absolute radial profiles of the isentropic efficiency are plotted on the right. The absolute values are "normalized" by subtracting the overall stage 3 efficiency at POS 3 taken from the PNEUM data.

The non-corrected PNEUM efficiency difference shows a good fit to the URANS results. Variations are found in the efficiency difference with peaks at maximum at -1.42 and +0.95 efficiency points. POS 3 gives the higher efficiency in the peaks at 18% and 85% span-height.

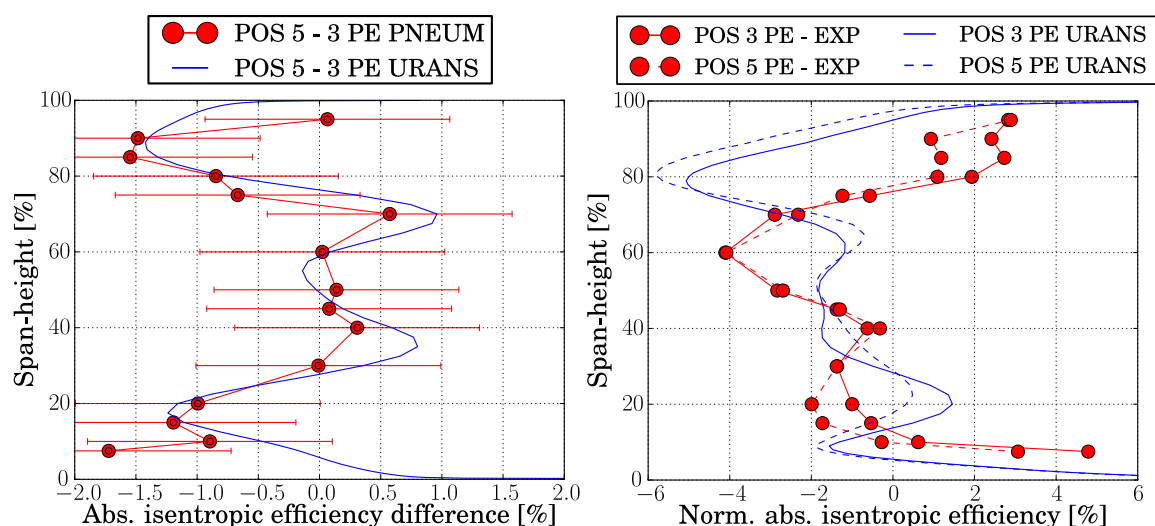


Figure 4.22 Isentropic efficiency differences between POS 5 and POS 3 for the stage 3 without correction (left) and as absolute isentropic efficiency (right, overall mean stage 3 efficiency at POS 3 taken from PNEUM subtracted from all given values) with PNEUM and URANS data

Between 35% and 70% span-height, the interaction between the stator 2 wakes and the separated stator 3 blade boundary layer leads to lower overall losses and consequently lower efficiency at POS 3.

The absolute isentropic efficiency profiles (on the right in the same Figure) show that the absolute PNEUM and URANS profiles do not fit in the radial distribution, contrary to the fit in the radial distribution of the difference. Certainly, important offsets are found in the casing boundary layer, where URANS shows a lower efficiency because of the over-prediction of the total temperature (and total pressure rise) in the rotors 2 and 3. Moreover this Figure shall show that the absolute isentropic efficiency difference variations do not solely originate from radial shifts in the profiles of the efficiency between two clocking positions. There are indeed real effects of the clocking but they can be positive or negative depending on the span-height. Those variations add up to an overall stage 3 isentropic efficiency +0.47 points higher for POS 3 than POS 5. That fits to the global change of +0.13 points for the whole compressor for POS 3 relative to POS 5 obtained with the PNEUM data but is still in disagreement with the global steady performance measurement (see Figure 4.2a).

#### 4.4.4 Unsteadiness in stator wakes

The stator blades are chopping periodically the incoming rotor wakes. This induces periodically local axial velocity deficits and flow angle increases in the stator blade boundary layer. This unsteadiness was reported to be damped when having a stator - stator wake interaction with the necessary clocking positions in the works of Walker et al. (1998), Key et al. (2008), and Jia et al. (2010). A characteristic modulation of a stator wake was presented in Section 3.4.1 related to the casing corner separation in the stator 2. An increase in the amplitude of the modulation is expected at the pressure side of a stator passage due to an accumulation of the rotor wake segments caused by the negative jet effect (introduced in Section 1.2.1). Furthermore, the modulation will be increased at the edges of the stator wakes which signifies their oscillation due to the modulation by the rotor wakes, as it was shown in Figure 3.37a.

Similar to the radial profiles of the losses and efficiency changes, the influence of clocking on the rotor induced modulation of the stator 2 and 3 flow field is traced in radial profiles in Figure 4.23. The modulation by the rotor wakes is calculated at each spatial measurement (or grid point) with temporal Fourier transforms. The first three harmonics of the rotor (5 blades) are taken into account. Their energy is calculated according to Equation 2.26, and its relative amplitude (square root of the energy) is shown for URANS and LDA results. The amplitudes are given relative to the mean axial velocity at the investigated plane in each respective database. The experimental values are based on LDA measurements and presented with the solid lines and dot symbols. The minimum and maximum amplitudes per clocking position and span-height are shown for the LDA results. The challenge for the LDA technique to capture accurately the rotor wake transport through the stators was discussed in Section 3.2.2. This can for one part explain offsets between the URANS and LDA results but are considered to influence the results for the two clocking positions globally in the same manner and therefore make this comparison still useful.



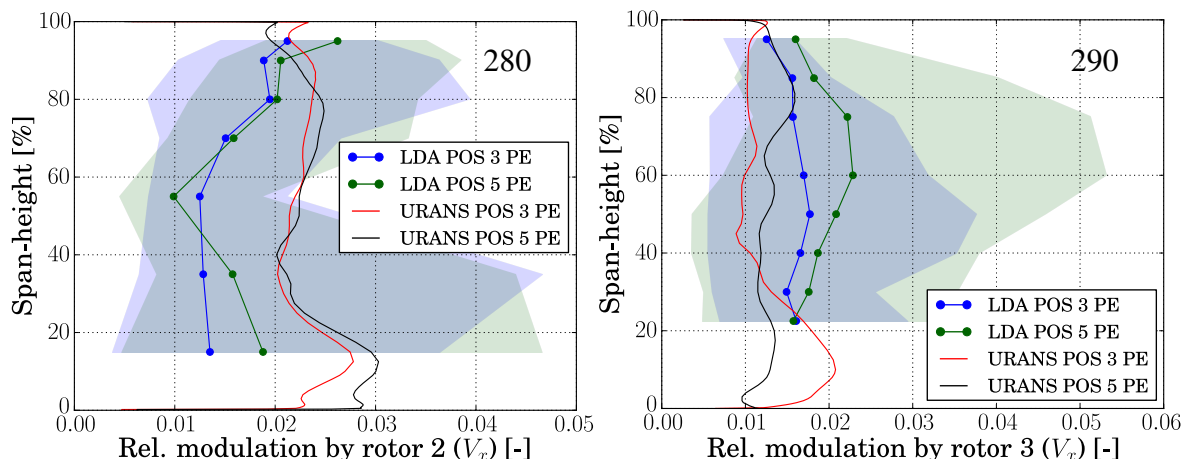


Figure 4.23 Axial velocity flow field modulated by the rotors 2 and 3 downstream the stator 2 (left) and stator 3 (right) respectively, given relative to local time mean  $V_x$

Globally, differences can be noted between the amplitudes at the two clocking positions for each respective database. These differences lie within 1% (of each respective mean axial velocity). Downstream of the stator 2, the mean variations cannot be conclusively pinpointed to a certain clocking position. For example, at 35% and 60% span-height, the modulation is the lowest with the POS 3. A stator wake-wake interactions occurs only at 35% but not at 60% span-height with the POS 3 (see also Figure 4.21 on the right).

In the stator 3 for the span-heights from 30% to 80%, the weaker modulation at POS 3 than POS 5 could be explained with the stator wake-wake interaction that has been identified here earlier. Likewise, the modulation is the weakest at the POS 5 below 35% span-height in the URANS results. Here, also the stator losses tended to decrease due to the wake-wake interaction (see Figure 4.22 on the left). Above 80%, the modulation and losses do not follow this pattern. That shall highlight just as in the stator 2, a clear rule between the relative circumferential stator wake position and the rotor modulations cannot be drawn conclusively for all span-heights. Though, variations between the modulations with the different clocking positions are observed.

#### 4.4.5 Changes in rotor flow field with clocking

The LDA measurements inside of the rotor passages allow the investigation of the effect of clocking on the rotor tip flow field over one stator pitch in space. A space-time diagram for the axial velocity at 95% span-height at the axial position 3 (about mid-chord, see Figure 3.24) inside of the rotor 3 is given exemplary on the left in Figure 4.24 for the POS 3. The circumferential coordinate is given on the vertical and one temporal period on the horizontal axis. The rotor blades appear as diagonal stripes partly with blank spots and partly with faulty low velocities, which are to be ignored. A diagonal stripe of low axial velocity (green) appears parallel to the rotor wakes. This corresponds to the location of the TCV. At about  $2^\circ$  (for the POS 3) a modulation of the TCV is found on a horizontal line, which represents the interaction between the incoming stator 2 wake and the TCV in the rotor 3.

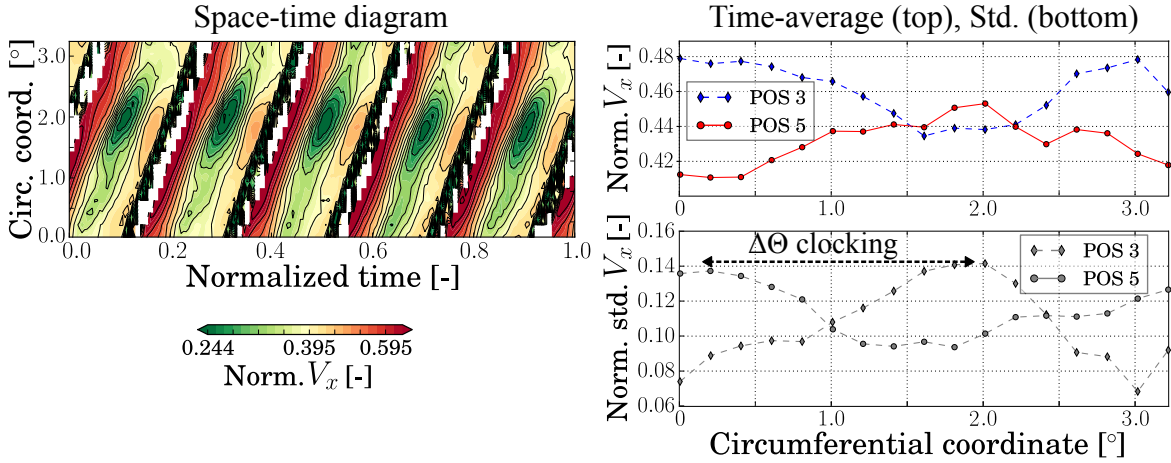


Figure 4.24 Space-time diagram of axial velocity measured at the axial position 3 (about mid-chord) inside the rotor 3 passage at 95% span-height (left) for POS 3, and time averaged data for the axial velocity and the ensemble avg. standard deviation (right)

On the right hand side in the same Figure 4.24, the time averages of the axial velocity (top) are given for the two clocking positions based on the LDA measurements for the same location in space as also used for the space-time diagram. The time averaged standard deviation (bottom) is presented in grey with the same symbols as for the time averaged axial velocity. The standard deviation is calculated on all registered particles in each respective ensemble average window. The following results are also verified to be valid at the same span-height for the axial position 4 (closer to the TE than the position 3).

The valley in the time averaged signals mark the incoming stator 2 wake at both clocking positions. The circumferential wake position changes with the clocking position by about  $-1.28^\circ$ , corresponding to the expected clocking effect from POS 5 to POS 3. The standard deviation, or non-deterministic unsteadiness, is the highest in the wake valleys. A weighted average of the standard deviation over the circumference yields a 0.008 normalized points higher one with at POS 5 than POS 3. This is a difference within the LDA measurement uncertainty. In a next step, this will be compared to changes in the rotor work and efficiency.

Figure 4.25a presents the total temperature rise across the rotor 3 as difference between POS 5 and POS 3. They are given for URANS and PNEUM results. Globally the total temperature evolutions vary by less than 1K between the clocking positions, which is largely within the measurement uncertainty range (indicated with red horizontal bars). Nevertheless, PNEUM and URANS show tendencies in the same direction in the tip region (above 70%) with higher work (negative difference) with the POS 3.

Figure 4.25b shows the isentropic efficiency for the rotor 3 as difference between POS 5 and POS 3. The efficiency variations are globally within a  $\pm 1$  efficiency point range. The URANS results show an elevated efficiency above 65% span-height for the POS 5. The PNEUM data is following this trend with only one outlier at 90% span-height.

A higher unsteadiness is expected in the case of a rotor that is working at a lower efficiency than a reference rotor. At POS 5, the higher unsteadiness has been noted but a lower total temperature rise and consequently higher efficiency is found in the tip region. This

is unexpected and illustrates that a possible effect of clocking on the unsteadiness in the rotor falls within the measurement uncertainty range of the LDA measurements. In conclusion, a link cannot be established between stator clocking and changes in the unsteadiness in the rotor flow field for the present compressor.

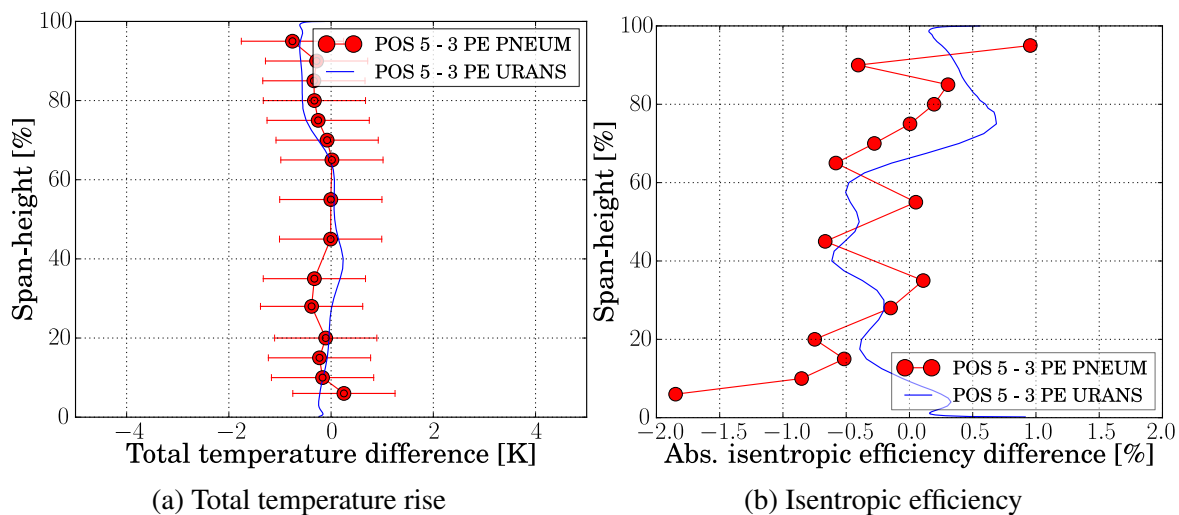


Figure 4.25 Flow evolution across the rotor 3 with PNEUM and URANS results

## 4.5 Conclusions about clocking

The results of this Chapter about the effect of clocking in a high-speed compressor are summarized hereinafter.

- The experimental and numerical estimation for the overall effect of clocking is within the indicated measurement uncertainty range. When ignoring the uncertainty range, a trend appears with a possible clocking position 3 as the critical and position 5 as the best clocking position. The steady performance measurements find a clocking effect of 0.3% efficiency points, and the URANS simulations an effect of 0.08% efficient points at the peak efficiency operating point. The same is valid for the loaded operating point with efficiency changes of 0.28% and 0.1% respectively.
- Evidence of the measurement uncertainty was highlighted with the help of the clocking position 6 which is theoretically identical with the position 1 but one stator pitch further.
- The limited spatial discretization for the experimental determination of the flow conditions at the compressor outlet were shown numerically to lead to a possible uncertainty of 0.13 efficiency points. Though, this was shown to be possibly over-estimated because of having the numerical outlet positioned one stator chord further upstream than in the real compressor.
- Furthermore, the measurement uncertainty of the total temperature with the pneumatic probes is critical for the estimation of the stage efficiency. All differences between the

clocking positions stay within the measurement uncertainty band for the total temperature. Measurement uncertainties were shown to lead easily to a mis-interpretation of a clocking effect at the example of the stage 1.

- The time mean dissipation (mixing out) of the stator wakes leaves only 8% of the initial (at the trailing edge) total pressure deficit before the interaction with the downstream stator blade row. This mainly explains the small effect of clocking.
- The instantaneous wake transport characteristics (deformation of wakes) along their flow path were shown to contribute to the weakening of the clocking effect. The numerical method needs to resolve this transport correctly for giving an accurate prediction of the clocking effect.
- A radial dependency of the effect of clocking was noted due to the differences in the radial shape between the upstream stator wakes and downstream stator blades. This was identified to lead to local positive and negative effects of clocking on the stage performance, which are eventually almost in equilibrium and cancel each other out in this compressor.
- As a consequence of clocking, a radial redistribution of the flow can occur. The clocking effect can therefore obtain locally an important radial component, which might make it difficult to understand the local effect of clocking.
- At certain span-heights, the wake-wake interaction, or "impingement" of an upstream stator wake on the leading edge of the downstream stator, causes lower losses than the mixing of a stator wake, which is passing at mid-passage through the downstream stator.
- The interaction between the incoming stator wake and the downstream stator blade suction side boundary can lead to higher losses than the mid-passage passing. Especially in the stator 3, the separation is growing due to this interaction at mid-span.
- A global conclusive link does not exist between a clocking position and local changes in unsteadiness or rotor modulation of the flow field in this compressor. If a link existed, it would be smaller than the measurement uncertainty.



# Chapter 5

## Flow instabilities

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The previous Chapters discussed flow field characteristics and effects at stable operating points. This Chapter deals with the instabilities arising in CREATE when throttling the compressor towards surge. Firstly, the tip flow field will be characterized for a loaded and near surge but stable operating point. Secondly, rotating disturbances which are occurring at stable operating points will be discussed with a focus on why numerical and experimental data do not show the same disturbances. In literature, these studies are mostly presented for low-speed compressors due to the constraints that come with a high-speed test rig. That makes it so interesting to see if the knowledge about rotating disturbances are transferable on a high-speed compressor case. Thirdly, the stall inception type of this high-speed compressor will be presented and compared to the rotating disturbances. The originality of this Chapter lies in the acquiring and understanding of a large volume of high quality numerical and experimental data for the study of instabilities in a high-speed multistage compressor. And new original insights will be given into numerically and experimentally identified rotating disturbances.

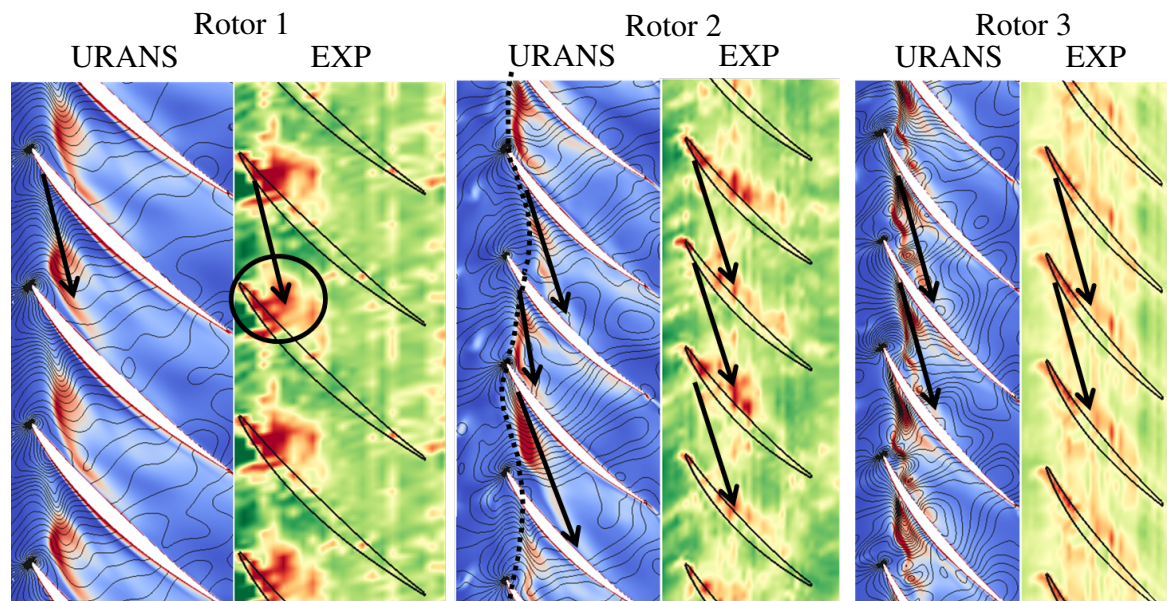


Figure 5.1 Rotors 1, 2, and 3 tip flow field at near surge OP, respectively for each rotor: standard deviation on casing pressure measurement (right), and numerical static pressure contour in black, and numerical rel. velocity gradient in red/blue (left); Red shows in either case a high value

## 5.1 Characterization of the rotor tip flow field at loaded operating points

The characteristics of the rotor tip flow field at loaded and near surge operating point are discussed in this Section, which will give the motivation for the studies about rotating disturbances in Section 5.2, and the spike type stall inception in Section 5.3. The characteristics will be introduced with the help of blade-to-blade cuts - as already presented for the peak efficiency operating point in Figure 3.21 - and a statistical analysis. The general characteristics of the tip clearance vortex trajectories in the blade-to-blade cut are very similar at loaded and near surge operating point. Thus only the blade-to-blade cuts for the near surge operating point are presented in Figure 5.1, showing the biggest difference to the peak efficiency operating point.

The instantaneous maps of the relative velocity gradient visualize the edges of the tip leakage flow induced blockage zone in case of numerical results. Black static pressure contours give additional information about the TCV trajectories. Time averaged standard deviation maps visualize all non-deterministic unsteadiness based on the high-frequency casing pressure measurements. More information about this type of flow field representation was already given in Section 3.3.1.

A steepening of the TCV trajectory is found in the experimental and numerical results for the rotor 1 from nominal to loaded and near surge operating point. In the experimental results, the spots close to the pressure side with elevated standard deviation (high unsteadiness) are moving upstream close to the first 20% of chord. The spatial "overlap" of the elevated



standard deviation spot close to the pressure side of the passages and the elevated standard deviation in the first 20% chord in the tip gap (circled in Figure 5.1) indicate an interaction between the tip leakage flows of two adjacent tip gaps. The numerical results do not indicate such an interaction in the rotor 1, judged by the TCV trajectories.

In the numerical results for the rotor 2, oscillation of the TCV blockage zone can be noted from one passage to another. This oscillation is highlighted with a black dashed line. In certain passages the blockage zone edge is aligned with the LE plane and hence the tip leakage flow of these passages influences the tip leakage flow in the adjacent tip gap. A forward spillage of the tip leakage flow from one passage to another does not occur. This would be visible on the static pressure contours for example, if a recirculation flow was impacting on the adjacent blade's pressure side. Likewise in the rotor 3, an oscillation of the TCV can be noted. The oscillation has a smaller amplitude in the rotor 3 than in the rotor 2, which makes it more difficult to detect in the blade-to-blade cut. The axial evolution of the amplitude will be discussed later on.

Only in the rotor 3, spots of high standard deviation can be found over the whole passage width downstream of the TCV trajectory in the experimental results. That indicates a high unsteadiness, hence blockage over the whole passages width, induced by the tip leakage flow in the rotor 3. That is comparable to what is found in the numerical results. No sign of backward spillage is found at the TE of the rotors 2 and 3 in the numerical or experimental results.

The validity of the characterization of the tip flow field (presented with the casing pressure measurements) shall be supported with a comparison between URANS results and LDA measurement data at the axial position 2 at 96.3% span-height in the rotor 2 passage at the near surge operating point. This span-height and axial position is chosen as an example, the other measurement locations in the rotor 2 tip show the same behavior. For this position, the axial velocity fluctuations are presented in Figure 5.2. Fluctuations are defined around the mean value of the signal of each respective database. The URANS results show in blue the time averaged data of one spatial period ( $\frac{2\pi}{16}$ ). The red line gives a instantaneous snapshot of the axial velocity fluctuations based on URANS data. The LDA data is given with the green dotted line and shows the axial velocity fluctuations over one temporal period ( $\frac{2\pi}{16\Omega}$ ). The comparison of data over time and space in the rotors has been demonstrated to be valid in Section 3.3.1. The influence of the circumferential position (stator influence) on the signal over time is small compared to the studied differences here.

Globally, the time averaged URANS and the LDA data show a good match in blockage induced by the tip leakage flow at this location. The tip clearance vortex trajectory matches depending on the passages. Differences between the passages can exist because of the rotor clocking (URANS) and RSI (LDA). The instantaneous snapshot shows an increased (almost doubled) blockage in certain passages. As already seen in Figure 5.1, the modulation of the rotor 2 tip flow field is characterized by a mode 2 over one spatial period. This is mainly seen with the instantaneous snapshot (red signal) as important velocity deficit in the passages 1 and 4 (from the left to the right). The shape of the velocity deficits is not identical because of the additional modulation by the RSI and a mode 2 in a rotor with 5 blades per spatial period extends twice over 2.5 passages.



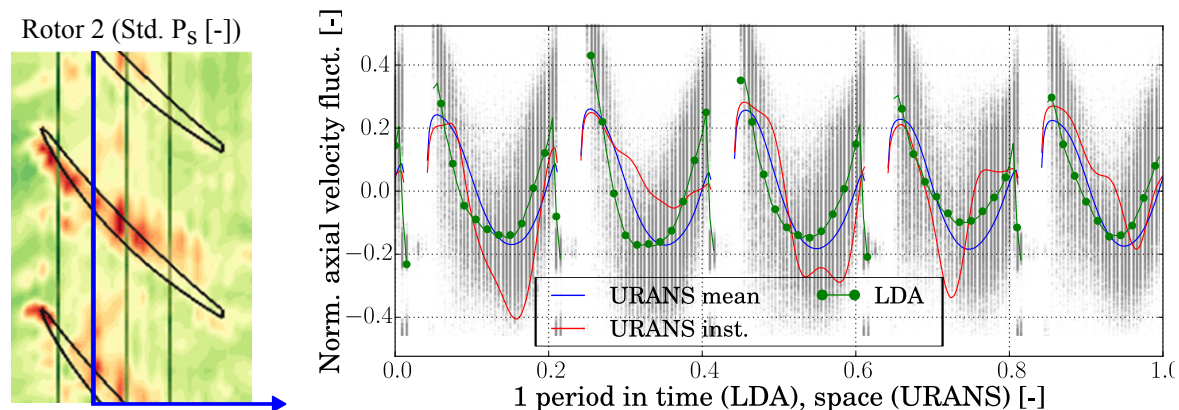


Figure 5.2 Axial velocity fluctuations at 96.3% span-height in the rotor 2 at the axial position 2 (indicated in the map on the left) in a comparison between URANS (time) mean, URANS instantaneous, and LDA data

There can be non-deterministic oscillations of the tip clearance vortex which cannot be captured by the LDA due to the measurement technique. However, the modulated tip flow field (instantaneous signal) overlaps with rather the edges of the grey particle envelope at certain locations. This shows that the modulation - especially the large velocity deficit at certain locations - does not represent a frequently reoccurring state of the measured flow field. Furthermore, the high-frequency wall pressure measurements do not reveal any such flow field modulation. In conclusion, the URANS method predicts a deterministic rotor flow field modulation, which does not seem to occur in the actual compressor (measurements).

### Statistical analysis of the tip unsteadiness

The unsteadiness of the rotor tip flow field shall be quantified for three different rotation speeds: 100% shaft speed ("100%  $\Omega$ "), 90%  $\Omega$ , and 80%  $\Omega$  over several rotations just before stall inception. As introduced earlier in Section 4.2.1, the stall/surge inception is reached by closing very slowly the outlet throttle valve, starting from the position for the last stable operating point. The throttle valve is closed with such a low speed that it allows quasi stationary flow conditions. This Section shall give an overview over arising instabilities at these different rotation speeds and introduce the subsequent Sections of this Chapter. High-frequency casing pressure measurements above the rotors 1 to 3 are used for this analysis. The standard deviation is computed on about 5000 registered data points in each of the 200 ensemble average window. This corresponds to a signal length of about 400 rotations with an acquisition frequency of 500 kHz (see Section 2.2.3). The temporal period will be then resolved with 200 standard deviation values. Per rotor tip, the casing pressure is measured at up to 12 axial positions. The standard deviation is presented in two different ways in Figure 5.3. Firstly, the mean standard deviation is calculated over one temporal period, and then for the sake of having a better overview, the averaged standard deviation is once more averaged over all axial positions. Secondly, the same steps are done but with the maximum

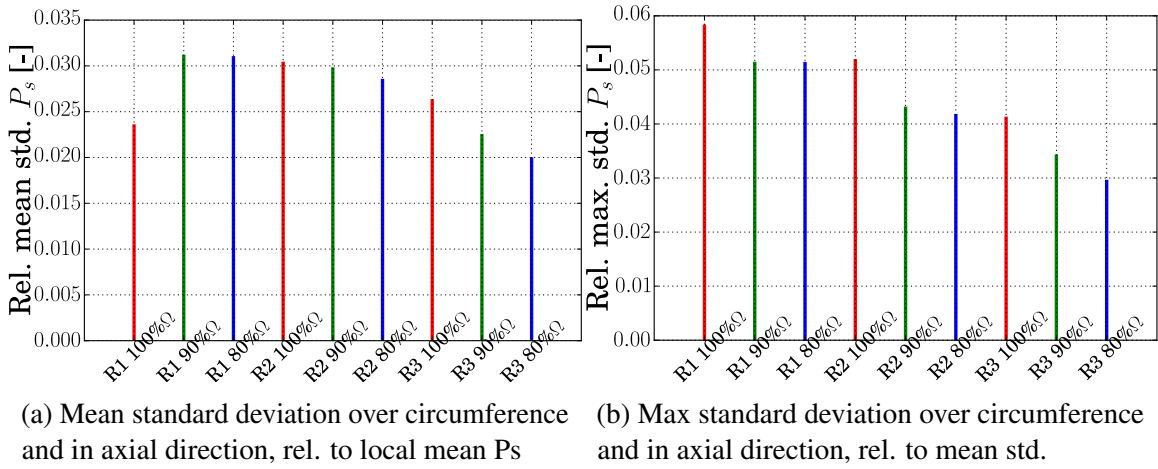


Figure 5.3 Tip unsteadiness in the rotors 1 to 3 based on standard deviation computed from casing pressure measurements

standard deviation. The standard deviation is calculated at all steps relative to the mean static pressure at the given position.

Note the decreasing of the mean standard deviation for the rotors 2 and 3 with the shaft speed. This is the most pronounced in the rotor 3. The rotor 1 stands for an exception, where the unsteadiness - standard deviation - goes to a higher level from 100%  $\Omega$  to an about constant level at the two speeds 90% and 80%  $\Omega$ . The increasing non-deterministic unsteadiness (standard deviation) is a sign for an arising instability, which will be discussed in Section 5.2.3.

The maximum standard deviation are always found at 100%  $\Omega$  (red bars), and the maximum standard deviation is decreasing with the rotation speed. The rotor 1 shows the highest values for the maximum standard deviation. Those maximum values origin from the first third of the chord (not shown here). Signs of this high non-deterministic unsteadiness was also indicated with the possible interaction between the tip clearance vortex and the adjacent tip gap leakage flow in Figure 5.1. A local very high standard deviation can occur because of small spike like reoccurring disturbances. The significance of the unsteadiness will be discussed in Section 5.3 related to the spike type rotating stall inception, which is detected at 100%  $\Omega$ .

## 5.2 Rotating disturbances at stable operating point

Evidence of rotating disturbances of different type are found in the URANS simulation at nominal shaft speed and in the casing pressure measurement at part speed. It will be shown that they have characteristics in common but the URANS simulations predict them in the rotors 2 and 3, while the measurements find them in the rotor 1.

### 5.2.1 Predicted rotating disturbance at nominal speed

The simulations for the loaded operating point were not conducted with the intention of studying a rotating disturbance. This explains why the simulation domain only contains a  $\frac{2\pi}{16}$  sector with periodic lateral boundaries as also for the peak efficiency simulations. A full annulus simulations would be maybe more adapted for avoiding a forcing of the periodicity to the flow field. It will be shown later on with the help of previous study results that the mesh extend does not have an important impact on the flow field characteristics in this compressor case. A global indicator of a rotating disturbance is found as mass flow rate oscillations, which cannot be explained by any combination of the blade row influences. The frequency spectra of the mass flow rate signals from the rotor 2 outlet (in the rotating frame) over one rotation are presented for the peak efficiency and loaded operating point in Figure 5.4 as an example. The frequencies are normalized by the shaft speed ( $\Omega$ ). The blade row related frequencies appear naturally as most important amplitudes here, for example the rotor 1 with 64 (= 64 blades), the rotor 2 and 3 with 80, and the stators with 112. These oscillations are induced by the rotating of one wake row relative to the others. Thus, these frequencies can also be understood as RSI frequencies.

A numerical probe would find that the rotating disturbance induces a temporal mode 23 in the stator reference frame, and mode 9 in the rotor reference frame (not shown here). Both frequencies appear in the mass flow rate spectrum as normalized ( $\frac{1}{\Omega}$ ) frequencies 9 and 103 (80+23). Their amplitude is more than 2.3 times smaller than the one induced by the rotor 1 influence. These frequencies do not correspond to any frequencies that could be caused by known RSI in CREATE. Their origin shall be discussed in the following Sections.

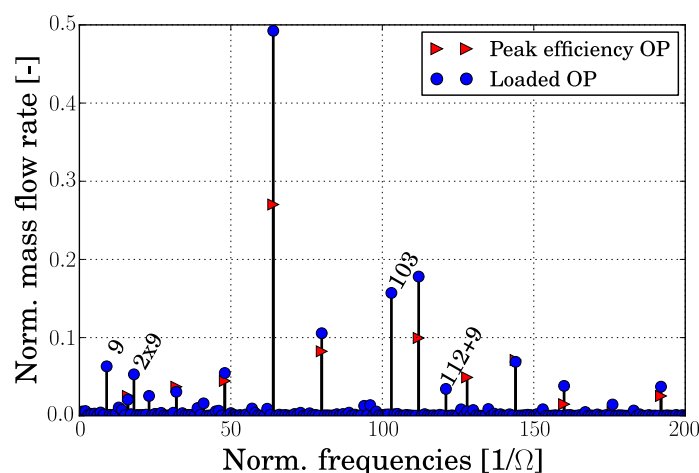


Figure 5.4 Frequency spectra of mass flow rate signal over one rotation for the peak efficiency and loaded operating point at the rotor 2 outlet

### Disturbance in flow field

The tip leakage oscillation (Figure 5.2) occurs only in the unsteady numerical results and not in the measurements (Figure 5.1) at loaded and near surge operating points. This is identified

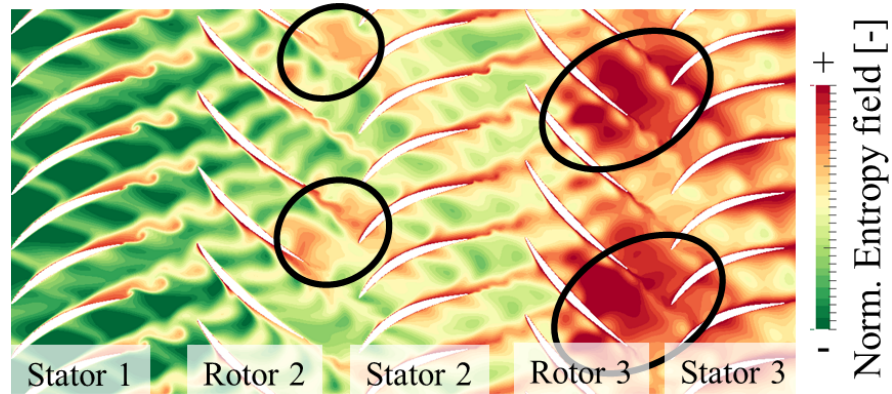


Figure 5.5 Normalized instantaneous entropy URANS field at 82% span-height, high entropy spots induced by the rotating disturbance are circled in black

to cause the previously mentioned mass flow rate oscillations. That will be elaborated in the following Sections of this Chapter.

The unsteady numerical results predict a rotating disturbance in the tip flow field (20% of span towards casing) in the rotors 2 and 3. This can be seen in the normalized entropy distribution at 82% span-height for the flow field from the stator 1 to 3 in Figure 5.5. The disturbance is found as 2 cells within the spatial period ( $\frac{2\pi}{16}$ ). This corresponds to 32 cells around the full annulus. The number of cells could correspond to a direct interaction between the surrounding stators and the rotors ( $112-80=32$ ). Note that the disturbance is most visible between the mid-chord of the rotor 3 and the stator 3 because of the chosen span-height. Each cell or elevated entropy spot extends over 2 to 3 rotor passages. A rotating disturbance with 2 cells is also identified in the rotor 2 but these cells are more difficult to detect on the entropy map at this span-height.

The flow field can be filtered with the help of temporal discrete Fourier transforms and their inverse transforms in order to keep only the contribution of these frequencies. That treatment is done at each grid point. It allows the estimation of the rotation speed of the disturbance. The phase change of the spatial mode 2 over time can be evaluated because it is only caused by the rotating disturbance in the filtered field. The rotation speed is equal to  $0.7188\Omega$  which corresponds to  $\frac{23}{32}\Omega$ .

Figure 5.6 presents the spatial distribution of the disturbance in a meridian map based on a filtered flow field. It is built using the mean absolute amplitude of the disturbance over the circumference. The disturbance reaches at maximum an amplitude magnitude comparable to the one of the BPF. The maximum amplitude is found in the tip region of the rotors 2 and 3 with a peak towards the LE. The shape of the zone with the increased amplitude shows the link between the disturbance and the tip leakage flow. The zone stays confined to the tip of the rotor blade row, and does not show any influence upstream of it. This would be expected in case of rotating stall in the rotor blade row as introduced in Section 1.4. The shape is characteristic for the tip leakage flow which is diving into the vane from LE to TE of the rotors. The "diving" explains why the elevated entropy spots are found in the aft part of the blade at 82% span-height in Figure 5.5.

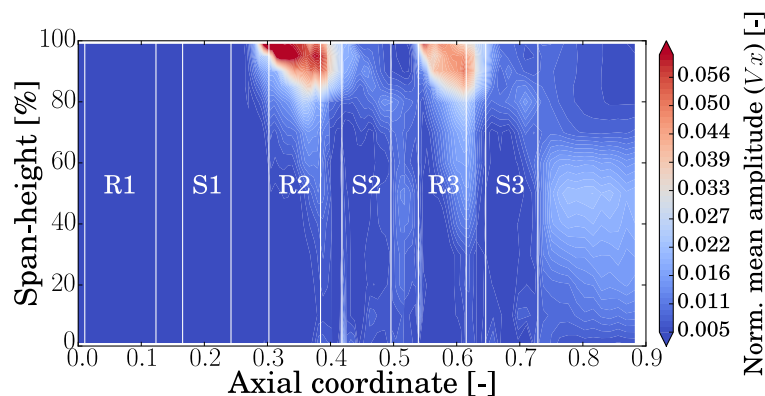


Figure 5.6 Spatial distribution of the rotating disturbances, based on URANS axial velocity at the loaded operating, LE/TE in white simplified solid lines

Only weak traces of the disturbance are found below 80% span-height. Though weak, a modulation of the mid-span flow field by the rotating disturbance can be noted starting from the rotor 2 in stream-wise direction in Figure 5.6. The light blue zone downstream of the stator 3 occurs because of the interaction between the rotating disturbance origin in the rotor 3 and a large separated blade boundary layer on the suction side of the stator 3 (see Section 3.4.2).

Gourdain et al. (2012) found numerically a similar rotating disturbance in the rotor 3 with a spatial extension of 20% on a previous version of CREATE (with different blade design), see Figure 12. Only the stators 2 and 3, and the rotors 1 and 3 had the same blade count as in the current version of CREATE (see Table 2.2). Moreover their simulation domain did not include the IGV. Nevertheless, the authors observed the same cell number (32) with a rotation speed of  $0.66\Omega$  (compared to  $0.72\Omega$  in the present work) and understood it as a rotating-stall phenomenon. Furthermore, they found that the disturbance would occur with the same number of cells not only in a simulation on the spatial period ( $\frac{2\pi}{16}$ ) but also in a simulation that takes the full annulus into account. However, the rotation disturbance was not studied in further detail because it was not the main objective of their publication.

A further numerical work was conducted by Crevel et al. (2014) on the same previous version of CREATE, taking into account the IGV in the simulation domain. They observed basically the same rotating disturbance but with 24 cells. In their second test case, only 16 cells occurred with the full test rig in the simulation domain. Here the authors noted rotation speeds of  $0.78$  and  $0.8\Omega$  respectively, which correspond to temporal modes of  $\frac{19}{24}\Omega$  and  $\frac{12.9}{16}\Omega$ . Crevel et al. (2014) suggested that including the IGV in the simulation domain could change the local response in terms of rotating stall cells from 32 cells to 24 or 16 cells. The present numerical work (32 cells) proves this statement wrong with the IGV in the simulation domain. Therefore the cause for the change of number of cells might need to be searched in the influence of the imposed spatial periodicity. For the present work, a spatial periodicity of  $\frac{2\pi}{16}$  was used compared to  $\frac{2\pi}{8}$  in the work of Crevel et al. (2014). A blade count change does not appear to have an influence on the cell number because the



numerical works of Gourdain et al. (2012) and Crevel et al. (2014) were conducted with an identical compressor version.

In any case, the number of cells between 16 and 32 appears one order of magnitude larger than expected for rotating stall (see Section 1.4.1). Furthermore, the meridian view (Figure 5.6) does not show any significant additional blockage induced by the rotating disturbance in average over the circumference. That would be expected from rotating stall. Standard tip gaps are used in this thesis work compared to increased tip gaps in the rotors 2 and 3 in the previous work. These increased gaps were believed to provoke the disturbance. Answering to the questions, whether or not the rotating disturbance is actually rotating stall, and what is the source of it, is the motivation of the next Sections.

### Independence of spatial discretization

Independently from the results of Gourdain et al. (2012) and Crevel et al. (2014) which have shown those rotating disturbances using different simulation meshes, the influence of the mesh has been evaluated for the present case with the help of a second mesh. This mesh has a slightly increased first cells size close to the hub and casing (ratio 5 to 1), which gives still a non-dimensional computed wall distance  $y^+$  of about 1 for most part of the computation domain. An increased first wall cells size allows the redistribution of the grid points. The grid is densified in the radial direction in the zone of the rotating disturbance while keeping an identical grid point number due to computation resource constraints. Furthermore, the outlet of the simulation domain is placed half a stator chord further downstream to reduce the likelihood of numerical reflections at the outlet boundary.

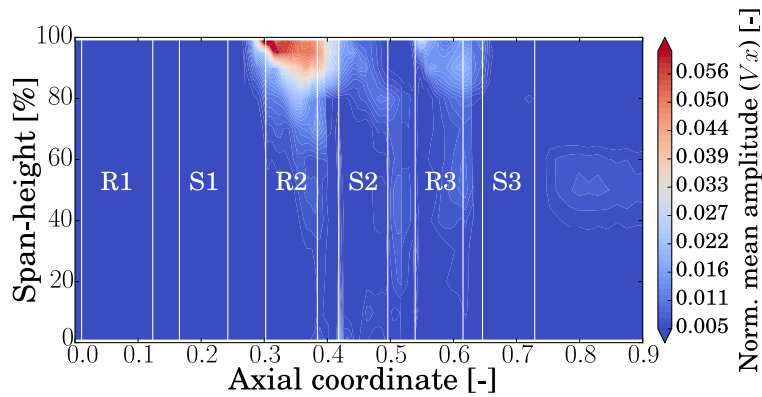


Figure 5.7 Spatial distribution of the rotating disturbance with redistributed grid points, based on URANS axial velocity at the loaded operating, LE/TE in white simplified solid lines

The resulting field at the loaded operating point shows the same rotating disturbance with 32 cells, with a small change in rotation speed from  $\frac{23}{32}\Omega$  to  $\frac{22}{32}\Omega$ . Furthermore, the amplitude of the disturbance decreased slightly in both of the rotors but is still present (smaller decrease in the rotor 2), see Figure 5.7. As a consequence the interaction is weaker between the disturbance and the separation at mid-span in the stator 3. A numerical identical valve position has been used for mesh independence study simulation, but the shift of the

outlet does change the system losses. The study has the object of verifying that the main characteristics of the rotating disturbance would not change with the mesh. Hence a small shift of +0.2% in mass flow rate has been accepted, which can explain the weaker amplitude of the disturbance in the rotor 3 as well. Consequently the interaction between the rotating disturbance in the rotor 3 and the separation might be weaker in the stator 3 at mid-span. The small shift in rotation speed cannot be explained before completely understanding the source and propagation mechanism of the rotating disturbance.

In summary, this study shows that independent from the mesh the rotating disturbance would occur with almost identical characteristics. Small changes are noted but a vanishing of the disturbance in the rotor 2 is not expected with an even finer mesh, judged by the small change in the disturbance amplitude here. The results are not elaborated any further because it does not give any additional insight in the rotating disturbance.

### Propagation mechanism

The propagation mechanism of the rotating disturbance has been previously indicated with the analysis of the rotor tip flow field (Figure 5.1). Mailach et al. (2000) observed a similar TCV trajectory oscillation in an experimental work, which is visualized in a sketch in Figure 5.8 together with the filtered static pressure field in the rotor 2 tip of CREATE.

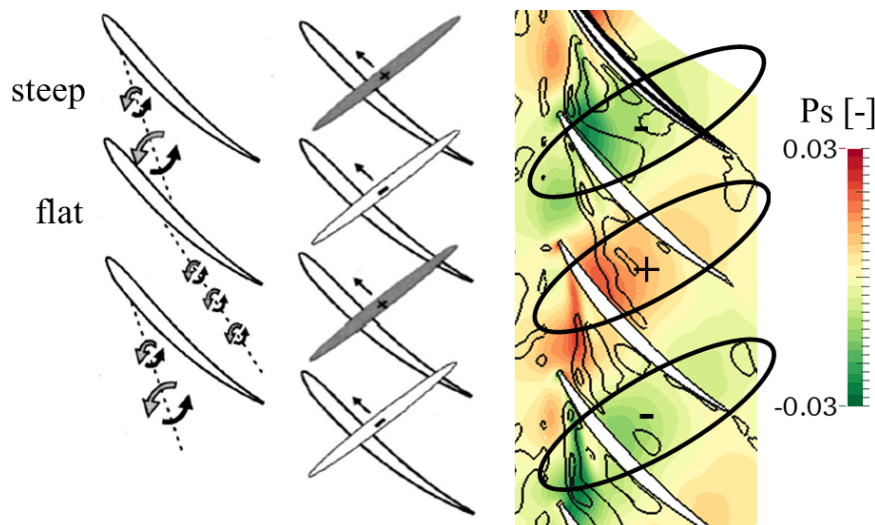


Figure 5.8 TCV oscillation (left), and propagation of "rotating instability" (middle) as proposed by Mailach et al. (2000), and filtered CREATE static pressure URANS rotor 2 tip flow field, 98% span-height (right)

The left sketch shows an alternation of steep and flat trajectories as well as a change in the chord-wise spillage point of the tip leakage flow. The authors related the steep trajectory (more aligned with the inlet plane) to a stronger TCV and the flat trajectory to a weaker TCV. The TCV oscillation causes alternating spots of low and high pressure, as illustrated with the sketch in the center. The steep TCV trajectory causes a low static pressure spot, and the flat trajectory the high static pressure spot. The same alternating spots are found in the filtered

(for disturbance) static pressure field (same filter mechanism as for Figure 5.6 and Figure 5.7). The filtered field is obtained with a blade-to-blade cut in the rotor 2 at 98% span-height. The tip clearance vortex location is indicated with contours of the relative velocity gradient. The link between low and high pressure spots and the steep and flat trajectories respectively can be noted in the filtered CREATE flow field as well. Nevertheless, the principal flow direction is still axial and this disturbance is only a modulation of the flow field.

The circumferential propagation mechanism shall be discussed in more detail with the help of Figure 5.9. Again, the blade-to-blade cut of the filtered (for disturbance) static pressure flow field is presented for the rotor 2. Black contours indicate the gradient in the relative velocity field. Three arrows mark in a simplified manner three different types of tip clearance vortex trajectories. They are labeled "flat", "transition", and "steep" trajectory. The trajectory is oscillating between these different types periodically. A steep trajectory leads to an alignment of the tip clearance vortex with the inlet plane.

At the top in the same Figure, the time mean leakage mass flow rate is given as fluctuations around the overall mean leakage mass flow rate for the first 20% of axial chord for each respective tip gap. The tip leakage mass flow rate variations are in the order of  $\pm 33\%$  of the time mean tip leakage mass flow rate. The leakage mass flow rate decreases when the trajectory is the steepest (gap 1 and gap 4). This goes together with an acceleration of the fluid in the circumferential direction, which is found as low static pressure spot (green). That causes a weakening of the pressure gradient through the adjacent tip gap and hence a weaker TCV here. This is then found as a transition type trajectory. The transition and flat trajectories (related to the high pressure spots, e.g. gap 2 and 3) cause an increase of the pressure gradient through the adjacent tip gap and consequently a steepening of the TCV. By modulating always the tip leakage flow in the adjacent tip gap, the TCV trajectory oscillation appears actually as a rotating mode. A spillage of the tip leakage flow does not occur from one passage to another. The characteristic of the tip clearance vortex in the rotor 3 are the same but with a smaller oscillation amplitude.

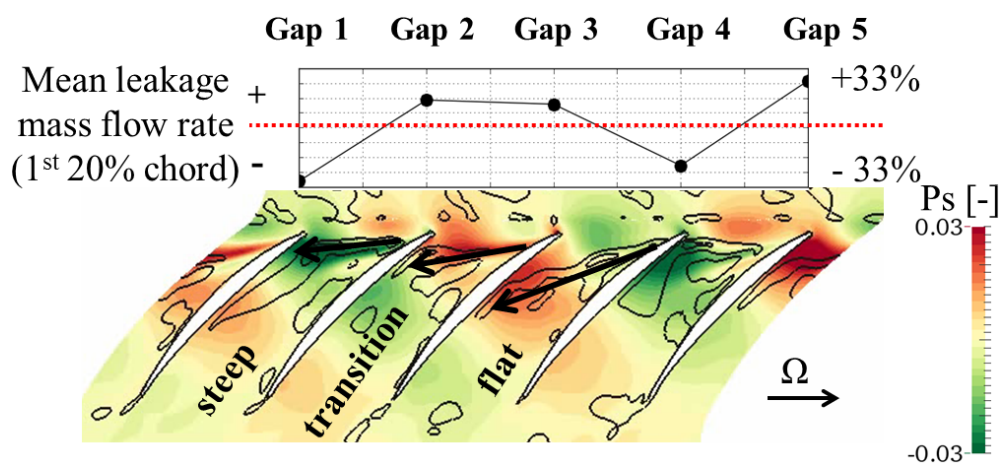


Figure 5.9 Circumferential propagation mechanism of the rotating disturbance explained with tip gap leakage mass flow rate variation at loaded operating point and the filtered (for disturbance) static pressure field in the rotor 2 at 98% span-height



The work of Mailach et al. (2000) shows that the resulting rotating disturbance is characterized by a bump in the frequency spectra of high-frequency static pressure sensors installed at the casing (not shown here). This suggests that the rotating disturbance changes the number of cells and speed in an unstable way over time. The authors call the rotating disturbance a "rotating instability" because of its unstable behavior. However, the compressor is still found to operate in a stable way. The frequency bump or rotating instability is reported in many other works but mostly for low-speed compressor cases with enlarged tip gaps as introduced in Section 1.4.3.

In the URANS results for CREATE, the frequency does not change over time but all other characteristics of the rotating instability apply. With the imposed spatial periodicity and time constant compressor inlet conditions, it is not surprising that the frequency stays constant in the present numerical work. Though the numerical work of Zhang et al. (2012) on a low pressure turbine stage resolved multiple frequency peaks for a rotating instability. Their full-annulus simulation domain contained only a 2D blade-to-blade section at 90% span-height. Consequently no tip gap is resolved. At low mass flow rate the turbine is working locally as a compressor and the rotating instability would form. A separated suction side blade boundary is shed non-synchronized to the rotation frequency into the rotor passages, and creates a circumferential non-uniform rotating structure. This mechanism appears to be different to the separation vortex which has been reported for compressors to interact with the tip leakage flow or a corner separation (Section 1.4.3). An URANS resolved frequency bump has not been reported for compressors in literature.

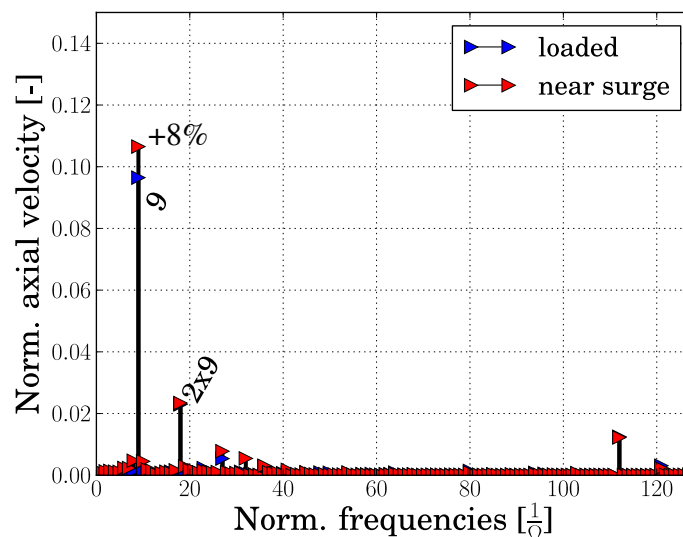


Figure 5.10 Amplitude of the rotating disturbance at loaded (blue) and near surge (red) operating point calculated with a signal of a numerical probe located downstream of the rotor 2 at 98% span-height in the relative frame (URANS results)

### Evolution of the rotating disturbance from the loaded to the near surge operating point

The near surge operating point is marked in Figure 3.1 and is close to the last numerical stable operating point with the given numerical boundary conditions, which do not allow backflow at the outlet of the simulation domain. The rotating disturbance is known to be present at the loaded and near surge operating point in terms of TCV trajectory oscillations. The number of cells and rotation speed is found to stay constant for the loaded and near surge operating point (not further elaborated here). Therefore a direct comparison of the amplitude of the temporal mode 9 (temporal rotating disturbance mode in rotor frame) is possible between the loaded and near surge operating point. Figure 5.10 shows the spectra obtained with discrete Fourier transforms on the signal of a numeric sensor placed downstream of the rotor 2 at 98% span-height. The amplitude of the rotating disturbance (mode 9) increases by 8% with decreasing mass flow rate from loaded to near surge operating point. The second harmonic of the rotating disturbance frequency (mode 18) does not change at this scale.

### Onset of the instability and influence of rotor-stator interactions

The previous works on CREATE (Crevel et al., 2014; Gourdain et al., 2012) assumed that the rotating disturbance would be initially caused by a RSI mode ( $112-80=32$ ) because of the fitting blade numbers. The RSI was assumed to create local changes to the incidence angle and provoking the rotating stall cells. No such relation between the blade number and the number of cells can be found for rotating stall or a rotating instability in further literature. As examples, März et al. (2002) noted for a rotating instability 30 to 35 cells in a compressor having 24 rotor blades and 17 stators blades. Mailach et al. (2000) observed 10 to 40 cells and the maximum disturbance amplitude with 28 cells in a compressor with 63 rotor and 83 stator blades. Furthermore, Pardowitz et al. (2015) found 18 to 25 cells, with 24 being dominant, in a compressor with 24 rotor blades and 12 stator blades.

The influence of the RSI on the development of the rotating disturbance shall be numerically analyzed by replacing the sliding mesh interface treatment (Mohan Rai (1986)) between the rotor 2 row and the surrounding stator rows by the mixing plane method. This method computes azimuthal averages based on Riemann invariants at the row interfaces before transferring the aerodynamic quantities to the neighboring row. This study makes use of the azimuthal average as filter for the axial (mixing plane at all row interfaces) and circumferential modes of the RSI.

Several cases have been investigated from which two will be presented here. In a first case only the circumferential RSI modes will be filtered, and in the second case the axial and circumferential RSI modes:

- Case 1 has mixing planes at both interfaces upstream and downstream of the rotor 2, and the simulation has been restarted with a peak efficiency flow field. This field does not contain the rotation disturbance frequency but the frequencies of the RSI and stator influence. This case filters only the circumferential modes at the rotor 2 interface. But the case lets pass all axial modes that are present in the compressor because of the RSI.

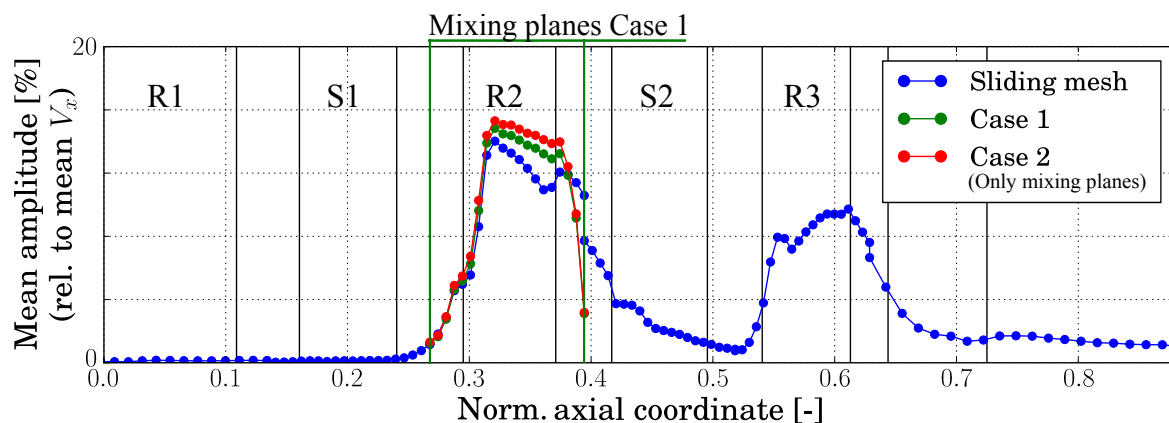


Figure 5.11 Axial evolution of rotating disturbance amplitude (axial velocity), LE/TE as black vertical lines, mixing planes indicated in red, 98% span-height (based on URANS results)

- Case 2 has mixing planes at all row interfaces but is nevertheless conducted as a URANS simulation (with a global time step). This simulation has been restarted from a steady RANS flow at the peak efficiency operating point (step 1). In a second step it was re-restarted from a steady RANS field at the loaded operating point (step 2). With mixing planes at all interfaces, axial modes cannot be produced by the RSI because the rows are no longer "rotating" relative to each other.

Figure 5.11 shows the axial evolution of the normalized rotating disturbance amplitude for the case 1 in green, the case 2 in red, and the original sliding mesh version in blue. The rotating disturbance is still present, basically unchanged, with the mixing planes upstream and downstream the rotor 2 (case 1 and case 2) compared to the original sliding mesh version. Therefore, the mode number and also the rotating disturbance itself cannot be a consequence of any circumferential RSI mode. In other words, the interaction "112-80=32" is not a possible explanation for the rotating disturbance mode number.

The axial RSI modes are caused by the relative motion of the blade rows in front of each other and cause a pulsation of the mass flow rate. Courtiade et al. (2012) state that whatever the spatial mode of a RSI, the temporal pulsation of a RSI is always a multiple of the blade number of the rotors. Here, the main axial modes are 64 and 80, which correspond to the temporal RSI modes induced by the blades of the rotor 1 (64) and the rotor 2 and 3 (80). The amplitude of these modes is given over the compressor axis using the mass flow rate signals at the row interfaces in the URANS domain in Figure 5.12a. The modes 64 (circles) and 80 (crosses) are presented on the vertical axis. With the case 2 (all mixing planes), the axial modes cannot form because the mixing planes filter all relative blade row motion influences. The interaction mode 80 gets surprisingly filtered with the mixing planes in the case 1, which means this interaction mode depends on the rotor 2 blade row. The stage 3 is nevertheless expected to induce an interaction mode 80 but this mode gets filtered here. The axial mode 64 is created by the stage 1, transported downstream through the compressor, and is not filtered by the mixing planes in case 1.

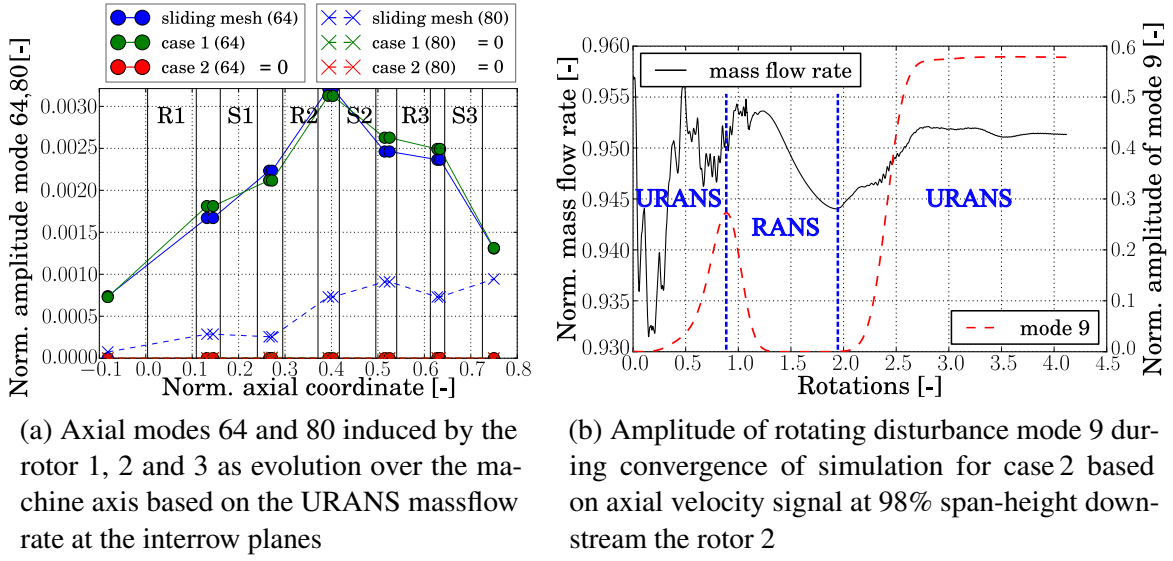


Figure 5.12 Study of source of rotating disturbance

The mass flow rate during the convergence of the simulation for the case 2 is given in Figure 5.12b on the left vertical axis. Furthermore, the amplitude of the mode 9 (temporal rotating disturbance mode in rotor frame) is given on the right vertical axis. The amplitude is obtained from a wavelet transform applied to the axial velocity signal obtained with a numerical probe placed at 98% span-height downstream the rotor 2 in the rotor frame. Just before the rotation "0", the numerical valve position is changed from the peak efficiency to the near surge operating point position. Looking at the mass flow rate, the convergence process (mass flow rate oscillation) is seen up to about 1.25 rotations (called step 1 earlier), where the simulation is conducted with URANS until rotation 0.9. Then during one rotation from 0.9 to 1.9, the simulation is conducted as steady one with a local timestep before being restarted as a URANS one (step 2). During the steps 1 and 2 the rotation disturbance mode 9 (red dashed line) rises in amplitude. As expected the rotating disturbance gets eliminated by the RANS simulation part. This was done to ensure that a possible axial mode induced by the valve position changing (before rotation 0) is not responsible for the occurring of the rotation disturbance. In conclusion, the case 2 successfully avoids the creation of the axial mode 64 but eventually not the rotating disturbance. Moreover, the rotating disturbance does not arise due to circumferential nor axial rotor-stator interaction modes. Another blade row is not necessary for the onset of this rotating disturbance, it can be considered auto-induced in the rotor blade row.

**Verification with a simulation containing only the rotor 2** The results of the auto-induced rotating disturbance have been verified with a simulation on a domain containing only the rotor 2 and extended inlet and outlet ducts following the actual compressor duct design. Mixing planes are positioned between the rotor mesh and the inlet and outlet ducts. Periodic circumferential boundaries are used. The meshes for the inlet and outlet blocks are gradually coarsen towards the extremities of the domain. This shall avoid the influence

of any reflections at the boundaries of the domain. Steady circumferentially averaged inlet conditions to this simulation domain have been derived from the flow field (at plane 270) at the loaded operating point from the simulation on the full compressor domain. The numerical outlet throttle valve has been adjusted so to match the mass flow rate of the full domain simulation at the loaded operating point. The rotor 2 mesh discretization and the numerical parameters are the same as introduced for the full domain simulations (Section 2.3).

Figure 5.13 shows the entropy fluctuations in a blade-to-blade cut for one spatial period of CREATE ( $\frac{2\pi}{16}$ ) at 98% span-height for the isolated rotor 2 simulation. Furthermore, the Figure presents black contours of the relative velocity gradient for the identification of the outline of the blockage induced by the tip leakage flow. The two spots of high entropy represent again the spatial mode 2 of the rotating disturbance, as already presented for the full simulation domain in Figure 5.5. Furthermore, the rotating speed has been identified (not detailed here) as unchanged compared to the full domain simulation ( $\frac{23}{32}\Omega$ ). Forward spillage of tip leakage flow occurs where the tip clearance vortex is aligned with the inlet plane of the rotor (high entropy zone ahead of the rotor blade LE). This does not occur in the full domain simulation and might be explained by small differences in the incidence angle to the rotor tip between the two simulation types. Though, the spillage does not have any influence on the rotating speed nor spatial mode of the rotating disturbance.

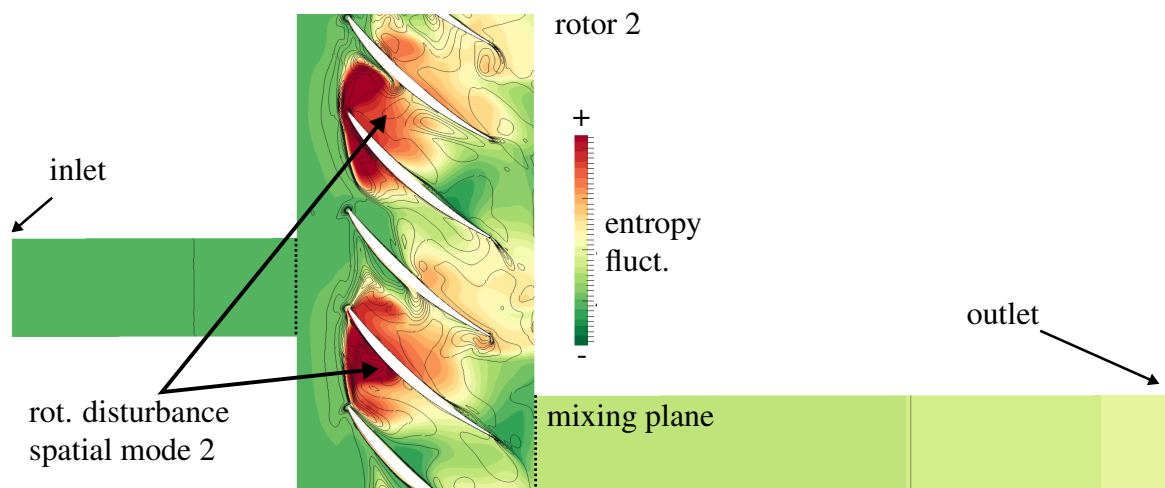


Figure 5.13 Entropy fluctuation field (color map, relative to overall entropy rise in the compressor) and relative velocity gradient field (black contours) at 98% span-height of the isolated rotor 2 simulation at loaded operating point mass flow rate

This study shows clearly that the rotating disturbance appears independent from any rotor-stator interaction in this high-speed compressor. Moreover, an isolated rotor simulation allows capturing almost unchanged characteristics of the rotating disturbance compared to the simulation on the multistage domain. Future studies about the rotating disturbance could be therefore conducted on an isolated rotor. That would allow less costly verification of the influence of different turbulence models and more advanced numerical methods such as ZDES on the rotating disturbance.

**Trigger of the rotating disturbance** A possible trigger for the rotating disturbance has been identified. In a case with mixing planes up and downstream of the rotor 2 (similar to the previously mentioned case 1), the URANS simulation is restarted from a steady RANS field (obtained at the peak efficiency operating point) with a numerical throttle valve position for the loaded operating point, and extractions of the flow field are realized during the convergence process. Several meaningful instants are selected and presented in Figure 5.14. The axial velocity field is presented as difference to an ensemble averaged reference passage for the rotor 2 (calculated at each presented instant). Flow structures are highlighted with a threshold for visualizing velocities from  $\pm 0.07\%$  to  $\pm 3.5\%$  of the mean axial velocity across the rotor. The latter one is a threshold close to the measurement uncertainties. Positive and negative velocity fluctuations are marked with blue and red respectively, but the sense of the fluctuations is of no importance here because they coexist. An entropy fluctuation field would give the same information.

- At  $t = 0$ , "imperfections" are found which are actually very small. The largest initial velocity differences reach a level of about  $0.07\%$  of the mean axial velocity. They form just after the restart of the simulation and are linked to the tip leakage flow. As they stay at a fixed location in a rotor passage, they actually rotate at rotor speed. Furthermore, smaller differences form in the blade boundary layers but are filtered because of the threshold applied for the illustration. They result from the modulated tip leakage flow.
- The zone of imperfections or the disturbance grows during 0.14 revolutions until the positive and negative zones overlap for the first time with a structure of considerable size (black circled at  $t = 0.14$ ). This marks the starting point of a rotating structure at a speed different than the one of the rotor.
- Between 0.14 and 0.3 revolutions the structures start rotating all together. A spatial mode 2 (or 32 over the whole circumference) has not formed yet.
- After 0.6 revolutions, the rotating disturbance mode 2 has formed from the initial very small imperfection or disturbance.

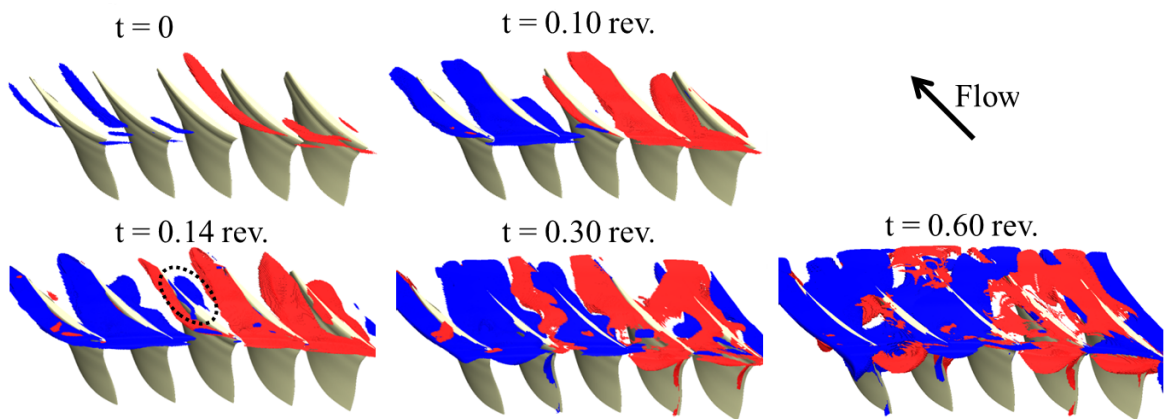


Figure 5.14 Rotor 2 axial velocity field compared to ensemble averaged reference passage after closing numerical valve from peak efficiency to loaded operating point, (blue: higher and red: lower than average)

The initially small imperfection (rotating at rotor speed) describes a possible trigger for the rotating disturbance which eventually rotates at a fraction of the rotor speed. The imperfection arises during the convergence process in simulations restarted from a steady or unsteady flow field and is therefore considered physical. The mesh has been verified and has been excluded as source for the imperfection. The error induced by the multiplication of the mesh from the single passage (RANS) to the multi passage mesh (URANS) is smaller than  $1 \times 10^{-17}$  m for the casing radius compared to the smallest cell size of  $1 \times 10^{-6}$  m for example. The instability excitation is favored by the tendency of the URANS method to form well-defined flow structures. Physically, the imperfection could be induced by variations of the blade tip gap size, stagger angle, or an eccentric casing. They have been demonstrated to be a possible trigger for instability inceptions. For example Weichert and Day (2013) could show that a spike type inception would be caused in 90% of the inceptions in the same passage. Likewise, Young et al. (2012) identified that the instability would form always at the rotor blade with the largest tip gap (compared to all other gaps in a rotor blade row). Note that the purely numerical imperfection in the case of CREATE does not cause any leading edge separation vortex as commonly observed during the onset of a spike or rotating instability (Figure 1.20). During the formation of a separation vortex, the disturbed boundary layer rotates at the rotor speed. That is a shared characteristic with the initial imperfection found in CREATE.

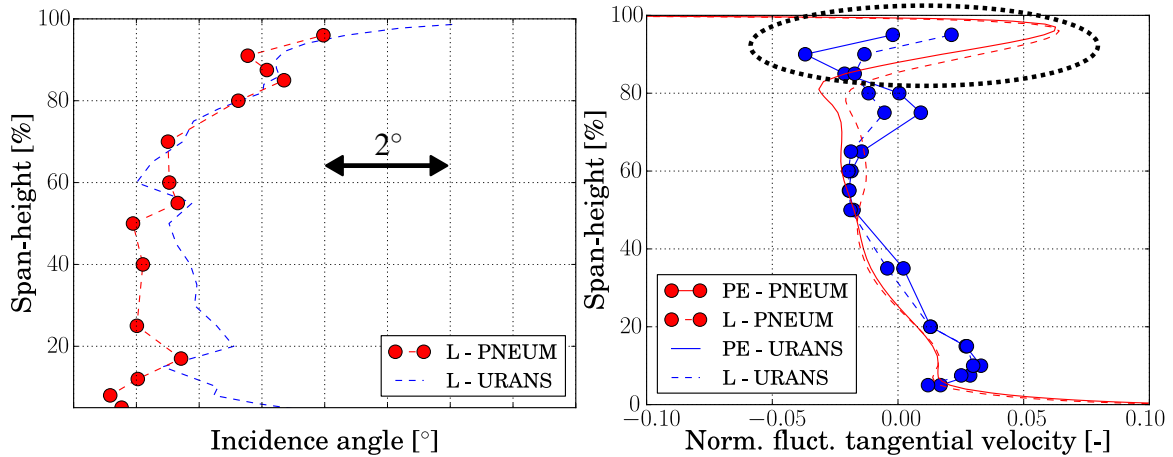
### 5.2.2 Discrepancy between measurement and prediction at nominal speed close to stall inception

An over-prediction of the blockage in the rotor tip regions was detailed in Section 3.3.1 for the peak efficiency operating point. Then it was shown with the blockage zone (relative velocity gradient) for the near surge operating point in the blade-to-blade cut in Figure 5.1. Hereinafter, the measured and predicted inlet conditions to the rotor 2 will be discussed. Furthermore, the radial extend of the over-predicted blockage zone is presented and its influence on the stall inception.

The mean inlet flow conditions to the rotor 2 are shown as radial distribution of the incidence angle for the loaded operating point in Figure 5.15a. The measurement uncertainty has been estimated to  $0.5^\circ$  by Courtiade (2012). Generally, the two data bases show a fit with differences smaller than  $1^\circ$ . Below mid-span, the differences can be explained by the simplification of the simulation domain. The shrouded stator hub-cavity leakage flow is not taken into account. Further simulations would be necessary to support this hypothesis. Above 80% span-height, the difference between the experimental and numerical data is smaller than the measurement uncertainty. The increasing incidence angle above 90% span-height is found in both data bases. Given the magnitude of the presented incidence angle differences, the inlet flow conditions to the rotor 2 are not believed to cause the rotation disturbance in the simulations. Further judgment about the inlet flow conditions, such as turbulence level, would require additional unsteady measurements.

Figure 5.15b shows the radial distributions of the tangential velocity with URANS and PNEUM data downstream the rotor 2. The 1D mean velocity has been subtracted from





(a) Radial distribution of incidence angle to the rotor 2, based on flow angle in the inter-row plane between S1/R2, measurement uncertainty is about  $0.5^\circ$

(b) Radial distribution of tangential velocity fluct. (around 1D mean) downstream the rotor 2, mis-prediction indicated with black dashed ellipse, (theor. meas. uncertainty is smaller than 0.01)

Figure 5.15 Tip flow field mis-prediction at loaded operating point

both databases at each respective operating point. Experimental (PNEUM) values are given with the red lines, and numerical values with the blue lines. Between peak efficiency (PE, solid lines) and loaded (L, dashed lines) operating points only small differences are found within the two respective databases. The over-prediction of the blockage zone results in an under-prediction of the axial velocity (not shown here) and over-prediction of the tangential velocity from 80% span-height to the casing, as shown in the dotted ellipse.

According to the Euler equation, the energy transfer of the rotor on the fluid is higher with the over-predicted tangential velocity. That can also be found as an over-predicted total pressure rise in the tip region (as shown for the peak efficiency operating point in Section 3.3.2). Consequently, the unsteadiness of the tip region flow field is probably over-predicted. Therefore it is believed that the tip regions in the rotors 2 and 3 have an over-predicted sensitivity to the formation of a rotating disturbance, which is not found in the measurements.

A different turbulence model was tested at the peak efficiency operating point and was shown to give no useful improvement concerning the tip flow field mis-prediction (see Section 3.3.2). A mis-prediction of the tip leakage flow is possible with the given limitations of the RANS turbulence models as shown by Denton (2010). A more accurate flow field prediction could be achieved with Large Eddy Simulations. Using this simulation technique, only a few but promising results of compressor like test cases could be presented in recent years, such as in the work of McMullan and Page (2012) on a compressor stage, and most recently by de Laborderie et al. (2016) on a previous version of CREATE. The little number of published results can be explained by the high computation costs when using the Large Eddy Simulation technique. Riéra et al. (2013) applied the Zonal Detached Eddy Simulation (ZDES) method for the simulation of the first rotor of the previous version of CREATE.



The results showed promising improvement to the tip leakage flow prediction but are still too costly for the simulation of the whole compressor.

In conclusion, the URANS method might predict the rotating disturbance due to a strong sensitivity of the tip flow field to small perturbations. This is additionally favored by the tendency of the URANS to predict well-defined flow structures. LES and ZDES are promising methods for an investigation if the instability is purely a numerical method problem or if its onset is predicted too early (too high mass flow rate on the compressor characteristic curve) by the URANS method. However, well resolved LES and ZDES simulations have yet to be demonstrated for multistage applications at reasonable costs for research.

### **Stall inception type and location**

Vo et al. (2008) suggest with their work that the alignment of the TCV trajectory with the leading edge plane is one possible condition for the formation of a spike-type stall inception (see also the spike introduction in Section 1.4.2).

This condition has been found to be almost fulfilled in the measurement for the rotor 1 at the near surge operating point (Figure 5.1). And it has been found periodically fulfilled with the TCV trajectory oscillation in the rotors 2 and 3 in the numerical fields at the near surge operating point. Further steepening of the trajectory is expected with further throttling towards surge inception. The simultaneous occurrence of back-flow at the TE from a neighboring passage would be a second condition formulated by Vo et al. (2008), which is not found in URANS results.

The actually measured stall inception process in this compressor is characterized by a short length-scale disturbance of spike type occurring first in the rotor 1. This will be discussed in detail with measurement data in Section 5.3. The important oscillation of the TCV trajectory found with the simulations in the rotors 2 and 3 on the one hand, and the measured spike inception in the rotor 1 on the other hand, raise the question about how the rotating disturbance mis-prediction might influence the stall and surge inception location. The onset of surge has not been simulated for the current version of CREATE. Though, it was shown that from the loaded to the near surge operating point the rotating speed and number of cells of the rotating disturbance remain the same.

Crevel (2013) studied numerically the surge inception for another version of CREATE. They found a rotating disturbance in the rotor 3 at a near surge operating point, similar to the one found in the current version in the rotors 2 and 3. The authors noted that when closing the numerical valve further, the rotating disturbance would gain in amplitude and circumferential size. Eventually short before surge inception, the number and size of the rotating disturbance cells would change and induce the blockage which leads to flow reversal and surge.

Therefore, it is believed that the rotating disturbance, found in the current version of CREATE at the loaded and near surge operating points, plays a major role in the numerical surge inception prediction. The tip flow field mis-prediction is expected to lead to a wrong prediction of the stall inception in terms of location and type.

### 5.2.3 Measured rotating instability at part speed

An increased mean unsteadiness was stated for the rotor 1 when going from the nominal to the part speed at an operating point short before stall inception (see Figure 5.3a). The source of this unsteadiness together with its influence on the stall inception will be discussed in this Section.

#### Frequency characteristics of disturbance

Always when approaching the stall limit at part speeds ( $80\% \Omega$  and  $90\% \Omega$ ) with the CREATE compressor, an increase of several amplitudes are observed in the frequency spectra of casing pressure signals at about 30% of the blade passing frequency (12 to 24 compared to 64). That is also the case in the measurements for a previous version of the compressor at part speed (but has not been noted before). Frequency spectra are obtained from signals of a probe located close to the rotor 1 leading edge and presented for three shaft speeds ( $80\% \Omega$ ,  $90\% \Omega$ ,  $100\% \Omega$ ) in Figure 5.16a. The input signals have a length corresponding to about 400 rotations, and the spectra are integrated over bands of  $\frac{6Hz}{\Omega}$  to reduce the noise.

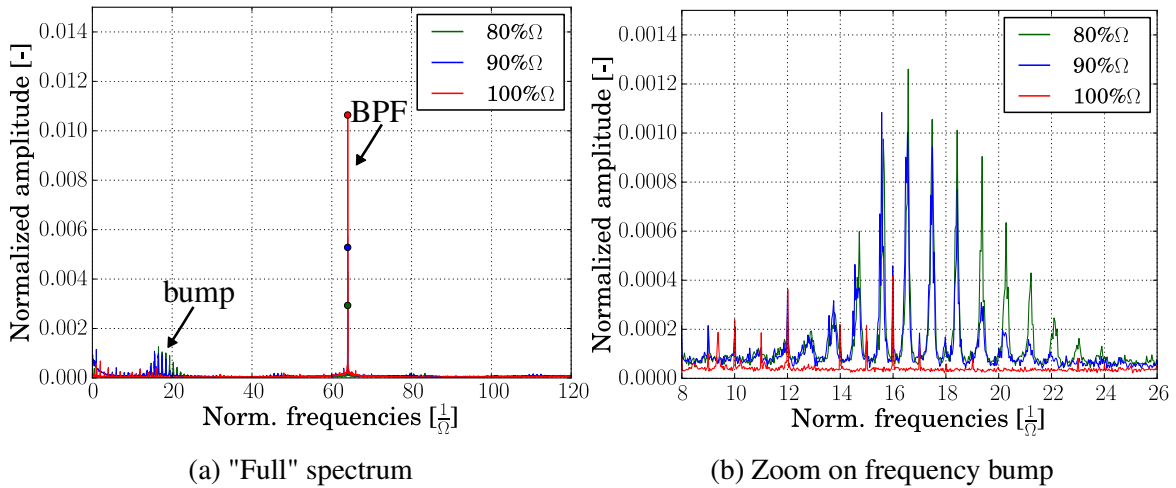


Figure 5.16 Frequency spectra of a casing pressure probe close to the rotor 1 LE in comparison between three shaft rotation speeds ( $80\% \Omega$ ,  $90\% \Omega$ ,  $100\% \Omega$ )

Apart from the BPF peak, a bump in frequencies can be seen at around normalized frequencies of 20. The maximum amplitude in the frequency bump reaches about 12% of the blade passing frequency of the rotor 1. The zoom on the frequency bump is shown in Figure 5.16b. At the nominal speed (red) only the shaft speed harmonics and the temporal periodicity (16) are found. At the part speeds, the disturbance in the flow field causes between 12 to 14 discrete peaks at frequencies which are not multiples of the shaft speed. The bump is not cleanly symmetric, and the maximum amplitude is not found at the same frequency between different measurements (not shown here). Though, the discrete peaks are found at about the same normalized frequencies comparing multiple measurements. This can be seen here as fitting peaks for the two part speeds. Small variations of the normalized RI

frequencies are in the order of  $\pm 0.08$ . The maximum amplitudes are found at either of the peaks: 15.56, 16.58, 17.5, 18.42, or 19.4. This depends randomly on the measurement. Lowering the speed from 90%  $\Omega$  to 80%  $\Omega$ , a general amplitude rise is found on the right side of the bump, whereas the left side of the bump is basically identical at both speeds. Lowering the speed means the "higher" frequency content of the disturbance gains in importance.

The frequency bump is reported as characteristic for the controversially called "rotating instability" (RI), see the introduction in Section 1.4.3. As a summary of the introduction, it remains unclear what is causing exactly the multiple peaks in the frequency spectra. The multiple peaks suggest a certain unstable behavior in terms of rotation speed and cell number of the disturbance. This RI has been observed to occur with and without a tip gap. Multiple works give evidence of an interaction between a separation vortex (Inoue et al., 2004; März et al., 2002; Yamada et al., 2013) and the tip leakage flow or a corner separation (no tip gap, Beselt et al. (2013)).

Exclusively the measurement data at 80%  $\Omega$  will be used in the following Sections to demonstrate the characteristic of the RI in CREATE. The main characteristics are the same at 80%  $\Omega$  and 90%  $\Omega$  but the amplitude is higher at 80%  $\Omega$  (Figure 5.16b), which allows a better visualization here.

Figure 5.17a presents the amplitude of the instability (modes between 11 to 24 normalized frequency) on the left vertical axis and the blade passing frequencies on the right vertical axis (red and blue) over the machine axis. This evolution is based on the high-frequency casing pressure measurements in the inter-row planes and above the rotors. The blade row leading and trailing edges are indicated with vertical black dashed lines. The scales of the left and right vertical axis differ by a factor of five. The maximum amplitudes of the blade passing frequencies are found close to the respective leading edges of the rotors (labeled R1 to R3). The integrated amplitude over the RI frequency bump has its maximum at the leading edge of the rotor 1, with a 10 times smaller amplitude than the one of the rotor 1 blade passing frequency. This corresponds well to what was presented in the frequency spectra

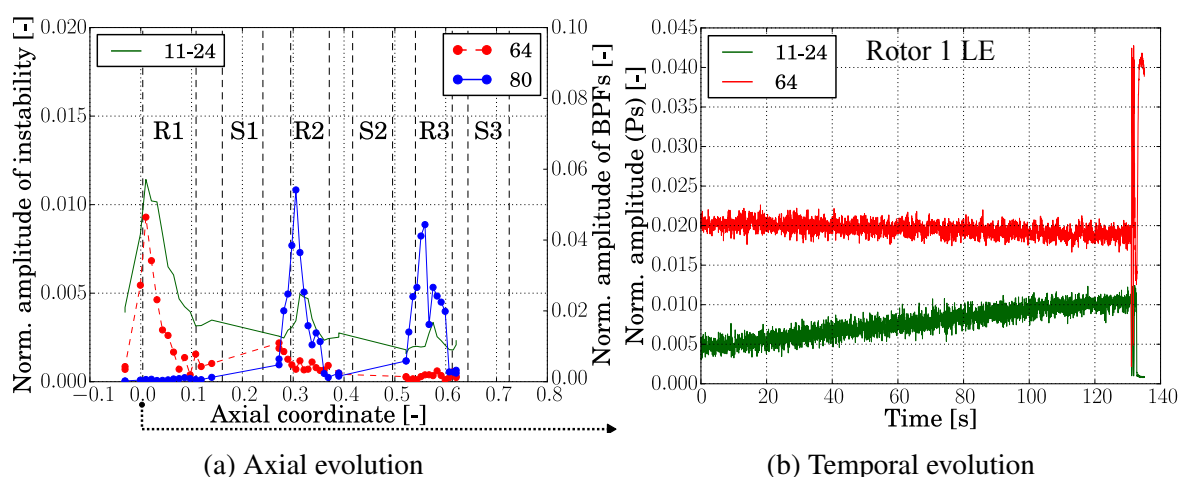


Figure 5.17 Axial and temporal evolution of amplitude of rotating instability modes (11-24) and rotor blade passing frequencies (64 and 80)

in Figure 5.16. The amplitude of the RI decreases rapidly in up- and downstream direction. The amplitude level downstream the rotor 1 trailing is considered as noise induced by the rotation frequency harmonics, which are taken into account with the integration over the bump's frequency band.

In Figure 5.17b, the temporal evolutions of the RI and rotor 1 BPF is presented based on the casing pressure signal from a probe positioned close to the rotor 1 leading edge. The measurements start at a stable operating point and the outlet valve is closed very slowly giving quasi steady operating conditions (introduced in Section 4.2.1). The compressor falls into stall and surge after about 131 seconds. During the valve closing, the BPF decreases in amplitude by less than 10% but the integrated amplitude of the RI doubles. Even though the RI amplitude is not reaching the level of the BPF, the RI can be seen as a precursor of rotating stall at part speed here. It reaches a critical level (close to stall) when it surpasses half of the BPF amplitude. Beyond this level stall inception can be expected, and it will be shown that the stall inception and the RI are related after a presentation of the main characteristics of the RI.

### Rotation speed estimation

The rotation speed of the RI can be obtained with two different methods. The casing pressure is measured with several probes around the circumference upstream of the rotor 1 at the plane 260. Taking the signals at two subsequent circumferential positions (e.g. 60° apart), the signals  $S_{Ps,1}$  and  $S_{Ps,2}$  can be correlated. A flow structure is sensed in the two signals with a time delay  $\tau$ , corresponding to the traveling time of the flow structure between the two probes. A cross-correlation function ( $X_{12}(\tau)$ ) between the two signals (of length  $T$ ) is formed with Equation 5.1.

$$X_{12}(\tau) = \frac{1}{T} \int_{-\infty}^{+\infty} S_{Ps,1}(t + \tau) S_{Ps,2}(t) dt \quad (5.1)$$

The cross-spectral density spectrum between  $S_{Ps,1}$  and  $S_{Ps,2}$  is obtained with the Fourier transform of the cross-correlation function with Equation 5.2 using  $X_{12}(\tau)$ .

$$G_{12}(f_x) = \frac{1}{2\pi} \int_{-\infty}^{+\infty} X_{12}(\tau) e^{if_x \tau} d\tau \quad (5.2)$$

A normalized magnitude of the cross-spectral density ( $G_{12}(f_x)$ ) is necessary to judge the importance of a correlated flow structure in the two signals. The normalized ( $C_{12}(f_x)$ ) coherence function is obtained from Equation 5.3 with the help of the cross-spectral density spectra  $G_{11}$  and  $G_{22}$ . If the signals are identical the coherence function is equal to one. If they are uncorrelated, the coherence function is equal to zero.

$$C_{12}(f_x) = \frac{|G_{12}(f_x)|}{\sqrt{G_{11}(f_x) G_{22}(f_x)}} \quad (5.3)$$

The first method for the estimation of the rotation speed between the two signals consists in the evaluation of the phase response of the rotating instability frequencies in Section 1.8.

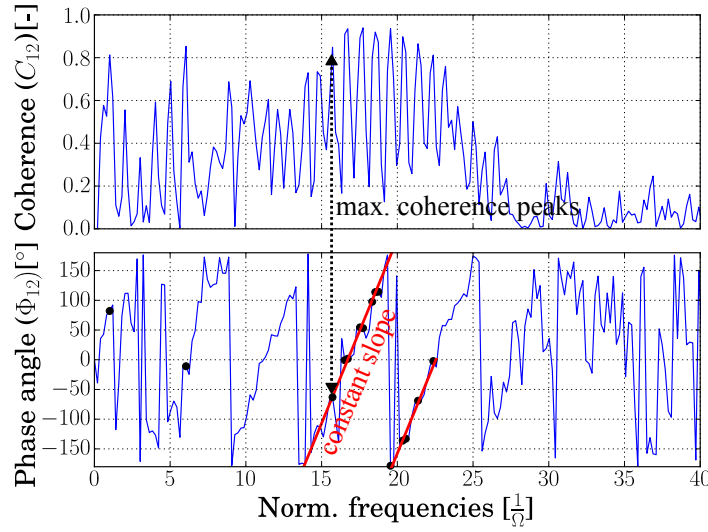


Figure 5.18 Characteristic coherence and cross-correlation phase angle spectra from two casing pressure probe signals at plane 260

The cross-correlation phase angle is obtained with Equation 5.4.

$$\Phi_{12}(f_x) = \arctan \frac{\text{Im}(G_{12})}{\text{Re}(G_{12})} \quad (5.4)$$

Figure 5.18 shows the coherence (top) and cross-correlation phase angle (bottom) spectra of two probes at plane 260. These probes are circumferentially placed at  $60^\circ$  apart. Signals with a length corresponding to forty rotations are taken, the Fourier transforms are conducted on windows of about eight rotations with an overlap of four rotations. These parameters are chosen so to obtain a clean coherence spectrum. The rotating instability frequencies are found as peaks of increased coherence levels up to 0.94. The shaft speed harmonics are found with a comparable coherence level. A coherence level of 0.8 is chosen arbitrarily for selecting the corresponding meaningful phase angles ( $\Phi_{12}$ ) in the rather noisy phase angle spectrum. These points are marked with black dots. A linear fit is computed over the selected points representing the rotating instability to obtain the slope  $\frac{d\Phi_{12}}{df_x}$ . Then the rotation frequency of the rotating instability (RI) is calculated with Equation 1.8 as introduced earlier, which yields a rotating frequency  $\omega_{RI,grad} \approx 0.96\Omega$ . This method depends strongly on the selected frequency resolution (length of signal), and the selected points in the phase spectrum. Taking a higher frequency resolution, the phase fluctuations are too high, and the resulting rotation frequencies can be completely unrealistic. If there was more probes available at the plane 260, several phase correlations could have been averaged.

A second method of estimating the rotation frequency considers the frequency distance between the constantly spaced peaks in the frequency spectra as rotating frequency of the rotating instability according to Equation 1.10. This gives a rotating frequency of  $\Omega_{RI} \approx 0.92\Omega$  at both compressor part-rotation speeds. This second method is a trade-off between accuracy in the frequency determination of each peak and the noise in the signal. The

noise can be reduced by integration over small frequency bands but decreases the resolution in the frequency space.

The difference between  $\omega_{RI,grad}$  and  $\Omega_{RI}$  reflects the uncertainty in the estimation of the rotating frequency with the two presented methods but is considered a good fit still. This estimation is only a challenge because the measurements are conducted in a high-speed and not low-speed compressor. The high-speed environment requires signals over several rotations to obtain a fine enough frequency resolution when conducting the Fourier transform. Rotating frequencies of 50% to 65% rotor speed have been reported in several works Baumgartner et al. (1995); Mailach et al. (2000); März et al. (2002); Vignau-Tuquet and Girardeau (2005), and 70% by Inoue et al. (2004). Note that the works of Baumgartner et al. (1995) and Vignau-Tuquet and Girardeau (2005) were conducted on high-pressure compressors. Also a rotation speed of  $90\%\Omega$  has been reported by Zhang et al. (2012) in a numerical work on a turbine working locally as a compressor at low mass flow rate. The  $\Omega_{RI} \approx 0.92\Omega$  appears high but not out of range of what has been reported in literature.

### Flow field modulation

A highly resolved instantaneous snapshot of the flow field is not possible to obtain with the available measurement techniques in CREATE. The twelve casing pressure probes above the rotor 1 can be used to reconstruct the flow field. Because of the unstable behavior of the rotating instability, the signals cannot be ensemble averaged for the reconstruction. Therefore instantaneous space-time diagrams are reconstructed from these high-frequency measurements and one is presented as an example in Figure 5.19. For building this diagram, one-rotation-segments are taken from the original signals. The twelve signals are registered synchronously at different axial position. Thus the vertical axis of the space-diagram shows the axial chord, and the horizontal axis shows the time as a fraction of a rotation. A time lag exists between the signals because they are not obtained at identical circumferential position. The time lag  $\Delta t_{12}$  of a flow structure spinning at the rotation frequency  $\omega_{12}$  between two probes positioned  $\Delta\Theta_{12}$  apart can be expressed with Equation 5.5.

$$\Delta t_{12} = \frac{\Delta\Theta_{12}}{\omega_{12}} \quad (5.5)$$

First the signals are decomposed with Fourier transforms. That allows the compensation of the time lag correcting the phase of selected frequency content. The compensation requires the knowledge of the rotation frequency of the flow structure which is described by the respective frequency content. The phase angles of the frequencies representing the rotating instability are corrected using  $\omega_{12} = \Omega_{RI} = 0.92\Omega$  (presented in previous Section). The phase angles of the remaining frequency spectrum are corrected using  $\omega_{12} = \Omega$ . This rotation frequency is true for flow structures rotating at shaft speed but not for RSI which can rotate at different speeds. This filters the RSI and let them appear as noise in the space-time diagrams. That is accepted because they are not subject of the study here.

Figure 5.19 shows the space-time diagram at the top for the rotor 1. Two blades are indicated exemplary with black contours. A black dashed line is indicated close to the leading edge. For this axial position, the temporal signal is shown at the bottom in the same Figure.

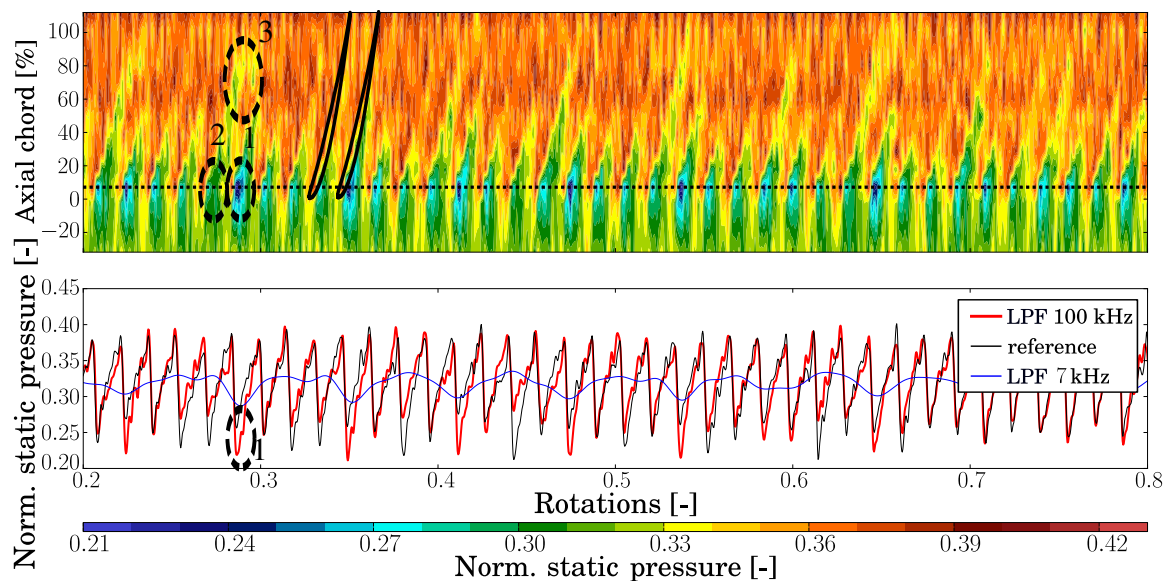


Figure 5.19 Space-time diagram of casing pressure field above rotor 1 showing rotating instability (top) and temporal signal of casing pressure at black dashed line (bottom)

The signal is given for the low pass filtered (LPF) signal at 100 kHz (above BPF, red), 7 kHz (below BPF, grey), and a reference signal (labeled ref, black). The reference signal shows the LPF 100 kHz signal from one rotation before. The acceleration of the fluid on the suction side of the rotor blades is found as low static pressure spot (blue) close to the leading edge. In chord direction, the static pressure increases (towards red values). A modulation of the low pressure spots is found as spots of higher (labeled 2) and lower (labeled 1) pressure than in the neighboring passages. The higher pressure modulation could represent a blockage in the respective passage. In the passages with the higher pressure at the leading edge, spots of static pressure systematically lower than in the neighboring passages are seen towards the trailing edge (labeled 3). This is not equally strong marked in all the "blocked" passages.

The temporal distance between the low pressure spots (labeled 1) at the leading edge is not constant. These spots can be easily identified with the help of the LPF 7 kHz signal as valleys. In the given time window, the temporal distance appears to be four passages up to the middle of the signal but changes to less clear distances later during this rotation. Globally the distance appears to change arbitrarily over all investigated rotations, which are of course not all presented here. The instability is not periodic to a shaft rotation, and therefore the low pressure spots are not located at the same point in time when comparing the signals of two subsequent rotations. This is seen when comparing the LPF 100 kHz and reference signals.

Assuming the temporal extend corresponds also to the spatial extend of the disturbance cell, the variation of the disturbance size is evaluated with a wavelet analysis. Figure 5.20 shows the casing pressure signals (a) low-pass filtered at 7 kHz and 100 kHz. The latter is used only for a visual reference for the amplitude of the RI. The wavelet power spectrum is shown with a zoom on the rotating instability frequencies (b). The variation of the disturbance's temporal extend is seen as a modulation in the wavelet spectrum. That causes



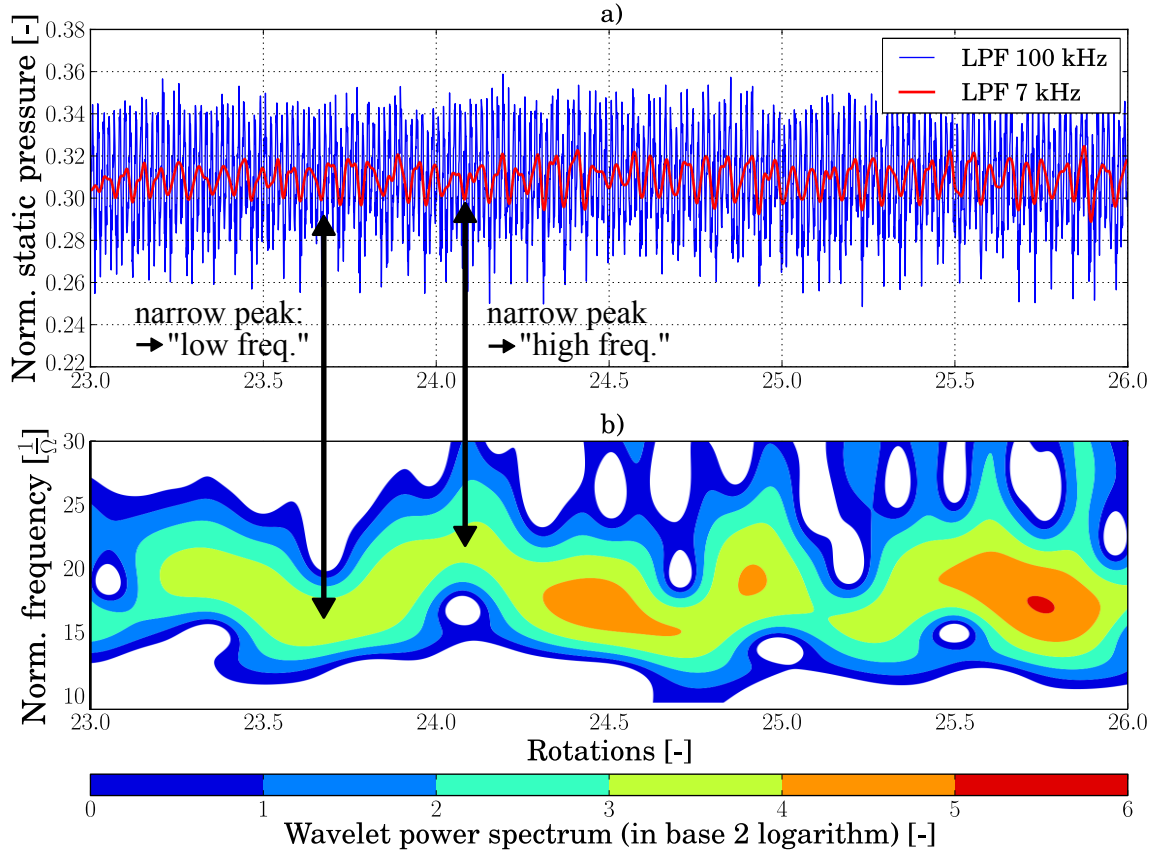


Figure 5.20 Rotating instability characterization with casing pressure signals over three rotations (a) and wavelet power spectrum with zoom on rotating instability frequencies (b)

the "up and down" of the maximum amplitude in the range of the RI frequencies. A high frequency is related to a narrow peak (narrower than the surrounding ones) in the LPF 7 kHz casing pressure signal. That is valid due to the relating of the temporal to the inverse of the frequency extend. At certain time points double peaks are found in the casing pressure signal, which are followed by a narrow peak. This illustrates the changing of the size of the disturbance, which could be caused by an interaction of the disturbance with the rotor 1 blade row. The temporal evolution of the frequency with the maximum wavelet power is presented in Figure 5.21a. About one peak is found per rotation. Figure 5.21b shows the frequency spectrum of this signal. The vertical axis gives the amplitude of the normalized frequency variation. Four peaks are found, where the most energetic is located at  $0.925 \frac{1}{\Omega}$  (verified with longer input signals as well), with its harmonics 1 to 3. This corresponds to the rotation speed of the rotating disturbance  $\Omega_{RI} = 0.92\Omega$ . That reveals a periodic behavior of the rotating disturbance which is linked to the rotation speed of the disturbance. The temporal period of the rotating disturbance is thus  $\frac{1}{0.925} = 1.081 \text{ rev.}$

It was introduced in Section 1.4.3 that the rotating instability might be understood as a rotating source with a non-uniform circumferential distribution as proposed by Kameier and Neise (1997b). This corresponds well to the size variations of the disturbance cells over a



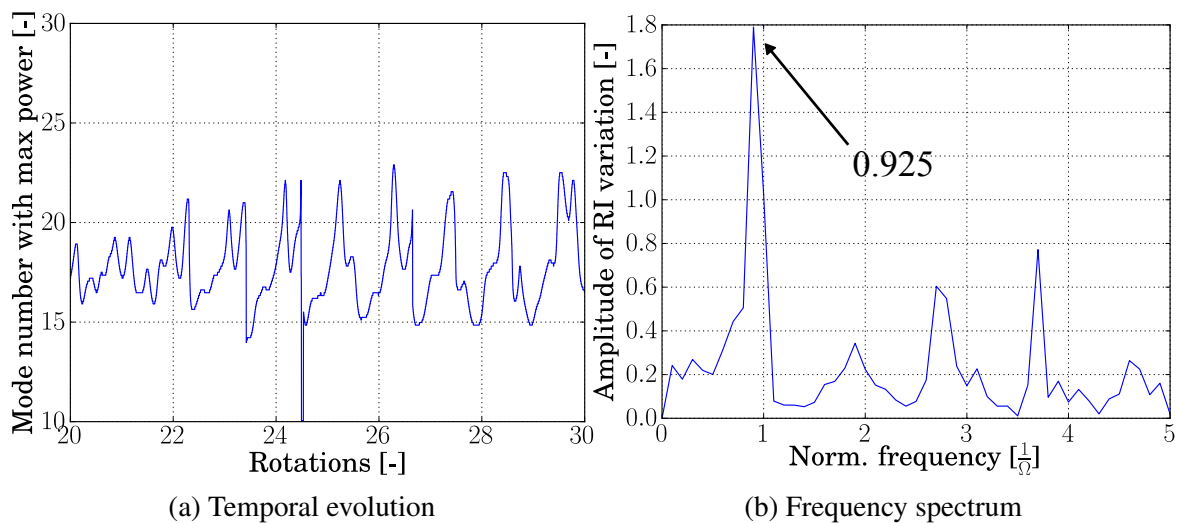


Figure 5.21 Frequency variation of rotating instability over time

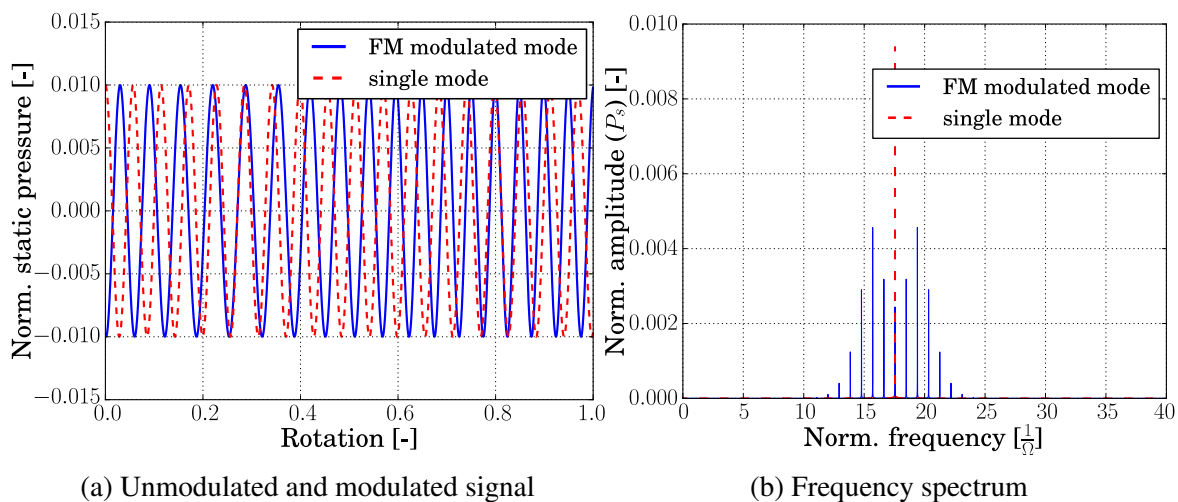


Figure 5.22 FM modulation illustrated with two fictitious pressure signals

rotor rotation as identified with the wavelet analysis (Figure 5.20). Taking the example of the widely known frequency modulation (FM), a carrier frequency ( $f_c$ ) and a modulation frequency ( $f_m$ ) are used according to Equation 5.6.

$$P_s(t) = A_c \cdot \cos \left( 2\pi f_c t + \frac{\Delta f}{f_m} \sin(2\pi f_m t) \right) \quad (5.6)$$

where  $\Delta f$  corresponds to the maximum frequency shift between the carrier frequency and the side peaks in one direction of the bump.  $A_c$  is the amplitude of the carrier frequency. The FM is known to produce a signal with multiple frequencies (wide and narrow peaks) as sketched in Figure 5.22a. The  $A_c$  corresponds to a realistic one for the rotating instability. For comparison, a single mode (pure cosine) is plotted with a red dashed line, whose frequency

corresponds to the  $f_c$ . In this first example, the chosen  $f_m$  and  $f_c$  shall not be important.  $\frac{\Delta f}{f_m} = 3$  is giving a good number of side peaks and corresponds to the three harmonics found in Figure 5.21b. The frequency spectrum of the FM modulated signal shows multiple peaks, whereas the single mode obviously only one peak (Figure 5.22b).

In the case of the RI, the carrier frequency could be a theoretical unmodulated rotating disturbance. The result of a classical RSI is a regular spaced (in time and space) lobed structure according to Tyler and Sofrin (1962). The RI is not a classical RSI because it is found as an irregular spaced lobed structure in time. That means its structure is not imposed by a blade row with constant blade pitches. The rotating disturbance might be caused by a separation vortex on the rotor blade suction surface and its interaction with the neighboring blade passages. The rotation of the disturbance relative to the blade row can cause the modulation of the disturbance over the circumference. This modulation is irregular around the circumference (or in time) but periodic to 1.081 revolutions, as stated earlier. A relative motion of the disturbance to the blades would be also observed in a compressor cascade. Thus this complies with observations of rotating instabilities in cascades (see Section 1.4.3). Due to the modulation, the rotating disturbance is no longer uniform over the circumference.

Figure 5.23 shows the casing pressure signals of two successive temporal periods of the rotating instability ( $T = \frac{1}{0.925}$ ). They are low-pass filtered below the BPF at 5 kHz. Globally a large number of valleys ("negative peaks") correspond with their location in time. That is a proof for a certain temporal periodicity of the RI. Figure 5.23 shows as well an amplitude modulation, which causes peaks of different amplitudes between two instability periods. The amplitude modulation is periodic to one shaft rotation. That is demonstrated with the help of Figure 5.24, which shows casing pressure data points from 600 rotations superposed together with the signals of two example rotations. The locations in time of the peaks in the two example signals do not fill well, which is another prove for that the periodicity of the frequency modulation is another than one shaft rotation. The amplitude modulation results in the grey envelope of data points which is found as a temporal mode 1 over one shaft rotation. The minimum modulation is found at about 0.4 shaft rotation and the maximum modulation

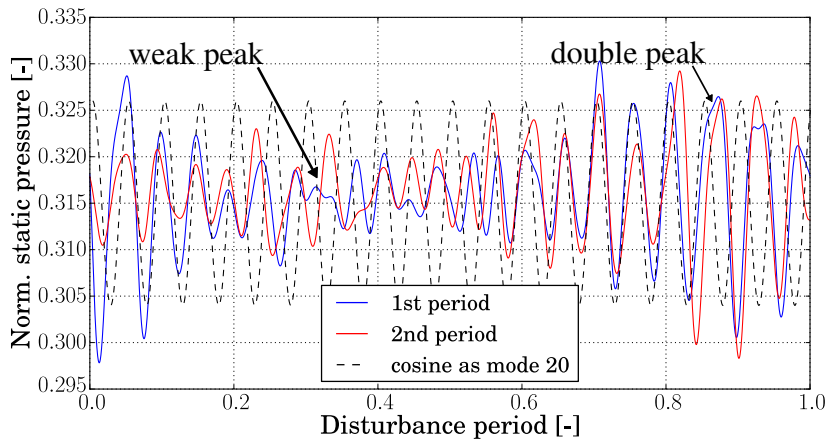


Figure 5.23 Two successive temporal periods of the rotating instability in the casing pressure signal from above the rotor 1 LE (low-pass filtered at 5 kHz, below BPF), and cosine as ref.

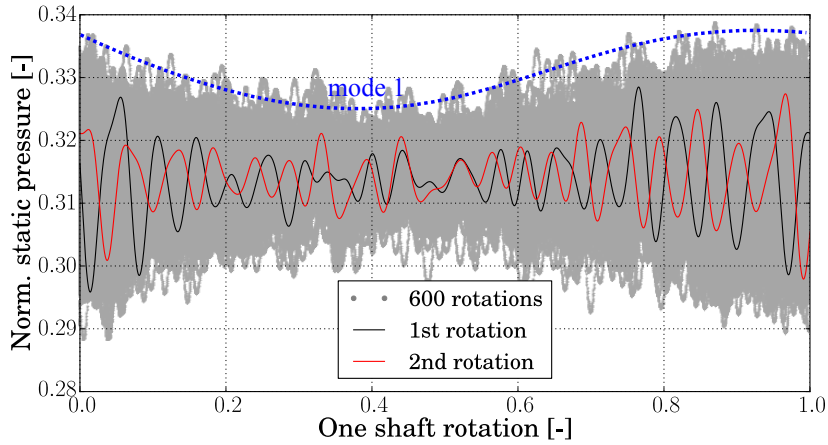


Figure 5.24 Casing pressure signal of 600 shaft rotations stacked and superposed on two example rotations (low-pass filtered at 5 kHz, below BPF) at a location above the rotor 1 close to the LE

at beginning or ending of the shaft rotations. This temporal mode also arises at 100%  $\Omega$ . It signifies the instability being modulated by the system for example due to an eccentric shaft.

Now, the focus shall be put back on the main differences between the signals in Figure 5.23. These are found where one signal shows "double" peaks and the other one only a single peak (for example at 0.86 rotations). That indicates the process of the changing in size of the separation cells over time (or circumference). Eventually, this can lead to locally very "weak" peaks as for example at 0.33 disturbance period in the blue signal. At the same time point, this is not found in the red signal, which is a sign for the high unsteadiness of the instability.

Counting the peaks over 600 disturbance periods (not shown here) yields to a mean number of peaks of 20.097 with a standard deviation of 1.02 peaks. Periods with 21 peaks are found to be followed by periods with 19 cells, which explains the standard deviation and also shows that there is probably not a significant change in number of cells or rotation speed of the instability. Moreover an ensemble average phase locked to the disturbance period would also yield a mode 20 (neither shown here). An idealized mode 20 is thus traced as an adapted cosine in Figure 5.23. The variation in matching between the peaks of the mode 20 (black dashed) and the actual signals (red/blue) visualizes the frequency and amplitude modulations. The number of peaks over one temporal period of the RI corresponds as well to the number of cells (20) of the RI.

Thus, the source of the RI is a spatial mode with 20 lobes that is rotating at a speed of  $0.925\Omega$ . The temporal mode or center frequency is consequently  $f_c = 20 \times 0.925\Omega = 18.5\Omega$ . The frequency modulation is caused by the rotation frequency of the RI ( $f_m = 0.925\Omega$ ). These parameters allow the construction of a signal whose frequency spectrum fits well to the one of a real example signal, as shown in Figure 5.25. The real spectrum is seen in black, and the reconstructed signal in red. The amplitude modulation of the RI does not have an influence on the frequencies in this spectrum, and could be caused by non-uniformity in the flow field for example caused by a small difference in blade tip gap sizes in the rotor 1.

In conclusion, the term "rotating instability" should be given a new thought because the rotation speed is found to stay constant and the number of cells might be constant as well. The unstable behavior is only found in the locally changing size of the disturbance cells which resembles a frequency modulation in time. The actual interaction, which causes the irregular modulation of the cell sizes, needs to be resolved with spatially higher resolved measurements in a future work.

Summary of frequency modulation parameters relative to a rotating instability temporal period of  $T_{RI} = 1.081 \text{ rev.}$ :  $f_c = 18.5\Omega$ , and  $f_m = 0.925\Omega$

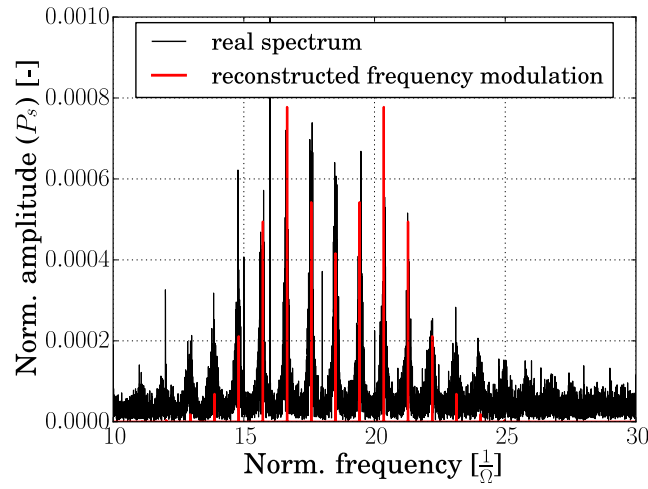


Figure 5.25 Frequency spectrum of real and reconstructed signal showing RI frequency range

### Statistical analysis of the interaction between the RI and the rotor 1 tip flow field

The analysis of the statistics on the casing pressure measurements above the rotor 1 is helpful for the understanding of the tip flow field modulation induced by the RI rotation relative to the rotor blades. Figure 5.26 shows the temporal evolution of the standard deviation and skewness (see definition in Section 2.4.2) calculated on the casing pressure data points falling in each respective ensemble average window. The ensemble average is calculated on the temporal period of  $\text{CREATE} \left( \frac{2\pi}{16\cdot\Omega} \right)$ . Here the statistics are calculated on about 5000 data points per ensemble average window. The horizontal and vertical axis present time and space respectively. The blade passing is indicated with the blue contours. The time is advancing to the right, thus the blades are rotating to the left. The RI is not periodic to the temporal periodicity of the compressor and therefore the statistics maps can only trace the modulation of the tip flow field by the RI in the rotor passages but not resolve the RI structure itself. An ensemble average phase locked to the RI would not be sufficiently resolved in space for a useful analysis because only 12 synchronous signals are available over the chord of the rotor.

A possible tip clearance vortex oscillation induces the elevated standard deviation spot at the pressure side of the passage in the first 20% of chord (indicated with black dashed line with the arrow head). An alignment of the tip clearance vortex with the inlet plane does not

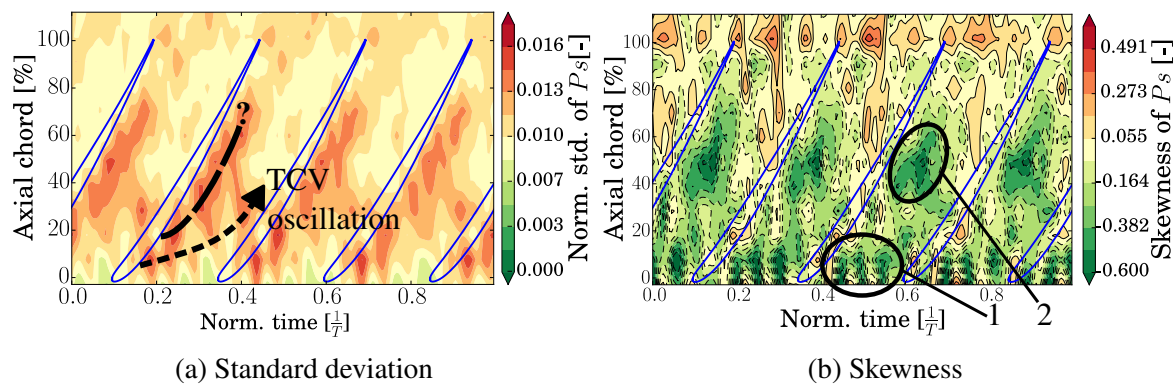


Figure 5.26 Statistics on casing pressure above the rotor 1 close to stall inception at 80%  $\Omega$

occur in any passage. This would otherwise cause a band of increased standard deviation at the inlet as shown in the works of Inoue et al. (2004) and März et al. (2002).

The increased standard deviation on the suction side of the channel, marked with the black long-dashed line and a question mark, are related to the intermittent appearance of the low pressure spots which were labeled with a 3 in Figure 5.19. This is a specific behavior at part speed and not found at 100%  $\Omega$  (compare to the 100%  $\Omega$  speed case in Figure 5.1). This zone is not caused by a tip clearance vortex breakdown. The consequently induced blockage would lead to an increase in static pressure as shown for example in the work of Inoue et al. (2004). Right next to this zone, a negative skewness is found highlighted with the black dashed circle and labeled with a 2 in Figure 5.26b. The negative skewness indicates infrequent pressure spikes below the average. Further spots of negative skewness are found in the inlet plane (labeled with a 1), whose source is unclear.

Inoue et al. (2004) demonstrated that a possible leading edge separation vortex can influence the pressure rise negatively in the whole concerned rotor passage. In CREATE, this might cause the noticeable pressure drop on the suction side of the rotor passages. More spatially resolved casing pressure measurements would be necessary to identify the structure which causes the pressure drop.

### Rotating instability evolves into rotating stall

The rotating stall inception is analyzed with a RMS deviation evolution during stall inception in Figure 5.27a. A reference ensemble average is calculated per signal of all inter-row plane casing pressure probes. The RMS deviation (RMSD) is calculated per revolution per signal. The RMSD gives a measure for how much the signal of a rotation deviates from the reference ensemble average, and given as normalized static pressure. The rotating stall inception is expected to be found as measurable local deviation from the reference signal. The average RMSD per inter-row plane is traced here for 140 rotations, where the surge inception occurs after about 127 rotations. The unsteadiness in the tip flow field stays constant up to the rotation 120, seven rotations prior to surge. Here the signal starts deviating from the reference at the plane 260 first, and subsequently at 26A downstream the rotor 1 and shows also the

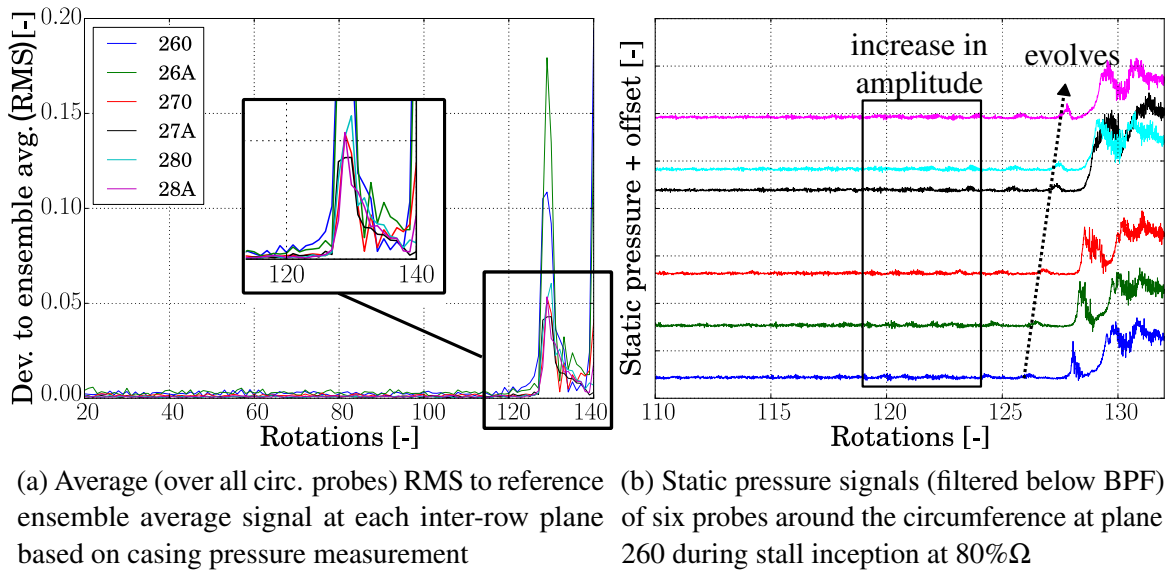


Figure 5.27 Rotating instability evolves into rotating stall cell

overall highest RMSD at the latter. This shows the occurrence of the stall inception in the rotor 1 prior to the formation of rotating stall cells in the downstream rotors.

Figure 5.27b shows the rotating instability evolving into a rotating stall cell in the rotor 1. Casing pressure signals (filtered below BPF) are shown for six probe positions around the circumference at plane 260. The offset between the signals takes into account the circumferential position of the probes. The fluctuations in the signals are mainly caused by the rotating instability. Rotation 131 shall be noted as surge inception time point. From rotation 118 on, the unsteadiness rises in the signals but a stall cell evolves from the rotating instability only from the rotation 126. The stall cell grows and the rotation speed decreases compared to the rotation speed of the RI. This shows a very fast evolution from the rotating instability to the stall and surge inception. No spike like disturbance is noted here.

Once a rotating stall cell is detected, the frequency trace of the rotating instability disappears as shown in Figure 5.28. The signals of two probes (positioned at 14' and 44' at plane 260) are analyzed with wavelet transforms, and the amplitude variations of the normalized frequencies up to the rotor 1 BPF are given in the two power maps. The signals are zoomed on about 13 rotations prior to rotation stall inception. The wavelet transforms are conducted with a temporal resolution which allows the detailed analysis of the amplitude evolution. A wavelet analysis is always a trade-off between frequency and temporal resolution. Thus the frequency resolution of the BPF is not ideal here, and found as a large horizontal band above 60 normalized frequency. Again, the rotating instability is found in the band of 11 to 24 normalized frequency.

The stall inception is seen as vertical stripe of elevated power (white) at rotation 13.5 at the top (14'), and 14 rotations at the bottom (44'). Note that the rotor is turning from probe 1 (14') to probe 2 (44'). So does the rotating stall cell. Close to the probe at 14' the rotating stall cell forms, and causes a diversion of the flow as also introduced by Emmons et al. (1955) (see Figure 1.12). As a consequence, the rotating instability is no longer seen by the probe



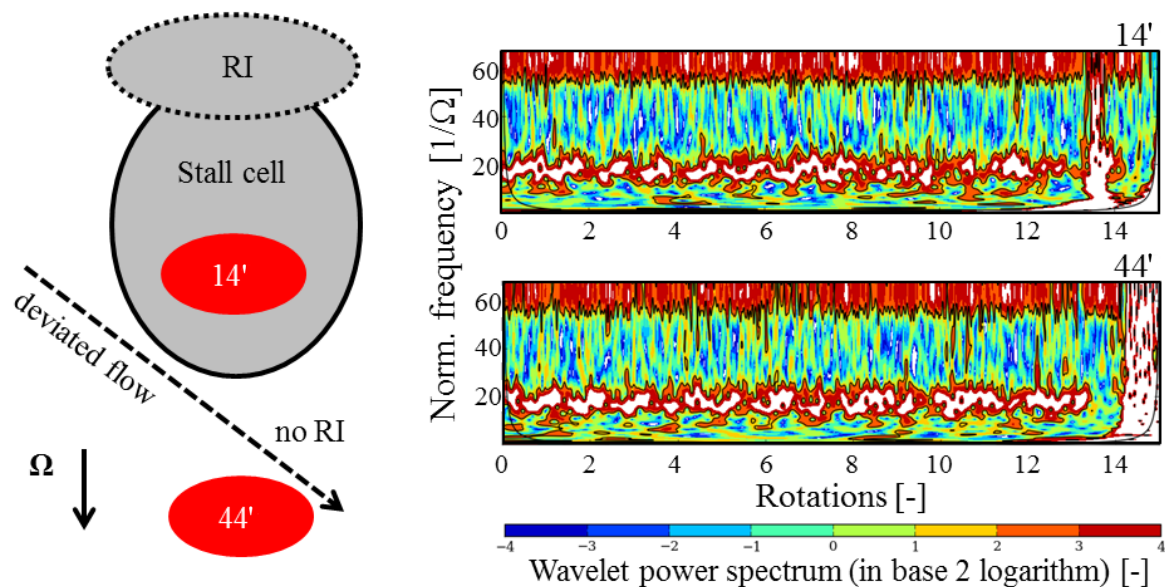


Figure 5.28 Rotating instability trace in wavelet analysis (right) and sketch (right) based on two signals of probes positioned at plane 260 at 14' and 44' (red ellipse)

at 44'. This behavior is summarized in the sketch in Figure 5.28 on the left. The rotating instability disappears thanks to the deviated flow. This shows the very fast response of the rotating instability to changes in the flow field. The rotating instability does not reappear once the rotating stall cell is established as can be seen after rotation 14 of the signal at 14'. The rotating stall must create a blocked and unblocked through-flow area in the rotor 1. The unblocked - "clean" - flow field appears unfavorable to the rotating instability because of the changed incidence angle. This might avoid a separated blade boundary layer which could be the initial source of the rotating disturbance frequency.

### 5.3 Spike type rotating stall inception

As a reminder, a previous version of CREATE exhibited precursor like rotating pressure waves evolving into rotating stall cells, see the work of Courtiade (2012). In the current version of CREATE, no stall precursor is found prior to stall inception at nominal rotation speed. With the help of Figure 5.29, the stall inception location is identified with an analysis of the casing pressure signals above the rotors during more than 200 rotations prior to surge inception. The signals are cut in segments representing each one temporal period. A cross-correlation is calculated between each segment and the corresponding one a rotation earlier (labeled "self-correlation"). Without a stall precursor and therefore no frequency trace, the calculation of a coherence spectrum would not make any sense (see Equation 5.3). Thus simply the cross-correlation function ( $X_{12}(\tau)$ , see Equation 5.1) is taken, and divided by the auto-correlation peaks of each input signals to obtain correlation coefficients up to 1 according to Equation 5.7. A correlation coefficient of 1 represents a perfect match between two signals, in other words the cross-correlation between a signal and itself (auto-correlation).

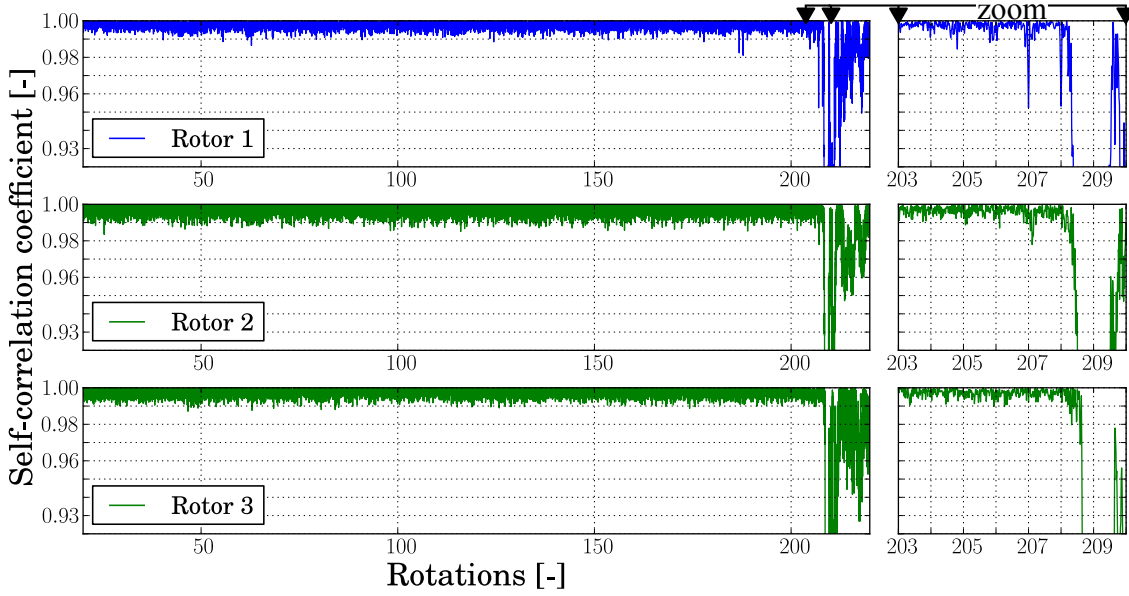


Figure 5.29 Disturbance detection with self-correlation of casing pressure signals at leading edges of the rotors 1 to 3 at 100%  $\Omega$

$$x_{12}(\tau) = \frac{X_{12}(\tau)}{\sqrt{X_{11}(0)X_{22}(0)}} \quad (5.7)$$

The maxima of the cross-correlations per temporal period are plotted in Figure 5.29. This gives a fine measure for changes in the flow field. The signals are low-pass filtered at the BPF (at 7 kHz) because otherwise the cross-correlation would not be able to detect small stall inception related disturbances. The cross-correlations are based on casing pressure probes close to the leading edge of the three rotors. Globally, the unsteadiness increases in flow direction. This is seen as increasing background noise or band of fluctuations between 0.99 and 1. The surge inception occurs between the rotations 208 and 209, which is found as drop of the coefficient below values of 0.92. In this example, a disturbance occurs about 20 rotations prior to the surge inception in the rotor 1 (understood as "pre-disturbance"). It causes the coefficient to drop to a value outside of the noise band. With the instrumentation setup, this is not always observed before surge inception in this compressor. In all surge inception measurement though, the "last" disturbance prior to surge occurs first in the rotor 1. This can be seen in the zoom in Figure 5.29 on the right at rotation 207. During the same rotation, a disturbance is also found in the rotor 2. In none of the measurements, the rotor 2 shows the disturbance prior to the one in the rotor 1. Furthermore the pre-disturbances occur only in the rotor 1 and never in the rotors 2 or 3. A disturbance arises always last in the rotor 3, only as a consequence of the disturbances from upstream. In conclusion, it is in the rotor 1 where a disturbance leads to the stall inception of the compressor.

Figure 5.30 shows the filtered (below BPF) casing pressure signals over eight rotations theoretically centered on the rotation 207 found in Figure 5.29. The signals are shown for two



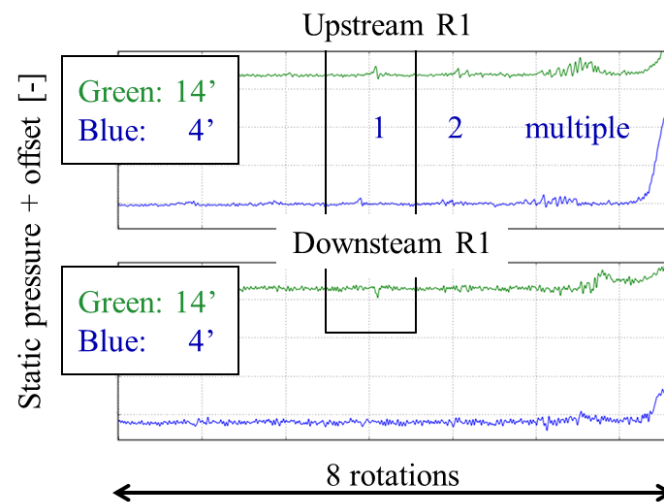


Figure 5.30 Spike captured by casing pressure probes upstream and downstream of the rotor 1, signals low-pass filtered at 7 kHz (below the BPF)

circumferential positions at plane 260 (upstream of the rotor 1) and at plane 26A (downstream the rotor 1). The distance between the signals takes into account the circumferential position of the probes (4' and 14'). The disturbance arises (highlighted with the rectangle) and is found as a up-and down of the static pressure (labeled with a 1). This is characteristic for a spike (see introduction in Section 1.4.2). Finding a spike like stall inception is conclusive with the probable alignment of the tip clearance vortex with the inlet plane of the rotor 1, which was shown in Figure 5.1. This alignment can cause a separation vortex on the suction side of the rotor 1 blades close to the leading edge (Section 1.4.2).

Downstream of the spike (downstream of the rotor 1) a characteristic suction peak is found due to the acceleration of the fluid. Only one rotation later the spike evolves and two peaks are found (labeled with a 2). This is no longer the classical signature of a spike, and could be either a double spike or rotating stall cell. The rotating speed of the spike is about  $90.9\Omega$  just after the onset, in this example case. It is difficult to estimate accurately the rotation speed due to the fast evolution of the spike over time. The applied cross-correlation method is introduced later on in Section 5.3.1. In either case the spike should be understood as the embryonic rotating stall cell here because of the very fast evolution over less than two rotations. Again only one rotation later, multiple peaks (labeled with "multiple") are found which signifies the disturbance of the flow field by the passing of a rotating stall cell. The stall cell grows and the surge is eventually seen upstream of the rotor 1 at 4' as important static pressure increase.

Camp and Day (1998) identified that the spike stall inception occurs when a critical rotor tip incidence is exceeded before the peak of the overall total-to-static pressure rise characteristic is reached (introduced with Figure 1.14). The total-to-static pressure rise coefficient is presented in Figure 5.31 for the rotation speeds 80%, 90%, and 100%  $\Omega$  of CREATE. The performance curves go until the last stable operating point close to surge. At 100%  $\Omega$ , the last stable operating point is clearly reached before the peak of the pressure rise

coefficient is attained. Finding a spike like stall inception in this compressor is thus also expected according to the widely accepted criterion. At 80% and 90%  $\Omega$ , the pressure rise coefficient has not reached the peak neither at the last stable operating point. Though, it is found very close to the peak because the coefficient reaches almost a flat part. The rotating instability (see previous Section) found at the part speed could be considered as multiple weak spikes because of their similar signatures in the pressure signal. Close to the peak of the pressure rise coefficient, small differences in the pressure rise coefficient can cause large mass flow rate variations. This can lead to a spike type stall inception according to Camp and Day (1998) or to rotating instabilities, as presented for CREATE.

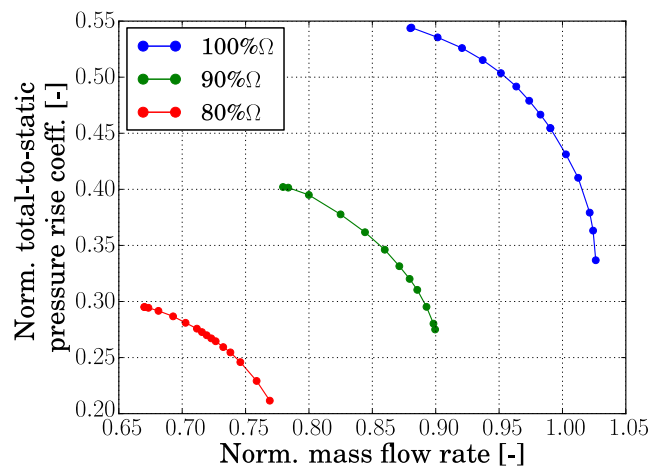


Figure 5.31 Nominal total to static pressure rise coefficient of CREATE for 80%, 90%, and 100%  $\Omega$

### 5.3.1 Characteristics of a spike

A wavelet analysis will give a fine description of the frequency trace of the spike. Figure 5.32 presents the casing pressure signal over 40 rotations up to the surge inception with low pass filtered signals at 100 kHz and 7 kHz (below BPF) at the top. In the same Figure at the bottom, the wavelet power spectrum is shown for normalized frequencies of up to 40, where the rotor 1 would induce a frequency of 64 (equal to the number of blades). This can be understood as a zoom on the stall inception already presented in Figure 5.29. The pre-spike is seen at rotation 17 as a peak in the filtered signal, and as a vertical stripe in the wavelet spectrum.

The corresponding spike like structure is shown in a zoom in Figure 5.33 along with one more exemplary occurrence of a small pre-spike and the "final" spike at rotation 38. The pre-spikes do not leave a concentrated frequency trace in the wavelet power spectrum because of their weak trace which almost does not sort out of the background noise. The final spike is seen as a clear trace in the wavelet spectrum with a maximum power at about 16 normalized frequency, and a second spot at about 8. That corresponds respectively to the inverse of the temporal extend of half and the full spike  $\Delta t_{spike} = \frac{1}{8}$ . Half the spike describes only the up or the down in static pressure. The temporal extend of the spike (red signal) can

be compared to the blade passing (blue signal) in the zoom on the rotation 38 in Figure 5.33. The eight rotor 1 blade passings are also found here.

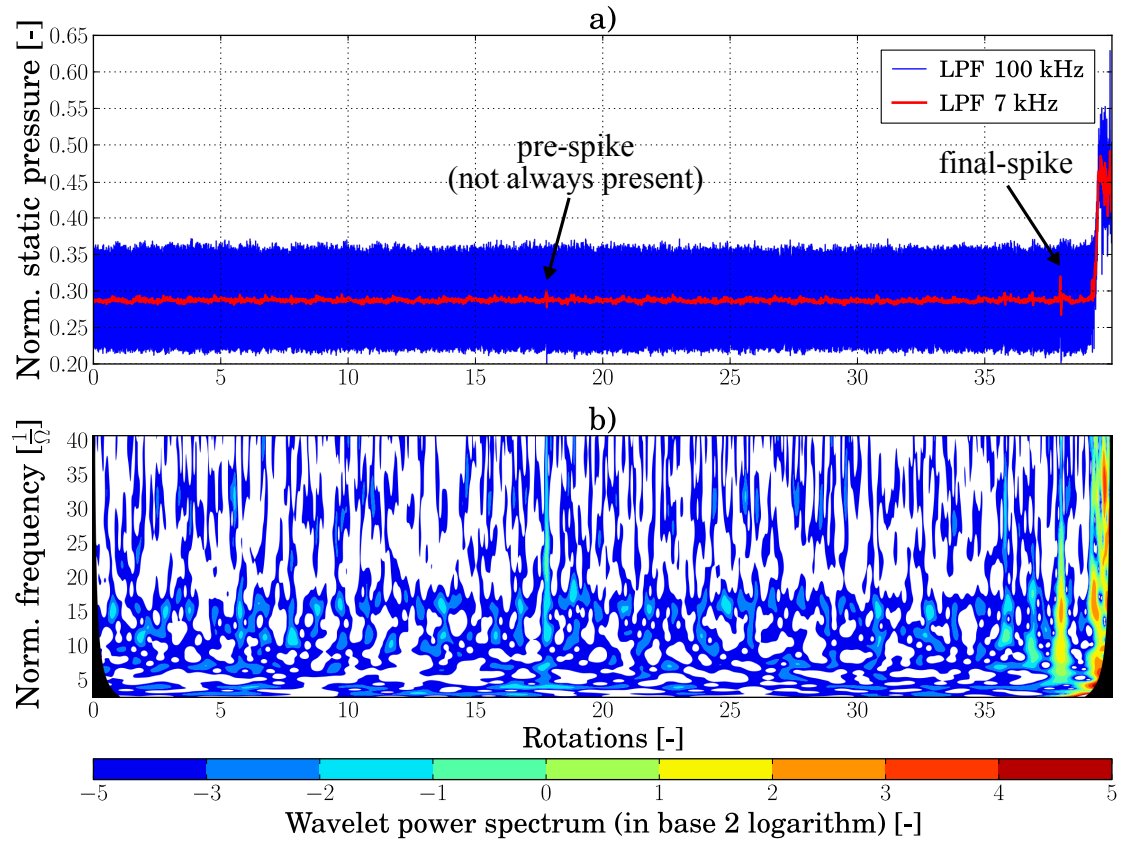


Figure 5.32 Wavelet analysis of spike inception in an example case with casing pressure signal (a) and zoom on wavelet power spectrum (b) for 40 rotations prior to surge inception

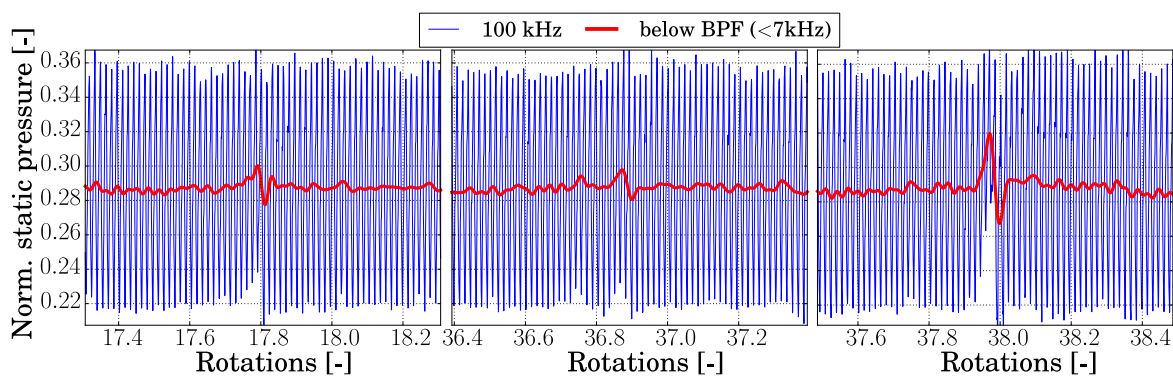


Figure 5.33 Pre-spike disturbance (left), intermediate disturbance (center) and final spike formation prior to rotating stall

Here, the spike cannot be taken as a stall precursor for an early warning system of stall onset. In most of the measurement cases, a clear trace of the spike is found only one

to five rotations prior to the surge inception. The wavelet analysis is nevertheless a good tool to detect the spike, which leaves its trace at frequencies far below the blade passing frequency due to its narrow temporal extend. A monitoring of the amplitude of the normalized frequencies between 5 and 40 would allow the detection of the spike.

Cumpsty and Greitzer (1982) developed a model for the prediction of rotating stall cell speeds based on extensive study of measurement data. For this model, the authors defined the static pressure spike ( $\Delta P_{s_{spike}}$ ) due to relative motion between a blade row and a stall cell with Equation 5.8. Interpreting the spike as a small part span stall cell, the model can be applied here.

$$\Delta P_{s_{spike}} = 0.59 \cdot \frac{c}{s} \cdot V \cdot (U - V_{spike}) \quad (5.8)$$

where  $c$  is the rotor blade chord,  $s$  the rotor blade pitch,  $V$  the flow velocity at the rotor tip (taken from URANS result field), and  $V_{spike}$  the circumferential velocity of the spike. The ratio of  $\frac{V}{V_{spike}}$  is equal to 13.7. This results in a pressure spike with a normalized amplitude of  $\Delta P_{s_{spike}}$  of 0.025, which corresponds well to the measured amplitude of 0.03 (see Figure 5.33). That means close to the expected amplitude of the spike, the flow analysis tools are able to capture the spike. And this final or strong level is reached only very close to surge inception (1 to 5 rotations). The pre-spikes do not reach this amplitude level and can therefore be defined weak spikes.

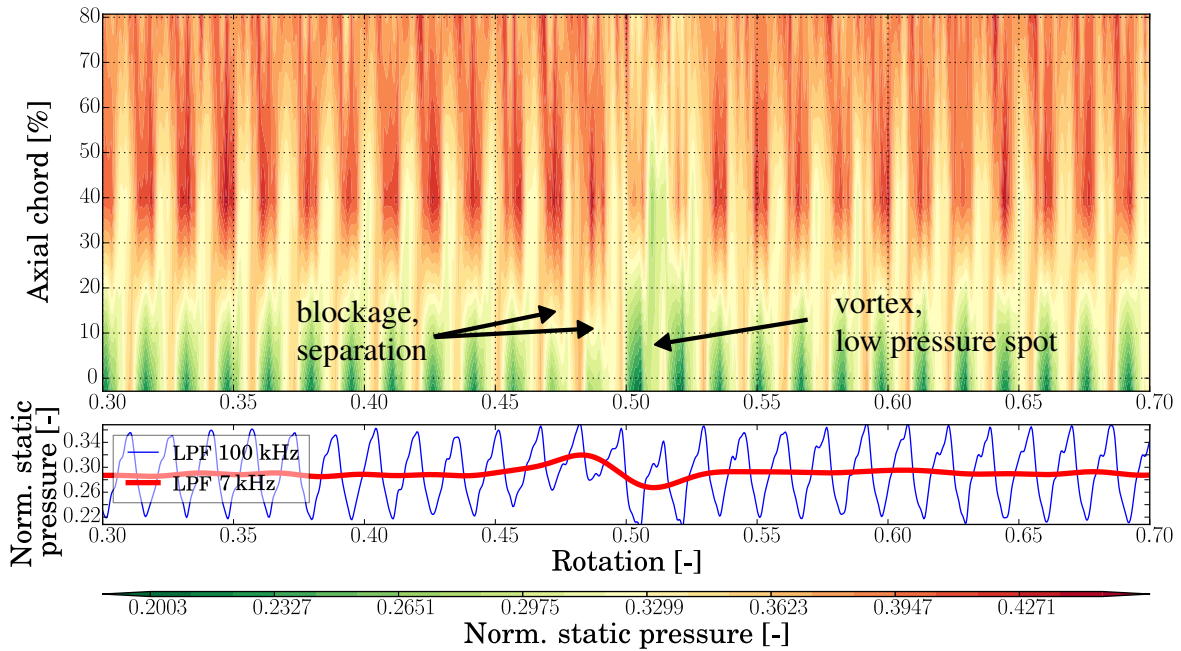


Figure 5.34 Space-time diagram (top) and and pressure signal (bottom) based on casing pressure measurements (at an identical circumferential location) above the rotor 1 with zoom on a spike cell at 100%  $\Omega$

The most detailed description of the rotating stall is limited to measurements with four casing pressure probes, positioned at four different axial positions but identical circumferential positions above the rotor 1. Remember that a time lag is caused by the probes being at different circumferential positions; this time lag cannot be well compensated for a spike event because it is not a periodic event.

Figure 5.34 presents at the top the space-time diagram zoomed on a spike in the rotor 1. The horizontal and vertical axes give the time (in rotations) and the space (axial chord) respectively. At the bottom of the same Figure, the corresponding casing pressure signal is given for the most-upstream measurement location (out of the four probes). Here the signals is given once low pass filtered at 7 kHz below the BPF and once at 100 kHz. The time axis is centered on the spike, where the filtered (red) signal shows well the up and down of the static pressure. At this zoom level, the spike appears rather flat but reaches an amplitude level of half the blade passing amplitude. The static pressure rise is caused by blockages that are found in the passages time-wise before the static pressure drop. Keep in mind that the time is advancing towards the right, and thus the rotor is turning towards the left. The blockage might be caused by a separation in the rotor passages. This perturbs the incoming flow and can induce a leading edge separation in the neighboring passage. The separation is found as static pressure drop which extends from about 10% to 50% chord. From literature it is known that a separation can form on the suction side blade surface and extend to the casing, where it is found as static pressure drop. Due to the poor spatial resolution, the axial extension might be over-estimated. In circumferential direction, one more passage is concerned by the static pressure drop. A weaker leading edge separation might be found here. Even though this is not a highly resolved instantaneous view of the flow field, it fits well to the experimental observations for example in a recent work of Pullan et al. (2015).

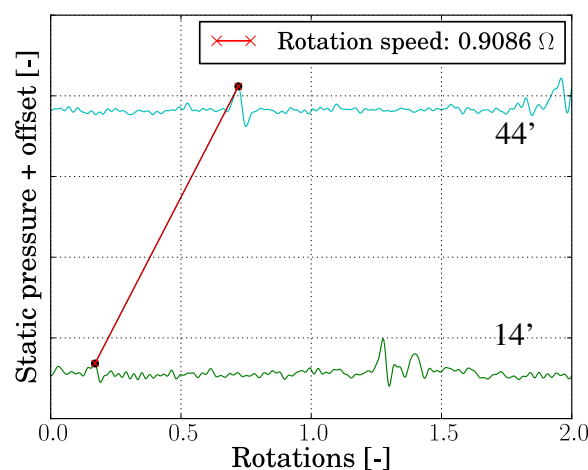


Figure 5.35 Spike rotation speed estimation with cross-correlation between two casing pressure signals at plane 260 for 100%  $\Omega$

### Rotation speed of the spike cell

The spike leaves only a very narrow trace in a measurement signal, and can evolve fast between two casing pressure probes. The rotation speed is estimated with the help of the cross-correlation between two pressure probes with a circumferential distance of 30°. Using Equation 5.7, the time lag between a flow structure passing can be found where  $X_{12}(\tau)$  reaches the maximum compared to the maximum of the auto-correlation  $X_{11}(\tau)$ . Due to the fast evolution of the spike structure, the exact rotation speed can only be estimated in a few measurement cases but they show always about the same high rotation speed. An exemplary estimation is shown in Figure 5.35. The casing pressure signals of two probes positioned at 14° and 44° are presented. The spike is detected with the highest coherence level during rotation 1 (marked with black dots). The rotation speed can be calculated based on the time lag information, knowing the circumferential distance between the probes, as gradient of the red line. The rotation speed of the spike is estimated to  $0.9086\Omega$ .

### 5.3.2 Spike compared to a rotating stall cell

The general characteristic and rotation speed of a stall cell in this compressor has been estimated geometrically with six probes around the circumference shown in Figure 5.36. The casing pressure signals are shown for the six probes around the circumference at plane 27A (downstream the rotor 2) for an old (left) and the current (right) of CREATE. This plane has been chosen for matching the previously published data for the previous version (see the work of Courtiade (2012)). The pressure signals are filtered below the blade passing frequencies for visibility reasons. In the previous compressor version, strong circumferential traveling pressure waves were found as a type of stall precursor with a maximum amplitude axially between the stator 2 and rotor 3. These pressure waves are seen as the pressure oscillations here until stall inception. These pressure waves are seen as the pressure oscillations here until stall inception.

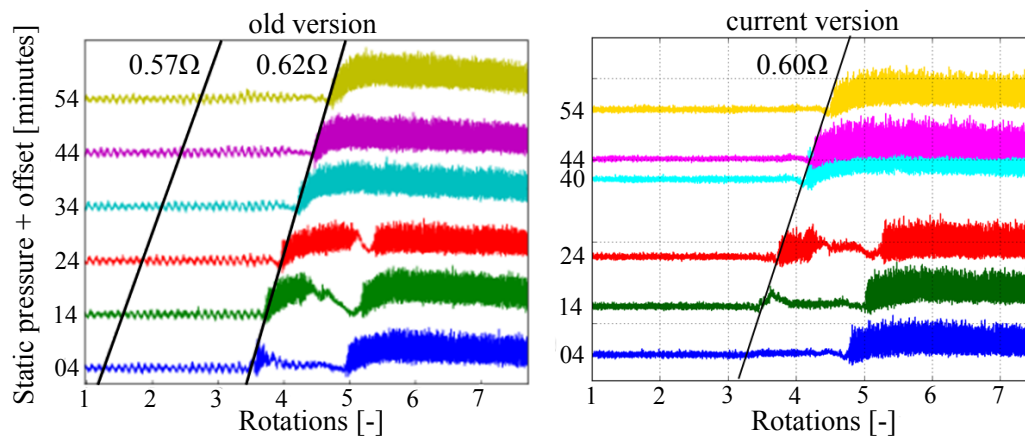


Figure 5.36 Casing pressure measurements a few rotations before surge at plane 27A (downstream the rotor 2) for an old version (left, Courtiade (2012)) and the current version (right) of CREATE



Courtiade (2012) developed an acoustic resonance model to explain the origin of the pressure waves. The growing of the full-span rotating stall cell is found between the rotations 3 to 5. In less than two rotations, the stall cell extends over the whole circumference and the compressor falls into surge. That marks a very abrupt onset of rotating stall, which is typical for high-speed compressors as in the recapitulating work of Day et al. (1999). The stall cell rotation speed has been estimated geometrically to  $0.62 \Omega$ . Here, the rotation speed is only based on the traveling of the LE of the cell. The TE travels slower because the stall cell is expanding over the circumference. In the current version of the compressor, the growing of the full-span stall cell shows the same characteristic and takes only two rotations. Also the rotation speed is basically identical with  $0.6 \Omega$ . The biggest difference lies in the behavior of the compressor prior to rotating stall. In the current version, the pre-cursor like pressure waves are not found. It has been verified that the conditions for the acoustic resonance have not changed with the current compressor version. That raises additional questions about the source of the pressure waves in the old compressor version. No new insights into answers to these questions are found with the current compressor version.

The difference between a spike and a rotating stall cell shall be discussed with the help of Figure 5.37. In Figure 5.37a, the static pressure is shown based on four casing pressure probes above the rotor 1. The horizontal and vertical axes give time and space as seen before. In Figure 5.37b, the static pressure signal is shown for one casing pressure probe close to the leading edge. The signal is given again in low-pass filtered signal below (7 kHz) and above the BPF (100 kHz). In Figure 5.37c, a wavelet transform is given based on the same signal from the leading edge. The normalized frequencies are shown up to the BPF (64). The time window shows only two rotations prior to surge. That is the short time span during which rotating stall can be observed in this high-speed compressor. This has been identified to be characteristic for high-speed compressors among others by Tryfonidis et al. (1995).

The rotating stall cell does not cause a spike like up- and down of the static pressure signal. Over a temporal extend of almost one rotation, a pressure rise occurs due to the presence of the rotating stall cell. At the top of the pressure rise, in somewhat the center of the stall cell, the blade passing frequency vanishes from the wavelet power spectrum. Here, a pressure gradient does not exist between pressure and suction side of the rotor passage. At this point in time, a characteristic pressure drop is found. This drop has a temporal extension of 4.5 rotor 1 passages, which is considered rather large. A spike was reported to affect only two rotor passages with a pressure drop. Furthermore the pressure drop is found up to the measurement probe at 80% axial chord. An acceleration of the fluid is not expected inside of the stall cell. This could otherwise explain a decrease in the static pressure. The acceleration would be expected to cause a healthy flow, which should lead to a pressure gradient between pressure and suction side, and therefore a trace of the blade passing frequency. This is not the case in the measurement of the stall cell in this compressor.

Veglio (2014) observed the same characteristic pressure drop in a rotating stall cell. The author believed in a local acceleration of the flow but had only velocity measurements at the hub to back up that theory. This raises doubts about the author's conclusions because the flow in a full span stall cell is not necessarily uniform. A velocity measurement is not available for CREATE, and therefore it is not possible to contradict the theory of a velocity

increase at the center of the stall cell. The experimental work from Levy et al. (2002) on a high-speed four stages compressor shows the pressure drop as well in a case that they call high frequency rotation stall. This characteristic is not found in the case with one rotation stall cell but only in the high frequency one with three stall cells. The pressure drop was not subject to their study though.

Judged by the axial occupation of up to at least 80% of chord, the pressure drop cannot be caused by a variation in axial extend over the circumference. This statement is valid

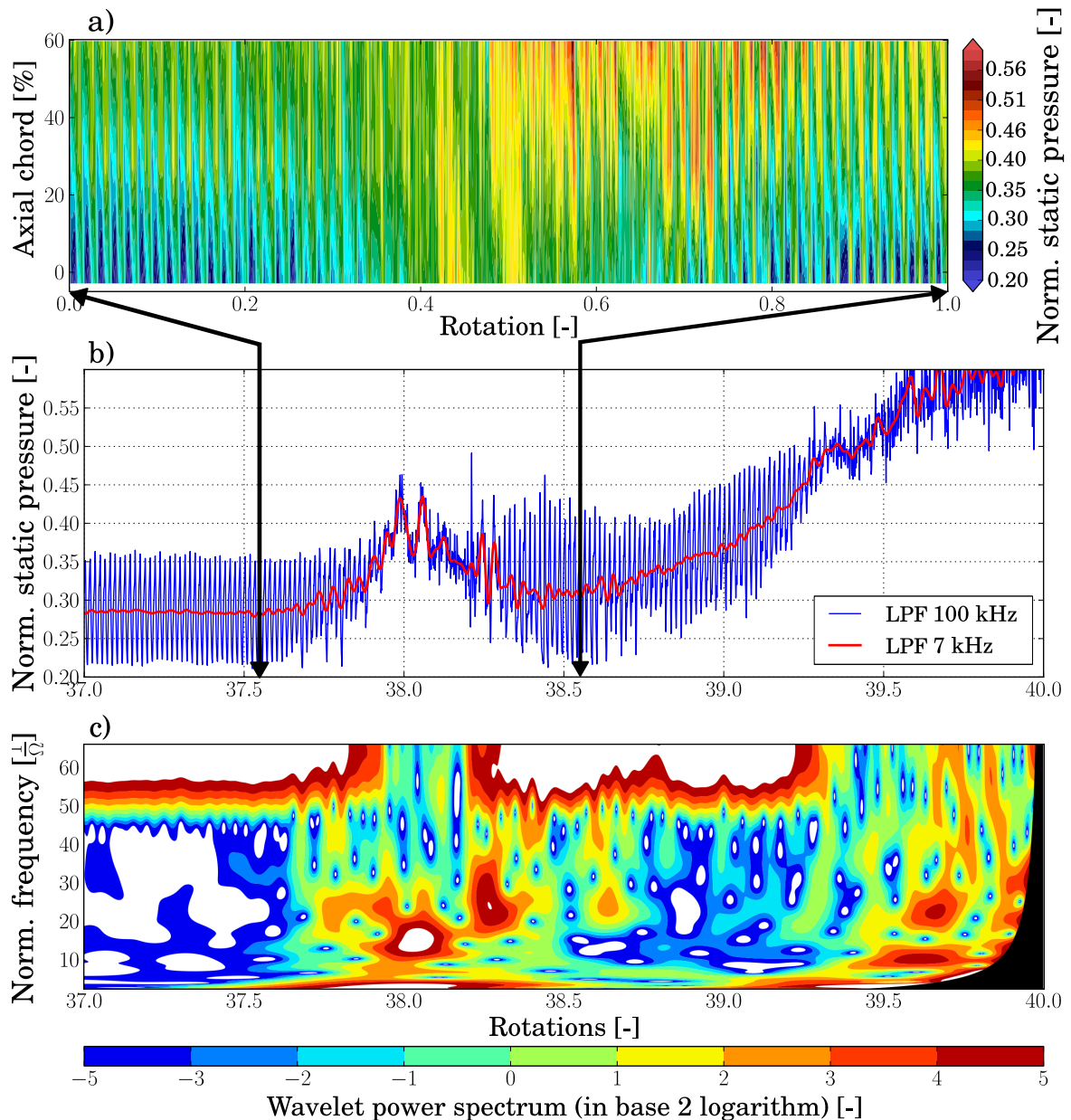


Figure 5.37 Characteristic of a stall cell based on casing pressure measurement: space-time diagram (a, top), temporal signal at LE (b, middle), wavelet power spectrum (c, bottom)



assuming that the rotating stall is formed in the rotor blade row. The variation in axial extent would lead to a probe sensing once the blockage upstream of the rotating stall cell, and once measuring the low pressure inside of the rotating stall cell. The large axial extent rather suggests a local unloading of the concerned rotor blade passages. There are also pressure drops at the side of the characteristic pressure drop. That can be seen in the red signal (7 kHz), and means that the rotating stall cell is not one structure with a smooth characteristic in circumferential direction. The pressure drop phenomenon can thus repeat in circumferential direction in a rotating stall cell.

The work of Day and Cumpsty (1978) gives rise to the idea that partially reversed flow in the stall cell moves upstream ahead of the rotor. The authors note a high velocity ahead of the rotor due to this recirculation because the fluid is reentering the rotor passage in the neighboring passages. The recirculated flow could explain a local static pressure drop due to its high velocity.

Time resolved velocity measurements could be conducted using hot-wire anemometers to characterize the velocity profile in the rotating stall cell. The cause of the main static pressure drop at the "center" of the stall cell is an intriguing open question because whatever the structure, it causes the vanishing of the blade passing frequency. That deserves an additional study of the velocity field in a future work. Nevertheless, the characteristic of a rotating stall cell is very different from the one of a spike. That can also be determined with these spatially limited measurements.

## 5.4 Summary about flow instabilities

In this Chapter, a fine description of the tip flow field was given and mainly two types of instabilities were identified in CREATE. The type depends on the operating point and the flow field investigation method (numerical or experimental). The general mis-match of the tip flow field between numerical and experimental data was identified to lead also to different instability types.

At stable operating points, the numerical data but not the measurements shows a rotating instability like rotating disturbance in the rotors 2 and 3. That was identified with a detailed description of the tip flow field based on casing pressure measurements and the unsteady RANS result field. A periodic oscillation of the tip clearance vortex trajectory was identified, leading to an interaction between the tip leakage flows in neighboring passages. This is characteristic for rotating instabilities occurring in compressor rotors. The rotating disturbance was shown to affect only very little the flow field away from the rotors 2 and 3 tip regions. Previously, the disturbance was wrongly reported as rotating stall in this compressor, which would have a larger effect on the surrounding flow field. The origin of the disturbance was pinpointed by filtering of blade-row interactions with the help of standard mixing planes in several test cases. The number of cells (32) suggested the influence of an interaction between a stator and the rotor 2 or 3 with the respective blade numbers of 112 and 80 ( $112 - 80 = 32$ ). Though the rotation speed of  $\frac{23}{32}\Omega \approx 0.719\Omega$  does not correspond to any known rotor-stator interaction. This work could prove that the origin of the disturbance does not lie in the rotor-stator-interactions even though the number of cells would suggest so.

The disturbance was demonstrated to occur in a flow field completely free of any rotor-stator interaction. A visualization of the simulation convergence process helped to identify that a disturbance rotating at rotor speed would emerge immediately at the beginning of the process. This disturbance was shown to evolve into the rotating disturbance, rotating eventually at a fraction of the rotor speed. The rotating disturbance is therefore completely auto-induced in the rotor 2, when using the URANS method.

The experimental data revealed a rotating instability but at part-speeds and only in the rotor 1 from a stable operating point until rotating stall inception. The first sign for this instability was identified with a globally increased unsteadiness of the rotor 1 tip flow field. The widely accepted common characteristics of rotating instabilities in low-speed compressors could be identified here in the high-speed compressor environment: A frequency bump was found in the casing pressure spectra, with equally spaced peaks. The spacing of the peaks corresponds to the rotating frequency of the disturbance and supports results presented in literature. The statistical analysis did not reveal any commonly reported alignment of the tip clearance vortex with the inlet plane of the rotor. This was reported in literature as a criterion but supports recent works that prove this criterion wrong. Also in the high-speed environment, the alignment is therefore not a requirement for rotating instabilities. The rotating instability was identified to have a constant rotation speed ( $0.925\Omega$ ) and number of cells. Only the cell size is varying periodically around the circumference. An amplitude modulation occurs as well and is periodic to one shaft rotation. The periodic cell size variation was identified to resemble to classical frequency modulation (FM), with a theoretical rotating disturbance frequency as a carrier frequency, and the rotation frequency of the disturbance as the modulation frequency. A statistical analysis of the casing pressure field revealed a highly intermittent behavior of the rotor tip flow close to the suction side of the rotor passages. An intermittent occurrence of a separation is possible here. The rotating instability was shown to evolve into a rotating stall just before the surge inception without the occurrence of a spike like disturbance.

At the operating range stability limit for nominal compressor speed, a spike type stall inception was identified, followed by a very abrupt onset of surge with only two rotations of rotating stall. A fine spike detection method was applied but the first signs of a spike like perturbation were identified arising usually only up to five rotations prior to surge. The characteristic up- and down of the static pressure signal due to the spike was quantified and resolved with little spatial information. The size of the spike was quantified with the help of a wavelet analysis to 16 rotor 1 passages for the up and down, and 8 passages for either the peak or valley of the spike. The respective normalized frequencies were identified in the wavelet power spectrum. The structure of the spike was compared in detail to the one of a rotating stall cell. A commonly known difference is that the spike size is measured in multiples of passages, where the stall cell size is measured in percentage of the circumference. Even though the spatial extend of the two structures is very different, misleading common characteristics were identified. Static pressure drops were identified in the rotating stall cell, which reminds of the pressure drop related to the spike. Though based on findings in literature, it is assumed that a recirculation is causing the pressure drop in the rotating stall cell compared to a vortical structure with the spike.



# Chapter 6

## Conclusions and perspectives

This experimental and numerical thesis answered to **three major objectives**, and the main conclusions are summarized hereinafter. Perspectives for future work are given alongside.

**Firstly**, the flow field of a new version of the 3.5 stages high-speed axial compressor CREATE has been characterized with a focus on the tip flow field and blade boundary layers separations. This description of the flow was accompanied by an identification of challenges for the measurement and simulation methods in such a high-speed compressor environment. The identification will help to advance with the methods in future works. This study showed that the trustworthiness analysis of experimental and numerical data needs to go further than only classical uncertainty calculations and requires deep knowledge of the flow field and the test rig.

A mis-interpretation of the measured steady performance data was demonstrated to occur easily and requires correction coefficients. The global performance of the compressor simulated with URANS computations is over-estimating the mass flow rate by 2.8%, the efficiency by 0.9 points and the pressure ratio by 3.24%. At some specific locations in the compressor, numerical and experimental exploration methods used in this study were found to give faulty flow field representations:

- The main mis-predictions of the URANS simulations were identified with the help of LDA measurements. It mainly concerns the over-prediction of the blockage induced by the tip leakage flow. Numerical results obtained with an advanced turbulence model did not yield a substantial improvement, and demonstrate the need for more advanced numerical methods such as ZDES or LES for the multistage compressor domain.
- Upstream of the stators, the measurements provided by the pneumatic pressure probes over-estimate the static pressure. This error is induced by the interaction between the stator potential field and the probe itself. Possible simplified test cases were proposed for a detailed study of that weakness. However, this important weakness of the pneumatic measurements, especially in the high-speed compressor environment with high Mach numbers, does not prevent good measurements of flow angles, the total pressures and total temperatures.

- Concerning the laser Doppler anemometry, the trustworthiness of the measurements downstream of the stators has been challenged. In this compressor, the transport of the rotor wakes through the stators might not be correctly captured with the seeding particles. That could lead to the over-estimation of the velocity with the major influence on the measured axial component downstream of the stators but not downstream of the rotors. The exact reason for this measurement problem has not been understood. In a future work, LDA measurements at several axial positions inside of a stator passage could help to identify the point of onset for the challenge to the LDA measurement technique.

A main drawback for the flow field reconstruction based on the casing pressure signals was caused by having probes at different circumferential positions distributed over the blade chord of the rotors. That causes a time delay between the circumferential spinning structures passing in front of the probes and needs to be compensated during the reconstruction. This leads to a quality loss for studies of highly intermittent flow structures. Smaller casing pressure probes positioned at identical circumferential positions over the rotor chord would allow the detailed identification of unsteady flow structures, as needed for the study of instabilities.

**The second objective** of this thesis was to investigate the effect of stator clocking in a high-speed research compressor. An identical blade count for the stators and a moving stator 2 ring allowed this study with a discretization of five clocking positions for one stator pitch. A similar blade count might be found in the rear stages of a high pressure compressor, and a detailed understanding of the flow physics of the clocking effect and identification of an optimal clocking position can thus be beneficial with no additional costs for the compressor.

Most of the available work about clocking was conducted on low-speed compressors and are yet to be proven on high-speed compressors. It was reported here that the global effect is small with both the experimental and numerical estimation. The experimentally determined maximum effect of 0.3 efficiency points lies within the measurement uncertainty band, and can only be understood as a tendency. Solely when ignoring the well emphasized uncertainty range, a trend was identified with a critical and best clocking position based on the performance data. Using URANS simulations, the prediction of the clocking effect was presented to depend on the simulation method, but stays small, with 0.1 efficiency points.

Several contributions to a weak global clocking effect were identified for eventually emphasizing that an ideal clocking position depends on the local flow conditions. The clocking position would need to be dynamically adapted to the current flow conditions (e.g. flow angles, separations) for obtaining always the best possible effect. The time mean mixing out of the stator wakes and the deformation of wakes along their flow path were shown to contribute to the weakening of the clocking effect. It is indispensable that the numerical method resolves this transport correctly for an accurate prediction of the clocking effect.

The local effect of clocking was identified to depend on the span-height because of the variation of the circumferential position of the wakes of a given stator relative to the leading edge of the downstream stator blades. Both the wake-wake interaction and the wake

passing at mid-passage of the downstream stator were shown to lead to positive effects of clocking (lower losses) depending on the concerned stator flow at the same operating point. A wake-wake interaction was identified to yield a positive effect only when there is no separated blade boundary layer present. This work showed that positive and negative effects of clocking are almost in balance in this compressor. A modified blade design could allow a matching of the leading edge shape of a stator blade to the 2D shape of the incoming stator wake for maximizing the clocking effect. This would require advanced numerical methods and prohibits any simplifications to the measurement domain for having the most accurate prediction of the wake transport. Contrary to other works, the unsteadiness in the flow field was shown to be not conclusively dependent on the clocking position.

**The third objective** of this work was to contribute to the understanding of instabilities arising close to the stability limit in axial high-speed compressors. Mainly two types of instabilities were identified in this version of CREATE: rotating instability and spike. Their study is so important because most of the published researches were conducted on low-speed research compressors or only numerically on high-speed compressors. Depending on the flow field investigation method, numerically or experimentally, different types of instability inception were found. Changes were also observed when operating the compressor at different rotation speeds. The general mis-match of the tip flow field between numerical and experimental data was identified to lead to the different instability types.

At stable operating points, a rotating instability like rotating disturbance was identified in the rotors 2 and 3 in the URANS simulations results. The rotating disturbance has been proven to be actually the periodic oscillation of the tip clearance vortex trajectory, interacting periodically with the adjacent tip leakage flow with rotating instability like characteristics. Affecting only slightly the flow of the surrounding blade rows, previous statements about labeling this rotating disturbance as rotating stall in this compressor could be proven wrong because it would affect the flow field by the important blockage it induces. Most importantly, the influence of axial and circumferential rotor-stator interaction modes on the disturbance could be excluded by filtering them with mixing planes. That is an important result because the number of cells (32 over  $360^\circ$ ) suggests wrongly the influence of the interaction between the stators and rotors 2 or 3 ( $112 - 80 = 32$ ). The source was pinpointed to a disturbance (rotating at rotor speed), which evolves into the rotating disturbance spinning at a fraction of the rotor speed ( $0.719\Omega$ ) within less than a shaft rotation.

As the major conclusion, the rotating disturbance is auto-induced in the rotor in the numerical results. A simulation of the isolated rotor 2 showed almost the same rotating disturbance characteristics as the full compressor domain simulation. The influence of the turbulence model on the rotating disturbance remains to be studied and could be therefore conducted with an isolated rotor simulation. This study might be achieved with a variation of the parameters for the used turbulence model (Wilcox  $k - \omega$ ) or a more advanced turbulence model of the type EARSM. Note that the flow solver does not allow yet the combination of the EARSM turbulence model and the sliding mesh rotor-stator interface method.

From stable operating points until stall inception, the experimental data revealed a rotating instability but only at part-speeds and in the rotor 1. The widely reported characteristics of rotating instabilities in low-speed compressors could be identified here (e.g. frequency bump with frequency rake, rotation speed, and tip unsteadiness) in the case of a high-speed compressor. As supported by recent works, it has been shown that the alignment of the tip clearance vortex with the inlet plane of the rotor is not a requirement for rotating instabilities. This was initially stated as a criterion in literature.

An original new result is that, in the case of CREATE, the rotating instability has a constant rotation speed ( $0.925\Omega$ ) over a large number of rotations and probably has a constant number of cells (20). The temporal period of the rotating instability has been identified to  $T = \frac{1}{0.925\Omega}$ . Only the cell extend is varying over time, which was shown with a detailed wavelet analysis. The cell extend variation was identified to resemble classical frequency modulation (FM), with a theoretical rotating disturbance frequency as a carrier frequency, and the rotation frequency of the disturbance as the modulation frequency. The relative movement between the disturbance and the blade row could modulate periodically the cell extend around the circumference. Additionally to the FM, the amplitude of the rotating instability is modulated periodically to one shaft rotation. That signifies the influence of the system on the instability amplitude over time and is also a new original result.

Statistical evidence of an intermittent occurrence of a separation was given but could not be resolved with the limited spatial discretization of the measurements. The modulation of the rotating separation vortices needs to be proven with higher spatially resolved measurements. In future collaborative works, the newly identified characteristics of the rotating instability could be verified on existing measurement databases for rotating instability cases presented in literature.

Eventually, the rotating instability was shown to evolve into a rotating stall cell just before the surge inception, without a spike like disturbance beforehand. Only two types of rotating stall inception exist widely accepted: modal or spike like inception. The rotating instability seems to suppress these stall inception types or could be a subtype of the spike inception. Their relation might be found in the separation vortex on the suction side of the rotor blades.

At nominal compressor speed, a spike type stall inception was identified, followed by a very abrupt onset of surge preceded by only two rotations of rotating stall. A fine spike detection method was applied but the first signs of a spike like perturbation were identified to arise usually only up to five rotations prior to surge. This is a characteristic of high-speed compressors and does not allow any stall warning system. The characteristic up and down of the static pressure signal due to the spike was quantified with a wavelet analysis and resolved with little spatial information. The difference between the pressure signature of rotating stall and a spike was demonstrated.

Once the problem of the mis-match between numerical and experimental data will be resolved for the onset of instabilities, a spike type stall inception might be studied numerically by triggering the spike onset with an imperfection such as a modified stagger angle or tip gap of one rotor blade. This numerical study might allow the spatial resolving of the spike structure in a high-speed multistage compressor.

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## AUTORISATION DE SOUTENANCE

Vu les dispositions de l'arrêté du 7 août 2006,

Vu la demande du Directeur de Thèse

Monsieur X. OTTAVY

et les rapports de

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**Monsieur SCHREIBER Johannes**

est autorisé à soutenir une thèse pour l'obtention du grade de **DOCTEUR**

**Ecole doctorale MECANIQUE, ENERGETIQUE, GENIE CIVIL ET ACOUSTIQUE**

Fait à Ecully, le 13 décembre 2016

P/Le directeur de l'E.C.L.  
La directrice des Etudes

