

On the aerodynamics and heat transfer of a high-speed turbine with advanced rotor tip geometries



 **UCLouvain**



**von KÁRMÁN INSTITUTE
FOR FLUID DYNAMICS**

Bogdan Cezar Cernat

Institute of Mechanics, Materials and Civil Engineering
Turbomachinery and Propulsion Department

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Abstract

Modern aero-engines development is primarily driven by the pursuit of maximized performance, reduced specific fuel consumption and increased components lifespan. The achievement of these multiple, conflicting objectives strongly depends on the performance and longevity of the high pressure turbine (HPT). The high-temperature, high-velocity leakage flow generated in the gap between the rotor airfoils and the casing is responsible for up to one third of the total stage losses and generates critical thermal stresses on the wetted components. An improved understanding of the flow physics established by advanced rotor tips is essential to refine the present design strategies and maximize the HPT performance.

The present thesis provides a comprehensive experimental and numerical characterization of the unsteady flow field of a HPT stage equipped with optimized rotor blade tip geometries. Two novel high-performance tip designs, one with a fully contoured shape and one presenting a multi-cavity squealer-type geometry, are compared against the baseline performance of a regular squealer tip.

The turbine stage is tested at engine-representative conditions in the short-duration turbine facility of the von Karman Institute (CT3). The one-stage HPT test section is heavily instrumented for steady and time-resolved aerodynamic and heat-transfer measurements, allowing the full aerothermal characterization of the over-tip flow field and of the stage out-flow. The HP turbine stage is operated in "rainbow rotor" configuration to allow the simultaneous testing of selected tip geometries featured in seven sectors along the rotor annulus. The multi-sector rotor minimizes the number of required tests and ensures an accurate tip-to-tip performance comparison by ruling off spurious test-to-test repeatability errors. Full-stage numerical flow simulations are employed to get a deeper insight of the HPT secondary flows and evaluate the aerothermal performance of the selected tips. The study exploits steady and unsteady flow computations, performed employing test-calibrated boundary conditions to ensure

full relevance with the experimental study.

Time-resolved casing pressure and heat transfer measurements are combined to high resolution flow predictions to characterize the impact of each tip design on the over-tip flow physics. The numerical dataset is exploited to track the development of the rotor vortical structures, correlating them with the measured level of random flow unsteadiness, and to evaluate the dominant loss generation mechanisms of each tip design. The thermal and aerodynamic contributions to the casing heat transfer field are quantified and low-order models are proposed to explain the temperature rise in the over-tip flow and the convection mechanisms at the rotor shroud. High-bandwidth aerodynamic probes are employed to investigate the unsteady total pressure and flow angle distributions at the stage outlet. Aerothermal measurements are employed in combination with CFD predictions to validate the relative flow quantities for the three rotor tip designs and endorse the tracked vortex trajectories.

A modal analysis methodology is developed to evaluate the flow unsteadiness sources within the HPT. The modal decomposition is applied to measured and predicted unsteady pressure fields at the rotor casing and outlet plane, demonstrating the relative impact of blade-rows interactions for the different tips.

A ranking of the three rotor tip designs is proposed, based on the aerodynamic and thermal performance improvement with respect to flat tips operated at relevant clearance. The multi-cavity tip is shown to outperform the other designs in terms of rotor efficiency, while the contoured profile significantly reduces the tip heat load with respect to the reference flat tip.

The rainbow-rotor configuration is characterized by combining time-resolved aerodynamic measurements with full-annulus unsteady computations performed on a HPT stage featuring a test-relevant multi-sector rotor. The flow periodicity established in each sector is demonstrated at the overtip rotor shroud and at the stage outlet plane, validating the multi-tip rotor testing concept.

The present research provides a comprehensive study on the aerodynamic and thermal performance of the advanced rotor tip designs, motivated through an in-depth analysis of the unsteady flow physics generated by

the different tip features. State-of-art measurement techniques are employed in synergy with numerical predictions which provide a high-fidelity replication of the experimental test conditions at improved spatial and temporal resolution. A solid cross-validation of the experimental and numerical methodologies is addressed while proving the validity of the tip optimization results.

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Chapter 1

Introduction

1.1 Research challenge

1.1.1 Tip leakage flow

Modern aeroengines, such as the turbofan represented in Figure 1.1, are essentially heat engines employing air as working fluid to provide thrust. Air at atmospheric conditions is first ingested by the fan located in the front part of the machine. A percentage of this air, which depends on the type and purpose of the engine, is ducted in the by-pass section whereas the remaining mass-flow is driven in the low-pressure and high-pressure compressor stages. The compressed air, which is already heated at 400 to 800 K by the work exchange in the compressor stages, is then injected in the combustion chamber. Here, heat is added at ideally constant pressure by introducing burning fuel, increasing the temperature of the working fluid by 700 to 1200 K [9]. The hot gas is then released into the turbine stages, which have the task of extracting energy needed to drive the fan, the compressor stages and the auxiliary systems. In order to achieve high cycle efficiency, modern engine designers strive to maximize the turbine inlet temperature while safeguarding the integrity of the turbine components through thermal coatings and energy-expensive cooling schemes.

Being located at the outlet of the combustion chamber, the high pressure turbine (HPT) stage is home to severe time-varying aerodynamic and heat transfer phenomena, which substantially impact the efficiency and lifespan of the machine.

High-enthalpy, high-velocity tip leakage streams (Figure 1.2a) are generated in the narrow clearance between the stationary shroud and the rotor tips causing large high-frequency variations of the over-tip aerothermal field [1, 10]. The fluid leakage is driven by the aerodynamic load of the blade tip, as shown in Figure 1.2b, and is responsible for severe thermal loads on the blade tips and on the stationary shroud.

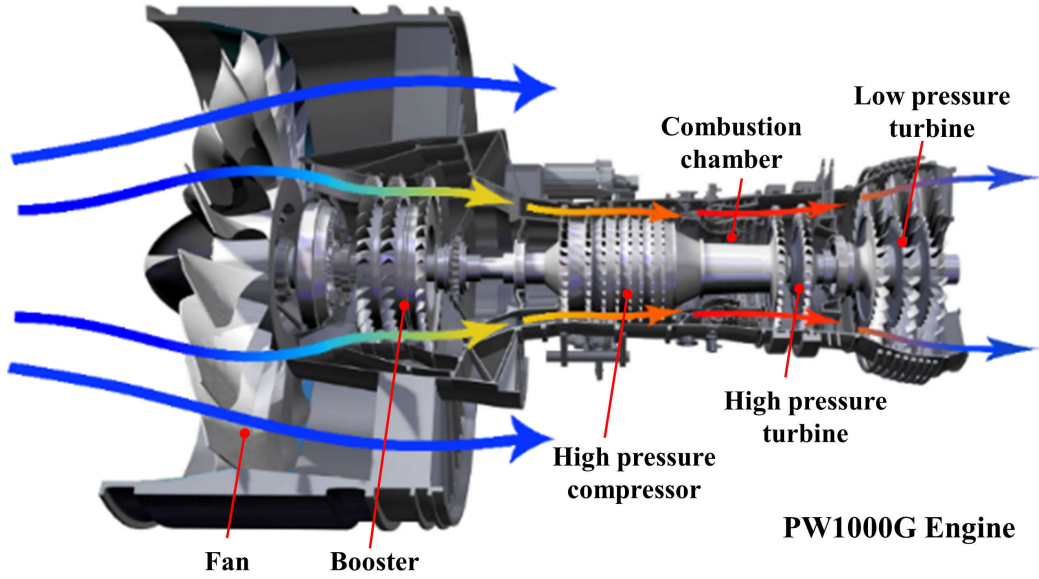


Figure 1.1: Turbofan engine overview (PW1000G)

Hence, energy-expensive cooling strategies are required to safeguard the integrity of the HPT components. Additionally, the thermal expansion of the rotor casing directly drives the operating tip clearance of the HPT rotor airfoils.

From the aerodynamic standpoint, the time-varying blade tip loading imposes transonic and even supersonic flow velocities in the tight tip gap. Generally, about one third of the total stage losses are ascribed to the gap shear flow and its subsequent mixing with the mainstream [11].

Historically, great research efforts have been devoted to the aerothermal study of leakage flows over flat rotor tips [12]. Numerical and experimental flow investigations [13] [14] confirm the inherent unsteadiness of the high-speed over-tip streams. The ejection of the leakage jet into the main flow at the suction side generates a vortical structure known as tip leakage vortex (TLV), which strongly interacts with the passage secondary flows [15].

In the continuous strive for improved engine performance and service life, industrial manufacturers demand for novel tip sealing solutions to limit the impact of the tip leakage flow on the performance and life-time of low aspect ratio turbine stages. Hence, in recent years, a variety of alternative tip geometries have been investigated to understand the impact of particular geometrical features on the local turbine flow physics.

Adding a shroud atop of the rotor blade tip was found to result in a significant reduction of the tip leakage loss [16]. This solution, however, imposes considerable limits

on the rotational speed due to increased mechanical stresses and poses prohibitive cooling requirements to counteract critical thermal loads.

Hence, substantial research efforts have been dedicated to develop advanced tip geometries for unshrouded HPT rotors, aiming to seal the tip gap and reduce the blade heat transfer [17] while maintaining high rotational speeds.

A mid-way between the two design approaches is represented by partial shrouds, or winglets [18], which were shown to provide a significant sealing improvement [19]. The increased performance, however, is obtained at the expense of increased airfoil weight, lower rotational speeds and energy-expensive cooling configurations [20].

A good potential was shown by non-conventional shroudless rotor tip designs in terms of sealing effectiveness and thermal field management [2, 21]. A significant body of published work performed in low-speed cascades [22] reported the dependency of the tip leakage flow topology [23] and loss production mechanisms [24] on the tip geometry. Numerical computations employed in support to linear cascade testing [25] allowed to identify significant variations in the heat transfer pattern over a selection of alternative tip profiles.

The rotor tip design most commonly adopted in modern engines is the squealer

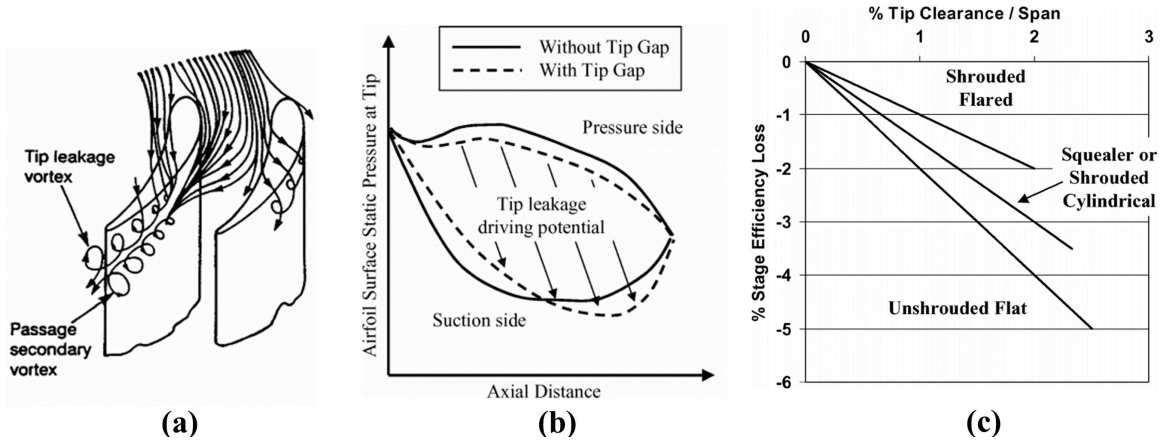


Figure 1.2: Tip leakage flow schematic (a), generic near-tip airfoil pressure field (b) and effect of tip clearance on stage efficiency (c) [1]

tip. This profile features a continuous or partial perimetral rim, which acts as a labyrinth seal and limits the over-tip fluid spillage [26]. The rim forces the leakage fluid conveyed in the squealer bath to turn in the main stream direction, delaying the flow ejection at the blade suction side [27]. Additionally, squealer tips allow for a tighter operative tip clearance, since the occasional rim rubbing on the over-tip casing represents a less severe damage compared to flat profiles. While [23] demonstrated

the effectiveness of suction-side rims in the reduction of the leakage mass flow, experimental results highlight a significant decrease of heat transfer on the cavity floor compared to classical flat tips [28].

The bottom section of Figure 1.3 shows a schematic of the leakage flow developing over a selection of squealer-type profiles with PS single rim (a), SS single rim (b) and full perimetral rim (c)[2]. Additionally, the heat transfer coefficient distribution measured in linear cascade testing is shown for the tip surface (center) and over-tip rotor casing (top). A substantial reduction of h_{tip} and h_{cas} is demonstrated over the SS-rimmed tip (b), in reason of a higher clearance value at the blade PS, where the flow is ingested in the over-tip gap.

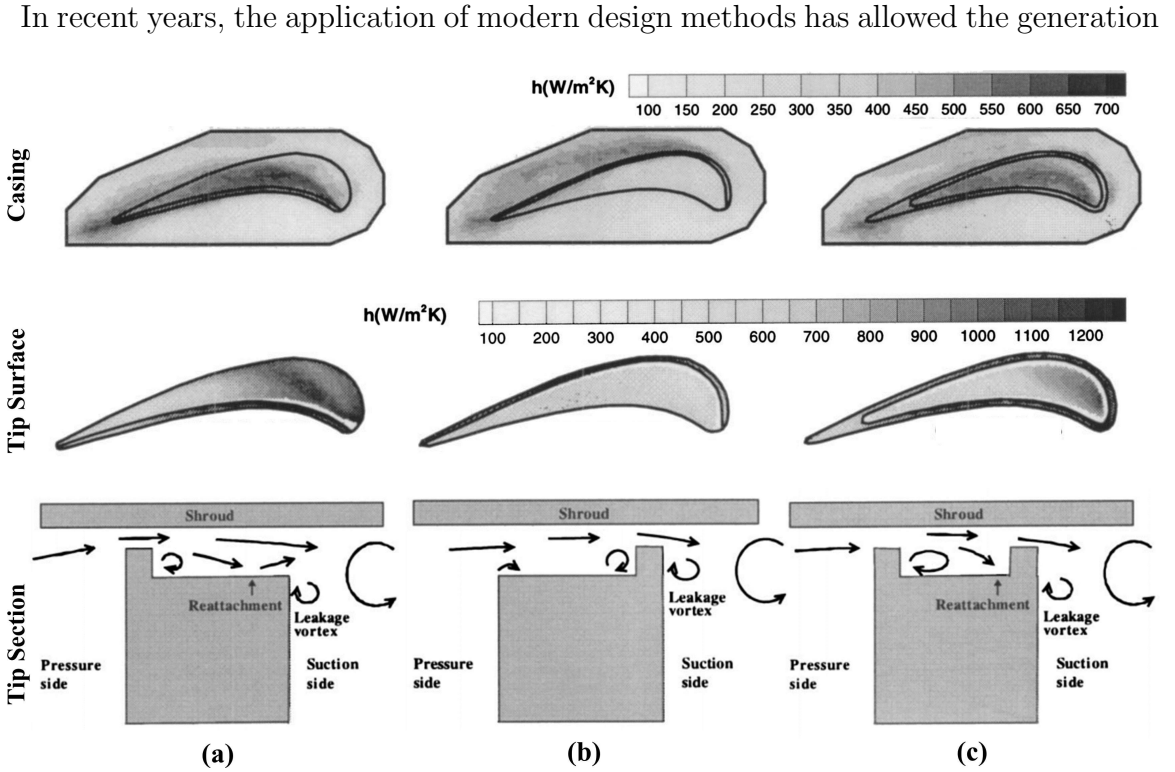


Figure 1.3: Tip leakage flow schematic (bottom) and heat transfer coefficient over tip surface (center) and rotor casing (top) for different squealer-type profiles: linear cascade measurements by [2]

of advanced rotor tip geometries with improved control of the tip leakage flow [29]. The adoption of non-conventional squealer-type profiles, characterized by complex rim patterns, allows to minimize and re-direct the fluid by-pass in the over-tip gap [30, 31] while limiting the average heat transfer on the tip surface.

Du et al. demonstrated that the adoption of arbitrary rimmed tips allows to minimize the average tip heat transfer, as presented in Figure 1.4. However, the number,

volume and topology of the tip cavities needs to be carefully tailored in order to avoid detrimental effects in terms of casing heat load and aerodynamic performance [3]. Numerical computations supported by linear cascade tests have demonstrated the tip clearance effect on the leakage mass flow and tip thermal load for flat and squealer-type profiles [25], showing a global increase in both quantities for larger operative clearances.

Consistent results were obtained numerically by [32] and partially validated in low-

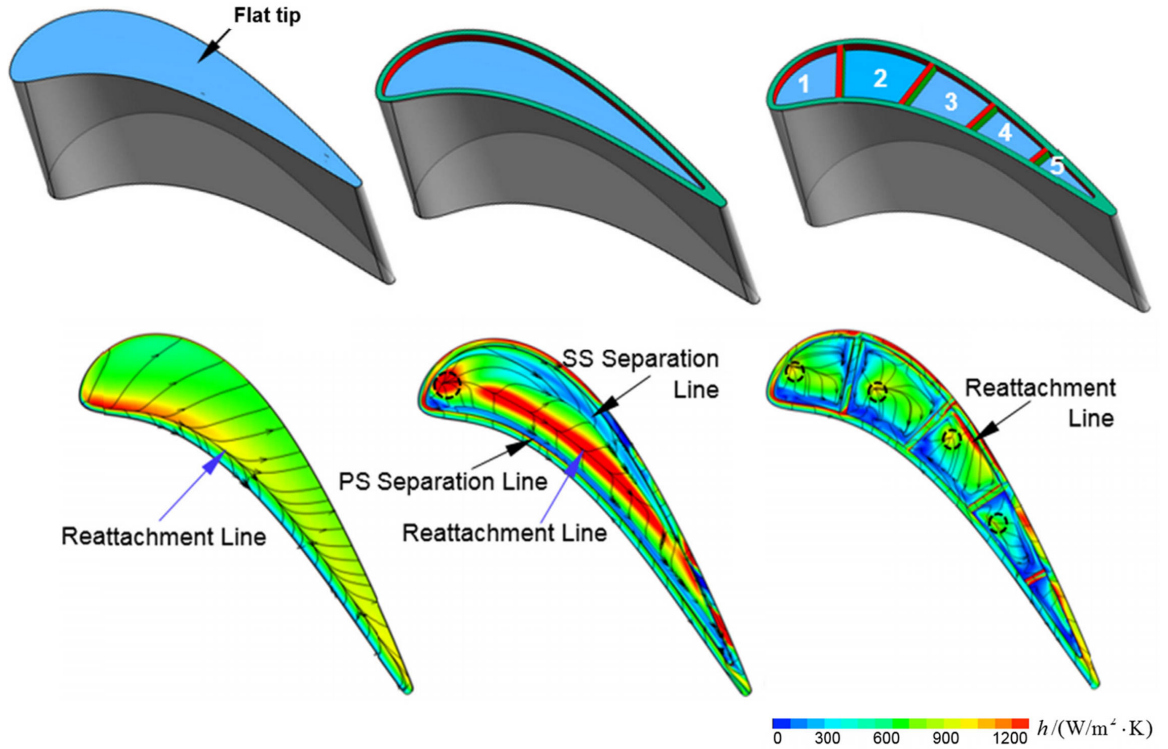


Figure 1.4: Flat tip, squealer and multi-cavity tip profiles (top) and corresponding heat transfer coefficient and streamlines on tip surface (bottom) [3]

speed linear cascade setups by [33] and [34]: blade tip and rotor shroud heat transfer increased with blade tip clearance following the increase of leakage mass flow. This behavior was observed for both flat and squealer tip designs. Bunker et al. [1] showed that the HPT stage efficiency deficit scales linearly with the increase of operative tip clearance. As shown in Figure 1.2c, higher slopes are obtained for flat tip geometries with respect to squealer designs, demonstrating the lower sensitivity of the latter to the gap dimension. Low Mach testing performed by Cho et al. [35] showed significant changes of the local heat flux pattern as a function of the tip clearance dimension (varied between 0.66% and 2.85% of the rotor chord) while minor variations were reported in the average value.

In the constant pursuit of higher power densities and weight reduction, the turbine blade loading and rotational speed is constantly increased generating transonic and supersonic tip leakage flow regimes [36]. The subsequent growth of aerodynamic losses within the tip gap and at the blade suction side, combined with strong variations of the tip heat transfer, make conventional tip geometries not suitable for high-speed flow conditions [37].

The use of carved tip profiles has been proposed to control the acceleration of the gap flow to choking conditions and desensitize the leakage mass flow from the blade tip loading [38]. A convergent-divergent nozzle was shown to promote the local acceleration of the flow in the front part of the tip up to sonic conditions, as presented in Figure 1.5 in comparison to flat tip designs operated at equal or tighter clearance (G). While desensitizing the leakage mass flow from the blade tip loading, this design was shown to significantly reduce the overall tip heat transfer [4].

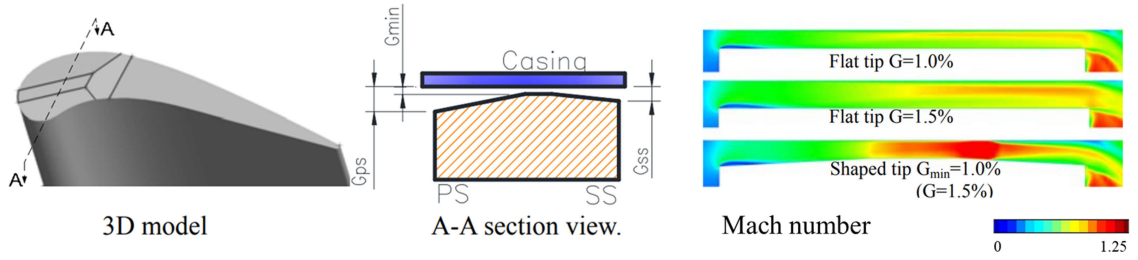


Figure 1.5: Carved tip design by Zang et al. (left), section view (center) and Mach number on section cut plane for flat and carved tip (right) [4]

1.1.2 Experimental methods

The design of advanced turbomachinery components needs to be experimentally validated at high degree of engine flow similarity before being introduced in real engines. Historically, academic and industrial research studies on gas turbine engines have been performed following a bottom-up approach, aiming at the fundamental understanding of decoupled flow phenomena through simplified experimental setups [39]. An accurate characterization of the HPT aerothermal field, however, can only be achieved in engine-relevant research facilities [40, 41] which allow to establish a highly coupled test environment operated at precise flow conditions [10].

Full-stage testing in rotating cascades is of prominent importance to allow the correct reproduction of blade-rows interactions and mimic the development of the casing

boundary layer periodically disrupted by the passage of the rotating airfoil tips. High-resolution experimental data collected in engine-representative test rigs are required to characterize the unsteady aerodynamic and heat flux pattern at the over-tip casing in order to quantify the tip sealing potential and tailor specific energy-efficient cooling schemes. Experience-driven guidelines, based upon expensive engine data and mainly pertinent to conventional tip geometries, hold no longer as a trustworthy and effective design strategy [42].

While a wide selection of studies are reported in open literature on the characterization of alternative tips in low-Mach linear cascades, scarce information are available on their actual performance at engine-relevant flow conditions.

Recent research works demonstrated the importance of correct non-dimensional flow parameters [43] and blade-casing relative motion [44, 45] for the establishment of engine-representative rotor tip flows. Dunn et al. [46] and Guenette et al. [47] were among the first to employ full stage rotating rigs to assess the inherent unsteadiness of the heat transfer and pressure field across the rotor casing. Time-resolved measurements allowed to document the strong bond between blade-induced aerodynamic fluctuations and unsteady casing heat flux [48].

Experimental studies by Thorpe et al. documented the presence of over-tip work exchange mechanisms [49, 50] which increase locally the absolute stagnation temperature of the flow above turbine entry levels [51], further demonstrating the critical necessity of testing at high degree of engine flow similarity to correctly mimic the over-tip thermal field.

A reliable, detailed evaluation of the differential performance of advanced rotor airfoils operated in engine-relevant test rigs poses an exceptional challenge in terms of measurement precision and turbine assembly tolerances. The resolution of the aerodynamic and thermal signature of several turbine rotors hosting different blade geometries and cooling schemes can be significantly affected by test-to-test repeatability errors in the turbine rig flow conditions. Additionally, manufacturing and assembly tolerances of the rotating turbine stage can introduce further uncertainties in the comparative flow studies which involve a multitude of test rig setups.

The adoption of rainbow rotors allows the simultaneous testing of several blade tip geometries mounted in different sectors of the same rotor disks. This approach allows to rule out all the test-to-test flow discrepancies and test rig assembly tolerances which may affect the quantitative resolution of the tip-dependent flow differences in different turbine setups. Due to the exceptional challenge posed in terms of turbine rig design, test section assembly, high-bandwidth aerothermal instrumentation and

data reduction routines, a very limited number of rainbow-rotor turbine flow studies are available in open literature.

Nickol et al. [52] have recently employed a multi-sector rotor in the short-duration Turbine Test Facility of the Ohio State University to assess the performance of multiple rotor blade cooling configurations. Mason et al.[53] reported a study on the unsteady heat transfer of a HPT stage operated with rainbow rotor in the short-duration Turbine Research Facility of the US Air Force. In this case, the turbine rotor was equipped with flat and squealer tip geometries presenting two different film cooling setups.

While the aforementioned studies suggest the potential of multi-sector testing for the simultaneous characterization of different rotor blade cooling schemes, no detailed assessment is available about the suitability of turbine rainbow-rotors to establish the correct flow periodicity and operative point in the individual sectors nor on the statistical convergence over short duration tests.

1.1.3 Research objectives

It is the purpose of this research to analyze the impact of a multitude of advanced rotor tip shapes over the unsteady aerodynamic and thermal field of a HPT stage operated at engine-relevant flow conditions.

The superior rotor tip designs were generated by a multi-objective optimization campaign targeting improved rotor efficiency and minimized tip heat transfer [54], performed through rotor-only steady computations which employed engine-scaled boundary conditions. In this work, the high-performance tip geometries are tested experimentally at high degree of engine flow similarity and further characterized by means of high-fidelity full stage RANS and NLH computations performed with test-calibrated boundary conditions.

The experimental flow characterization is performed simultaneously for all the tip designs by operating the HPT stage in rainbow-rotor configuration in the high-speed rotating turbine facility of the von Karman Institute [55]. The turbine annulus is divided into seven different sectors, each one hosting a dedicated tip geometry but all presenting an identical airfoil profile. The one-stage HPT test section is heavily instrumented with high-bandwidth instrumentation that allows the aerothermal characterization of each individual sector.

In this thesis, a contoured blade tip resulting from a Bezier surface-based parametrization, and a squealer-like geometry, featuring an optimized pattern of rims and cavities,

are compared against a classical single squealer design [56].

The simultaneous tip testing allows to accurately evaluate the tip-dependent flow features and quantify the tip-to-tip differential performance while ruling out all the experimental errors which may derive from test-to-test repeatability or assembly tolerances. The combination of time-resolved measurements and high-fidelity flow predictions allows to cross-validate the numerical and experimental techniques, monitor the performance of the advanced tip geometries and validate the rainbow-rotor approach for the simultaneous testing of multiple blade tip shapes.

In view of the critical knowledge gaps and pending challenges in gas turbines aerothermal research, the main objectives of the current study can be summarized as follows:

- To cross-validate high-frequency aerothermal flow measurements performed in engine-relevant flow environment with relevant high-fidelity turbine flow predictions;
- To develop and validate data reduction and analysis procedures for a high-speed rotor with multiple sectors: data synchronization, sector-to-sector comparison, unsteady flow analysis (modal analysis and advanced phase-lock averaging), rainbow rotor flow periodicity.
- To develop advanced data extraction and data processing routines for the analysis of steady and unsteady high-fidelity flow computations;
- To validate the aerothermal performance of two advanced rotor tip geometries generated by a multi-objective optimization campaign targeting improved rotor efficiency and minimized tip heat load against experimental aerothermal data collected in engine-scaled turbine tests;
- To quantify the accuracy of rotor-only and full-stage steady-RANS flow predictions against high-fidelity full-stage NLH computations;
- To assess the local impact of the individual tip geometries on the unsteady aerodynamic and heat transfer field of a HPT stage operated at high degree of engine flow similarity;
- To accurately quantify the impact of blade-rows interactions on the unsteady flow field of the HPT stage;

- To gain a deeper understanding of the dominant loss generation mechanisms produced by each individual tip design;
- To demonstrate the relevance, reliability and accuracy of rainbow rotor testing in short duration turbine facilities.

1.2 Research methodology

In order to fulfill the scientific objectives of the study, the following challenges are addressed:

- The setup and fulfillment of a turbine test campaign (150 tests) in a short-duration rotating rig operated at engine-relevant flow conditions ;
- The development of advanced data treatment techniques to convert high-frequency measurements into comprehensive sector-resolved information on the blade-periodic and random aerothermal flow fluctuations generated by each tip design;
- The development of dedicated post-processing routines for the analysis of high-fidelity steady and unsteady full-stage turbine flow simulations;
- The development of a modal analysis tool for the quantification of the blade rows interactions impact on the unsteady turbine flow field;
- The combined analysis of unsteady flow measurements and high-fidelity flow predictions to provide a detailed characterization of the tip-dependent turbine flow features;
- The analysis of the measured stochastic flow unsteadiness to validate the results of the vortex tracking algorithm applied to the CFD dataset;
- The characterization of the flow periodicity and operative flow conditions established by the rainbow rotor within the individual sectors by exploiting sector-mean time-resolved measurements and full-annulus unsteady flow predictions;
- The assessment of the correct ensemble averaging domain for each individual sector under investigation and the demonstration of the statistical convergence reached over the short-duration turbine test.

1.3 Thesis Outline

The present treatise is divided in five chapters which describe in detail the methodologies and results of the doctoral project.

Chapter 1 presents the main research topic tackled by this study. After a brief introduction on the latest developments and challenges in high pressure turbines research, the focus is moved on the hot gas path flow physics. The objectives and methodologies of the research are then characterized, clarifying the scientific and technological relevance of this research.

Chapter 2 is dedicated to the description of the experimental setup. The short-duration turbine test rig and the test article are analyzed in detail. The rainbow rotor configuration is addressed together with the advanced blade tip designs under investigation. A detailed description of the instrumentation employed for the aerothermal flow measurements is provided, including an in-depth explanation of the data reduction and data conditioning procedures.

Chapter 3 offers an overview of the numerical methodologies employed to analyze in detail the HPT stage flow mechanisms. Steady and unsteady CFD computations are addressed by providing a detailed explanation of the numerical setup and data analysis routines.

Chapter 4 reports the combined experimental and numerical results of the aerothermal study on the HP turbine stage. The time-average and time-resolved aerodynamic and heat transfer fields are cross-validated at the rotor casing and at the stage outlet, allowing to quantify the impact of the tip geometry on the unsteady turbine flow field.

Numerical results are employed to quantify the aerodynamic and thermal performance of the three tip designs. A physical insight into the secondary flows and loss generation mechanisms of each tip profile is provided by a detailed analysis of the numerical predictions.

Full-annulus rainbow-rotor unsteady computations are exploited in combination with time-resolved experimental data in order to assess the flow periodicity established by the multi-sector configuration. A modal decomposition is performed on the time-resolved numerical and experimental data in order to characterize the main sources

of unsteadiness in the HPT stage for each individual tip design.

Chapter 5 summarizes the main achievements of the study. The performance of the different HPT rotor tip designs is assessed from the aerodynamic and thermal standpoint, validating the results of the optimization campaign. The effectiveness of the rainbow-rotor setup for simultaneous multi-tip testing in short-duration facilities is evaluated together with the experimental techniques employed for the HPT flow study. The fidelity of the steady and unsteady flow predictions employed in the study is finally quantified.

Chapter 2

Experimental Methods

2.1 Rotating Turbine Test Rig

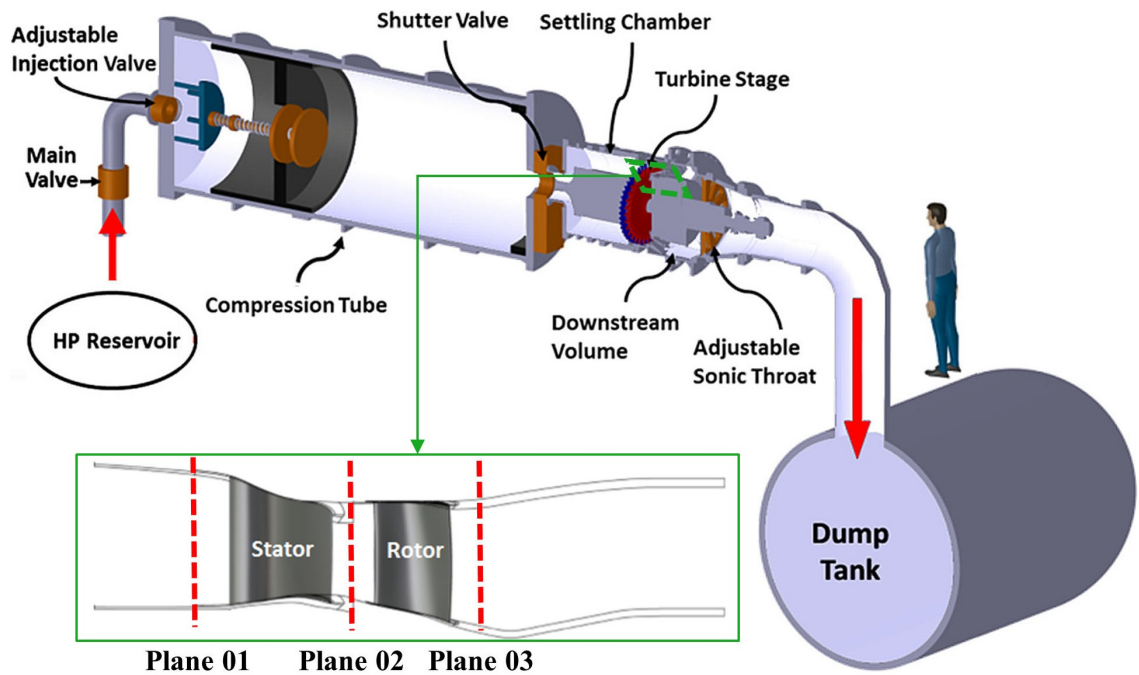


Figure 2.1: Turbine rig and test section

The experimental part of this research project is conducted at engine-representative conditions in the rotating turbine rig of the von Karman Institute. The short-duration facility operates on the principle of the compression tube [57] and allows testing a high-pressure turbine (HPT) stage at a larger scale while preserving the actual engine Mach and Reynolds numbers for a flow-on time up to 250 ms[55]. The thermal flow transient established by the blow-down of hot gas over the cold turbine article enables accurate heat transfer measurements at engine-like gas-to-wall temperature ratios.

The test rig, presented in Figure 2.1, consists of three main components: the compression tube, connected to the high pressure air supply, the test section and the downstream dump tank. A meridional view of the turbine research facility is visualized in Figure 2.2. The turbine test section is divided from the upstream cylinder by a fast-actuation shutter valve while an adjustable sonic throat at the outlet allows the fine tuning of the test mass-flow.

2.1.1 Test cycle

The typical test evolution of the flow pressure at selected locations of the short-duration test rig is presented in the top section of Figure 2.3. The short duration test allows to maintain the main aero-thermal parameters constant during a segment of the flow-on period (200-250 ms). The pressure ratio between the tube and dump tank environment sets the Mach number. The upstream absolute pressure value can be tuned to set the gas density and, therefore, the Reynolds number.

Prior to each test, a vacuum pump is employed to decrease the absolute pressure

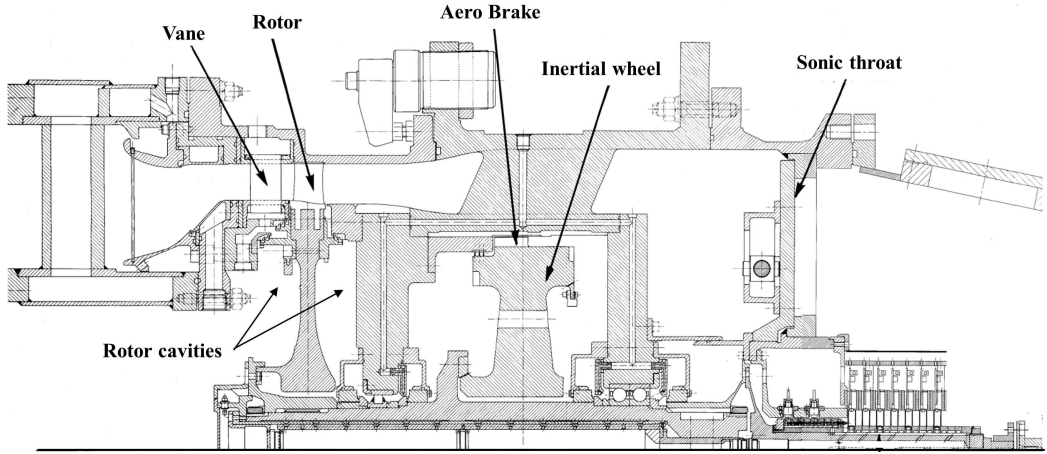


Figure 2.2: High speed turbine test rig (CT3): meridional view [5]

in the test section and dump tank environment down to 25 mbar. With the shutter valve initially closed, the pressure ($p_{i,tube}$) in the compression tube environment is regulated by means of a pneumatic ejector while the temperature ($T_{i,tube}$) is kept at ambient value. An auxiliary turbine, supplied by a 40 bar pressurized air line, is employed to gradually spin up the rotor and stabilize the rotation regime at 90% of the nominal speed. High pressure cold air is thereafter released inside the compression tube through sonic throats that control the incoming mass-flow. The cold inflow, supplied by the 300 bar pressure line, causes the light piston to slide forwards and compress the air volume ahead in a quasi-isentropic manner (Eq. 2.1).

$$T_{f,tube} = T_{i,tube} \cdot \left(\frac{p_{f,tube}}{p_{i,tube}} \right)^{\frac{\gamma-1}{\gamma}} \quad (2.1)$$

As the correct conditions of pressure ($p_{f,tube}$, black solid line in Figure 2.3) and temperature ($T_{f,tube}$) are achieved, the auxiliary turbine is disengaged and the fast opening shutter valve is opened (70 ms operation time) to allow the blow-down of hot air over the test article ($t \sim 0.9s$ in the figure). A settling chamber is interposed between the shutter section and the HPT stage, acting as a pressure-loss driven damping device that limits the flow oscillations due to the fast opening of the gate valve. The total temperature and pressure (blue dotted line) at the stage inlet rise abruptly to reach their nominal test value, which will be maintained throughout the test duration (210 ms in the present test). During the testing period, bounded by dashed gray lines, the maximum inlet total pressure variation amounts to $\pm 1-1.5$ mbar. Similarly, the static pressure at the rotor inlet and outlet endwalls, presented by dashed black and blue lines respectively, is maintained within $\pm 1-1.5$ mbar for the entire duration of the test.

A typical unsteady total pressure signal recorded in the flow path at the rotor outlet (Plane 03) is presented in Figure 2.3 decomposed in its low- and high-frequency components: the low-pass filtered signal (dash-dotted orange line) shows a good level of stability through the testing period and the AC component (solid green line) shows the magnitude of the unsteady pressure fluctuations present at the HPT outlet.

The mass flow transiting through the test section is regulated by the downstream sonic throat, finely adjusted prior to each test to match the volumetric mass-flow injected in the compression tube. Since the dump tank is initially under vacuum, this sonic throat remains choked from the shutter opening until the critical pressure ratio is reached, determining the end of the test. Hence, having the knowledge of the approximate discharge coefficient of the sonic valve and tuning the sonic valve aperture, the HPT mass flow can be estimated as by Equation 2.2.

$$\dot{m} = c_D \cdot S_{throat} \cdot \sqrt{\left(\frac{\gamma}{R} \cdot \frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{\gamma-1}} \cdot \frac{p_0}{\sqrt{T_0}}} \quad (2.2)$$

Table 2.1 presents the operational parameters adopted for the sonic valve in the present test campaign.

By considering the average P_{03} and T_{03} recorded by thermocouple rakes and pneumatic probes at midspan of the HPT outlet, it is possible to estimate the average mass

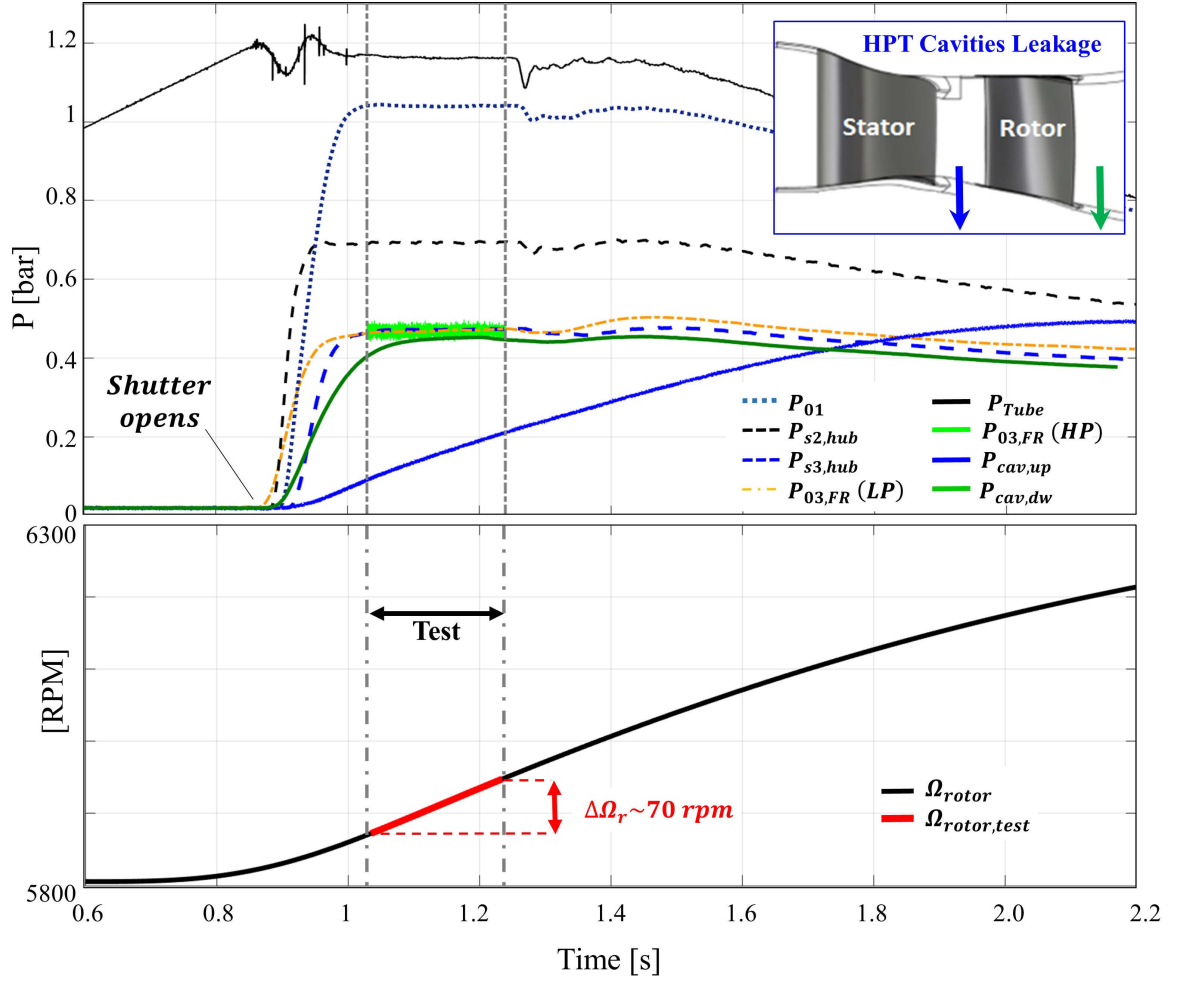


Figure 2.3: Turbine operation: aerothermal parameters

flow established for each test. In concordance with previous campaigns performed in the same test rig, a pressure loss of 10% and a total temperature drop of 5% are considered between the HPT outlet measurement plane and the sonic orifice. Following this procedure, an average mass-flow of $\dot{m}_{tot} = 5.9 \pm 0.2 \text{ kg/s}$ is estimated over a selection of 10 relevant tests.

During the blowdown, a small percentage of the the hot gas flowing through the test section leaks towards the cavities which exist upstream and downstream of the rotor disk (blue and green arrow in Fig. 2.3 respectively). In the present setup, the upstream cavity (rotor-vane gap) has a volume of 128 liters, while the downstream cavity (rotor-exit duct gap) has a volume of 12.46 liters. The pressure and temperature evolution is monitored in the two cavities throughout the entire duration of the test. The pressure variation in the two cavities, indicated by dark blue and green solid lines in the top section of Fig. 2.3, and thermocouple measurements are used to

	Quantity	Units
c_D	0.825	-
S_{throat}	0.072837	m^2

Table 2.1: CT3 downstream sonic orifice parameters

estimate the leakage mass-flow as from Equation 2.3. The transient signal recorded by the thermocouples is used to correct the slight delay produced by the pneumatic lines on the pressure recordings (0.03 s for upstream chamber and 0.07 s for downstream chamber).

$$\dot{m}_l = \frac{d}{dt} \left(\frac{P_0 \cdot V}{R \cdot T_0} \right) \quad (2.3)$$

The fluid spillage rate towards the upstream cavity is equal to 1.6-1.7% $\dot{m}_{tot}$ and does not vary significantly through the testing period in reason of the choking conditions reached at the inlet of the cavity (upstream rotor disk labyrinth). The leakage towards the downstream cavity amounts to $\sim 0.2\% \dot{m}_{tot}$ at the beginning of the test, while reaching a quasi-null value by the end of the testing period (cavity pressure equalized to flow path). The test-average leakage value, in this case, is $\sim 0.1\%$ of the total mass-flow, providing therefore a negligible impact on the total mass-flow budget of the stage.

An inertia wheel, mounted on the same shaft as the rotor disk, serves to increase the

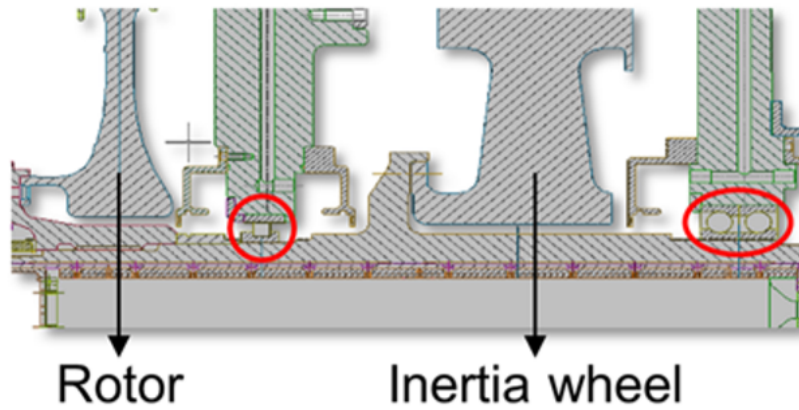


Figure 2.4: Rotor shaft bearings

total inertia of the rotating assembly (rotor, shaft, inertia wheel) to 17.8 kgm^2 and control the acceleration of the HPT rotor during the blowdown [58]. As indicated by the rotor velocity chart presented in the bottom section of Figure 2.3, this system

allows to limit the increase in rotational speed due to work extraction to $\sim 65\text{-}70$ RPM during the testing period, for an average acceleration ranging between 320 and 350 RPM/s.

During the entire facility operation time (~ 9 minutes/test), lubrication to the three rotor shaft bearings is ensured by means of an open-loop oil circuit while vibration levels are constantly monitored by two distinct accelerometers, located upstream and downstream of the inertia wheel (Figure 2.4).

2.1.2 Test Section and Rainbow Rotor Setup

For the present experimental campaign, the rig has been refurbished to host a high-subsonic uncooled turbine test article, representative of the first stage of a two-stage aeroengine HPT. A meridional view of the turbine stage is presented in Figure 2.5a. The test section is purposely designed and heavily instrumented for tip flows aerothermal research. The turbine stage counts 34 three-dimensional vanes, mounted in a convergent inlet duct, and 48 rotor airfoils.

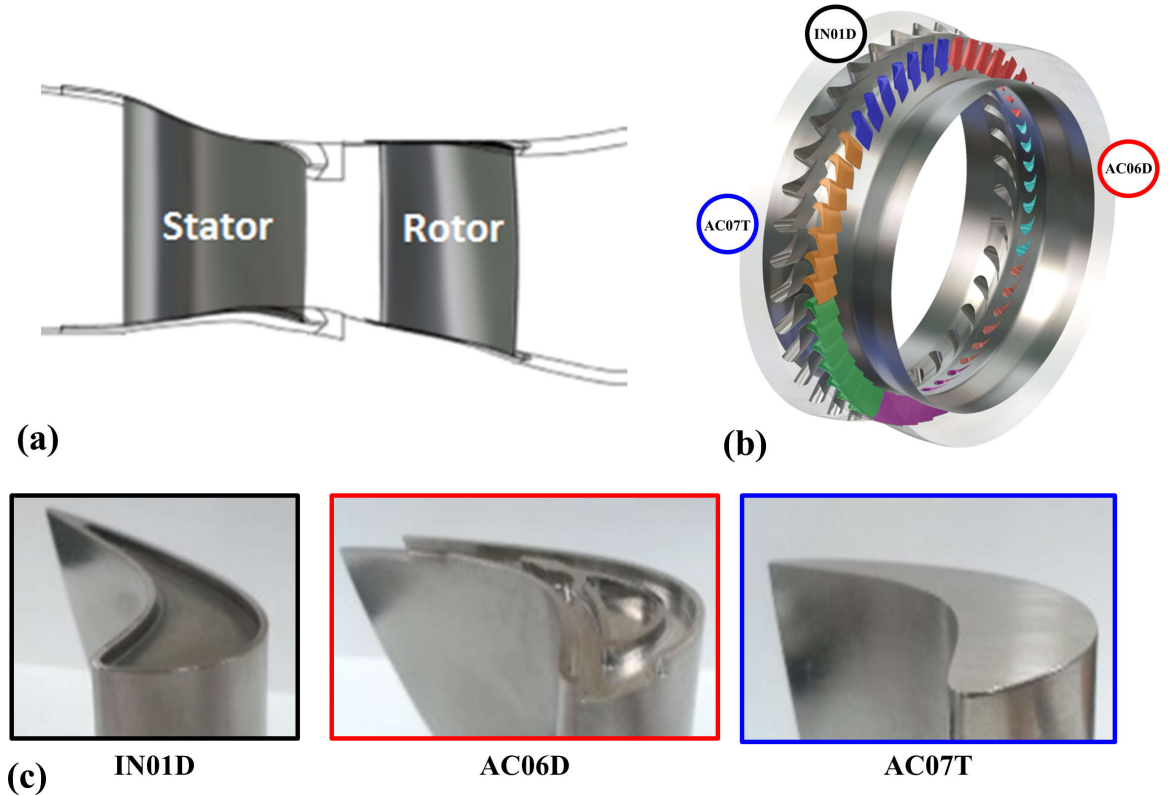


Figure 2.5: HPT stage meridional view (a), rainbow rotor setup (b) and blade tip designs under investigation (c)

Multiple optimized blade tip geometries are simultaneously evaluated in the one-stage test section by employing a "rainbow rotor" configuration (Fig. 2.5b). The rotor disk is divided in seven distinct circumferential sectors, each one presenting six or seven blades mounting the same tip design in order to ensure optimal flow periodicity. All the rotor airfoils incorporate the same baseline blade profile. The alternative tip designs were obtained from a previous multi-objective optimization effort, targeting improved aerodynamic and thermal performance [59, 54]. The rainbow rotor approach allows to minimize the measurement error due to test-to-test variations in the rig operating conditions, reduces significantly the number of experiments and offers a cost- and time-effective solution for testing multiple rotor tip geometries. The test section is purposely instrumented with high-frequency aerothermal instrumentation and dedicated data reduction techniques are developed to allow the characterization of the flow established in each individual rotor sector.

The presence of different tip geometries in the distinct circumferential sectors combined with blade-to-blade manufacturing tolerances can generate non-negligible eccentricities of the rotor assembly center of gravity. As a consequence, significant vibrations can develop during operation of the turbine test article with rainbow rotor setup, leading to fatigue failures of the turbine bearings or shaft. To prevent this risk, the rotor airfoils were weighted individually prior to installation with an error lower than $\pm 0.01\%$ of their own weight. While a maximum variation of 4% the global mean weight was detected within the set of 48 rotor blades, the airfoils sharing the same tip design did not differ by more than $\pm 0.05\%$ of the nominal value [6].

An optimal placing of the seven sectors along the annulus was adopted to reduce the rotor unbalance, expressed by U_{tot} . The vector-sum of the rotor unbalance is defined as in Equation 2.4, where every airfoil acts over the rotor assembly as a mass (m_i) concentrated at the radial position of its own center of gravity (R_i) and pointing in the direction of its azimuthal position (θ_i).

$$U_{tot} = \sqrt{\left[\sum_i m_i R_i \sin(\theta_i) \right]^2 + \left[\sum_i m_i R_i \cos(\theta_i) \right]^2} \quad (2.4)$$

Four mass-based patterns were considered for the sector-constrained airfoils sets. Two monotonous distributions were evaluated, with the mass increasing or decreasing from the first to the last airfoil of the sector, together with two symmetric patterns presenting respectively the lightest and heaviest airfoil in the central position of each sector.

The International Standard ISO 1940/1 was taken as reference to evaluate the final

rotor balancing during operation. Following the norm, a limit of $3.67 \mu m$ was computed for the vibration amplitude in order to satisfy a balance level of G2.5. Vibration levels higher than $5.7 \mu m$ were measured by the two accelerometers during a run-up of the rotating test rig performed with the optimal blade placement configuration, showing the necessity of an in-situ balancing of the rotor assembly.

A four-run strategy [60] was employed to this aim, correcting the residual unbalance with balancing weights (19g) positioned along the rotor disk. The vibration level was reduced to less than $1 \mu m$.

Prior to the manufacturing of the rotor airfoils, a geometrical modification of the

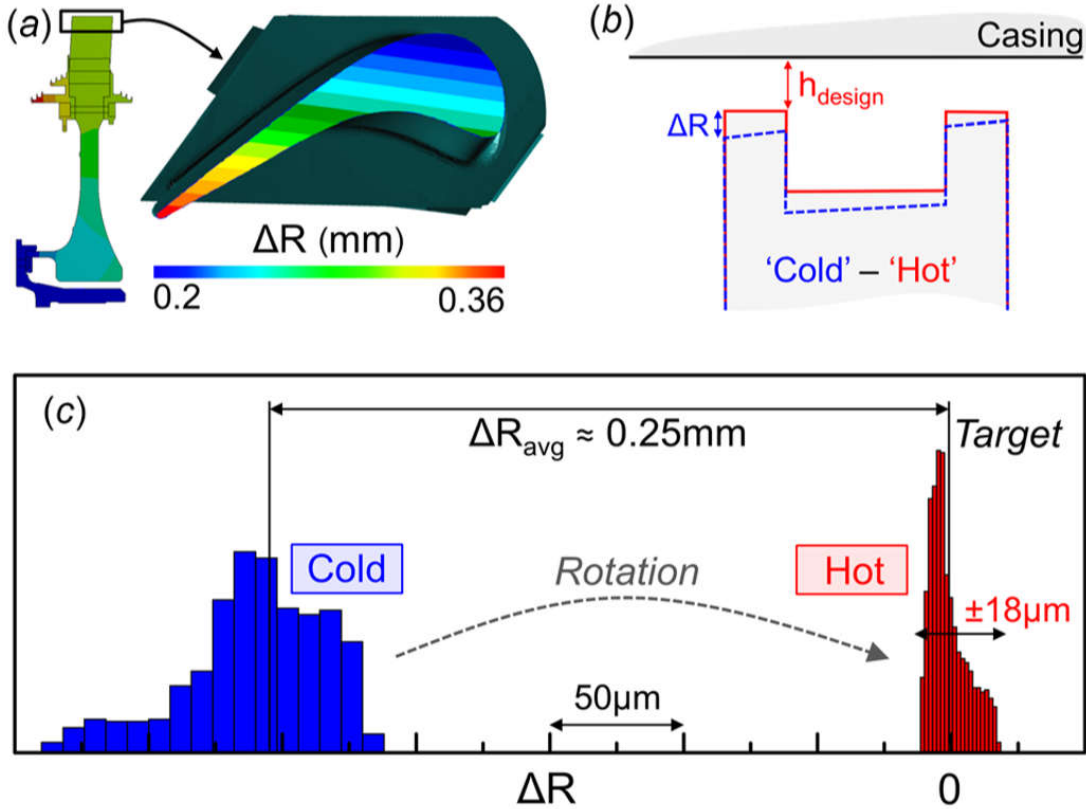


Figure 2.6: Hot cold blade transformation procedure: finite-element analysis (a), deformation compensation (b) and final radial deformation (c) [6]

selected blade tips had been operated in order to compensate for the radial deformation of the rotating assembly. A hot-cold transformation was applied in order to establish the nominal blade tip design and tip gap size under the centrifugal stresses at nominal rotational speed. As reported in [6], a finite-element (FE) analysis was performed on the rotating assembly, including disk and rotor airfoils, considering a flat tip configuration. The maximum radial deformation was evaluated to be about

40% of the nominal tip clearance value, while the deformations in axial and peripheral direction were considered negligible, being one order of magnitude smaller. The aerodynamic forces and thermal dilatation effects experienced by the airfoils in operation, computed by CFD analysis, were judged to provide comparatively negligible deformation values (several order of magnitude smaller).

In order to preserve the design height and shape of the tip gap channel during operation, the spatial variations of the tip surface radius, reported in Figure 2.6a, were employed to retrieve the cold geometry. To this aim, the reverse deformation was applied to the design tip geometries, trimming the nominal surface in function of the local deformation. The FE analysis was performed on the cold geometry of a complex squealer design to verify the validity of the procedure. The plot in Figure 2.6c shows the radius deviation from the design value at cold (blue) and hot (red) conditions. The height of the bars is proportional to the area of the tip surface characterized by the particular value of deformation. When operated at nominal speed, the average blade radius matches the nominal value within $1\ \mu m$ and the design tip geometry is matched within a range of $18\ \mu m$. The blade tip design does not substantially affect the tip radial deformation, as long as the geometry changes are confined in the near tip region (upper 4.5% of the blade span).

A titanium alloy (Ti6Al4V) was selected to manufacture the rotor airfoils to prevent the establishment of spurious electromagnetic effects on the casing measurement instrumentation [61].

2.1.3 Optimized blade tip geometries

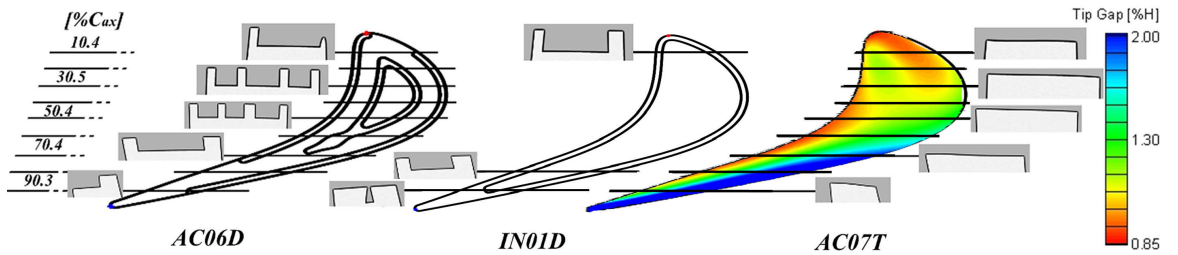


Figure 2.7: Investigated tip geometries: optimized squealer-type tip (left), baseline squealer (center) and carved profile (right)

The selection of high performance tip designs was previously generated through the optimization procedure, extensively described in [7], targeting the simultaneous increase of the aerodynamic efficiency (intended as shaft power over isentropic stage enthalpy drop as from Eq. 4.24) and the reduction of the heat transfer on

the tip surface. The optimization routine, developed at the von Karman Institute, included differential evolution (DE) and genetic algorithms (GA). Steady-state Reynolds-Averaged Navier-Stokes (RANS) simulations were employed for the evaluation of the performance of the different profiles. The flow domain, in the optimization campaign, included a single rotor blade passage. The complex tip geometries were meshed by multi-block unstructured grids and engine-scaled boundary conditions were imposed at the domain boundaries.

Two distinct parametrization strategies were used to design the tip shapes. The contoured tips [8] were parametrized through a 48 control-points general rectangular Bezier surface mapped onto the blade tip surface, as illustrated in Figure 2.8. The surface was discretized at eight sections orthogonal to the camber line where the contouring was adjusted independently at six different control points, obtaining a smooth clearance variation along the tip. The maximum deformation depth was set to 5.6% of the blade span. This tip shaping strategy allows the generation of carved designs particularly suited for operation at tight clearances, promoting the control of the tip flow acceleration up to sonic conditions and actively limiting the tip heat flux distribution [62].

The arbitrary squealer-like geometries, conversely, were designed starting from a

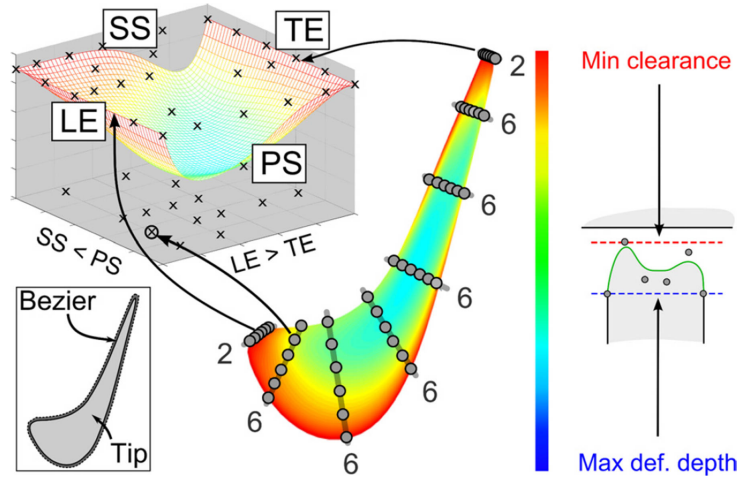


Figure 2.8: Illustration of the contoured tip parametrization [7]

block-based parametrization of the tip surface. The design philosophy, in this case, aimed at the generation of a wide range of squealer-like designs with arbitrary rims defining multiple cavities on the tip surface [63]. During the first part of the optimization, shown in Figure 2.9 as binary approach, each of the 200 blocks could evolve into a rim or cavity. In order to minimize the number of design variables, a

level-set approach was employed by mapping a Bezier surface onto the tip surface and evaluating each block center (blue dots in Fig. 2.9) with respect to a predefined threshold. Selected control points of the Bezier surface (red dots) were arbitrarily combined together in one single design variable based on the local tip thickness. The design of experiments (DoE) for the DE optimization was first created with the level-set approach. Subsequently, the best individuals were chosen to initiate the GA optimization performed with the binary approach [7]. To account for the manufacturability of the resulting designs, the cavity depth and minimum rim thickness were set to respectively 4.5% and 2% of the rotor blade span.

Of the 700 contoured tip profiles and 800 advanced squealer geometries evaluated

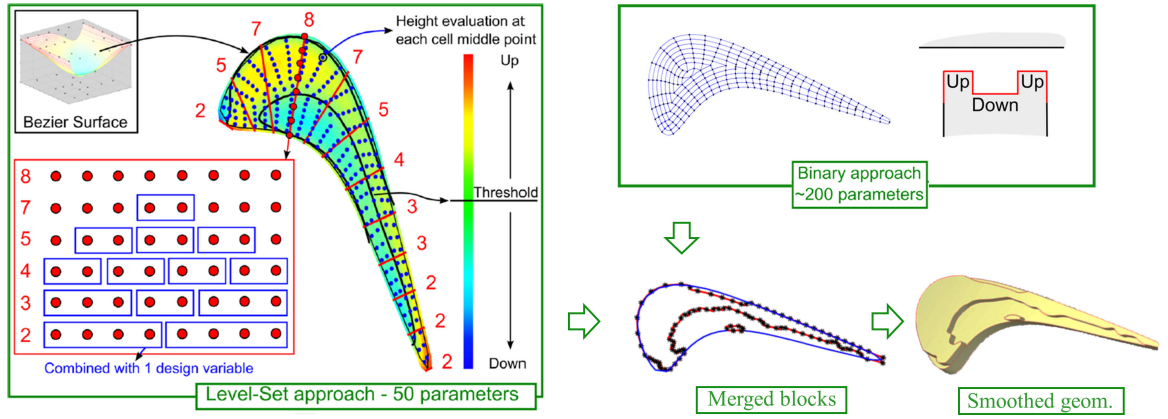


Figure 2.9: Illustration of the squealer tip parametrization [7]

in the multi-objective optimization, six advanced tip geometries were chosen for the implementation in the rainbow rotor setup. The combined experimental-numerical flow study presented in this thesis addresses three tip designs, visualized from left to right in Fig. 2.7: a multi-cavity squealer-like profile (AC06D), a conventional baseline squealer geometry (IN01D) and a contoured profile (AC07T). The schematic provides a detail of the tip cross-sections at selected axial positions along the rotor axial chord. The AC06D geometry features discontinuous lateral rims, as presented on the left-hand side of Fig. 2.7. The suction side rim presents two cuts on the rotor leading edge and trailing edge regions, extending for about 10% of the rotor axial chord. The pressure side wall opening covers the rear 30% of the rotor axial chord. Additionally, an island closed rim is present in the middle of the outer squealer cavity.

The AC07T profile, presented on the right-hand side of Figure 2.7, is a three-dimensional contoured surface with no sharp corners and a maximum tip clearance variation of $\sim 3\%$ of the rotor blade span. In the front part of the profile, the shaped surface mimics a squealer-like design. Considering a stream-wise section of the gap, the aft part

of the rotor tip presents a rounded pressure side knife-edge followed by a progressive opening of the tip channel.

The squealer-type profiles are tested with a nominal gap dimension of 1.38% of the blade span, while the carved tip geometry operates at a tight clearance value of 0.85% of the blade span.

2.1.4 Tip clearance

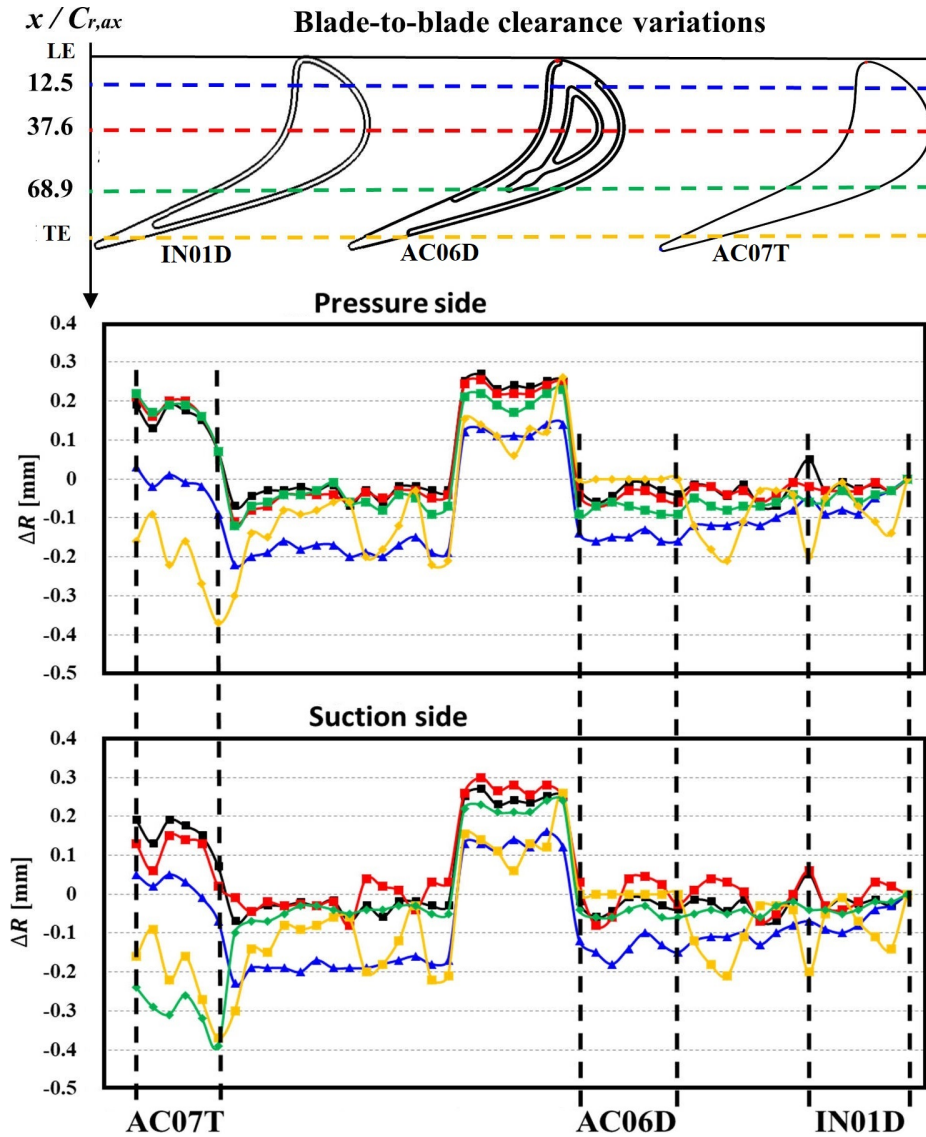


Figure 2.10: In-situ dial-gauge measurements at selected tip surface locations

In order to accurately characterize the cold geometry of each blade, in-situ measurements were performed by dial gauge on each airfoil of the rainbow rotor. This

procedure, operated on the final test assembly, allows to account for manufacturing tolerances and was performed at a selection of axial locations along the rotor chord where relevant geometrical features are located.

Figure 2.10(top) reports the selected axial locations, while the radial measurements are shown in the center and bottom sections of the figure for the blades pressure and suction side respectively. The horizontal axis refers to the blade number along the rotor annulus. The vertical axis reports the radial deviation measured on each airfoil tip with respect to the first tip of sector IN01D (last blade on the right). Within the three sectors of interest, the average sector-wise variation is about $\pm 0.05\text{mm}$. At those locations where the feature dimension could not be compared with a relevant feature in the reference blade (i.e. in the trailing edge region of blade AC06D), a null value is reported. The maximum blade-to-blade variation is detected at the trailing edge of blade AC07T and corresponds to $\pm 0.1\text{mm}$.

The tip gap established during turbine operation must be estimated along the rotor

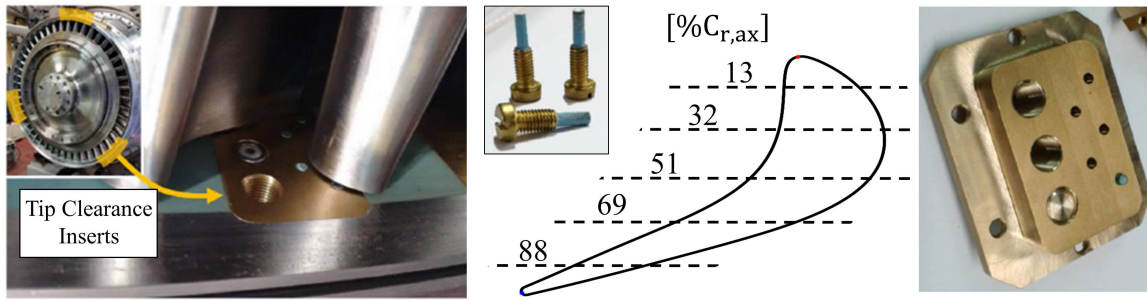


Figure 2.11: Tip clearance measurement instrumentation: measurement slots (left), wear gauges axial locations (center) and instrumented insert (right)

annulus to demonstrate the correct assembly of the turbine test article. Casing-to-tip proximity measurements are performed at five axial locations (between 13% and 88% of the rotor axial chord) by means of instrumented inserts located at three equally-spaced circumferential positions in the over-tip shroud. As presented in Figure 2.11, each insert is equipped with five wear gauges and one capacitive proximity sensor and flush mounted with the abradable line.

The wear gauges are abradable-made cylinders attached to Monel tubes of 1.0 mm diameter and fitted on a drilled screw DIN84 M5x10. During the rotor acceleration, as the blades deform under the action of the centrifugal force, the tip of each abradable pin is gradually machined.

Prior to the test campaign, a calibration of the operational tip clearance is performed, spinning up the rotor to different peak speeds in consecutive tests. For each

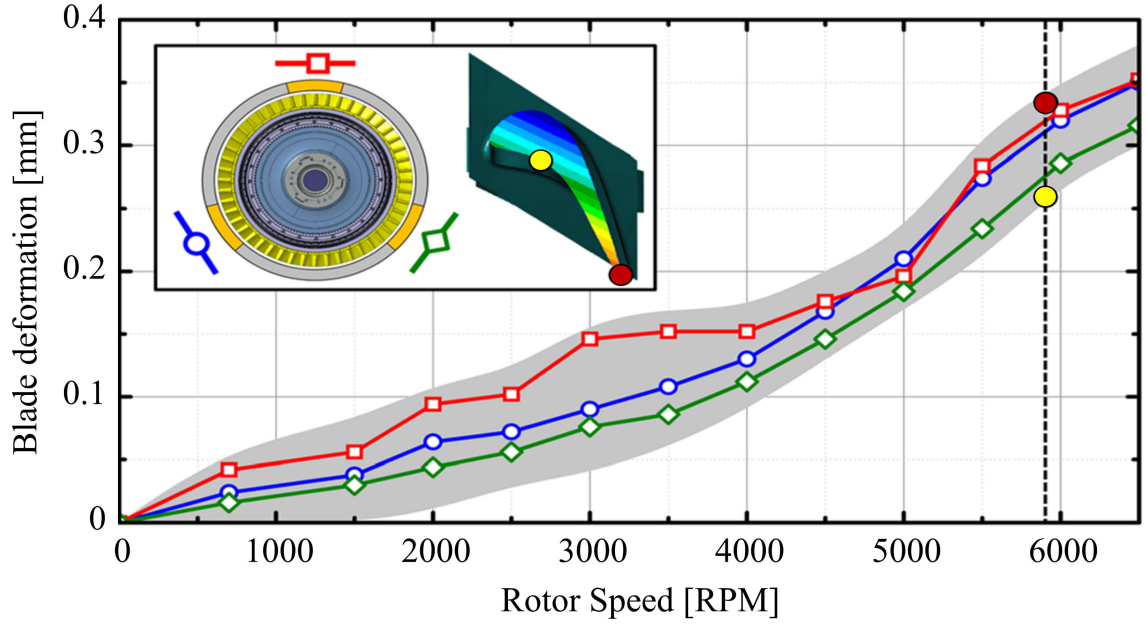


Figure 2.12: Rotor airfoils deformation at increasing rotational speed values

rotational speed, the height of the pin will be representative of the gap established by the longest of the 48 rotor airfoils. After each test, the pin extension from the casing endwall is measured by means of a dial gauge (systematic error of ± 0.01 mm) to compute the minimum gap dimension for that particular rotational speed. The wear gauge relatively large diameter (3 mm) provides a sufficiently large datum surface for an accurate dial gauge measurement.

The airfoil deformation in function of the maximum test rotational speed is presented in Figure 2.12. The three curves indicate wear gauge measurements performed at the three different circumferential slots, while the gray area covers the minimum-to-maximum deformation recorded across the rotor axial chord at each rotational regime. The maximum and minimum deformation values predicted by FE calculations are also reported for the design testing speed of 5900 RPM, indicated by a red and yellow circle respectively. A significantly good agreement with the experimental data is observed, with the tip trailing edge showing the largest deformation and the smallest radial displacement located in the early pressure side (40% of the rotor axial chord). The maximum and minimum displacement values correspond to 0.33 mm and 0.26 mm respectively.

The running tip clearance as a function of rotational speed is reported Figure 2.13 for the three measurement slots, with the dotted curves representing the minimum and maximum deformation registered by the wear gauges along the rotor axial chord. The slot-to-slot clearance variation, presenting a maximum value of 0.3 mm, depends

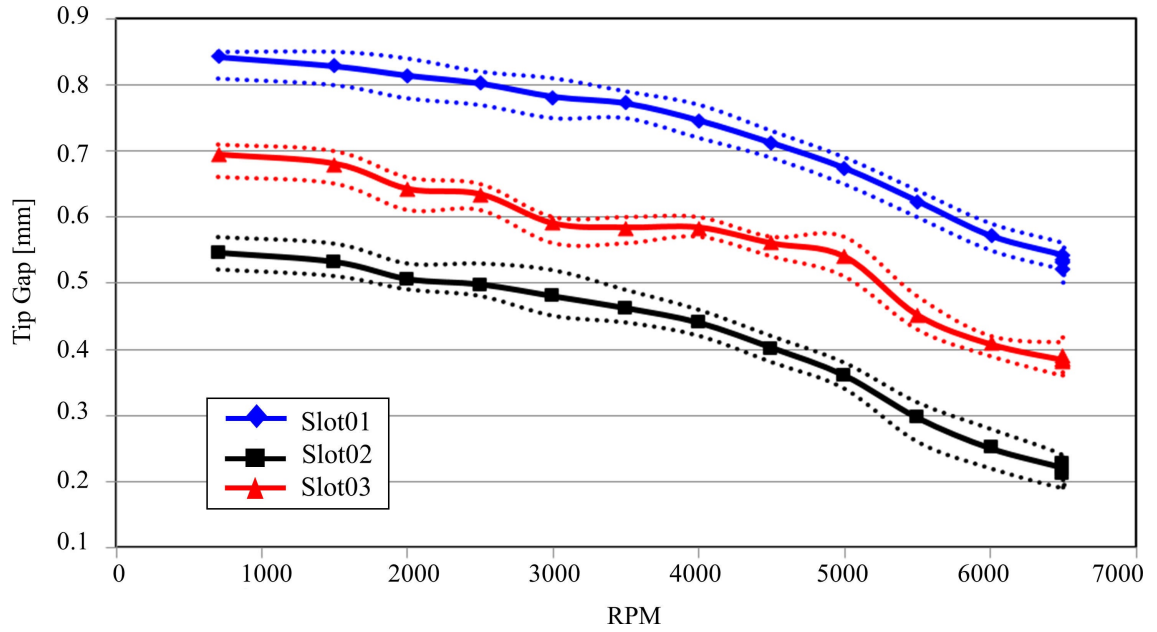


Figure 2.13: Minimum tip clearance at increasing rotational speed values

on the mutual alignment of the rotor disk and the external static shroud and on the shape tolerances of the casing ring. The clearance variation along the rotor blade, additionally, is shown to slightly decrease for increasing rotational speeds, indicating the correct transition of the blade tips from the cold to the hot geometry.

The radial distance between the static casing and the leading edge of the longest airfoil tip was measured in "cold" conditions along the rotor annulus. Knowing the dimensional difference between the rotor longest blade and the average measured gap of the "design-" and "tight-gap" airfoils, and exploiting the minimum running clearance evolution presented in Figure 2.13, the "tight" and "design" running tip clearance values were estimated along the annulus as in Figure 2.14. The dotted lines, in the same plot, represent the theoretical values of tip clearance for the two configurations. The occurrence of an isolated clearance peak at about 145 degrees is due to the presence of one tip clearance measurement insert.

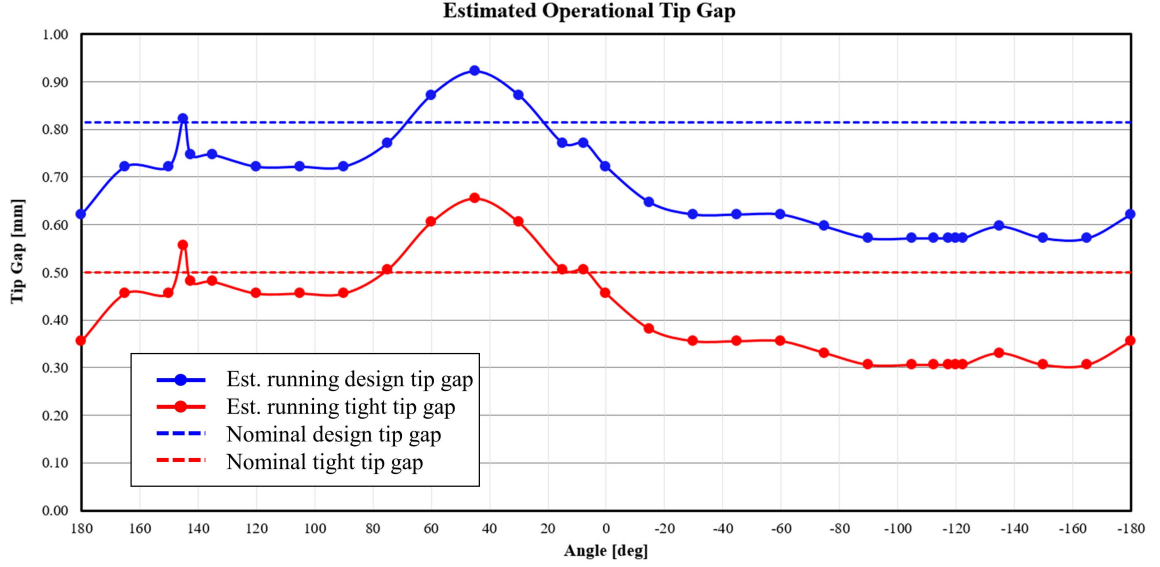


Figure 2.14: Estimated running tip clearance along full annulus

2.2 Aerothermal measurements

This section presents the aerodynamic and thermal measurement techniques employed to investigate the HPT stage flow field.

The inlet of the test article is equipped with reference probes and measurement rakes to accurately determine the pressure, temperature and turbulence intensity established at each test. The rotor shroud is heavily instrumented with high-bandwidth pressure and heat transfer sensor arrays which allow the time-resolved characterization of the over-tip flow field established by the individual tip profiles. The stage outlet flow field is studied by periodically traversing fast-response aerodynamic probes over an investigation region which covers the full spatial periodicity of the HP stage (1 vane pitch) and extends over the full blade span.

A comprehensive list of instrumentation, a plan of the instrumented rig and a table of relevant measurement uncertainties are provided in Appendix A.

2.2.1 Inlet flow conditions

The stage inlet flow is monitored by means of reference probes, presented in Figure 2.15a, which are positioned at midspan for the whole duration of the campaign. The probes present a Kiel pitot head and a bare K-type thermocouple of $50 \mu\text{m}$ diameter. The temperature recovery factor of the thermocouple, defined as the ratio $T_{0,meas}/T_0$, has been characterized experimentally to be 99.94% at $M=0.1$, corresponding to a temperature correction of 0.25 K at the inlet flow conditions ($T_{01} = 440\text{K}$). The

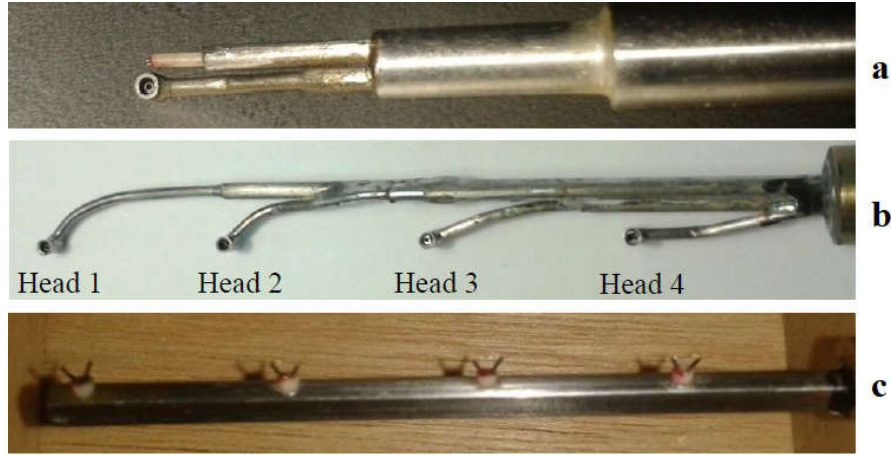


Figure 2.15: Plane 01 aeothermal probes: reference probe (a), total pressure rake (b) and total temperature rake (c)

pressure recovery factor of the Kiel probe was measured to be $p_{0,meas}/p_0 = 99.99\%$ up to $M=0.3$. Both recovery factors are presented in Figure 2.16.

Radial profiles of time-mean total temperature, total pressure and turbulence intensity are recorded by measurement rakes which can be displaced at six different circumferential locations to monitor the flow uniformity around the turbine annulus. The 4-head total pressure rake, traversed over the full radial span of the inlet channel, is shown in Figure 2.15b. Every head presents a pressure pipe of 0.6 mm internal diameter, surrounded by a Kiel geometry of 2 mm external diameter to desensitize the total pressure measurement for small flow angle variations. The pressure recovery factor was characterized in a controlled calibration nozzle, showing values above 99.9% for all the four heads at $M=0.1$ in a range of $\pm 30^\circ$ yaw angle and of $\pm 20^\circ$ pitch angle.

Three 4-head thermocouple rakes (Figure 2.15c) are employed for the measurement of the inlet total temperature profile. Each head mounts a bare $50\ \mu\text{m}$ -diameter K-type thermocouple, with a wire length-to-diameter ratio of 80, ensuring a frequency response of several tens of Hz while minimizing the unsteady conduction effects of the leads. A recovery factor of 0.9996 was measured in a calibration bench at $M=0.1$, corresponding to a temperature correction of 0.15 K at the inlet flow conditions, as presented in Figure 2.16.

Two hot-wire rakes, presented in Figure 2.17, were employed to characterize the stage inlet turbulence intensity. Each rake presents two wires radially spaced by 37 mm, allowing to cover the entire radial span, and a thermocouple which allows to measure the flow total temperature.

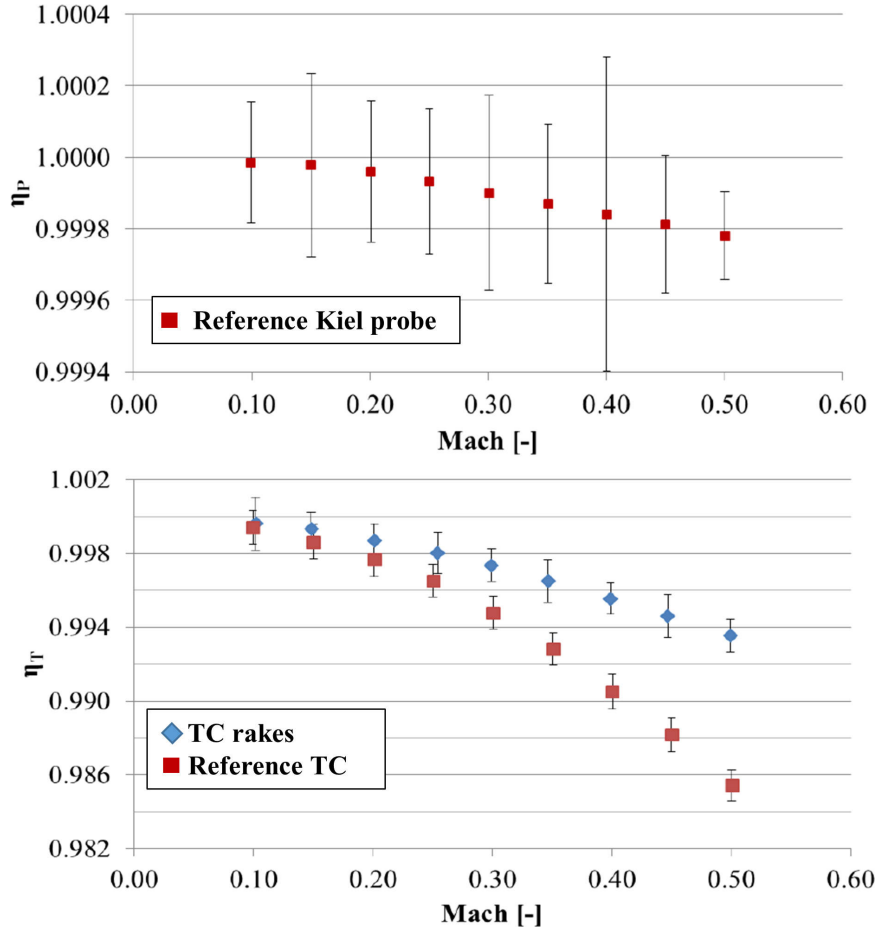


Figure 2.16: Pressure (a) and temperature (b) probe recovery factors

The wires present a diameter of $9 \mu\text{m}$, ensuring optimal structural strength and ageing resistance, and a total length of 2 mm. The extremes of the wires are plated with copper to prevent interference with the prongs, leaving an active wire length of 1 mm exposed to the flow.

The wire length-to-diameter ratio allows to minimize the errors induced by non-axial flow conditions. A Nickel-Platinum alloy was employed for the wire in order to reduce oxidation effects in high temperature flows and allow operational wire temperatures of 500K.

2.2.2 HP Vane

The vane pressure profile is measured on two distinct airfoils instrumented with pressure taps at respectively 50% and 90% of the span, as presented in Figure 2.18.

A total of 24 taps are employed, 14 at the vane suction side and 10 at the pressure

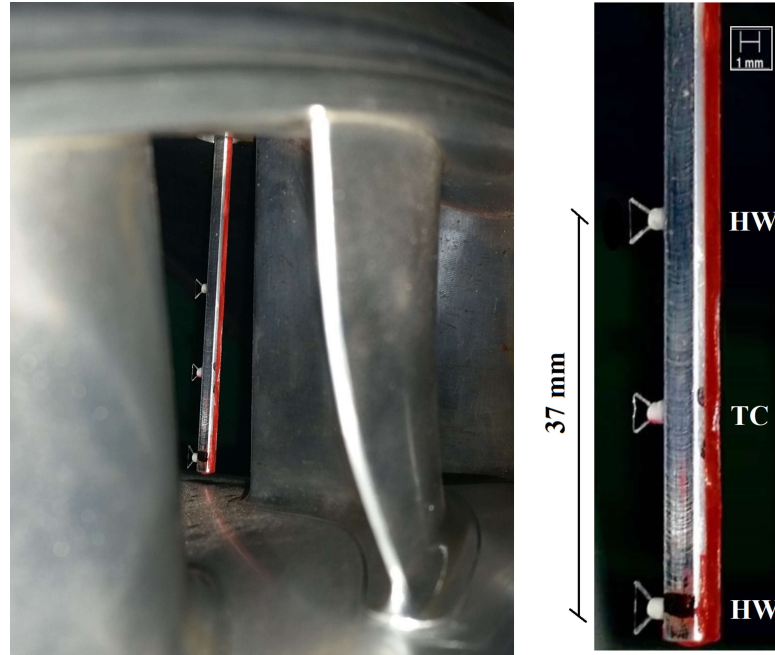


Figure 2.17: Plane 01 hot-wire rakes

side. Each tap presents a diameter of 1 mm, with the last three taps near the trailing edge being drilled in flush-mounted pressure tubes due to the reduced thickness of the aft part of the airfoil. Pneumatic lines with an internal diameter of 1 mm convey the pressure information to a remote scanner system (Scanivalve scan module ZOC 33/64PX 15 psid).



Figure 2.18: Vanes instrumented with static pressure taps at 50% span (a) and 90% span (b)

2.2.3 Vane outlet endwall pressure

The hub endwall static pressure is measured downstream of the vane in Plane 02. A total of 12 pneumatic taps and 6 fast response sensors (Kulite XCQ-062 25A) are embedded in a metallic insert which covers two vane pitches and is flush mounted at

the hub endwall. All the pressure taps present a diameter of 0.8 mm. The Kulite sensors are recessed by 0.2 mm to ensure a safe operation of the transducers without compromising the frequency response.



Figure 2.19: Instrumented pressure insert - hub endwall plane 02

2.2.4 Rotor Casing

Casing Static Pressure Instrumentation.

The over-tip static pressure field is investigated by means of the instrumented insert equipped with 66 pressure taps shown in Figure 2.20.

The instrumentation array is composed of 33 fast-response piezo-resistive pressure sensors (Kulite type XCQ-062 25A) and 33 pneumatic lines sampled by an external pressure scanner (Scanivalve® scan module ZOC 33/64PX ± 15 psid controlled by a DSM4000 unit). All taps have a diameter of 0.6 mm to ensure an optimal spatial resolution of the blade tip features. The fast-response pressure sensors were recessed 0.2 mm inside the metallic block to reduce the probability of damages from the near rotating blades while preserving a frequency response above 90 kHz. A dedicated study presented in [64] demonstrated that such bandwidth allows resolving over 95% of the information present in the unsteady casing pressure field established by a flat tip operated at design clearance ($1.38\% h_r$). The pneumatic pressure measurement setup ensures a frequency response of about 50 Hz thanks to a reduced pneumatic pipe length (< 500 mm).

The pressure sensors are arranged in a regular array composed by 6 measurement points in the circumferential direction and 11 measurement points over the axial direction. The measurement pattern covers one vane pitch and extends between -5% and +105% of the rotor axial chord. Thanks to the symmetric arrangement of the gauge grid, a rotation of the instrumented insert by 180 degrees allows to swap the positions of the pneumatic taps with those of the piezo-resistive sensors. For the same

spatial resolution, this design halves the number of high-frequency transducers and facilitates their packaging into the instrumented block.

The semi-conductor sensors are calibrated in-situ exploiting a pressure and thermal transient generated in the turbine rig [65]. The pneumatic taps provide the reference pressure value. The calibration procedure allows for compensation of the temperature effects and reduces the systematic uncertainty of the sensors to $\pm 3\text{-}5$ mbar (20:1) with a precision error of about ± 0.1 mbar.

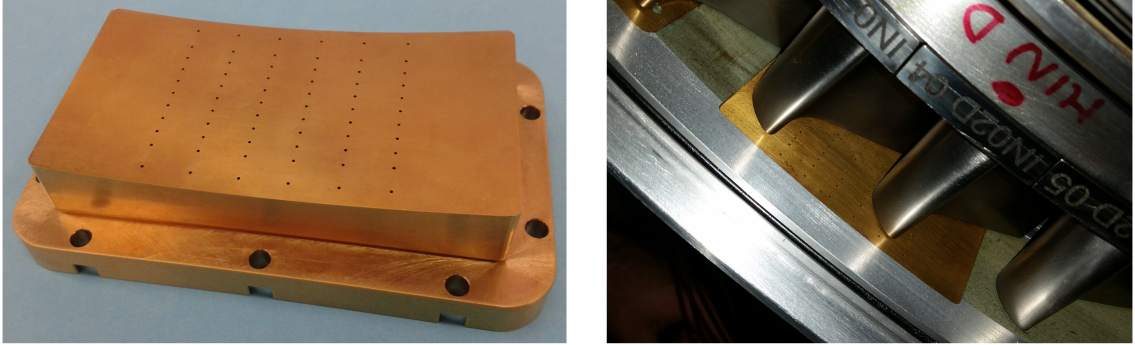


Figure 2.20: Static pressure casing insert

Casing Heat Transfer Instrumentation.

The unsteady casing heat flux is investigated by a measurement array of 35 double-layer thin film gauges, presented in Figure 2.21. The heat transfer measurement points replicate the location of the pressure instrumentation, symmetrical along the insert diagonal.

The high-frequency temperature gauge is composed of a 200 nm-thick Nickel element with a resistance of ~ 50 ohm deposited on a thermally insulating Upilex[®] Polyimide layer. The instrumented plastic sheet is glued on a metallic insert fitted on the rotor casing. The radial positioning of the metal support, made of aluminum alloy, is adjusted to ensure the thin-film sensors are flush-mounted with the casing wall. The thin-film sensor is a linear element with a length of 1.36 mm and a width of 0.5 mm, as schematized on the right-hand-side of Figure 2.21. The sensor length is aligned with the axial direction to maximize the spatial resolution over the rotor pitch.

The thin films are controlled by newly custom-built amplifiers. The gauge resistance is first converted into a voltage signal through an Anderson loop circuit that offers large-amplitude linear outputs for relatively low values of the excitation current (15 mA). The conditioning units can monitor the direct or the differential component of the gauge resistance, and boost the sensor voltage by a DC gain variable between 1 and

100. The sensor signal is further modulated by an analog filter stage with a gain profile increasing with frequency. The frequency-dependent amplification varies from a factor of 2.3 at 100 Hz up to a peak gain of 75 at 150 kHz (about +12 dB/decade).

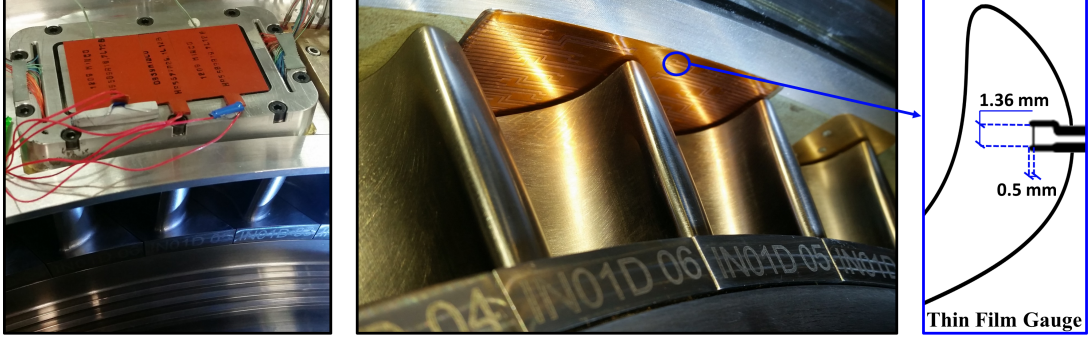


Figure 2.21: Heat transfer casing insert

The heat flux is derived from the unsteady wall temperature measurements using a Crank-Nicholson scheme to solve the 1D unsteady heat conduction equation for a two-layer substrate with the assumption of a semi-infinite boundary condition at the back wall [66]. The heat transfer insert has been calibrated in a controlled-temperature chamber between 295 K and 365 K to determine the temperature-resistance relationship. The thermal properties of the polyimide sheet and of the metallic substrate were experimentally determined through a transient calibration that retrieved the thermal thickness (l/k), and the thermal product ($\sqrt{\rho ck}$) of the two layers [67]. The measured thermal properties of the heat transfer instrumentation are reported in Table 2.2. The heat flux measurement suffers from a bias error of approximately $\pm 6\text{-}7\%$, mainly due to uncertainties in the thermal properties of the substrate. The precision error in time-resolved heat flux is of the order of $\pm 2\text{-}3\%$ [68].

	Polyimide film	Al7075T6	Units
$\sqrt{\rho ck}$	835.2	22976.0	$Jm^{-2}K^{-1}s^{-0.5}$
l/k	5.16×10^{-4}	-	m^2KW^{-1}

Table 2.2: Thermal properties of the heat transfer sensors

2.2.5 Stage Outlet

The stage outlet aerodynamic field is investigated 50% $c_{r,ax}$ downstream of the rotor trailing edge. The investigation region extends circumferentially over a full vane pitch, allowing to fully resolve the spatial periodicity of the machine, as shown in Figure 2.22.

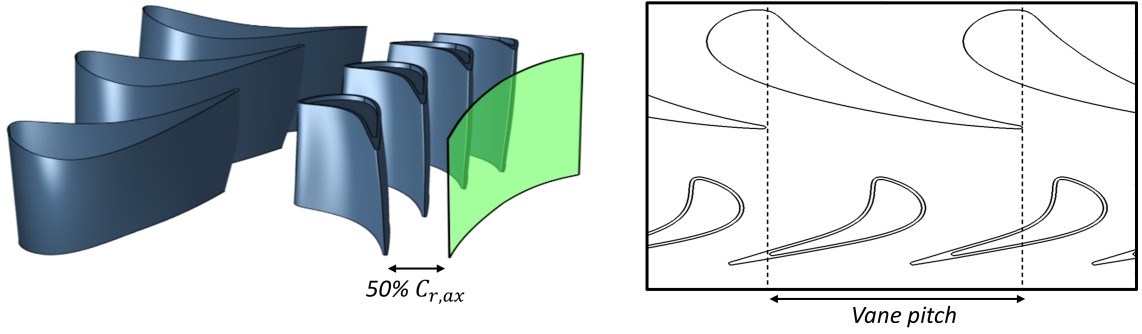


Figure 2.22: HPT stage outlet flow investigation region

Figure 2.23 presents the aerothermal probes employed for the characterization of the stage outlet flow field. Two reference probes, presenting a Kiel pitot head and a bare K-type thermocouple of $50\ \mu\text{m}$ diameter, are employed to monitor the midspan total temperature and pressure in each test (Figure 2.23a). Additionally, 4-head thermocouple rakes (Fig. 2.23b) and kiel rakes (Fig. 2.23c), presenting the same exact design of the ones operated at the stage inlet, are employed to characterize the outlet radial distributions of total temperature and total pressure respectively.

Double-head fast response sensor rakes, visualized in Figure 2.23d and equipped with semi-conductor measurement units (Kulite XCQ-062-25A), are employed to perform large-bandwidth total pressure measurements with a precision of $\pm 0.01\%$ of P_{01} (0.1 mbar). The stage outlet plane is interrogated by the fast response pressure rakes at 20 radial locations and 6 different peripheral positions along the vane pitch, for a total of 120 measurements.

The time-mean flow direction, static and total pressure and Mach number are mon-

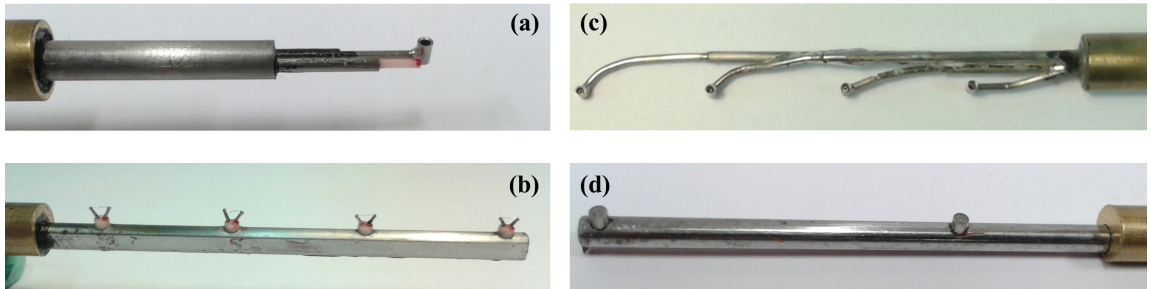


Figure 2.23: Outlet plane aerothermal probes: reference probe (a), thermocouple rake (b), pneumatic Kiel rake (c) and fast response pressure rake (d)

itored by employing two pneumatic five hole probes of equal head geometry (Figure 2.24). The two probes present stems of different length and are employed to investigate, respectively, the upper and lower 50% of the rotor blade span. The probes head

presents a conical geometry with cone angle of 62 degrees and an external diameter of 3.2 mm. The lateral taps are perpendicular to the head surface while the central hole is drilled with a conical shape of 25 degrees to desensitize the total pressure measurement for small flow angle variations.

Because of the design of the probes, the measurement matrix extends from 3% of the

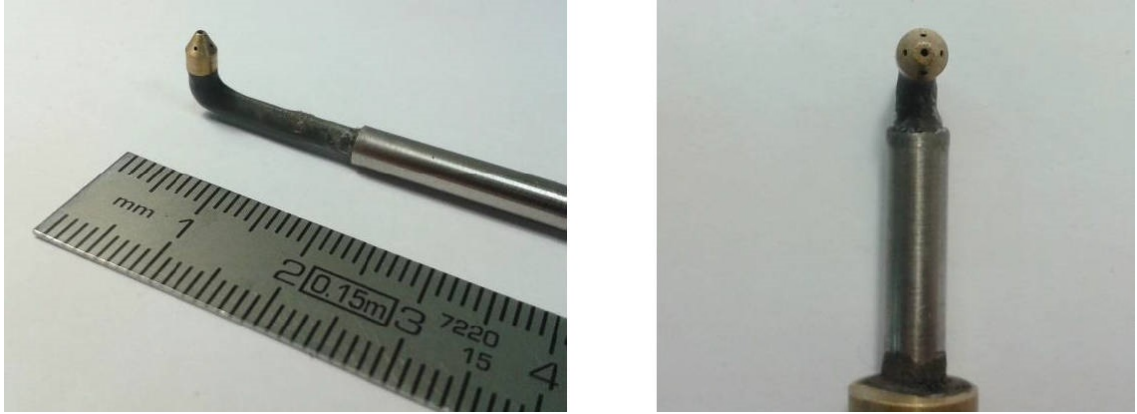


Figure 2.24: Pneumatic five-hole probe

span (near-hub region) up to 92% of the span (near-casing region). The aerodynamic calibration of the probes has been performed in a jet calibration facility between Mach 0.1 and 0.5 within a yaw flow angle range of ± 30 degrees (axial-peripheral plane) and a pitch flow angle range of ± 20 degrees (axial-radial plane).

Miniaturized single-hole probes [69], operated in "virtual three-hole probe" mode [70], allow to replicate the flow angle sensitivity of real multi-hole systems (Figure 2.25). To this aim, a periodic rotation is applied to the probe in three consecutive tests at the exact same radial location. Calibration maps are employed to retrieve the time-resolved histories of flow yaw angle (axial-peripheral plane) by combining the three ensemble-averaged unsteady signals.

The probes were calibrated at different Mach numbers (between 0.1 and 0.5) within a yaw range of ± 60 degrees and a pitch range of ± 20 degrees. To improve the spatial resolution of the measurement, the probe head presents a 0.2 mm diameter tap, connected to the semi-conductor sensor (Kulite XCQ-062) by a miniaturized cavity which ensures optimal frequency response (> 20 kHz). The measurement matrix of the virtual three-hole-probe covers only the upper 50% of the rotor blade span over 9 radial stations at five different peripheral positions along one vane pitch for a total of 45 interrogation points.

The dynamic calibration of the miniaturized fast response aerodynamic probe was performed by shock tube testing [71]. The resulting transfer function is presented in

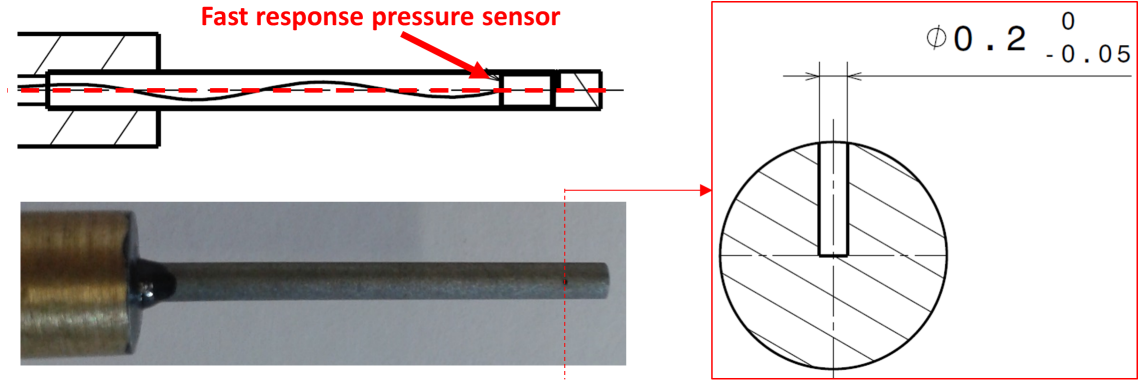


Figure 2.25: Miniaturized single-hole probes for V3H measurements

the top section of Figure 2.26 (blue line). The natural frequency of the line-cavity system in front of the fast response sensor is indicated by the peak present at 24 kHz. The transfer function is modeled by a linear second order model (red line), which provides the correct damping coefficient and natural frequency. While this was not done in this particular work, the modelled transfer function can be employed to demodulate the high-frequency signals and broaden the frequency band of the probe well above the natural frequency of the line-cavity system (35-40 kHz in the present case).

The frequency analysis of a typical short-duration turbine test is presented in the bottom section of the same figure. The red line indicates the power spectral density of the unfiltered signal, while the blue line refers to the signal processed by a digital Butterworth filter with cut-off frequency equal to 30 kHz. The filter was applied to the time-resolved dataset of the V3H probe for noise reduction purpose, before performing the ensemble-averaging. The blade passing frequency can be noticed as the highest peak in PSD, at 4.74 kHz, while the natural frequency of the probe line-cavity system is indicated by a weaker local maximum at ~ 24 kHz. As demonstrated in [64], this frequency response is enough to fully resolve over 95% of the correlated pressure unsteadiness established at the outlet plane of the HPT stage by a flat-tip rotor operated at design clearance (fluctuation frequencies globally below 25 kHz).

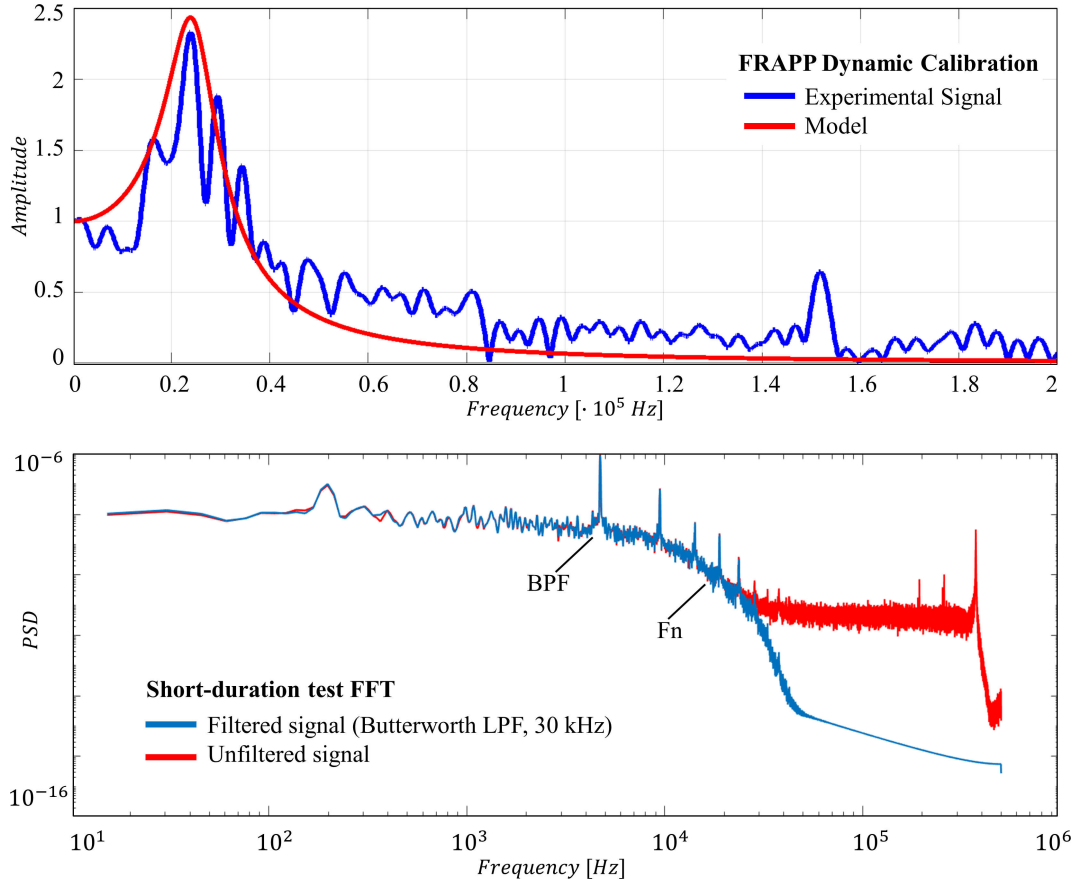


Figure 2.26: Dynamic analysis of miniaturized FRAPP

2.3 Data treatment

2.3.1 High frequency signal acquisition.

Before digitization, the high bandwidth signals are split in two components, as indicated in Figure 2.27, to enhance the resolution of the fluctuations at the time scale of the blowdown duration (0.5 s) and at the blade passing frequency ($\sim 200 \mu\text{s}$). The time-average signal is low-pass filtered at 500 Hz and sampled at 10 kHz for 10 seconds at 16-bit resolution. The high-bandwidth signal is conditioned through an analog band-pass filter allowing the information transfer in the range between 60 Hz and 200 kHz. The AC component is further amplified by a factor of 10 before being sampled on a 16-bit data acquisition system at 1 MHz for 3 seconds. With such sampling frequency, 210 measurement points per blade passage are taken. In the case of the high frequency measurements at the rotor casing, this is equivalent to a spatial sampling of $\sim 0.25 \text{ mm}$ over the rotor tip area. The static gain and the frequency response of the entire measurement chain are calibrated from 0 Hz to 1 MHz to generate

the de-boosting functions used for the signal data reduction.

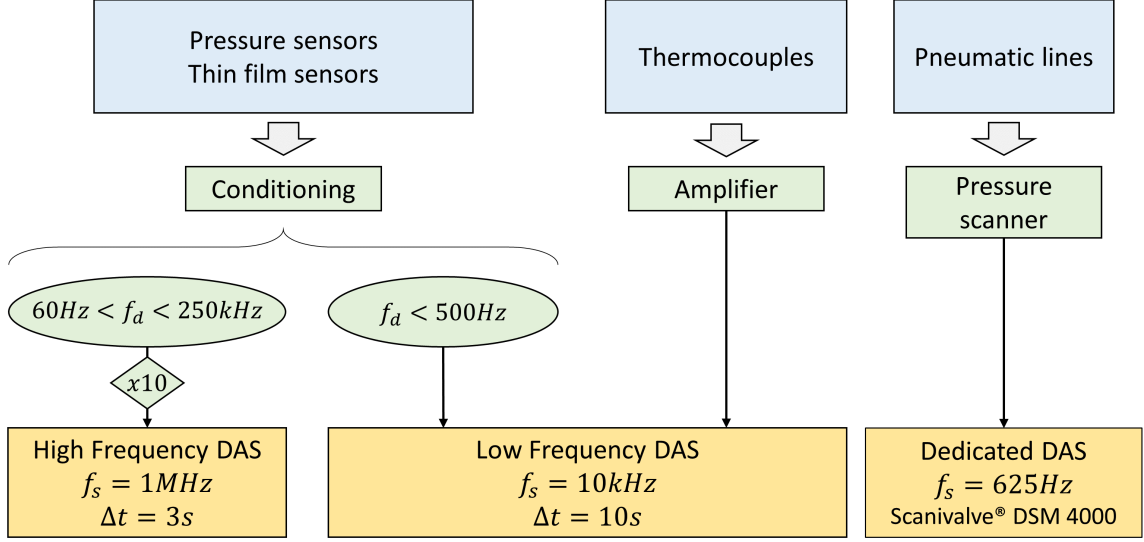


Figure 2.27: High- and low-frequency data conditioning and acquisition

2.3.2 Low frequency signal acquisition.

The pneumatic probes and endwall taps employed in the facility are connected to a high precision pressure scanner, Scanivalve® scan module ZOC 33/64PX ± 15 psid controlled by a DSM4000 unit. The length (1 m max) and internal diameter (1 mm) of the pneumatic lines allow to maintain a frequency response relevant to the blowdown rig setup, ensuring that the measured pressure signal reaches steady conditions during the short duration of a test.

The output signal of the thermocouples is conditioned by means of dedicated amplifier units and acquired at 10 kHz with 16-bit resolution. The acquisition time, in this case, is set to 10 seconds.

2.3.3 Data Reduction.

The time-resolved measurements are ensemble averaged to isolate the deterministic and random fluctuations that characterize the flow unsteadiness [72]. The deterministic component is periodic with the rotor blade passage while the non-periodic (stochastic) unsteadiness is mainly generated by turbulence and revolution-to-revolution flow

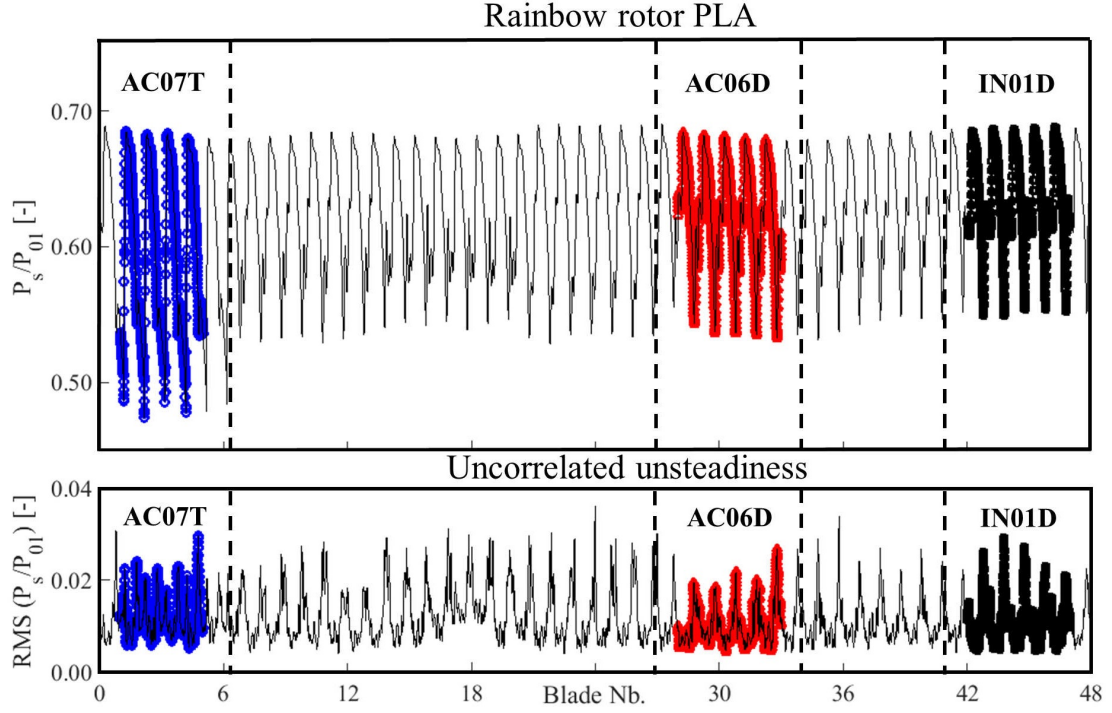


Figure 2.28: Phase locked averaging applied to casing static pressure signal

variations. The rainbow-rotor configuration imposes a flow periodicity of one complete disc revolution and the unsteady signals were phase-locked averaged over a rotor phase equivalent to 360 degrees.

The phase locked average (PLA) of a typical rotor casing pressure signal is presented in Figure 2.28. The raw unsteady data are averaged over 23 rotor revolutions corresponding to a time window of ~ 250 ms. The unsteadiness uncorrelated with the blade period is quantified by the root-mean-square (RMS) of the raw signal against the time-resolved PLA, calculated with a 95% confidence interval.

A sector-bounded analysis is further performed on the phase-locked data by pitch averaging the ensemble-averaged pressure histories of each single sector, as presented in Figure 2.29. In order to exclude all sector-to-sector flow interactions and ensure the highest flow periodicity between rotor channels, the outer passages of each sector were removed from the averaging procedure. Thus, the sector average is performed considering five blade passages at each revolution, corresponding to $23 \times 5 = 115$ blade periods.

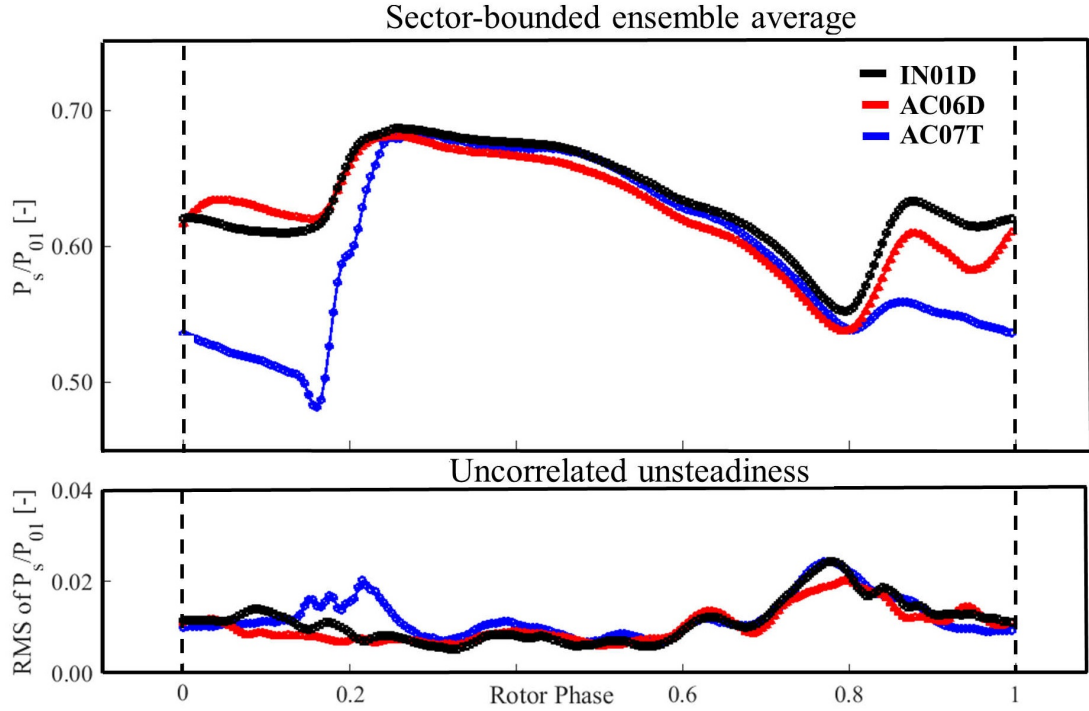


Figure 2.29: Sector ensemble averaging applied to casing static pressure signal

2.3.4 Statistical convergence

The statistical convergence of the phase-lock averaging procedure was verified by monitoring the maximum PLA amplitude and the average random unsteadiness of the rainbow rotor with increasing number of disc revolutions.

Figure 2.30 presents the results of this analysis performed on all the casing static pressure measurements. The maximum sector-wise PLA variation (averaged on all the casing measurement points) decreases monotonically in amplitude with increasing number of revolutions. However, the relative change in amplitude is below 1% when the ensemble average is performed on more than 15 revolutions. Averaging on more than 18 revolutions brings the relative variations in gauge-average rotor RMS value below 2% (~ 0.3 mbar). Such results show the statistical convergence of the sector data averaging for the short duration test.

The equivalent analysis is presented, in Figure 2.31, for the high-frequency casing heat transfer measurements. A higher variation of the PLA signal is found, in this case, within the first 15 revolutions. Consistently with the static pressure trend, the maximum heat flux variability is shown to decrease monotonically, reaching $\sim 5\%$ of the final value when the ensemble averaging is done over more than 18 revolutions. Regarding the uncorrelated unsteadiness content, averaging on more than 18 revolu-

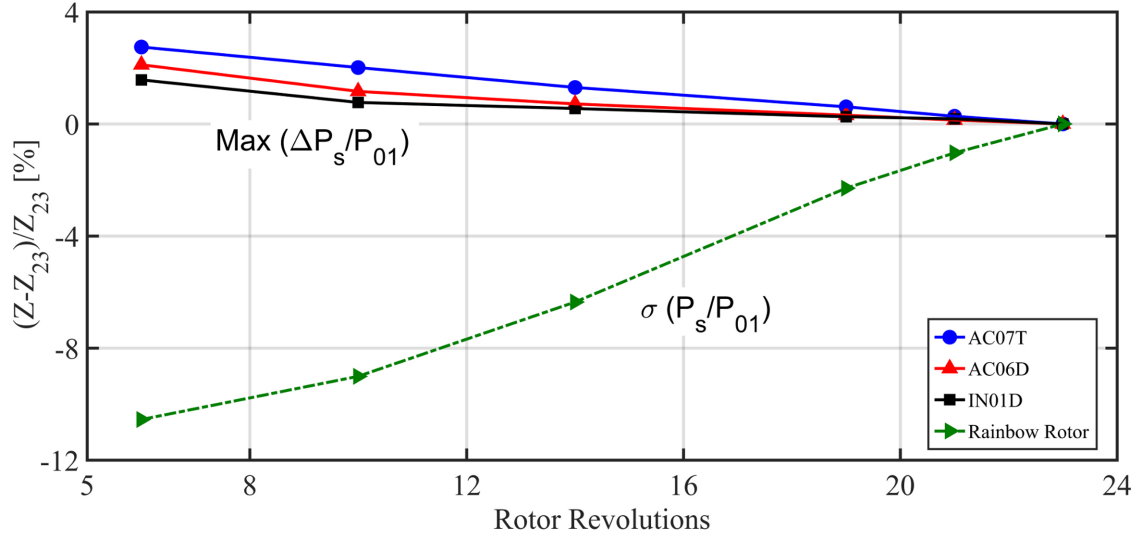


Figure 2.30: PLA statistical convergence: casing static pressure measurements

tions brings the relative variations in gauge-average rotor RMS value below 3% of the final value.

An overall uncertainty on the fluctuating part of the phase-locked signals has been evaluated from the analysis of the complete datasets of pressure and heat transfer measurements. The error estimation takes into account the sensor precision and resolution, test-to-test repeatability, and the passage-to-passage repeatability within each tip sector. Typical uncertainties in fluctuating static pressure and heat transfer were found to be approximately $\pm 0.3\%$ of P_{01} and $\pm 1.5\text{-}2\text{ kW/m}^2$ respectively.

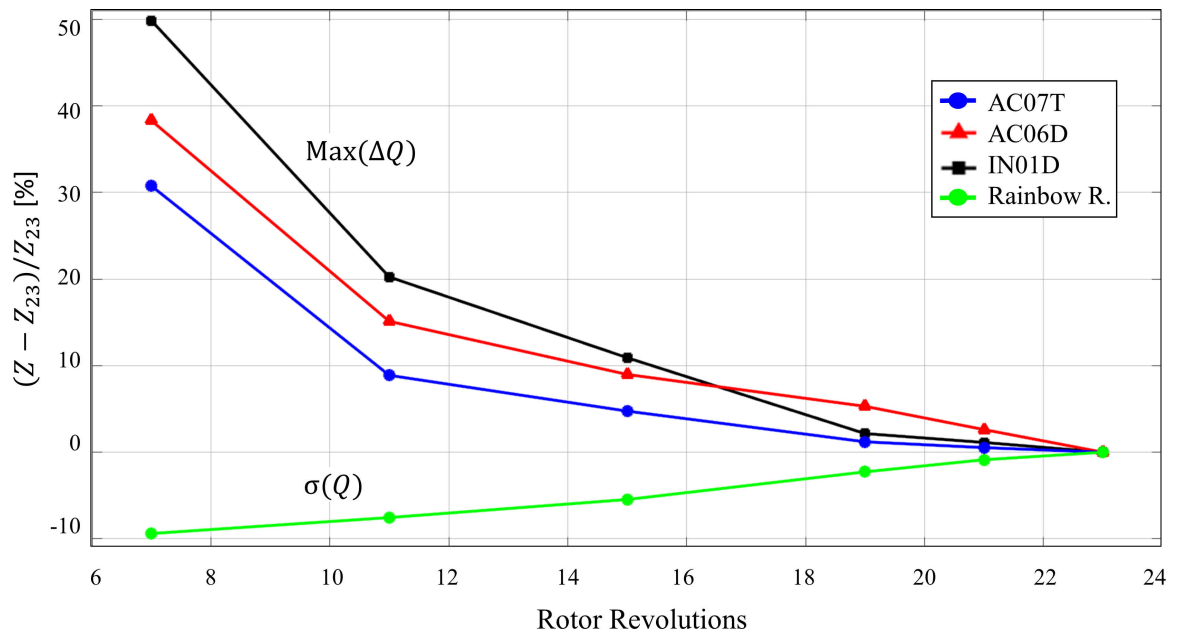


Figure 2.31: PLA statistical convergence: casing heat transfer measurements

Chapter 3

Numerical Study

From the numerical standpoint, RANS simulations remain the most employed method for the numerical characterization of turbomachinery flows, as the prohibitive computational cost of LES calculations makes them difficult to apply for industrial design purposes [73].

In spite of the inherent unsteadiness of the turbine flow field, highly modulated by pressure waves, shock waves and blade rows interactions [74], the mixing plane approach is commonly adopted at the vane-rotor interface to perform steady simulations at a low computational cost. Detailed comparative studies showed, however, slight discrepancies in terms of aerodynamic performance [75] and endwalls heat transfer [76] between steady and unsteady flow computations.

In order to resolve the flow interactions between adjacent blade rows and overcome the computational cost of full-annulus unsteady simulations, numerous researchers choose a domain-scaling approach. This methodology consists in modeling a limited number of airfoils per blade row with periodic peripheral boundary conditions [77], adopting a simplified vane-to-rotor blade-count ratio. As turbomachines are designed with uneven number of airfoils per row to minimize the risk of aeroelastic oscillations, substantial inaccuracies were demonstrated to arise in the scaled flow predictions when the blade-ratio change is substantial [74].

A more reliable approach for the prediction of blade-rows interactions is represented by phase-lagged methods which exploit the periodicity of the flow in space and time (i.e. chorochronic periodicity). With this method, a single blade passage per row needs to be modeled as phase-shifted boundary conditions are applied at the peripheral boundaries [77]. The memory demand, in this case, is way higher compared to a steady solver because the flow information needs to be stored and re-phased before being reintroduced at the domain boundaries.

With the unsteadiness of turbomachinery flows mainly modulated by the blade passing event, the nonlinear harmonic calculations (NLH) represent a valid alternative to the aforementioned unsteady methods. The NLH methodology was originally developed by [78] and allows to solve the time-dependent RANS system in the frequency domain, decomposing the unsteady flow in a time-average and a perturbation component. The flow unsteadiness is therefore resolved at a limited number of harmonics of the blade passing frequency, reducing the computational time by more than ten times compared to a phase-lagged simulation but providing a comparable degree of accuracy [79].

This section addresses the numerical flow simulations which were originally performed by Paty [80]. The extensive numerical dataset has been entirely post-processed and analyzed for the present thesis to gain a deeper insight into the flow physics established by the investigated tip geometries.

Steady RANS, performed on a single-passage, single-geometry fluid domain representative of the HPT test article, are interrogated. Additionally, full-stage, single-tip unsteady computations are employed to assess the impact of the blade-row interactions on the stage flow field and on the performance of the single tips.

Full-stage, full-annulus unsteady simulations, finally, are interrogated to validate the rainbow-rotor configuration and assess the flow periodicity established by the multi-sector setup.

Test-calibrated boundary conditions were imposed in the CFD studies to ensure the relevance of the flow predictions and direct comparability with the experimental dataset.

3.1 Single-tip CFD

3.1.1 Flow domain

The flow domain adopted for the numerical characterization of the single tip designs includes one vane passage and one rotor blade and is presented in Fig.3.1.

The domain inlet, referred to as *Plane 01a* in the figure, is located $1.33 c_{v,ax}$ upstream of the vane leading edge while the outlet (*Plane 05*) is positioned $4.66 c_{r,ax}$ downstream of the rotor trailing edge. The experimental measurement locations, in blue in Figure 3.1, are included in the numerical domain, so that an iterative procedure is necessary to match the experimental boundary conditions. Table 3.1 presents an overview of the axial locations of the experimental and numerical planes of interest.

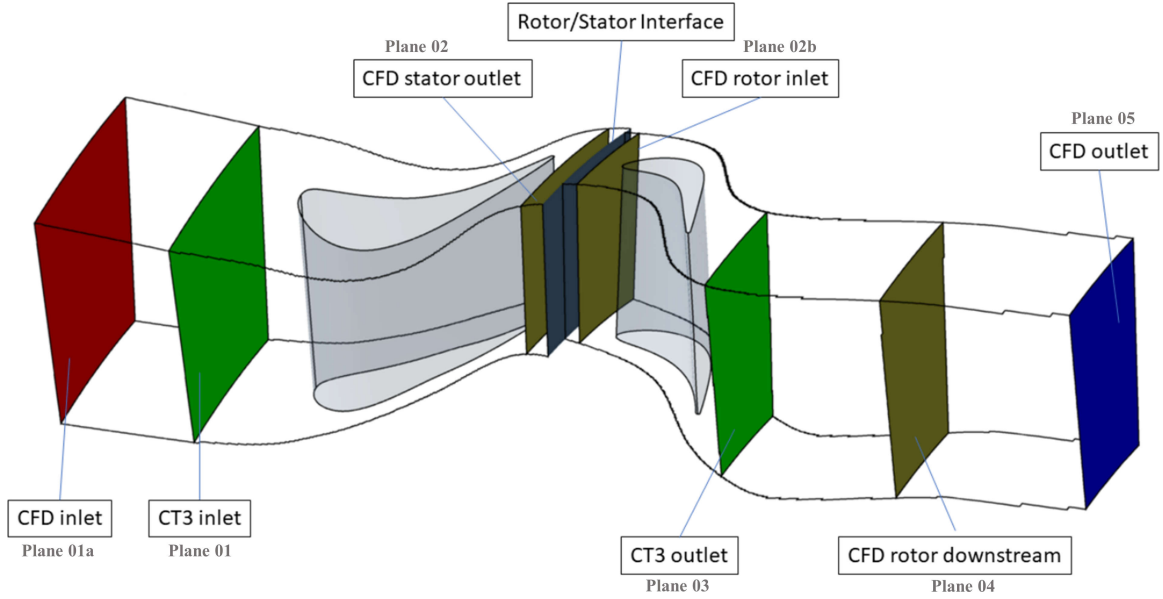


Figure 3.1: CFD domain: single passage vane / rotor simulations

Definition	Name	z-coord.	Rel. position	Exp.	CFD
Domain inlet	Plane01a	-0.0717309	$-1.33c_{v,ax}$		X
Stator inlet	Plane01	-0.02657	$-0.49c_{v,ax}$	X	X
Stator outlet	Plane02	0.05951	$1.10c_{v,ax}$	X	X
Rotor inlet	Plane02b	0.07	$-0.17c_{r,ax}$	X	X
Rotor outlet	Plane03	0.13621	$1.49c_{r,ax}$	X	X
Domain outlet	Plane05	0.2628	$4.66c_{r,ax}$		X

Table 3.1: Axial location of experimental and numerical planes of interest

3.1.2 Mesh

An unstructured, fully hexaedral, meshing strategy is adopted in this study by reason of the significant complexity of the rotor blade tip shapes under investigation.

The grids presented in Figure 3.2 were generated in FINE/Hexpress 6.1, the meshing tool of the commercial software Numeca. The final meshes count between 21.1×10^6 and 23.8×10^6 cells depending on the tip geometry.

The tip region is discretized with a minimum of 100 cells in axial and azimuthal directions while 50 cells were used in the radial direction. Additional boundary layer refinements lead to a first cell thickness of $1.3 \mu\text{m}$ with y^+ below 0.7 on the entire tip surface, allowing an optimal resolution of the significant flow gradients in the tight tip channel.

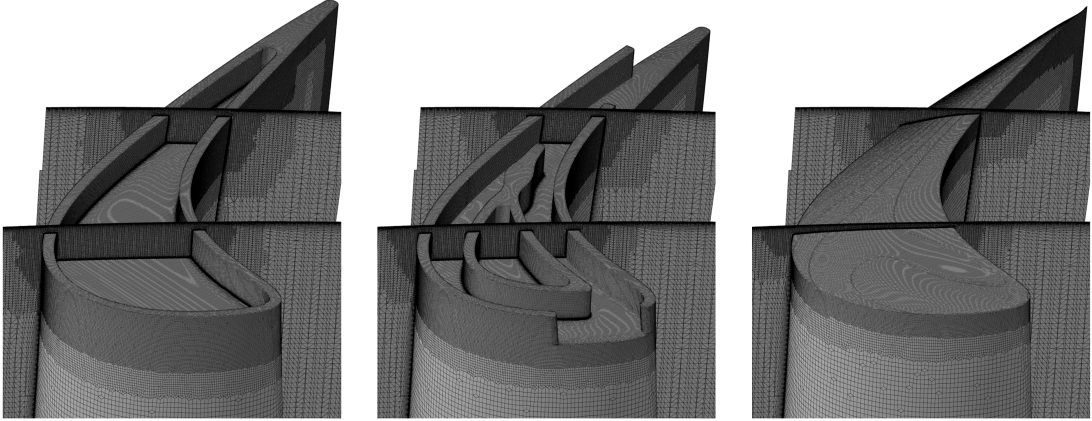


Figure 3.2: Rotor tips unstructured mesh

3.1.3 Steady simulations

Steady-state RANS computations are performed in Numeca FINE/Open 6.1. A constant wall temperature of 300 K is imposed on all wetted surfaces, respecting the experimental gas-to-wall temperature ratio of ~ 1.5 , representative of engine operation. A perfect gas hypothesis is considered for the working fluid.

The CFD solver and the numerical settings have been previously validated on a highly loaded turbine cascade featuring cooled squealer tips [7]. Periodic conditions are assumed at the boundaries with the adjoining passages, while the mixing plane approach is employed at the vane-rotor interface. This technique allows to account for the interactions between vane and rotor by exchanging pitchwise-averaged flow quantities at the interface. For this particular study, the software-default option "Conservative Coupling by Pitch-wise Rows" is adopted to establish the flux conservation through the interface (exact conservation of mass flow, energy and momentum [81]).

Test-calibrated mass-flow averaged total temperature is matched at Plane 01 by employing a scaling factor on the temperature profile imposed at the numerical domain inlet. The Pianko-averaging method [82] is employed for the total pressure. The latter averaging methodology, developed for turbomachinery, targets the calculation of a mean total quantity providing an equivalent isentropic work exchange as the non-uniform inlet distribution, given the same outlet static pressure field. In both cases, the experimental conditions are matched at the measurement planes, included in the numerical domain, by applying a scaling coefficient to the numerical inlet profiles.

Although the choice of the turbulence model was deemed to have limited impact on the quantification of the aerodynamic performance of the profiles, several studies show its significance in the correct prediction of the heat transfer field [83]. After a

preliminary study, the Spalart-Allmaras model is chosen as turbulence closure [84]. Imposing the measured inlet conditions of turbulence intensity ($Tu_{01,avg} = 13.3\%$) and integral length-scale ($L_{01,avg} = 0.019m$), however, leads to a turbulence viscosity ratio of $\mu_t/\mu = 3567.24$, well outside the validation range of the model ($\mu_t/\mu = 0$ to 100). Thus, a separate study with two-equation models ($k-\epsilon$ and SST) is performed with test-calibrated inlet conditions, imposing consistent inlet turbulent kinetic energy and dissipation rate, and compared to Spalart-Allmaras results. Based on the best agreement with two-equation models in terms of integral performance parameters (total pressure losses and average wall heat flux), an average inlet turbulent viscosity ratio of $\mu_t/\mu = 93.15$ is finally chosen for the SA model.

The exit hub static pressure is adjusted to match the experimental values of end-wall pressure in *Plane 03*. The scaling of the boundary conditions is based on the Spalart-Allmaras cases for all flow quantities except the turbulent kinetic energy, which is determined in the SST computations.

The evaluation of the convergence is based on the residuals and global performance parameters monitoring. Convergence is reached after 3000 iterations (2 days on one node of cluster with 64 processors, 2.3GHz clock speed), with values of the residuals varying from tip to tip, but always of the order of 10^{-5} .

3.1.4 Unsteady simulations

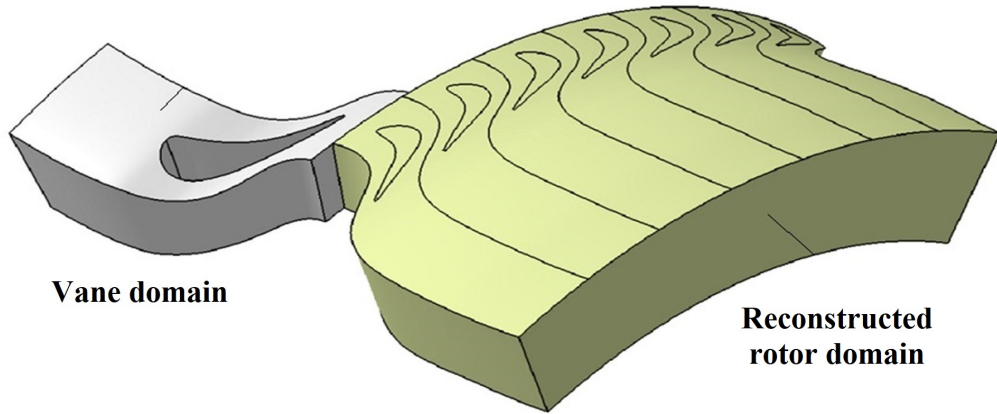


Figure 3.3: NLH reconstructed fluid domain

The HPT time-varying flow field is analyzed by means of nonlinear harmonic computations (NLH) performed in Numeca FINE/Open 6.1, exploiting the exact same set of test-calibrated boundary conditions as in the steady CFD. The NLH method is an advanced, CPU-efficient tool to solve time-periodic flow computations

which in recent years has been increasingly employed to model unsteady blade rows interactions in multistage turbomachines [85, 86, 87].

This numerical method consists in decomposing the flow into a time mean component and several perturbations located at the harmonics of the blade passing frequency. The unsteady conservative quantity U will therefore be defined as the sum of a time-average contribution and a series of periodic unsteady flow perturbations.

$$U(\vec{r}, t) = \overline{U}(\vec{r}) + \sum U'(\vec{r}, t) \quad (3.1)$$

The flow perturbations are defined as Fourier series presenting a number of harmonics of the blade passing frequency and the first-order linearised Navier-Stokes are solved in the frequency domain as explained in detail in [87].

The user can control the accuracy of the unsteady solution by tuning the order of the Fourier series, with the number of relevant harmonics chosen depending on the unsteady content of the investigated flow and on the desired resolution. With the flow equations cast in the frequency domain it is possible to obtain the transport equations for each particular time frequency. Along with solving the time-average flow steady-state equations, two additional sets of conservation equations are solved at each frequency, representing the real and imaginary part of the related harmonic. The nonlinearity is introduced in the flow solution by adding the deterministic stresses, computed from all the solved frequencies, into the time-averaged flow solver.

This procedure enables to resolve the periodicity of a multi-row machine by only modeling one single blade passage per blade row. The time harmonics model the perturbations originated from the relatively rotating adjacent blade rows, including the time-mean solution (regarded as harmonic of order 0). Hence, the flow solver allows to resolve vane-rotor interactions by greatly reducing the necessary fluid domain dimension (and, thus, computational expense) with respect to a full-annulus U-RANS simulation [81].

The one-equation Spalart-Allmaras (SA) formulation is chosen to model turbulence for its robustness in turbomachinery flow environment while adiabatic conditions are imposed at the walls. Due to a limitation of the software employed, conversely, no unsteady flow simulations could be performed with isothermal wall conditions. The perfect gas hypothesis is assumed for the working fluid and the NLH computations are initialized starting from the converged steady-state solution on the same fluid domain.

Based on the residuals and global performance parameters history, the simulations were considered converged after 10000 iterations. Each iteration resulted 15 times

more costly than in the case of the steady-RANS calculations performed with the exact same computational resources.

After an initial parametric study, based on integral performance parameters and on the study of the unsteady solution, five harmonics of the blade-passing frequency are considered to adequately represent the turbine correlated unsteadiness since negligible variations are detected in the flow predictions for higher numbers of harmonics.

A time reconstruction method is employed to retrieve the time-accurate flow solution over multiple rotor domains (Figure 3.3). The rotor blade passing period, defined as the time needed to travel a full vane pitch, is discretized in 140 time steps. At a blade passing frequency of 4.7 kHz, one time-step corresponds to $\sim 1.5 * 10^{-6}$ s.

The resulting dataset is post-processed by means of dedicated Matlab routines, purposely developed in order to characterize the unsteady turbine flow both in the vane- and rotor-relative frames of reference and enable relevant comparison with time-resolved experimental results.

3.2 Full-annulus CFD

Full-stage, full-annulus unsteady simulations are performed in order to model the rainbow rotor configuration, as presented in Figure 3.4. The NLH method allows, in this case, to resolve the unsteady flow interactions established between the vane row and the multi-sector turbine rotor by solving the Navier-Stokes system in the frequency domain. Hence, one single vane passage needs to be included in the fluid domain along with the full rainbow rotor.

The NLH solution allows to fully resolve the rotor-correlated unsteadiness up to the fifth harmonic of the blade passing frequency [88]. By comparing the numerical flow predictions, solely correlated with the blade passing event, with the unsteady measurements it is therefore possible to clearly distinguish the rainbow-induced flow variability in each sector from spurious flow differences induced by blade manufacturing tolerances or stochastic flow residuals.

As for the single-tip geometries, an unstructured, fully hexahedral mesh is employed to discretize the numerical flow domain, which includes one single vane passage and the complete multi-sector HPT rotor. Due to the software limitations in processing the complete rotor annulus, the rainbow disk is divided in seven sub-domains, each one featuring a specific tip design. Each sub-domain is then converted in Parasolid format and imported in FINE/Hexpress 6.2 to produce the unstructured mesh.

The number of cells per blade passage needs to be reduced, compared to single-tip simulations, to achieve a reasonable computational demand. As the NLH method is built on the spatial Fourier decomposition of the flow in azimuthal direction, however, there is a minimum amount of cells in azimuthal direction that should be provided to the solver at each blade passage. Following the guidelines provided in [81], the minimum number of grid points per blade channel in the vane ($N_{p,v}$) and rotor ($N_{p,r}$) domains is computed as follows:

$$N_{p,v} > 15 * (N_{b,r}/N_{b,v}) * N_H = 106 \quad (3.2)$$

$$N_{p,r} > 15 * (N_{b,v}/N_{b,r}) * N_H = 54 \quad (3.3)$$

where $N_{b,v}$ represents the number of vanes, $N_{b,r}$ the number of rotor airfoils and N_H the selected number of harmonics. Thus, each rotor blade passage is discretized by a minimum of 95 and 70 cells in radial and tangential direction respectively, while 200 cells are used in axial direction. Each vane passage counts, in azimuthal direction, a minimum of 114 points. The final full-annulus mesh counts a total of 75.2 million cells.

Table 3.2 presents the discretization of the rotor blade passage in the single-tip and

Rotor passage cells count			
	Radial	Tangential	Axial
Full Annulus	95	70	200
Single Passage	150	100	325

Table 3.2: Minimum number of cells per rotor blade passage

full-annulus grids. The Spalart-Allmaras approach is used for turbulence modelling. The Extended Wall Function (EWF) model by Numeca is employed at the solid boundaries, targeting a y^+ of 30 in the grid viscous layers generation. A validation study over steady-state, full-annulus simulations was performed to obtain a comparison of "wall-resolved" SA results against the SA-EWF approach, finding this latter to be the optimal compromise between computational expense and flow prediction accuracy.

As for the single-tip cases, direct comparability with the experimental results is ensured by matching the measured boundary conditions at the corresponding locations in the numerical flow domain. The converged solution of the steady-state computation, achieved in 550 iterations with a density residual root-mean-square (RMS) below 10^{-5} , is employed as initial solution for the unsteady flow field. The convergence evaluation is based on both residuals and global performance parameters (axial

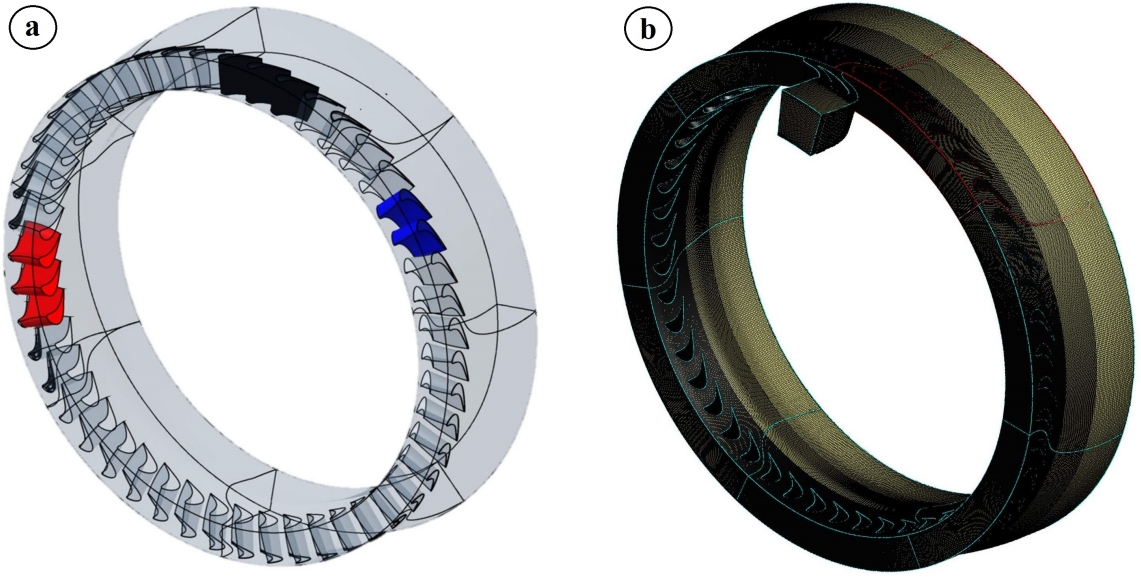


Figure 3.4: Full annulus domain (a) and unstructured mesh (b)

thrust, mass-flow difference, efficiency, torque).

Thanks to the steady-state initialization, all NLH residuals exhibit an order of magnitude below 10^{-8} after 1500 iterations, with a RMS value of the order of 10^{-4} . As the computation is performed in parallel on 48 processors, a limit imposed also by the available memory (640 GB in the present case), the steady state calculation is finalized in one day while the NLH, contrarily, requires 10 days (9 minutes per iteration).

3.3 Vortex tracking algorithm

A clear identification of the vortical structures affecting the rotor flow field is of primary importance in order to provide a critical evaluation of the aerodynamic performance of each profile. A vortex tracking methodology was developed for this purpose by Paty [89] and applied on the steady RANS predictions. A brief theoretical overview of the tracking algorithm is provided in this section of the thesis.

Fluid motion can be generally regarded as a combination of three components: rigid body rotation, irrotational straining and pure shear[90]. A classical approach for vortex line identification consists in decomposing the velocity gradient tensor $\nabla \vec{u}$ into a strain-rate tensor $\mathbf{S}$ and a rotation tensor $\mathbf{\Omega}$, to which the vorticity is directly related. Vortex tracking algorithms are usually based on the analysis of the stream-wise [91]

or axial [92] component of the vorticity vector.

$$\nabla \vec{u} = \mathbf{S} + \mathbf{\Omega} \quad (3.4)$$

This methodology, however, does not allow for a clear distinction between swirling flow structures, intended as regions of fluid rotating as a rigid-body around an axis, and pure shear flow regions [93]. This limitation was overcome, in this case, by employing the triple-decomposition method proposed by Kolar [94], which separates the fluid motion into three fundamental contributions: pure shearing, rigid rotation and irrotational straining.

After subtracting the pure shear contribution, the residual velocity gradient tensor is further decomposed into a symmetrical and an asymmetrical component, representing the residual strain and residual vorticity. Hence, the residual vorticity tensor can be employed to define the vortical structures as connected flow regions with non-zero magnitude of residual vorticity. For the present application, a threshold of $\omega_{min} = 3500$ was set for the vortex identification, corresponding to less than 1% of the maximum vorticity value detected in the fluid domain. Such low threshold allows to capture the early stages of vortex generation and still prevent the detection of spurious flow structures. The center of the vortex is defined as the point of maximum residual vorticity magnitude [94].

The residual vorticity contours over the axial cut at $0.7c_{r,ax}$ of blade IN01D are presented in Figure 3.5a [80]. On the left, the total magnitude of the vorticity vector is presented, with vorticity peaks corresponding to the tip leakage vortex (TLV), upper passage vortex (UPV) and lower passage vortex (LPV). Uniformly high levels of residual vorticity are present, however, in the shear layer region along the airfoil SS, which could prevent the clear discrimination of the UPV and LPV.

Hence, a modification is operated on the model, subtracting the radial component of the vorticity vector, particularly high in the near-SS shear. Based on the consideration that the vortical structures of interest for this study are generally well aligned with the flow, the blade-to-blade component $\vec{\omega}_{RES}^{b2b}$, defined in Equation 3.5 and visualized on the right in Figure 3.5a, is employed to monitor the vortex location.

$$\vec{\omega}_{RES}^{b2b} = \vec{\omega}_{RES}^{\theta} + \vec{\omega}_{RES}^z \quad (3.5)$$

The vortex boundaries are defined as vorticity isolines where $\vec{\omega}_{RES}^{b2b} = \omega_{min}$. The strength of the vortex is thus expressed as the integration of the residual vorticity over the vortex section, as from Equation 4.13.

$$\Gamma_{RES} = \int_A \vec{\omega}_{RES}^{b2b} \cdot \vec{n} dA \quad (3.6)$$

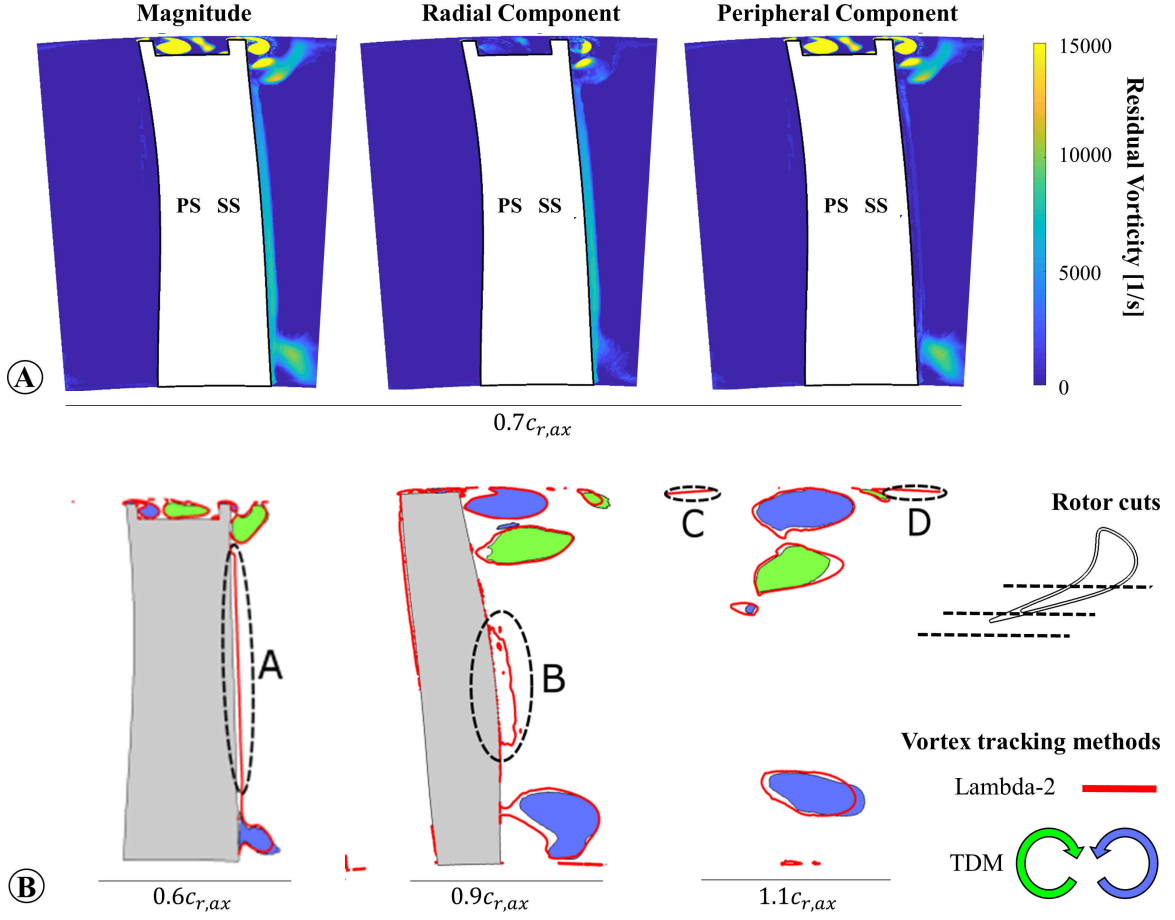


Figure 3.5: Residual vorticity (a) and vortex identification results for different techniques (b)

The rotational sense of the vortex is then assessed, at each axial location, with reference to the tangent of the blade camberline, as from Equation 3.7. In this frame of reference, a vortical structure swirling in clockwise direction will present $\hat{\omega}_{sign} < 0$.

$$\hat{\omega}_{sign} = \text{sgn}(\cos(\vec{\omega}_{\text{RES}}^{b2b} \cdot \vec{t}^{\text{camber}})), \quad t_z^{\text{camber}} > 0 \quad (3.7)$$

In order to validate the TDM method for turbomachinery flows, a comparison with the classical λ_2 criterion is proposed in Figure 3.5b at three axial locations along the rotor axial chord. This vortex tracking method, widely validated in turbomachinery flows, identifies swirling structures where the second eigenvalue of the $\mathbf{\Omega}^2 + \mathbf{S}^2$ tensor is negative. As for the TDM method, a lower threshold was set, defining vortices as regions with $\lambda_2 < -5 \times 10^7$. The vortex boundaries identified by the λ_2 criterion are shown here as red curves, while green and blue patches indicate the clockwise and counterclockwise vortical structures identified by the TDM respectively.

The global agreement between the two techniques is good at the three axial locations.

The λ_2 criterion, however, detects near-SS (A, B) and near-casing (C, D) vortices which are clearly motivated by shear rather than rigid-body flow swirling. Additionally, with the λ_2 quantity represented by a scalar, it is not possible to introduce a directional correction as done for the TDM. Finally the TDM provides additional knowledge on the sense of rotation of the vortex, its strength and the position of its axis.

Chapter 4

Stage aerothermodynamics

The present chapter addresses the impact of the three individual rotor tip shapes over the aerodynamic and thermal field of the HPT stage.

The flow conditions established in the experimental campaign, which also represent the boundary conditions of the numerical study, are first characterized. The inlet total quantities, measured at several circumferential locations across the turbine annulus, are presented together with endwall static pressure data collected at several axial locations of the turbine test article to evaluate the sector-resolved degree of reaction and the operating point of the multi sector rotor setup. The aerodynamic performance and thermal budget established by each tip seal design are evaluated in comparison to flat tips operated at relevant clearance conditions and a breakdown of the loss generation process is produced.

Time-resolved measurements collected at the over-tip casing region and at the stage outlet plane are then combined to unsteady numerical predictions for each tip design. The flow study in the rotor domain allows to isolate the impact of the individual tip geometries on the rotor passage flow field, disregarding the vane impact on the circumferential flow gradients. Steady and unsteady flow predictions are validated and exploited to gain a deeper insight of the turbine flow field. A vortex tracking technique is employed to trace the generation and propagation of the tip leakage vortex and its interaction with the rotor-induced secondary flow structures as a function of the tip shape.

A modal analysis tool is developed and employed to characterize the flow unsteadiness across the turbine stage. The analysis of space- and time-resolved measurements and numerical predictions allows to quantify the blade rows interactions impact over the unsteady aerodynamic and thermal field at the rotor casing and at the HPT outlet. A critical analysis is performed to justify the tip-to-tip differences in terms of modal energy content and flow periodicity type at different locations in the fluid domain.

Full-annulus unsteady simulations are compared to time-resolved experimental data to evaluate the periodicity established by the multi-sector rotor setup. A detailed study is performed in each individual sector to quantify the influence of the design concept and operative tip clearance over the passage-to-passage flow variability. A parametric study is proposed in order to assess the adequate number of blade channels to be considered in the flow ensemble averaging performed in the rotor domain.

4.1 Flow conditions

Parameter	Mean	Repeatability	Units
$P_{01,ms}$	1040	$\pm 1.5\%$	<i>mbar</i>
$T_{01,ms}$	436.3	± 4.0	K
$P_{01}/P_{s3,Hub}$	2.196	$\pm 0.5\%$	-
$P_{01}/P_{s3,Tip}$	2.222	$\pm 0.8\%$	-
T_{01}/T_{03}	1.239	$\pm 1.4\%$	-
$Tu_{01,ms}$	9.03	$\pm 3.0\%$	%
N	5925	$\pm 0.3\%$	<i>rpm</i>
BPF	4740	-	<i>Hz</i>
$Re_{2,NGV}$	6.93×10^5	-	-
$M_{3,rel}$	0.78	-	-
$Re_{3,Cax}$	2.76×10^5	-	-
T_{gas}/T_{wall}	1.50	-	-
Running tip clearance			
<i>IN01D & AC06D</i>	1.30	± 0.10	% <i>Span</i>
<i>AC07T</i>	0.77		

Table 4.1: Experimental test conditions

One single flow regime, representative of an HPT operated at engine cruise conditions, is considered for the present study. Engine-scaled thermodynamic flow conditions are established through the test article. The turbine rotor is operated at a mean rotational speed of 5925 RPM, corresponding to a thermodynamic speed of $281 \text{ rpmK}^{-0.5}$, with an average increase of $\sim 60\text{-}70$ RPM during the testing period (200-250 ms = 20-24 rotor revolutions). The sonic throat of the turbine test rig is finely regulated to establish an average total-to-static stage pressure ratio of 2.2, producing high subsonic flow conditions at the stage outlet ($M_{r,03} = 0.78$ and $Re_{03,c_r,ax} = 276000$). The operating parameters of the HP turbine stage, summarized in Table 4.1, have been constantly monitored for the full duration of the experimental campaign, corresponding to 150 short-duration tests.

For each test, the inlet mid-span total pressure $P_{01,ms}$ is computed by averaging the measurements of the two reference probes and of the two central heads of the Kiel

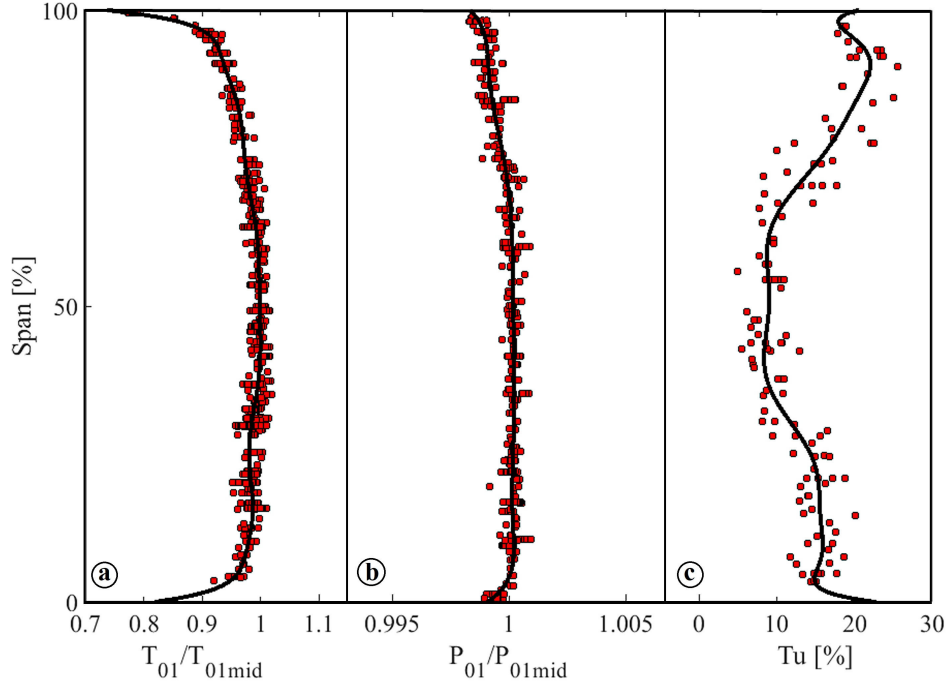


Figure 4.1: Experimental inlet profiles: normalized total temperature, total pressure and turbulence intensity

rake (Figure 2.15a,b), operating between 40% and 60% of the blade span. Similarly, the value of $T_{01,ms}$ is obtained by combining the signals of the two reference probes and of the central heads of the thermocouple rakes (Figure 2.15c). The two reference probes positioned in Plane 03 serve to assess the mean $P_{03,ms}$ and $T_{03,ms}$, while the mean endwall static pressure at the same axial location is recorder by endwall pressure taps.

The turbine inlet is characterized by axial inflow. Figure 4.1 presents radial profiles of total temperature, total pressure and turbulence intensity measured at the stage inlet, $50\%c_{v,ax}$ upstream of the vane leading edge. These distributions, plotted in the form of experimental points complemented by fitting curves, served as boundary conditions for the numerical study. All the plots are normalized by their mid-span values reported in Table 4.1.

Figure 4.1a shows how the inlet total temperature profile is considerably affected by the presence of the endwalls, with a measured temperature drop up to 15% $T_{01,ms}$ in the near tip region. The spatial resolution in proximity of the lower endwall is, however, affected by the thermocouple rake design, with the lowest measurement point positioned at 4.4% of the inlet span. The data scatter includes measurements performed through the full experimental campaign at three different circumferential

positions, demonstrating a high level of temperature uniformity across the annulus within the test-to-test repeatability of $\pm 1.5\%$ of the mid-span value.

The inlet total pressure profile is visualized in Fig.4.1b, showing a uniform radial trend and an approximate drop of $0.1\% P_{01,ms}$ (i.e. ~ 1 mbar) in close proximity to the upper and lower endwalls. The measurement points have been recorded through the 150 tests at two different circumferential positions along the annulus (352° and 150° respectively, with respect to vane 01). Hence, the radial distribution confirms a circumferential uniformity of the inlet pressure profile within the measurement uncertainty of the pressure scanner, corresponding to $\pm 0.1\%$ of the mid-span value. A test-to-test repeatability of $\pm 1.5\% P_{01,ms}$ was achieved, as specified in Table 4.1.

The turbulence intensity profile shows an average mid-span value ranging between 7% and 12% . A substantial increase in turbulence intensity is registered in proximity to the upper and lower endwall, with maximum values of 25% and 20% respectively, confirming the trends reported in [95] for the same turbine inlet setup.

4.1.1 Vane pressure profiles

The vane static pressure distributions at mid-span ($50\%h_v$) and near-casing sections ($90\%h_v$), normalized by the stage inlet total pressure, are presented in Figure 4.2. Point-wise experimental data, recorded over 25 tests by pneumatic taps on the instrumented vane, are plotted together with 95% confidence level error bars, which are however within the resolution of the visualized points (below $0.1\% P_{01}$).

Steady state RANS and unsteady NLH results are also visualized for comparison. The NLH distributions are presented for both the IN01D and AC06D single-tip cases, characterized by a finer grid of the vane domain, and for the Full-Annulus computation. The RANS results are presented for blade IN01D and averaged between all the seven different rotor blade geometries of the rainbow rotor.

The numerical predictions show a perfect agreement with the experimental data at the blade pressure side, while higher discrepancies are visualized in the aft part of the NGV suction side. While the location of the suction side pressure minima is adequately captured by all the numerical predictions, only the single-tip NLH data present an adequate quantitative match of the experimental results, with no difference between the IN01D and AC06D predictions.

At 50% of the NGV span, the minima predicted by the RANS (IN01D and rainbow-average case) and full-annulus NLH results underestimate the measurements by $2.2\% P_{01}$ and $3.9\% P_{01}$ respectively.

This discrepancy indicates the relevance of the unsteady numerical approach com-

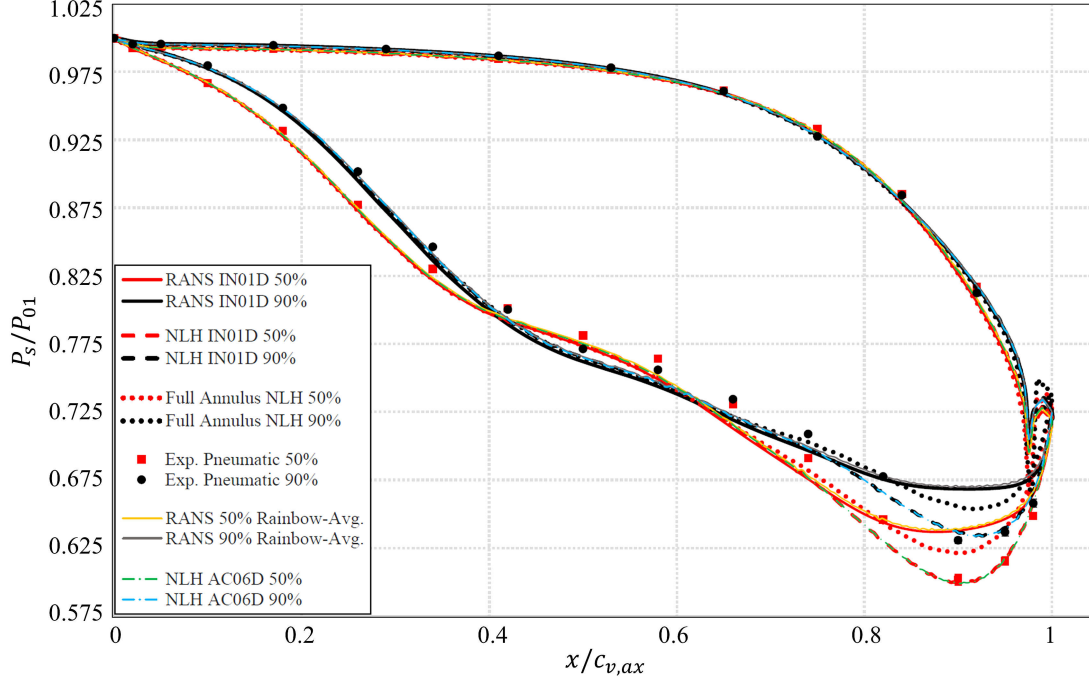


Figure 4.2: Measured and predicted vane static pressure distribution at 50% and 90% h_v

pared to the less computationally-expensive steady RANS calculation for the evaluation of the time-average NGV blade load. The better matching of the single-tip NLH computation is shown both at 50% and 90% h_v , for both blades IN01D and AC06D, excluding any potential influence of the rotor tip design. In support to this, no variations are shown between the individual RANS results of tip IN01D in comparison to the pressure distribution resulting from the RANS rainbow-average.

The lower accuracy shown by the full-annulus NLH predictions is therefore to be ascribed to the coarser mesh refinement in close proximity to the solid walls ($y^+ \sim 30$) adopted for these computations.

4.1.2 Vane outlet

The time-average casing static pressure at the vane outlet (Plane 02) is displayed in the top section of Figure 4.3, with the horizontal axis indicating the peripheral position along the NGV pitch. The zero-phase position corresponds here to the vane trailing edge.

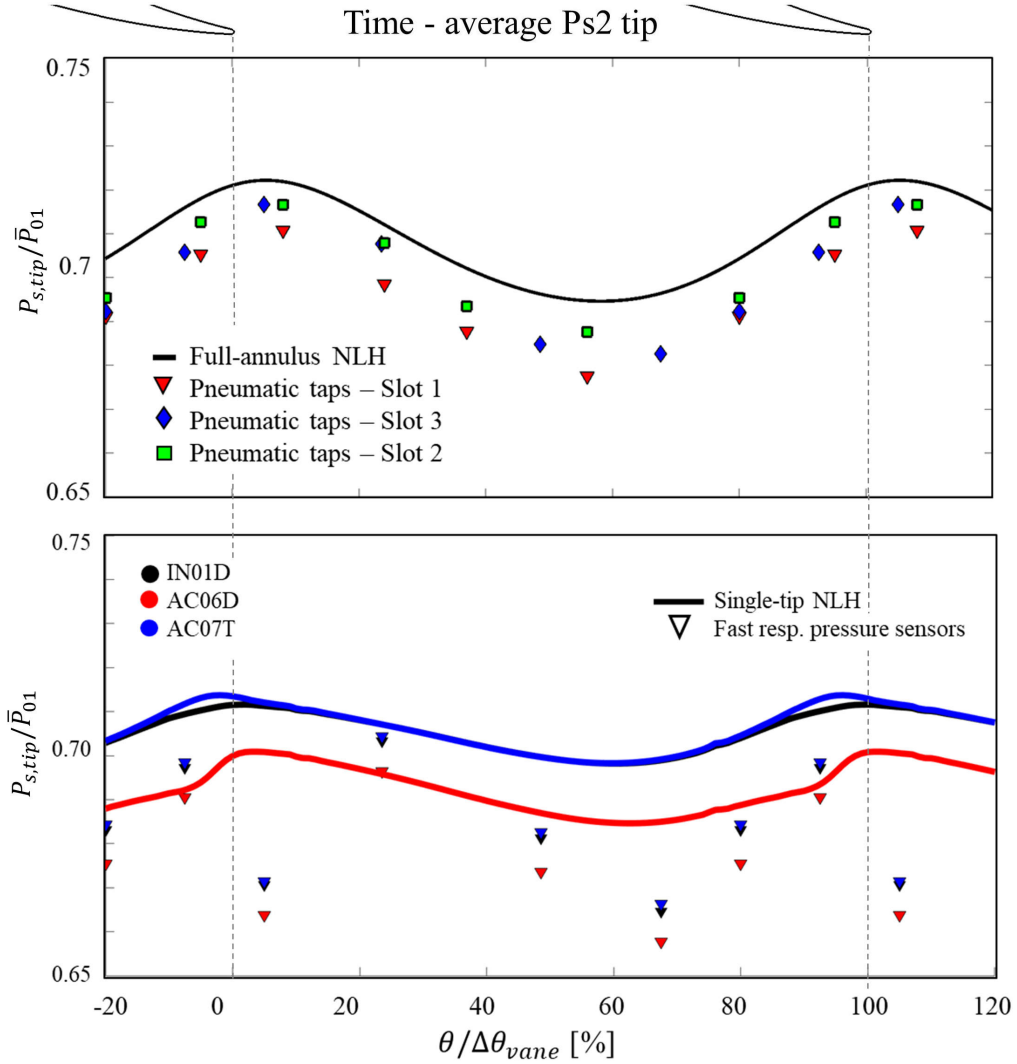


Figure 4.3: Time-average casing static pressure distribution at the vane outlet (Plane 02): experimental vs. numerical results

The experimental distributions are recorded at the three different circumferential locations along the annulus (slot 1, 2 and 3) by means of pneumatic taps (slot locations reported in Appendix A). The numerical dataset is extracted from the full-annulus CFD computations.

The potential effect of the HP vane is visible as a static pressure variation of $\sim 2.5\%$ of the inlet total pressure, with the maxima located between 0 and 10% of the vane pitch. A good qualitative and quantitative match is observed between numerical and experimental results. The offset between the predicted and measured data is below 1% of P_{01} and of the same order of magnitude of the offset displayed between pressure measurements at different experimental slots.

The experimental dataset of Slot 01 (red triangles) exhibits a constant negative offset in the mean pressure value of about $1\%P_{01}$ with respect to Slot 02 (green squares). The reason for this discrepancy is the peripheral position of the measurement slots. Slot 01, positioned at 130° along the turbine annulus represented in Figure 2.11, is characterized by a larger estimated running rotor tip gap ($+0.16$ mm) with respect to Slot 02, centered at about -115° in the same figure. Slot 03, located at 0° and thus characterized by an intermediate clearance value, shows a higher correlation with Slot 02 in the near-PS region (up to 40% of the vane pitch) and a closer agreement with Slot 01 between 50% and 100% of the vane pitch.

The time-mean casing static pressure at Plane 02 is displayed for each tip pro-

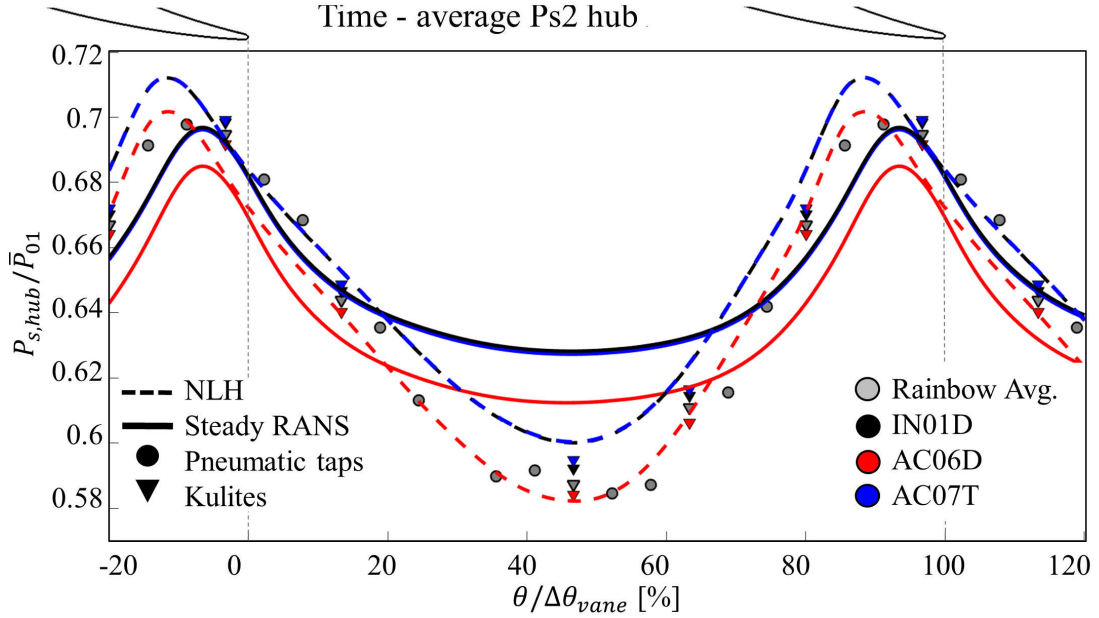


Figure 4.4: Time-average hub static pressure distribution at the vane outlet (Plane 02): experiments vs. RANS and single-tip NLH

file in the bottom section of Figure 4.3. The experimental dataset, collected by fast response pressure sensors along one full vane pitch in Slot 03, is compared to time-average distributions elaborated from full-stage single-tip NLH computations. The sector-to-sector variations are confirmed by the experimental results, with blade AC06D characterized by a measured pressure value up to $1\%P_{01}$ lower than the other blades, against a predicted pressure deficit of $\sim 1.3\%P_{01}$. This result is consistent with the trend highlighted between the different measurement slots: higher clearance values determine lower casing static pressure levels at the rotor inlet. Globally, the predicted distributions overestimate the mean measured pressure level by $\sim 1.5\%P_{01}$

for all the profiles.

The time-average static pressure variation at the hub endwall is presented in Figure 4.4, with measurements by Kulites and pneumatic taps compared to Steady-RANS and single-tip NLH predictions. The rainbow-average measurements are indicated in gray. The influence of the NGV potential field, at this location, is more significant than at the casing, with measured variations of amplitude $12\%P_{01}$. The experimental and RANS datasets agree on the position of the pressure peak, located between 85 and 100% of the vane pitch. However, the RANS predictions overestimate the minimum value by $4\%P_{01}$ in reason probably of the mixing-plane approach adopted at the stator-rotor interface.

The full-stage NLH predictions, conversely, appear to match well the position and

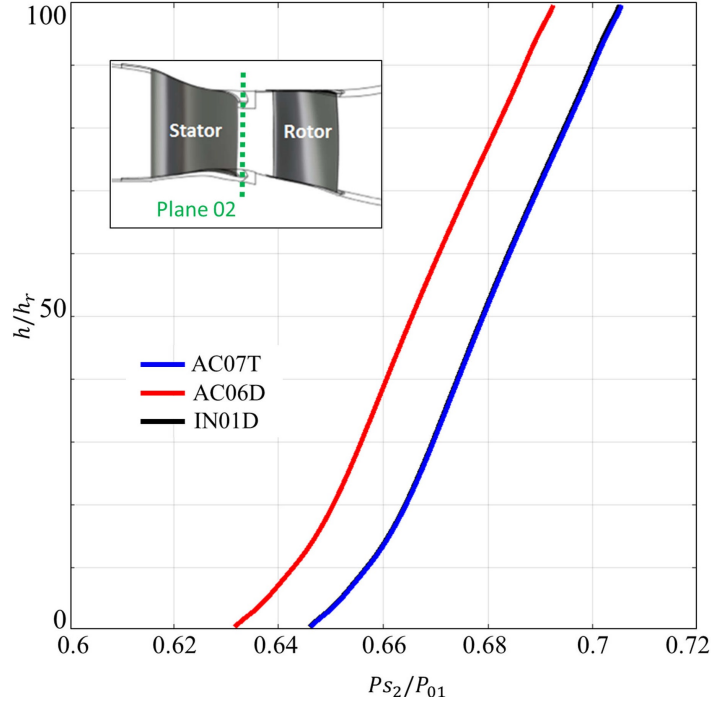


Figure 4.5: Predicted radial profiles of pitchwise-average static pressure at the vane outlet (Plane 02), single-tip NLH

amplitude of the pressure minimum, as well as the tip-to-tip variations. The sector-resolved dataset confirms that the mean pressure level of AC06D is up to $2\%P_{01}$ lower than the one of blades AC07T and IN01D in correspondence of the pressure minimum, while the gap reduces to $1.5\%P_{01}$ at the pressure peaks. A slight peripheral shift is present, however, in the position of the NLH-predicted pressure maximum (-10% of the vane pitch with respect to the experimental dataset), which also exceeds

the RANS predictions in amplitude by $1\%P_{01}$. The low-spatial resolution measurements suggest that the correct amplitude of the pressure peak lies in between the two predicted values.

The radial distribution of pitch-average static pressure by single-tip NLH at the vane outlet is presented in Figure 4.5 for the three tips. The static pressure is seen to monotonically decrease from the outer to the inner endwall. The negative offset of blade AC06D appears unchanged across the blade span, confirming the endwall measurements.

To provide a more comprehensive evaluation of the potential field at the outlet of

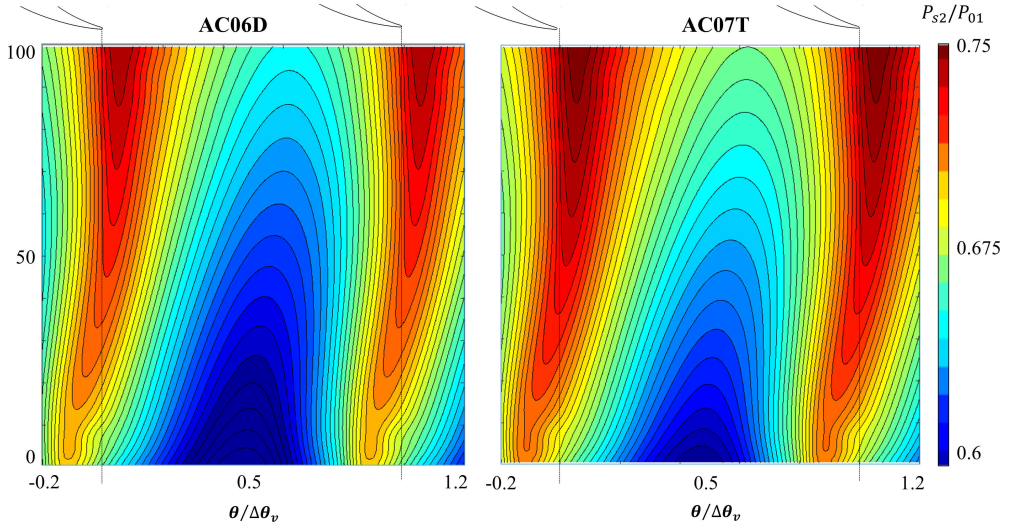


Figure 4.6: Time-average static pressure contours at vane outlet for blades AC06D and AC07T, single-tip NLH

the HPT vane, Figure 4.6 displays the time-average static pressure contours predicted by single-tip NLH computations for blade AC06D and AC07T (considered the two extreme cases given the previous analysis). The horizontal axis represents the peripheral position with respect to the NGV trailing edge, with the coordinate normalized by the vane pitch. The vertical axis, conversely, indicates the radial position and extends over the full span of the flow path in Plane 02.

A periodic high-pressure region, produced by the three dimensional vane geometry and leaned towards the direction of rotation of the rotor blades (left to right in the figure), indicates the fluid leaving the NGV pressure side. Blade AC06D shows, as expected, a globally lower mean pressure level than blade AC07T, with a negative offset of about $2\%P_{01}$ which confirms the endwall pressure analysis and indicates the influence of the open-rim rotor tip profile on the upstream potential field. The pressure

variation pattern, however, correlates quite well between the two profiles, indicating the uniform pressure offset as the only discrepancy between the two blades.

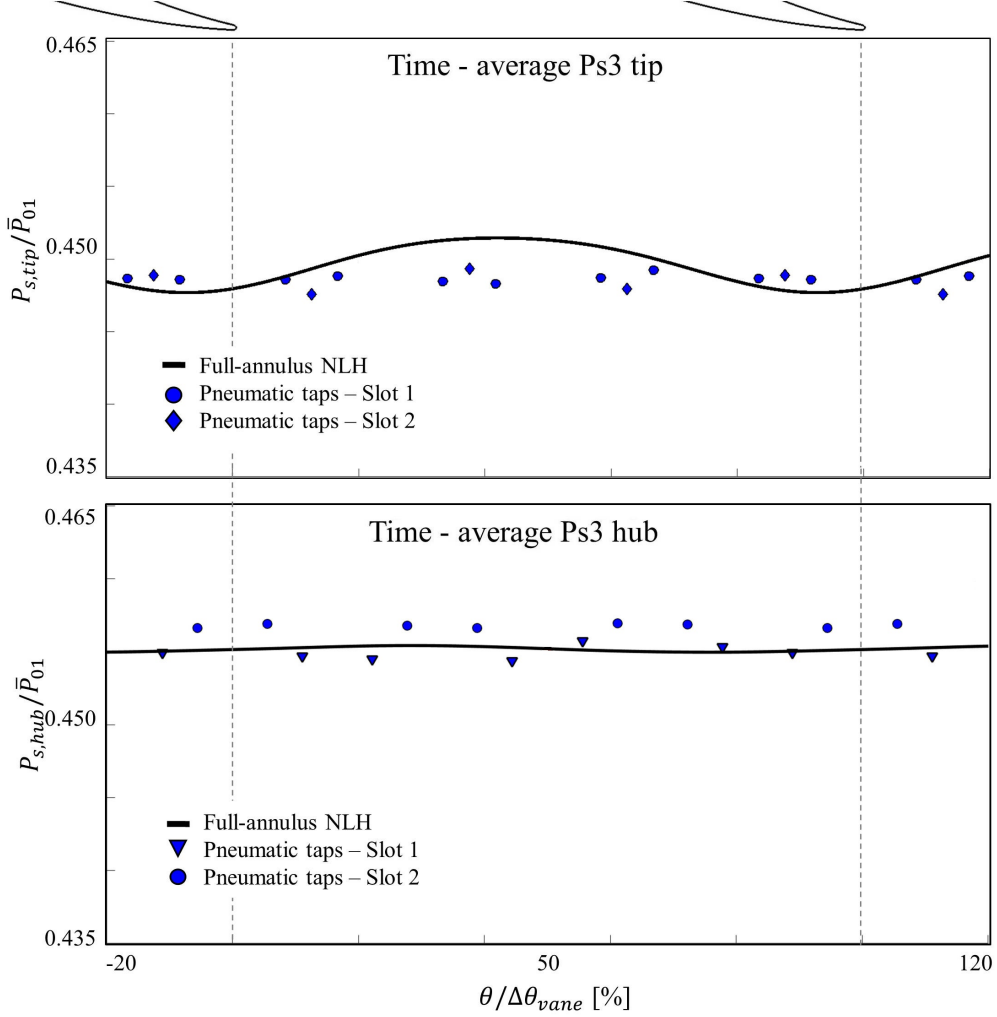


Figure 4.7: Time-average tip static pressure distribution at the rotor outlet

4.1.3 Rotor outlet

The time-average casing and hub static pressure distributions at the rotor outlet (Plane 03) are visualized in Figure 4.7. Pneumatic taps measurements are displayed in comparison to numerical data extracted from the full-annulus NLH computations. No time-average predictions are available, in this case, by the RANS simulations performed in the rotor domain.

The pressure variations registered at the two endwalls are very mild compared to what displayed in Plane 02, with little influence of the potential field of the upstream vane. The maximum pressure variation measured at the outer and inner endwalls is

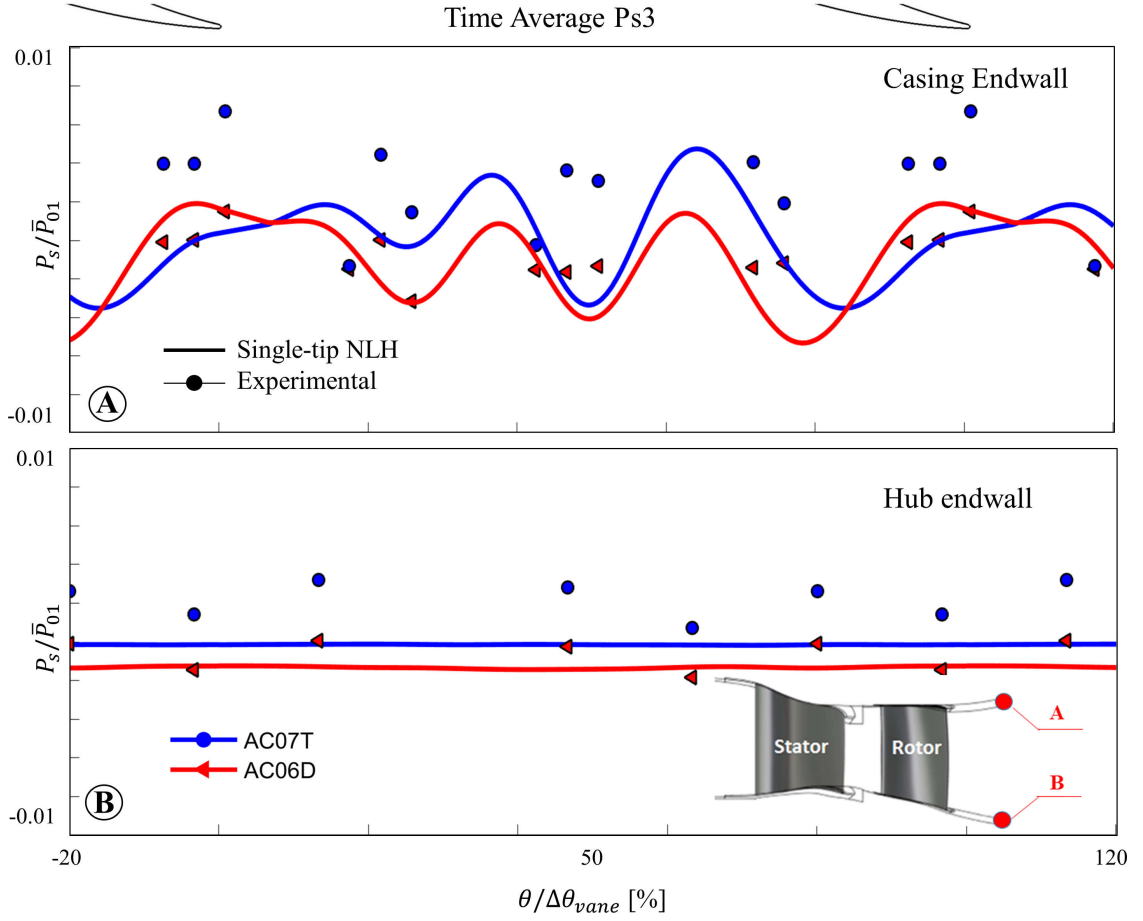


Figure 4.8: Sector-resolved time-average hub and casing static pressure at rotor outlet

below $\pm 0.1\% P_{01}$, significantly close to the measurement uncertainty of the pneumatic taps ($\pm 0.05\% P_{01}$).

The sector-resolved time-average pressure distributions at Plane 03 are visualized in Figure 4.8. Fast response measurements are compared to time-average pressure distributions extracted from single-tip NLH computations, represented as the variation with respect to blade IN01D. This representation allows to discard the systematic uncertainty of the fast response sensors, amounting to $\sim \pm 3 - 5$ mbar, and highlight the sector-to-sector variations with a precision error of ± 0.1 mbar.

Blade AC06D confirms a lower mean-pressure level at the hub endwall with respect to blade AC07T, with a predicted difference of $0.15\% P_{01}$ and a measured offset of $0.3\% P_{01}$. Blade AC07T matches perfectly the pressure distribution of IN01D, while showing a slightly higher pressure level in the experimental results.

At the casing, the single-tip NLH predicts a more variable distribution in comparison to the rainbow-average full-annulus CFD (Figure 4.7). A lower mean pressure differ-

ence is shown between blade AC06D and IN01D, with a maximum negative offset of the optimized squealer of $\sim 4\text{mbar}$. Blade AC07T shows a positive offset with respect to blade IN01D, up to $0.8\%P_{01}$, in reason of the tight operative tip gap.

4.1.4 HPT operative point

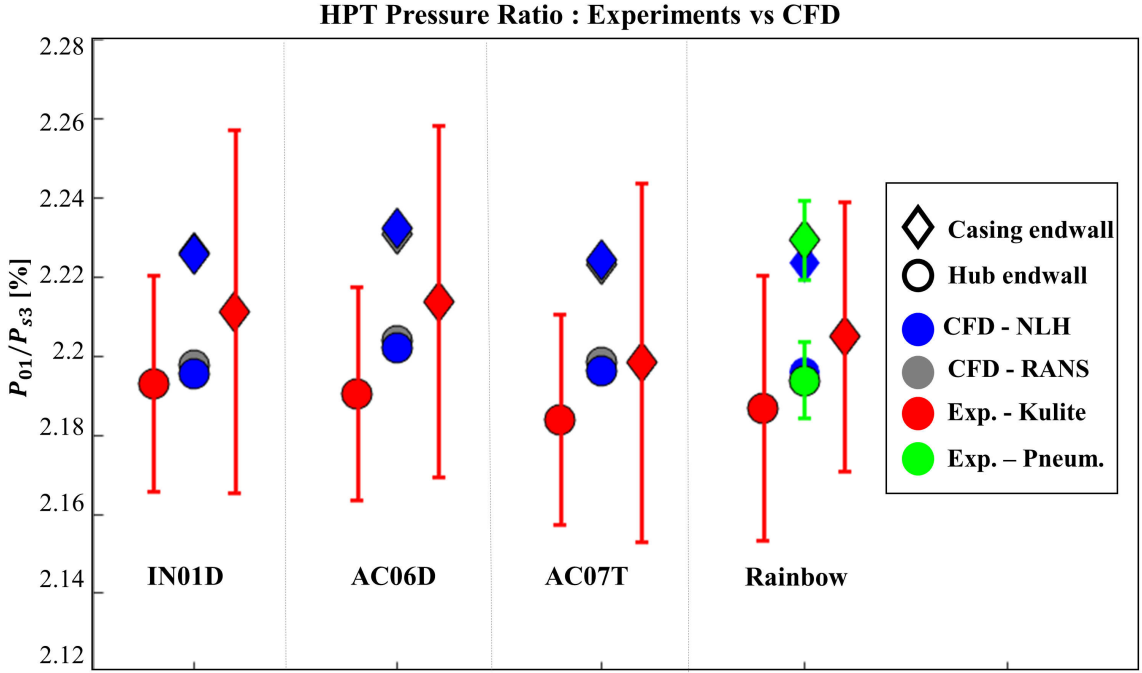


Figure 4.9: HPT stage total-to-static pressure ratio: sector- and rainbow-average results

Figure 4.9 presents the HPT stage pressure ratio averaged over the three investigated sectors and over the full rainbow rotor. The ratio is computed between the mean inlet total pressure at mid-span and the average static pressure measured at the casing and hub endwalls in Plane 03. The experimental results by the fast response endwall pressure sensors are reported in red, while the pneumatic taps measurements for the full rotor annulus are reported in green.

The sector-averaging is performed over four central blades of sectors IN01D and AC06D and over three central blades for tip AC07T. The choice of the blade passages will be further motivated in the rainbow periodicity study (Section 4.7). The error bars indicate the measurement repeatability with 95% confidence level computed for each tip on a selection of 15 tests. A mean test-to-test variability of ± 0.045 is computed at the casing for the three sectors, while a lower repeatability error is computed

at the hub (within ± 0.025). The numerical predictions refer to single-tip NLH (blue) and steady RANS computations (grey). The rainbow-average results, also reported in blue, are generated by the full-annulus NLH computations.

A good agreement is detected between the sector-to-sector variations predicted by the CFD and the experimental data, with a maximum difference within 0.8% of the average pressure ratio between the three tips. The casing pressure ratio is higher than the hub value in both experimental (+2%) and numerical (+3%) results. No significant inconsistencies are noticed between steady-RANS and NLH results.

The rainbow-rotor pressure ratio from pneumatic measurements presents a higher repeatability with respect to the fast response sensors, with a computed test-to-test variability below 1% of the mean value. At both measurement locations, the numerical predictions fall well within the uncertainty range of the measurements but the level of agreement with the pneumatic taps results is considerably higher.

The experimental pressure ratio variability between the different profiles, below 0.8% of the mean rainbow-average value, is in good agreement with the numerical predictions. This result indicates that the usage of the rainbow configuration in the turbine test rig, where the total stage mass flow is imposed to achieve the operative pressure ratio, allows to establish the correct operative point for the three tips, in concordance with flow predictions where the stage pressure ratio is set as boundary condition.

The HP stage degree of reaction is computed individually for the three tips and for the full rainbow rotor following the definition in Equation 4.1. The numerical and experimental results, computed at the hub and casing endwalls, are presented in Figure 4.10.

$$r_{stage} = \frac{\frac{P_{s2}}{P_{01}} - \frac{P_{s3}}{P_{01}}}{1 - \frac{P_{s3}}{P_{01}}} \quad (4.1)$$

The experimental sector-resolved degree of reaction, based on measurements by fast response sensors at the vane and rotor outlet, is shown to be in good agreement with the numerical predictions. The sector averaging is performed over the same passages considered for the pressure ratio study and on a selection of 15 tests. The repeatability propagation has been computed individually for each tip profile by Taylor series expansion. The final variability, computed with 95% confidence level, is below 1% for the fast-response sensors dataset and below 0.2% of the mean value for the pneumatic measurements. Sector AC06D shows, in general, a slightly lower degree of reaction (-2%) with respect to tips IN01D and AC07T, probably due to the larger effective tip clearance and in good agreement with the CFD predictions.

Numerical and experimental results show a consistently higher degree of reaction at

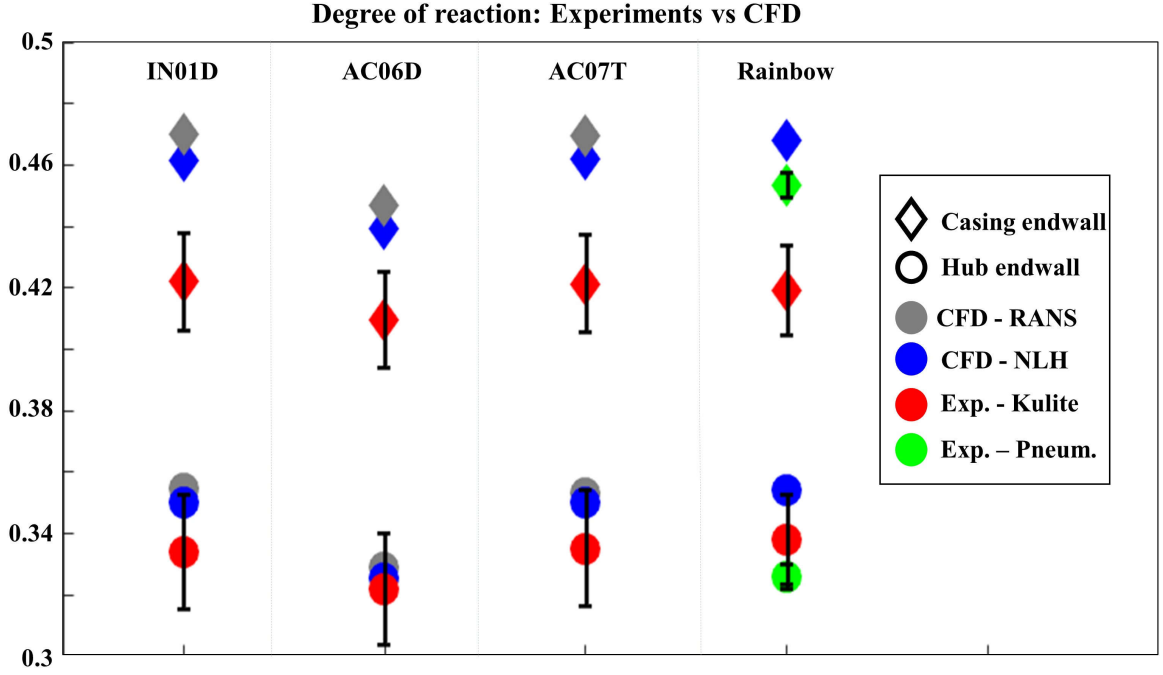


Figure 4.10: HPT stage degree of reaction: sector- and rainbow-average results

the outer endwall with respect to the turbine inner wall (~ 0.1) for the individual profiles and the full-rotor average. The pneumatic measurements present a significantly better agreement with the numerical predictions at the turbine casing with respect to the fast-response measurements. The substantial offset of the sector-resolved experimental degree of reaction at the turbine casing is most probably due to the absolute measurement error which affects the Kulite sensors, producing an overestimation of the outlet casing static pressure (Figure 4.7), combined to the underestimation of the casing static pressure at the rotor inlet (Figure 4.3).

The NLH-predicted casing degree of reaction shows a 1% negative offset with respect to the RANS results, consistent within the three sectors. This discrepancy is motivated by the better match of the NLH-predicted static pressure minimum at the vane outlet presented in Figure 4.4.

The average rotor passage mass flow predicted by steady and unsteady flow computations is presented in Figure 4.11. The results from single-tip RANS (grey circles) and NLH (red squares) are compared here to the mean passage mass flow computed over the complete sectors (7 passages for IN01D and AC06D, 6 passages for AC07T) in the full-annulus NLH computations. The latter results are plotted for the rotor outlet (Plane 03, blue diamonds). The rainbow-average mass flow is extracted on the full-annulus NLH and computed as a weighted average for the single-tip NLH predictions.

The mass flow discrepancy between blade IN01D and AC07T results below 0.1% in the single-tip predictions, with good agreement between steady and unsteady computations. Both tips show a mass flow in line with the rainbow-average value while an increase of about 1% is visualized for blade AC06D, motivated by the larger effective tip clearance generated by the front and aft rim apertures of this tip.

In the full-annulus NLH study, an equal increase of $\sim 1\%$ is shown in the mass flows of blades IN01D and AC07T, as well as in the extracted rainbow-average value. The discrepancy between blade AC06D and the other profiles, in this case, increases to about $\sim 2\%$ of the average value.

With the stage pressure ratio set as stage boundary condition in the full-annulus NLH computations, the larger effective clearance of blade AC06D promotes a slight redistribution of the mass-flow along the rotor annulus. In the experimental campaign, however, the total mass flow of the HPT stage is set by the sonic throat of the facility and the individual sector mass flows cannot be measured. The measured sector-resolved pressure ratio, however, demonstrates only a mild discrepancy between blade AC06D and the other profiles, indicating that the operating point of the different sectors is uniform within acceptable limits. A further validation of the multi-sector setup could be obtained by performing a full-annulus NLH computation at a prescribed stage mass flow, to fully replicate the flow conditions imposed by the short-duration turbine test rig.

4.2 Time-average flow

4.2.1 Rotor Casing Static Pressure.

The sector-resolved time-averaged static pressure distributions measured at the over-tip rotor casing are presented in Figure 4.12(top) as a function of the axial position over the rotor axial chord. The time-resolved pressure signals, acquired by the 66 fast-response sensors, are first phase-locked averaged over one full rotor revolution and subsequently ensemble averaged over each sector, discarding the extreme passages at the sectors boundaries to ensure a high level of flow repeatability. Hence, five passages were considered for sectors IN01D and AC06D (7-blades sectors) and four passages were considered for sector AC07T (6-blades sector).

The time-mean pressure values recorded at different peripheral positions along the vane pitch were then pitch-averaged at each axial location to remove the potential effects of the upstream NGV. The dataset is normalized by the turbine inlet total

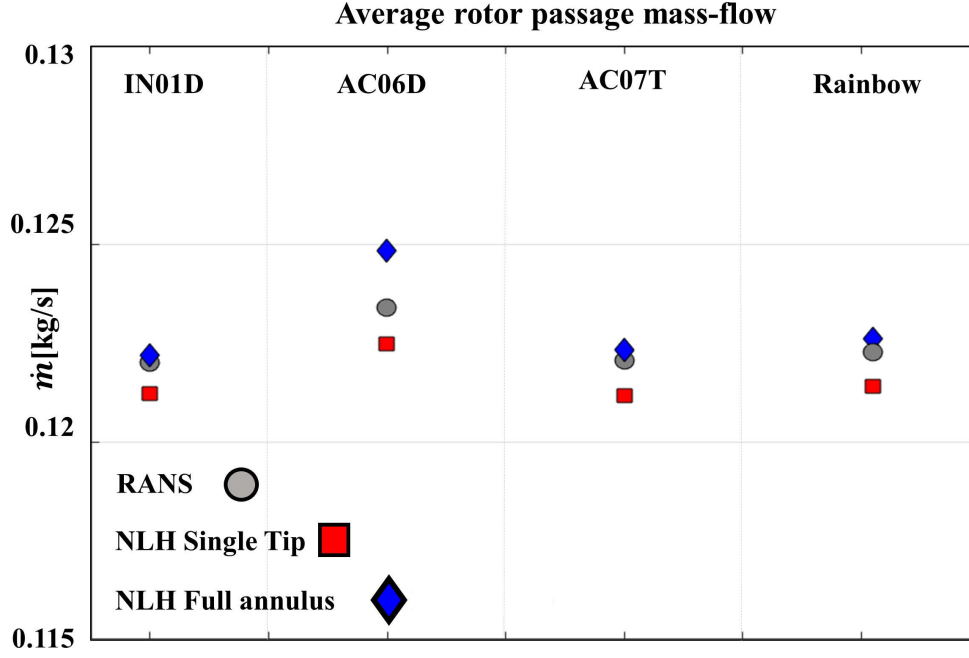


Figure 4.11: Sector- and rainbow-average rotor passage mass flow at Plane 03

pressure value at mid-span. The corresponding time-average pressure distributions computed on the single-tip NLH dataset are presented as a solid line in the same plot, while the steady pressure distribution predicted by the RANS computations in the rotor domain is reported as a dashed line.

The axial static pressure distributions of the three profiles exhibit the typical decreasing trend along the rotor axial chord due to the gas expansion. The highest values of pressure are observed upstream of the rotor airfoil leading edge, before the flow incurs in a strong acceleration imparted by the blade turning. After a mild pressure decrease in the first half of the blade axial chord ($\sim 10\%$ of upstream value), the largest flow acceleration takes place in the aft 50% of the rotor axial chord. In this region, the normalized static pressure drops by more than 30%. The carved profile (AC07T) presents steeper acceleration (40% to $70\%c_{r,ax}$) and deceleration (90% to $105\%c_{r,ax}$) rates than the squealer-type profiles.

In general, the RANS and NLH computations show a very high level of agreement, demonstrating the accuracy of the steady computations in capturing the time-average static pressure field at the rotor casing. The experimental dataset provides a good validation of the sector-to-sector flow differences, showing however a globally lower mean pressure level with respect to the CFD predictions. The pressure offset, amounting to $\sim 2 - 2.5\%P_{01}$, is consistent among the three tip profiles and tends to decrease moving downstream along the rotor chord.

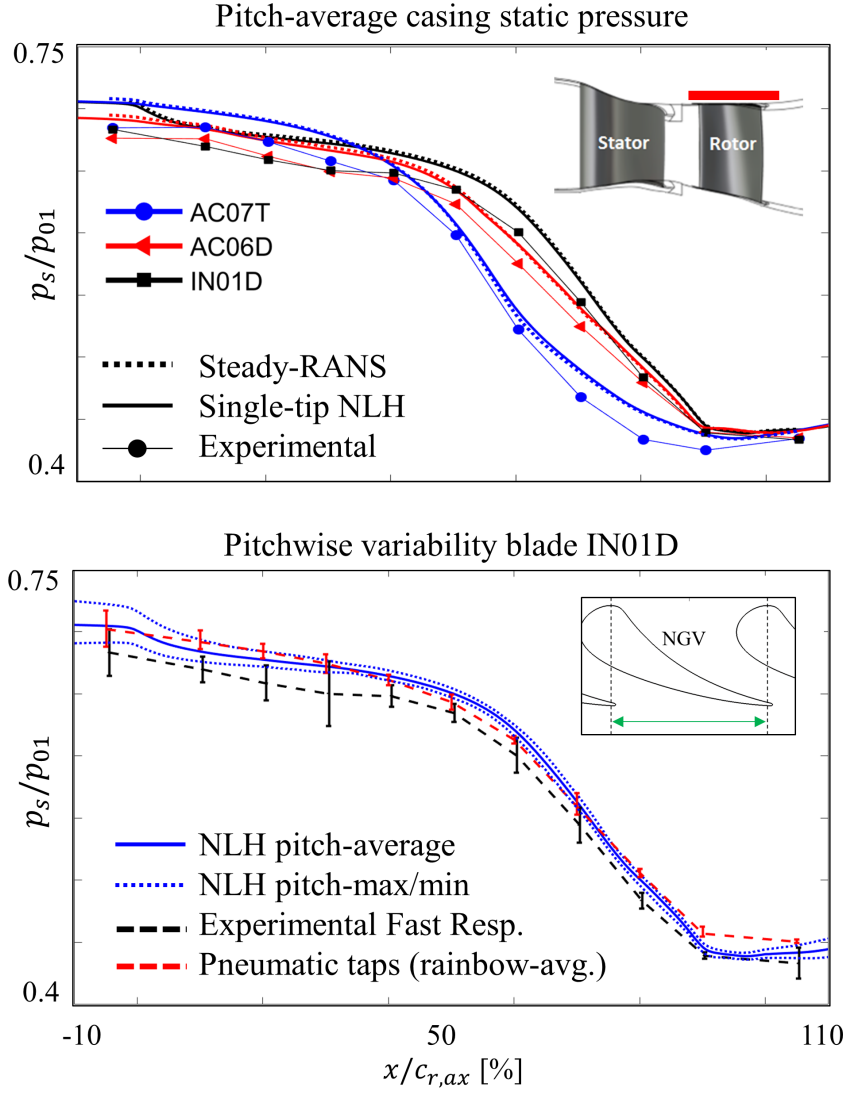


Figure 4.12: Sector-resolved pitch-averaged mean casing static pressure distributions (top) and pitchwise variability for sector IN01D (bottom)

In order to quantify the impact of the vane potential field, the pitchwise variability of the axial static pressure distribution is presented in Figure 4.12-bottom in form of error bars. The sector-average variability of blade IN01D, measured by Kulite sensors, is reported in black, while the rainbow-average pneumatic measurements are reported in red for reference. Similarly, the vane-induced variation predicted by the single-tip NLH for blade IN01D is presented in form of blue dotted lines. The measured variability, both by pneumatic pressure taps and fast response sensors, amounts to $\pm 2.5\% P_{01}$ at the rotor leading edge and shows a global decreasing trend moving downstream along the rotor chord. At 30% $c_{r,ax}$, an increased variation is visualized for the fast response measurements, probably due to a slight offset of one of the sensors. The

variation is, globally, in very good agreement with the NLH predictions, confirming the effectiveness of the computational approach in matching the time-mean effect of the blade rows interactions at the rotor casing.

The sector-resolved time average pressure contours measured at the rotor casing by

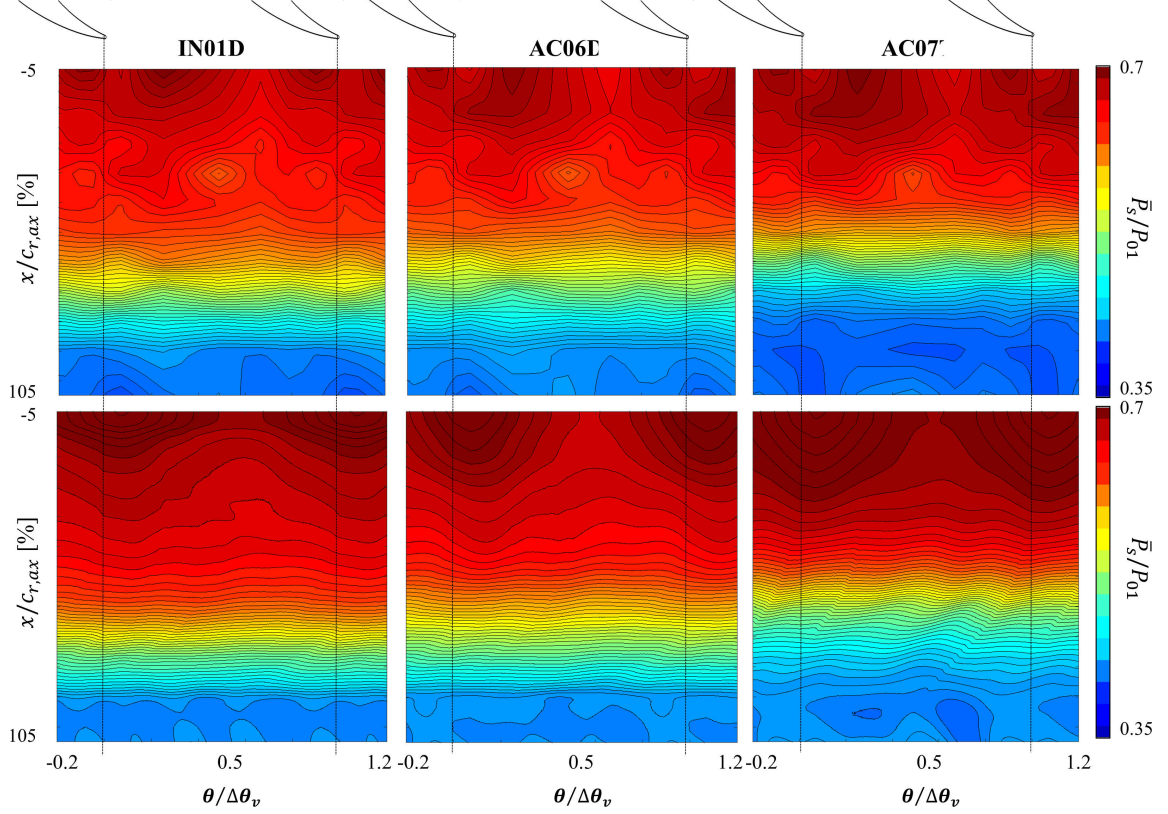


Figure 4.13: Sector-resolved time-average casing static pressure: experimental (top) and single-tip NLH (bottom) results

fast-response sensors are presented in Figure 4.13 (top) and compared to numerical predictions produced by single-tip NLH computations (bottom).

The plots report the pressure variations in the stationary frame of reference, between -5% and 105% of the rotor axial chord (referred to the rotor LE) and over a full NGV pitch. The potential field of the upstream vane is evident in the first 40% of the rotor axial chord for all the rotor tips, with a periodic static pressure minimum appearing between 50% and 60% of the vane pitch (as discussed in Figure 4.3). Among the three profiles, AC06D reaches the lowest inlet static pressure level, due to the front rim aperture which produces a larger tip clearance in proximity of the rotor LE.

The over-tip region is characterized by a progressive decrease of static pressure and a desensitization to the vane-relative peripheral position. Blade AC07T shows the highest axial gradient between 40% and 70% $c_{r,ax}$, in reason of the high tip flow

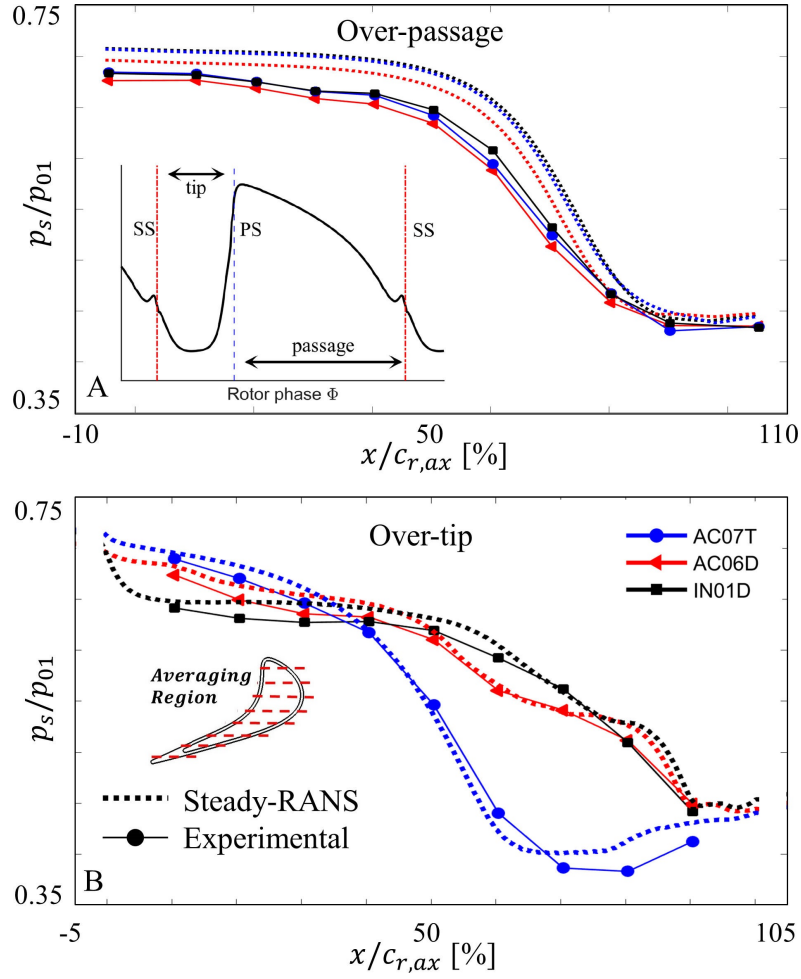


Figure 4.14: Sector-resolved pitch-averaged mean casing static pressure: over-passage (a) and over-tip (b) distributions

acceleration promoted by the contoured surface, and the earliest disappearance of vane-dependent pressure variations in peripheral direction. In general, the experimental contours show a high level of agreement with the NLH results, validating the amplitude and phasing of the time-average pressure variations for the three tips.

In order to offer a deeper insight into the impact of the blade tip design on the casing pressure field, the time-resolved data have been partitioned into passage and over-tip region. This procedure is graphically shown in the detail of Figure 4.14(a), that illustrates a typical blade passing event in terms of rotor casing pressure field. The over-tip area is identified as the time when the casing gauge is directly exposed to the blade tip surface. To account for tip-induced fluctuations occurring in the close proximity of the blade tip perimeter, the over-tip region has been extended by 5% of the blade pitch in both suction and pressure side directions. The sensor data are

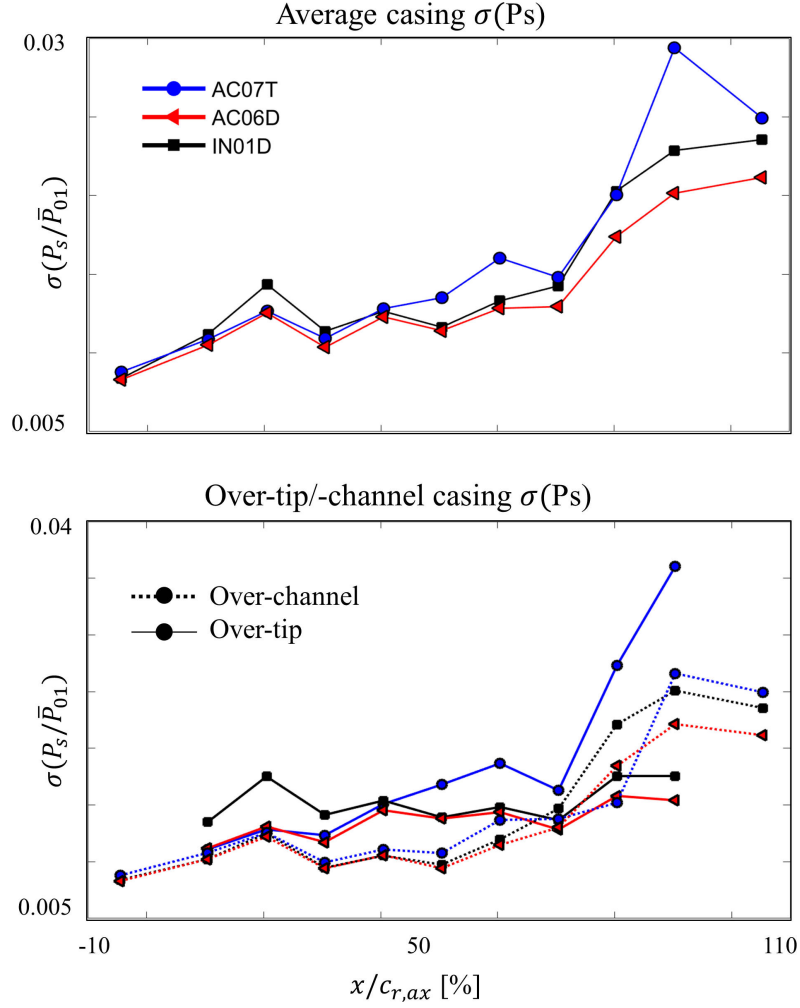


Figure 4.15: Sector-resolved pitch-averaged casing static pressure: stochastic component

area-averaged at each temporal phase to obtain the average values associated with the passage and the over-tip regions. The over-passage and over-tip axial distributions are presented in Figure 4.14(a) and Figure 4.14(b) respectively.

As expected, the passage static pressure is mildly affected by the tip geometry. Interestingly, even in the over-passage region, blade AC06D presents a slightly smaller casing static pressure (about -2% of P_s/P_{01}) than the other two blade tips. While this is in good agreement with what commented in Figure 4.3, the influence of this tip geometry also on the passage region confirms that the discontinuous rim generates a mild increase in stage mass flow (+1%) with respect to the IN01D and AC07T cases. The numerically predicted axial pressure evolution compares well with the experimental data, although the mean pressure offset is still present for the three tips, showing a decreasing trend with the axial chord.

Figure 4.14(b) presents the tip-driven static pressure distributions. The effect of the tip design is significant and the three geometries show diverse pressure evolutions with axial distance. In the region upstream of the blade leading edge, IN01D shows a steep decrease of static pressure, which then remains constant up to 30% of the axial chord. A more gradual static pressure drop of $\sim 5\text{-}10\%$ P_s/P_{01} is registered for both AC06D and AC07T in the same axial interval. This can be explained considering that these latter profiles feature an open gap that promotes the flow acceleration across the over-tip channel, while the close rim of IN01D accelerates the flow at the blade LE.

In between 30% and 50% $c_{r,ax}$ both squealer-type tips present a plateau in static pressure, consequence of the aerodynamics established by the squealer design over the central cavity. The contoured surface of tip AC07T promotes, conversely, a strong flow acceleration across the gap width. The large difference in over-tip flow acceleration rates between squealer and carved profiles motivates the variation in pitch-average static pressure observed in Figure 4.12. The RANS-predicted tip pressure compares well with the experimental data. A slightly lower mean pressure is measured in the front part of the three tips, but generally the offset is lower than $1\%P_{01}$, that is half of what detected in the over-passage region. The general concurrence on the pressure axial evolutions, however, is satisfactory: the relative differences among blades are correctly captured by the simulations at all axial positions.

Figure 4.15(top) displays the pitch-averaged time-averaged random unsteadiness levels ($\sigma(P_s)$) measured over the three tip profiles. An increasing trend of mean stochastic fluctuations is registered for the three tips, starting below $1\%P_{01}$ at the rotor leading edge. The two squealer-type profiles are shown to reach a maximum of $2.5\%P_{01}$ at the rotor trailing edge, with blade IN01D slightly exceeding blade AC06D in the aft part of the rotor chord (70-105% $c_{r,ax}$). The contoured shape of blade AC07T promotes a first local peak in of $\sigma(P_s)$ at 60% of the axial chord and a global maximum at 90% $c_{r,ax}$ which exceeds by almost $1\%P_{01}$ the level reached by AC06D at that same axial station.

To clarify the origin of the tip-to-tip discrepancies, Fig. 4.15(bottom) presents the over-passage and over-tip average distributions of $\sigma(P_s)$. The two squealer-like tips show a very slight increase of stochastic fluctuations across the rotor tip, with a global difference below $0.8\%P_{01}$ between the blade LE and TE. A more significant increase of $\sigma(P_s)$ is detected, conversely, in the over the blade channel ($1.5\%P_{01}$ from 60% $c_{r,ax}$ to blade TE), as a result of the TLV generation and of the unguided flow in the rotor passage. Interestingly, a steep increase in $\sigma(P_s)$ is detected in the aft over-tip

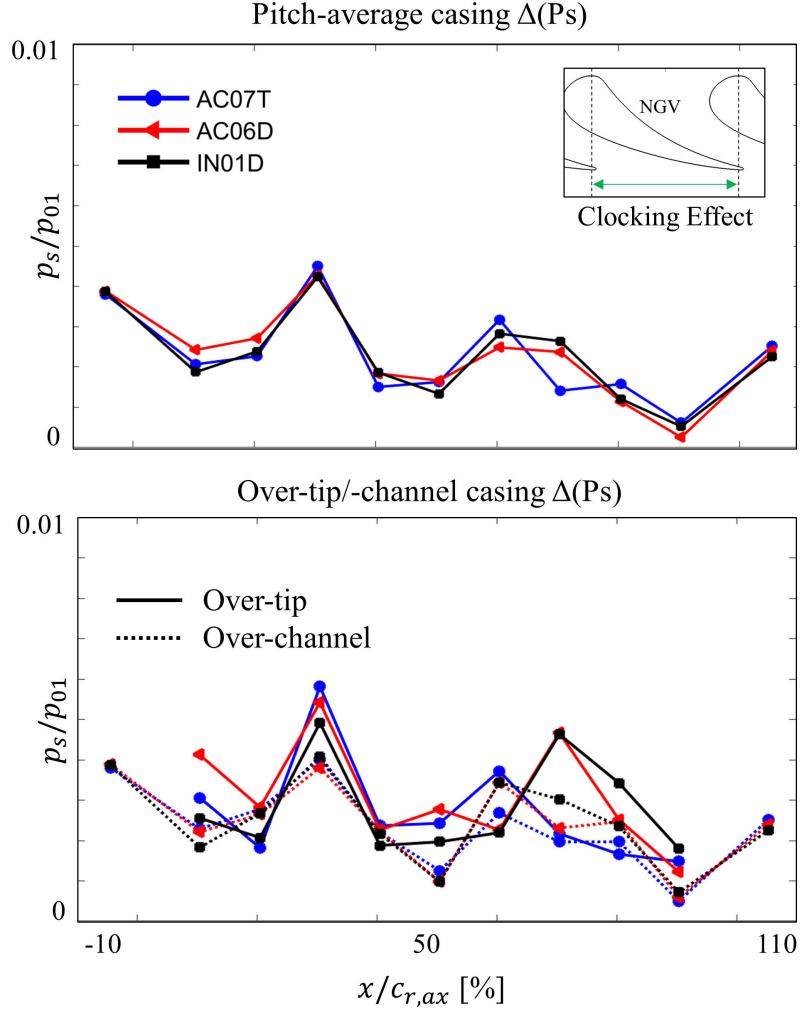


Figure 4.16: Sector-resolved pitch-averaged casing P_s/P_{01} : NGV clocking effect

region of AC07T. In this region of the tip, a flow separation occurs which will be discussed in detail in Figure 4.61, contributing to rise the stochastic fluctuation levels of $\sim 2.5\%P_{01}$. From $80\%c_{r,ax}$, a significant increase of $\sigma(P_s)$ is detected also over the blade passage ($+1\%P_{01}$), as a result of the TLV.

Fig. 4.16(top) displays the pitch-averaged time-averaged difference in static pressure measured at six distinct peripheral locations along the vane pitch ($\delta(P_s)$). The circumferential pressure variation between different sensors at the same axial position allows to quantify the effect of the NGV potential field on the near-casing pressure distributions of the three profiles.

Overall, the vane clocking is shown to have a limited effect on the mean pressure field, within 5% of the inlet total pressure. The variations are seen to drop from the blade LE to the TE. The over-tip and over-channel averages are presented, also in this case, on the bottom of Figure 4.16. Although no significant variations are detected

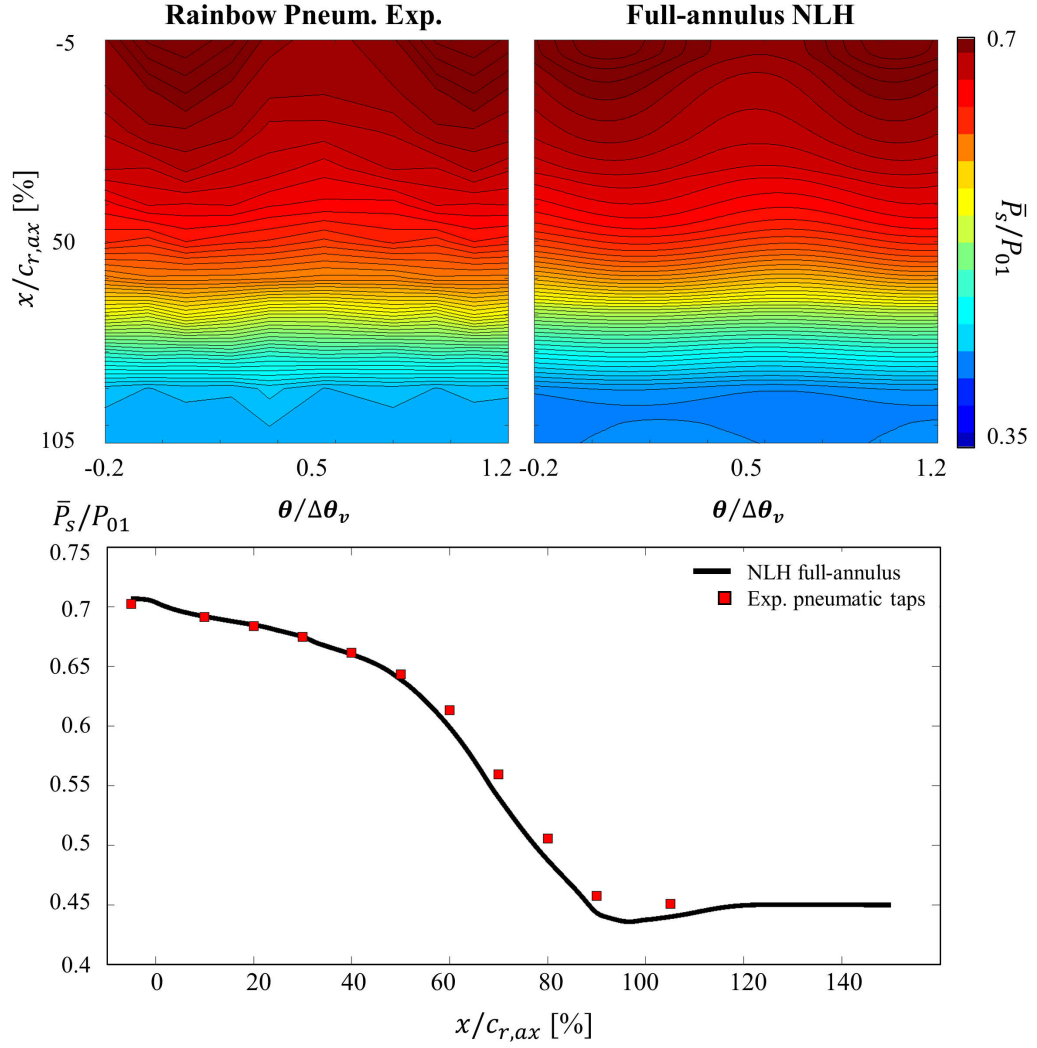


Figure 4.17: Rainbow-average time-average casing static pressure contours (top) and axial distribution (bottom): pneumatic taps measurements vs. full-annulus NLH results

between the axial trends in the two averaging regions, a higher variability is observed in the rear over-tip of the squealer-like profiles with respect to blade AC07T. This effect could be due to the relatively larger running tip clearance of blades IN01D and AC06D, which allows a certain degree of vane-modulation in the over-tip pressure field, while the flow chocking established in the gap of AC07T contributes to desensitize the leakage stream to the potential effects of the NGV-rotor interactions.

The rainbow-average static pressure contours measured by pneumatic taps are compared, in Figure 4.17, to the full-annulus NLH results averaged over a full rotor revolution. The same data-sets are then exploited to retrieve the pitch-average static pressure distribution presented over the full axial chord at the bottom of the same

figure. The agreement between experimental and numerical results is, in both cases, very good. The potential field of the vane, already discussed in Figure 4.12 for the three tips, is well captured by the unsteady simulations and presents a higher degree of similarity with the squealer-type profiles in terms of pitchwise and axial gradients intensity. The numerical predictions are shown to clearly match both the pressure field topology and the amplitude of the NGV-induced pressure variations.

The NLH dataset matches well the pitch-averaged measurements in the front part of the axial chord, while reaching a slightly lower level ($-1\% P_{01}$) in the near-TE region.

4.2.2 Rotor Casing Heat Transfer.

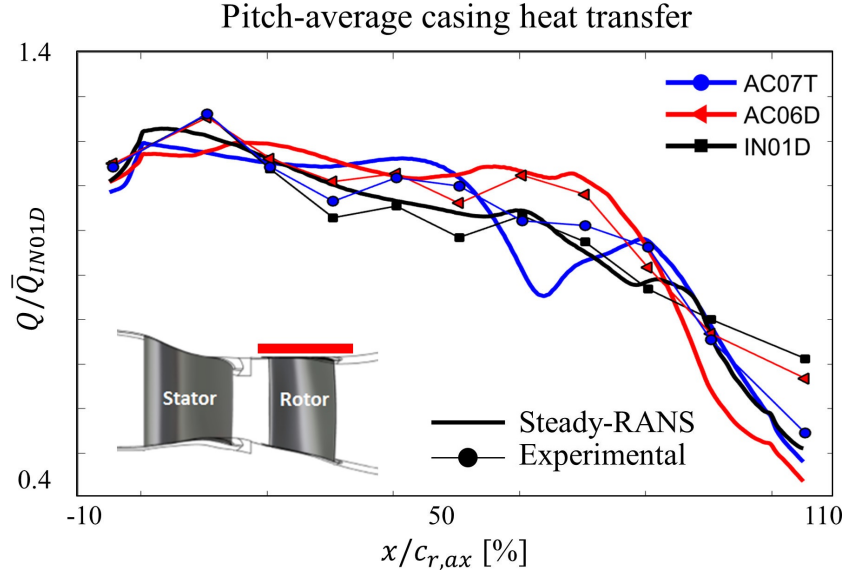


Figure 4.18: Sector-resolved pitch-averaged casing heat flux: measurements vs RANS

Measured and numerically predicted time-average heat transfer distributions on the rotor casing are reported in Figure 4.18, normalized by the averaged heat transfer measured over one vane pitch and the full axial chord of blade IN01D.

The casing heat transfer (Q) is highest in the early 20% of the rotor axial chord, about 10% higher than the value measured upstream of the rotor leading edge. Between 20% and 70% $c_{r,ax}$, Q gradually decrease by about 20% for the three tips. In the aft part of the rotor chord, the heat flux drops rapidly until reaching 60% of the inlet value downstream of the blade trailing edge.

The numerical predictions accurately model the heat transfer trends and the relative axial evolutions for the three geometries. A pronounced local minimum is for

blade AC07T at 60% of the axial chord, the origin of which will be further examined in Figure 4.44. Discrepancies of about 5-10% of Q_{IN01D} are registered between the measured and numerical distributions in the over-tip area, while a slightly higher difference is registered at the last measurement point, 5% $c_{r,ax}$ downstream of the rotor trailing edge.

The choice of the numerical turbulence model, as demonstrated by Paty [84], might be the driving cause of the partial discrepancies between predicted and measured mean heat transfer values up to the blade TE. A comparison between two-equation models (SST and $k-\epsilon$) and the Spalart-Allmaras model resulted in a change in casing heat transfer up to 13% of the average value among the different cases. None of the investigated turbulence models, however, resulted able to match the mean heat flux level at 105% $c_{r,ax}$ of the squealer-type profile (IN01D and, in this case, AC06D), generated by the combined effect of the unguided rotor outflow and of the tip leakage vortex (TLV).

The sector-resolved time average heat flux contours measured at the rotor casing are

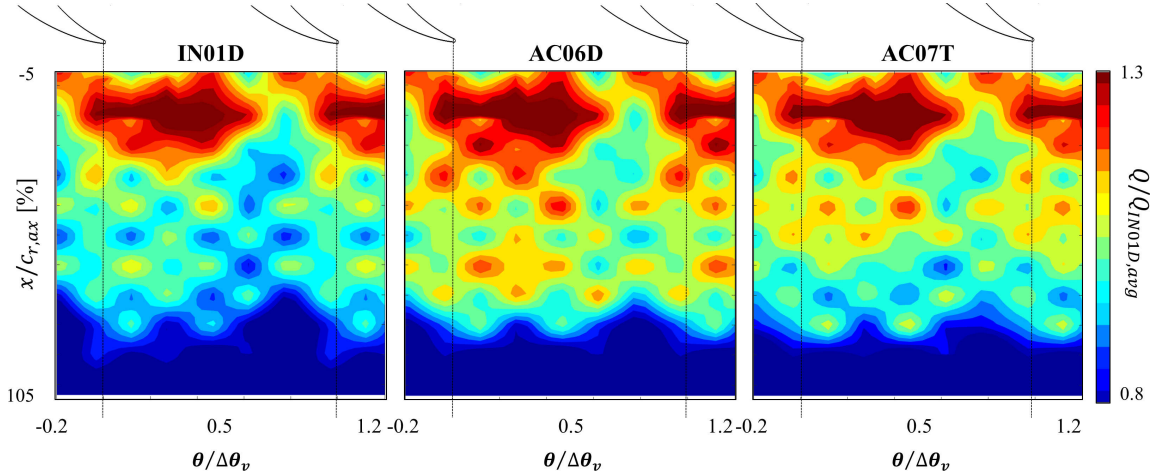


Figure 4.19: Sector-resolved time-average casing heat transfer

presented in Figure 4.19. A local region of high heat transfer is shown between -10% and 60% of the vane pitch for the three profiles. The axial and peripheral position of this high Q zone, which extends up to 30% of the rotor axial chord, correlate significantly well with the high static pressure regions visualized in Figure 4.13. The heat flux minimum corresponds to a the periodic low static pressure region indicating the peripheral position of the vane wake. Between 30 and 70% of the rotor axial chord, blade IN01D shows the lowest time-mean heat transfer, confirming the trend visualized in Fig.4.18, while AC06D show the highest heat flux levels at mid-chord.

The lower interpolation quality shown in these contours with respect to the pressure dataset (of equivalent spatial resolution) presented in Figure 4.13 is due to the aforementioned higher bias error of the heat transfer sensors (6% of the mean heat flux value).

As for the static pressure field, the contributions to the total casing heat transfer by

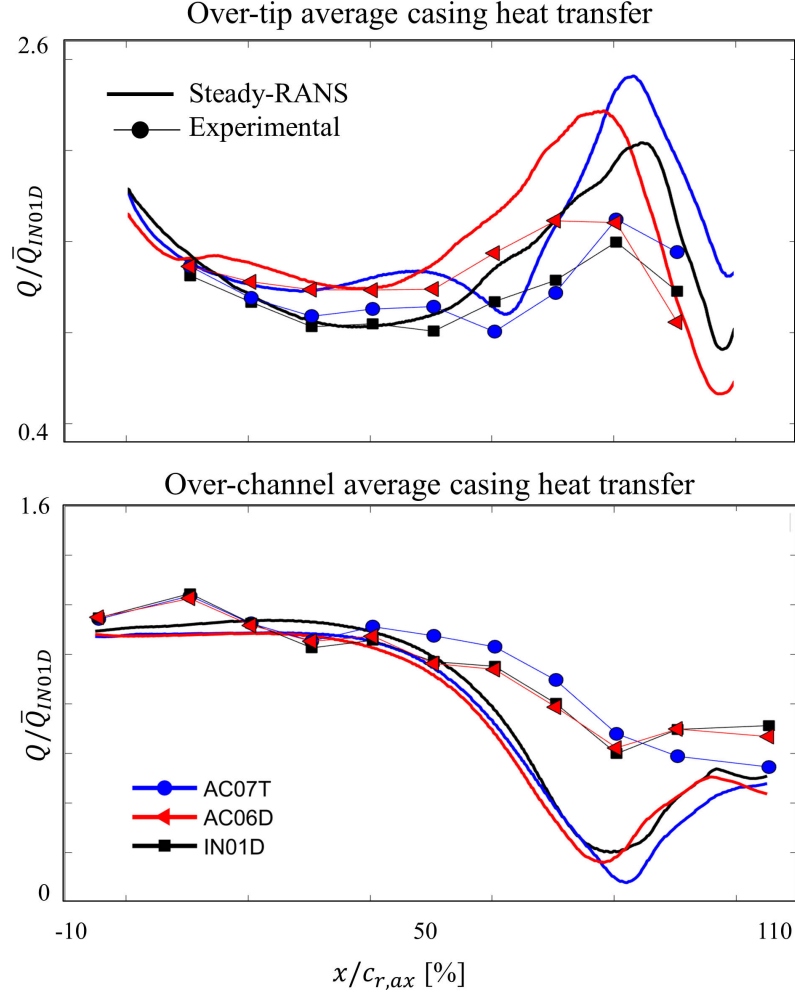


Figure 4.20: Sector-resolved pitch-averaged mean casing heat flux: over-passage and over-tip distributions

the passage and over-tip flows have been separated and are presented in Figure 4.20. The predicted passage-averaged heat transfer, shown in Fig. 4.20(bottom), remains nearly constant in the early 40% of the rotor axial chord, whereas the experimental data show larger variations (in the order of 10-15% Q_{IN01D}). The global minimum is registered at 80% $c_{r,ax}$ (50% heat flux drop) in the numerical predictions, with the experimental data confirming the position of the minimum showing a lower absolute Q decrease.

Overall, CFD predictions and experiments agree on that the three blade geometries have a similar evolution along the first 75% of the rotor axial chord. The squealer-type profiles show higher mean heat flux levels in the aft part of the blade ($+10\%Q_{IN01D}$) in reason of the stronger over-channel convection promoted by the TLV as demonstrated by the analysis performed in Figure 4.47.

The blade tip shape is observed to have a much larger impact on the over-tip Q , as shown in Figure 4.20(top). Between the rotor inlet and the rotor mid-axial chord, the heat transfer reduces gradually and remains below the rotor inlet levels for all the investigated geometries. In this region, the heat flux generated by the tip passage of blade AC06D is generally 10% higher than the ones of blades IN01D and AC07T. The close proximity to the casing of the multiple squealer rims of tip AC06D promotes large casing heat transfer rates, whereas the large squealer bath of tip IN01D and the smooth surface of blade AC07T mitigate the over-tip Q fluctuations.

At 50% of the rotor axial chord, the over-tip casing heat flux starts rising steeply up to $90\%c_{r,ax}$. The rates of increase are similar for the three tip geometries, but blades AC06D and AC07T reach higher absolute levels than the baseline squealer tip. The largest heat transfer generated by the tip traverse occurs between 70% and 80% of the rotor axial chord. In this area, both blade AC06D and AC07T reach values 30% higher than at the rotor inlet.

In the region close to the TE, Q drops drastically on the three blade tips. However, the decrease exhibited by tip AC06D is about 3 times larger than the Q drop in IN01D and AC07T. The numerical predictions of the heat transfer rate in the over-tip region show a significant mismatch with the experimental data. In the first half of the rotor axial chord, this difference is generally small, below 10% of the rotor inlet value. In the second half of the blade, Q is consistently over-predicted by CFD, with differences as high as 50% of the inlet value. At this location, the transit of the thin rear-blade tips in close proximity to the casing wall generate a localized heat transfer surge only partly captured by the thin-film sensors with limited frequency bandwidth and spatial resolution [96].

Figure 4.21(top) displays the pitch-averaged time-averaged random unsteadiness ($\sigma(Q)$) measured over the three tips. The stochastic fluctuation contribution is, in average, way higher than in the static pressure field. Up to $70\%c_{r,ax}$, $\sigma(Q)$ is uniform for the three profiles at about 40-45% of Q_{IN01D} , with a moderate increase in the squealer-type tips in the last part of the rotor chord ($+5\%Q_{IN01D}$). A steeper rise is shown by blade AC07T at $70\%c_{r,ax}$, with a global peak of stochastic fluctuation at $90\% c_{r,ax}$ amounting to $55\% Q_{IN01D}$.

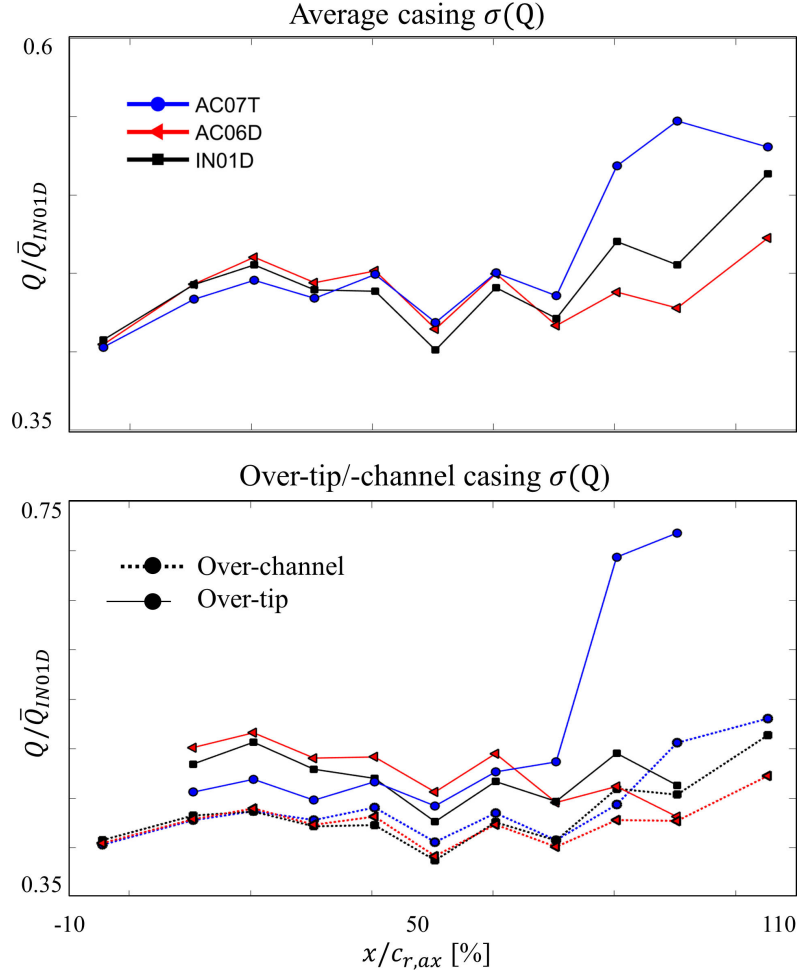


Figure 4.21: Sector-resolved pitch-averaged casing heat flux: stochastic component

To clarify the origin of this discrepancy, Figure 4.21(bottom) presents the over-passage (dotted lines) and over-tip (solid line) distributions of $\sigma(Q)$. The two squealer blades present a constant over-tip $\sigma(Q)$ over the rotor chord and a slightly increasing trend localized in the aft part of the over-passage region. Blade AC07T shows a slightly higher increase of average heat transfer fluctuation in the aft part of the channel and an abrupt increase of over-tip $\sigma(Q)$ at $70\%c_{r,ax}$ ($+25\%Q_{IN01D}$). This trend is due to the aft-tip flow separation highlighted for this profile in Figure 4.43 and which will be analyzed in detail in the unsteady flow analysis in Sections 4.4.3 and 4.5.2.

4.2.3 Stage Outlet.

Measured and numerically predicted time-average pitch-average distributions of outlet total pressure are presented in Figure 4.22. The experimental sector resolved data are recorded by means of fast response pressure rakes (circles) and virtual-three-hole (V3H) probes (triangles). The numerical dataset is generated by single-tip NLH computations (solid lines), in which the mean total pressure field produced over one rotor revolution was averaged over a full vane pitch to mimic the experimental results, and RANS performed in the rotor domain (dotted lines).

A first comparison between the predicted and measured pressure profiles shows a very good agreement in terms of tip-to-tip flow differences. The mean total pressure level measured by the V3H probe matches closely the CFD-predicted values for the three tips, while a negative offset of $\sim 15\text{mbar}$ is visualized in the pressure profiles recorded by the FRAPP rakes. This pressure bias is due to the effect of the abrupt temperature variation of the bare sensors of the FRAPP exposed to the hot blow-down flow, not fully compensated by the transient in-situ calibration realized in both pressure and temperature. The bias is not present in the encapsulated sensors of the V3H and, however, does not affect the precision of the measurement (tip-to-tip flow differences are well captured).

In the near-casing region, the presence of the tip leakage flow produces a total pressure peak in the three profiles (95% of the rotor span) which is maximized by the increased clearance of blade AC06D ($+ \sim 20\text{mbar}$ with respect to the other tips). The subsequent pressure minimum, between 85% and 90% h_r is particularly evident for the squealer-type profiles, which show a higher pressure fluctuation generated by the TLV, as demonstrated in the time-resolved results (Section 4.4.4, Figure 4.49). The CFD data, in this case, tend to overestimate the pressure minimum of blade AC07T, which in the measurements appears less significant than in the other blades. The FRAPP rakes data tend to slightly underestimate the amplitude of the near-casing radial pressure variation (-5mbar for all the tips) with respect to the virtual-three-hole probe which, conversely, validates well the CFD predictions. This difference might be due to the lower radial resolution of the bare-sensor FRAPP rakes, which averages the flow gradients over the full sensor diameter (1.7 mm) with respect to the miniaturized pressure taps of the V3H (0.2 mm diameter). At about 70% of the rotor span, a pressure variation of lower intensity is generated by the presence of the UPV.

In the lower 50% of the rotor span, a pressure variation of 10mbar , probably related to the presence of the LPV, is predicted by the numerical computations ($+5\text{ mbar}$

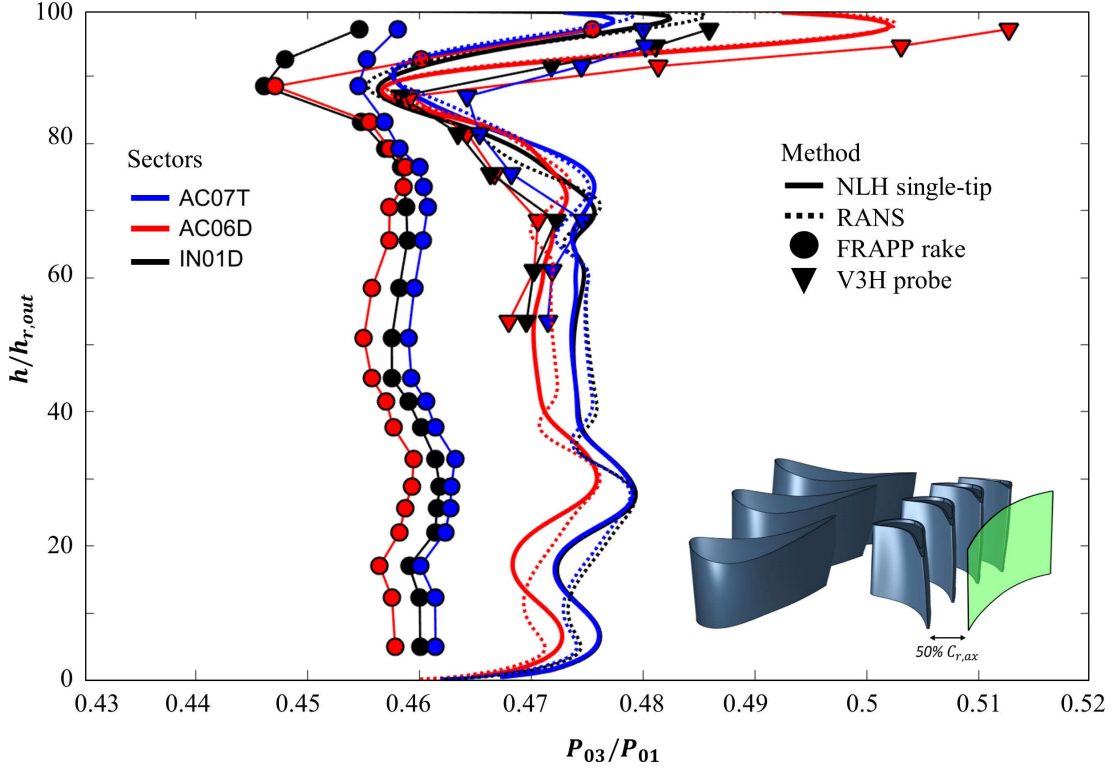


Figure 4.22: Sector-resolved time-average total pressure distribution at stage outlet

at 25% h_r , -5 mbar at 15% h_r with respect to the mean value), slightly sharper in the NLH dataset. The experimental data show a lower intensity pressure variation ($\sim \pm 2.5 \text{ mbar}$), which better correlates in radial position with the NLH predictions. This better matching with the full-stage unsteady flow predictions demonstrates a major impact of the blade-rows interactions at this radial location.

The influence of the tip profile is predicted and measured to be significant also in the lower 50% of the rotor span. While the shape of the time-mean distributions is mainly affected by the tip design in the upper 30% h_r , a residual pressure difference of $\sim 3 \text{ mbar}$ between blade AC06D and blade AC07T persists down to the inner endwall, in reason of the difference in mass flow experienced by this profile. The discrepancy between blade AC07T and IN01D, on the other hand, is virtually negligible in the numerical predictions and within $\sim 1 \text{ mbar}$ in the experimental data.

Rainbow-average outlet total pressure profiles are presented in Figure 4.23. Steady experimental data, recorded by pneumatic total pressure rakes (Kiel heads) and five-hole probes, are compared to time-mean measurements by FRAPP rakes and V3H probes averaged over the full rotor revolution. The numerical dataset, shown by means of a solid line, is produced by applying the equivalent rainbow-averaging procedure on the time- and space-resolved outlet pressure field predicted by full-annulus

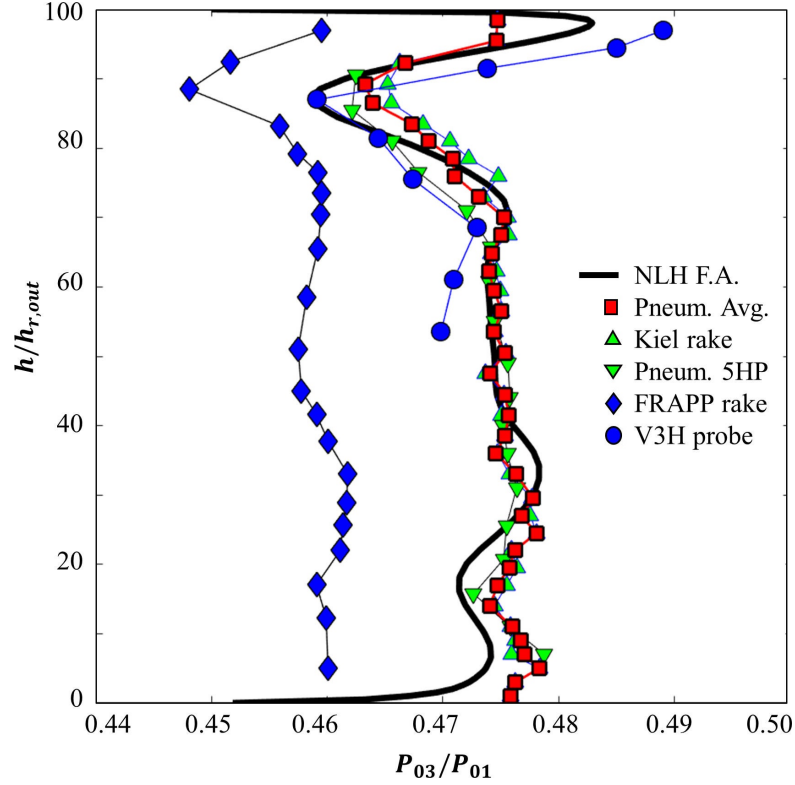


Figure 4.23: Rainbow-average time-average total pressure distribution at stage outlet

NLH computations.

A high degree of agreement is shown between numerical and experimental datasets, with the exception of the calibration bias of the FRAPP rake. A significant qualitative and quantitative validation of the full-annulus CFD is provided by the pneumatic measurements. The absolute pressure level and the radial gradients commented in detail in Figure 4.22 are well captured by the numerical predictions, with only few discrepancies.

The position of the LPV pressure gradient ($\sim 20\%h_r$) appears slightly shifted upwards ($\sim 4\%h_r$) in the CFD with respect to the pneumatic measurements, while being partially validated by the FRAPP measurements. This discrepancy between the pneumatic and fast-response measurements could be due to a slight axial misalignment between the probe heads, with the measurement point located slightly upstream ($\sim 1.5 - 2$ mm) in the case of the 5HP. The radial discrepancy is, indeed, almost negligible in the case of the total pressure Kiel rake.

The pressure deficit at $85 - 90\%h_r$ appears less pronounced in the pneumatic and FRAPP rakes measurements. The V3H data, however validate the CFD, thanks to the increased spatial resolution offered by the 0.2 mm diameter pressure tap.

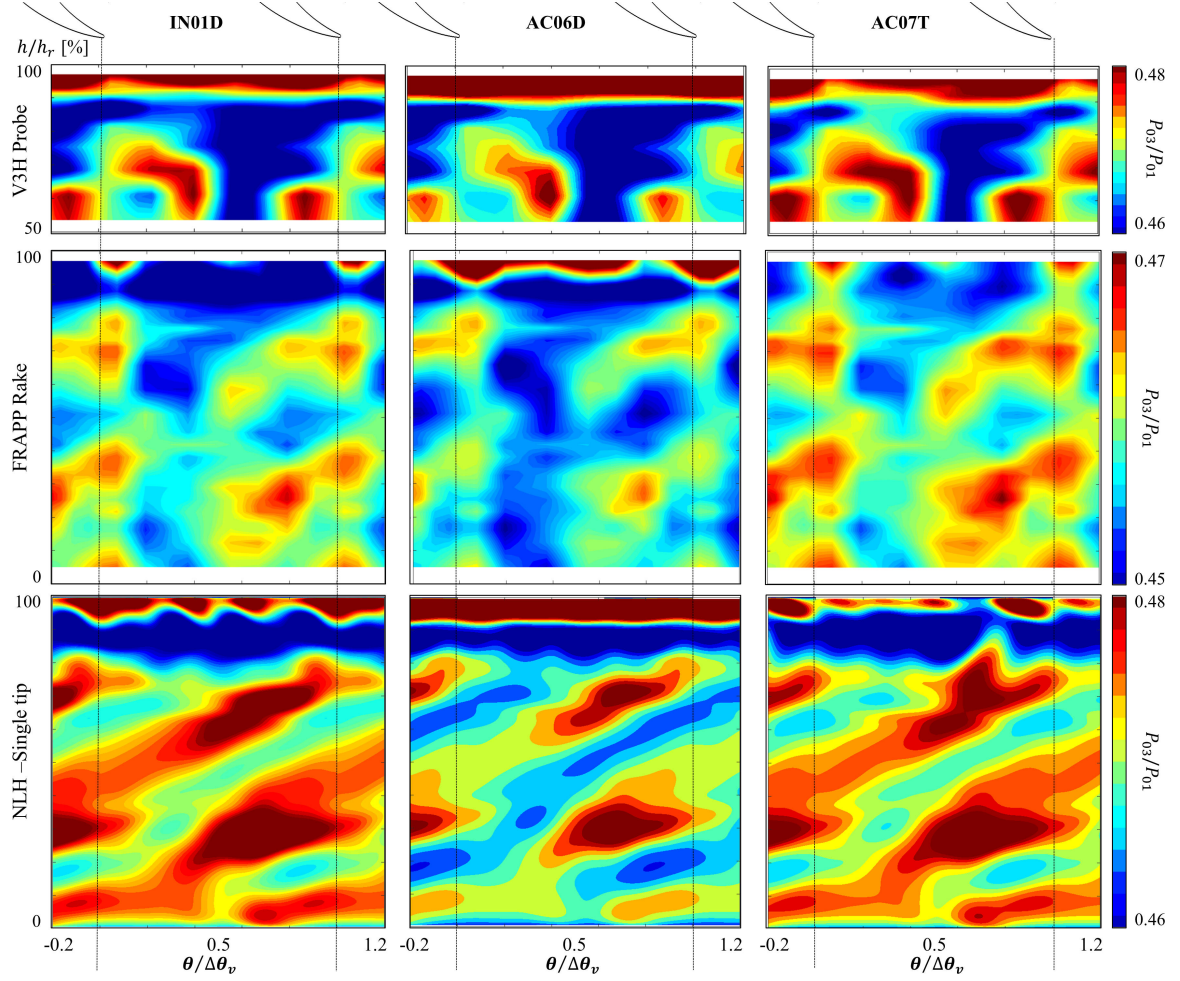


Figure 4.24: Sector-resolved time-average total pressure contours at stage outlet

Time-average outlet total pressure contours are presented in the stationary domain in Figure 4.23: experimental measurements by V3H probes (top) and FRAPP rakes (center) are compared with single-tip NLH computations (bottom). The V3H and CFD dataset are presented with consistent colormap ranges, while the range of the FRAPP contour is modified to compensate for the measurement bias highlighted of this probe. The horizontal axis represents the peripheral position with respect to the NGV trailing edge ($\theta = 0$) normalized by the vane pitch, while the vertical axis extends over the full blade span.

The near-casing region is characterized by high pressure levels, maximized in the case of AC06D due to the high clearance value. The improved spatial resolution offered by the V3H probe allows to capture precisely the near-casing pressure pattern, validating the steep radial pressure gradients predicted by the CFD, while the FRAPP rake shows significant limitations in the near-endwall area. A subsequent horizontal

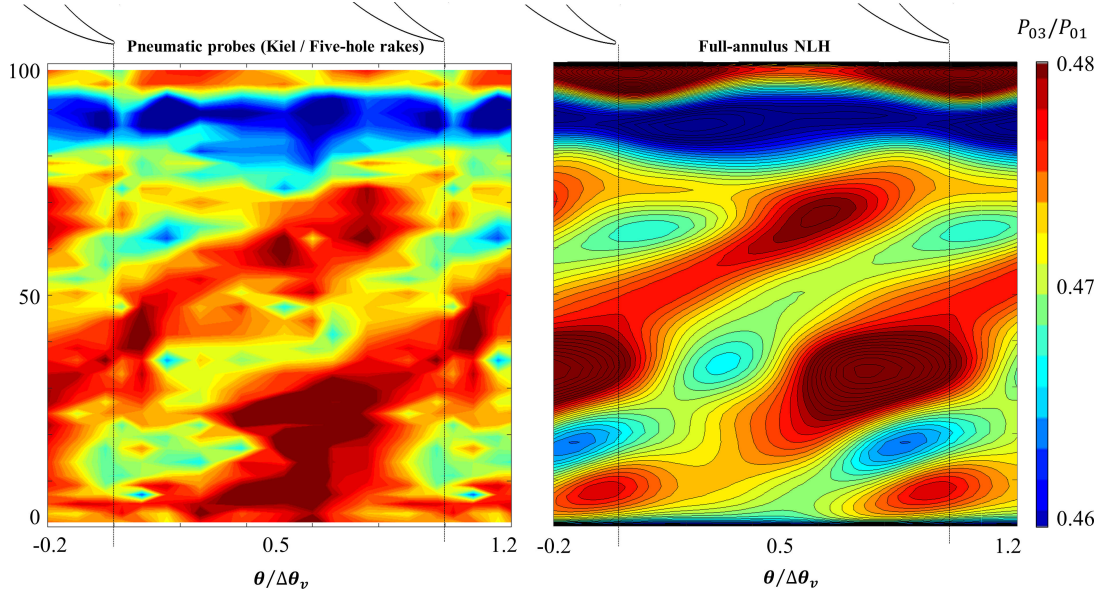


Figure 4.25: Rainbow-average time-average total pressure contours at stage outlet: pneumatic probes vs. full-annulus NLH

stripe of low pressure is visualized between 85% and 90% of the blade span. The measured pressure deficit is, in both cases, less severe and uniform in blade AC07T in partial discrepancy with the CFD results, which show equal pressure minima within the three profiles.

The presence of the vane wake signatures is highlighted by the periodic alternation of inclined low and high pressure regions which cross the investigation region between 20% and 80% of the rotor span. A higher level of mixing is detected, however, in the experimental data, with a measured peripheral variation of $\sim \pm 5 \text{ mbar}$ at constant radial position against a predicted value of $\sim \pm 9 - 10 \text{ mbar}$. In agreement with Figure 4.22, blade AC06D and AC07T show, respectively, the lower and higher mean pressure levels.

The rainbow-average time-average outlet total pressure contours are shown in Figure 4.25. Pneumatic measurements performed by Kiel rakes and five-hole probes are here combined to obtain the experimental contour (left), while the numerical distribution is obtained from the full-annulus NLH computations (right).

The two datasets are here presented with the same colormap range and the agreement in terms of mean pressure levels and flow topology is remarkable. The increased spatial resolution obtained by combining the two measurement probes allows to identify with enhanced accuracy the position of the vane wake signature, which matches significantly well the numerical predictions. The periodic change in radial extent of

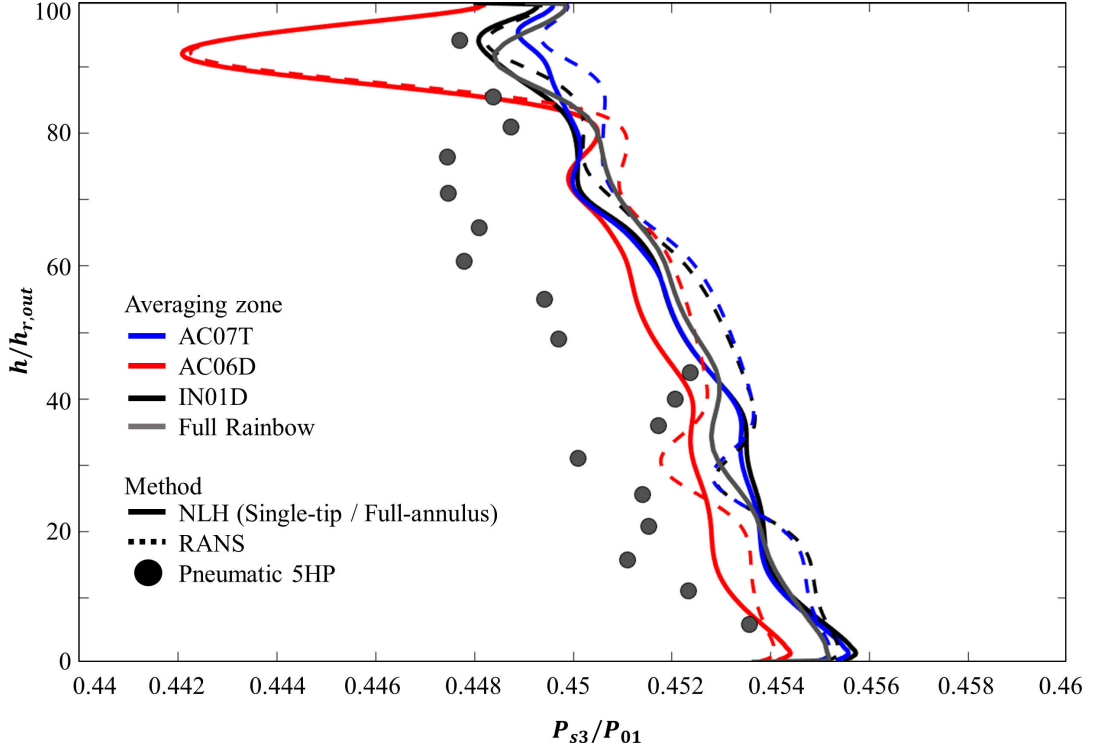


Figure 4.26: Sector- and rainbow-average time-average static pressure distribution at stage outlet

the pressure deficit in the near-casing area, confirmed by the experimental results, suggests an influence of the vane-relative position on the intensity and extension of the TLV. The predicted and measured inclination and phasing of the vane-induced peripheral and radial pressure gradients appear in good agreement. A slightly higher level of mixing is still detected, however, in the experimental results, with circumferential pressure variations of $\sim \pm 0.7 - 0.8\% P_{01}$. The presence of low total pressure patches centered at 0.8% of the vane pitch at 15% of the blade span, validated by the experimental contours, suggests a high degree of vane modulation in that area, which corresponds radially to the position of the lower passage vortex.

Measured and predicted time-average distributions of outlet static pressure are presented in Figure 4.26. The rainbow-mean experimental dataset (grey dots), recorded by the pneumatic five-hole-probe, is compared here to the rainbow-average distribution predicted by full-annulus CFD (gray solid line). Additionally, tip-resolved distributions produced by single-tip NLH and steady-RANS computations are presented in order to quantify the impact of the tip design on the outlet static pressure field, which effectively sets the operational pressure ratio of the HPT stage.

Globally, the predicted trend of the rainbow-average static pressure profile is con-

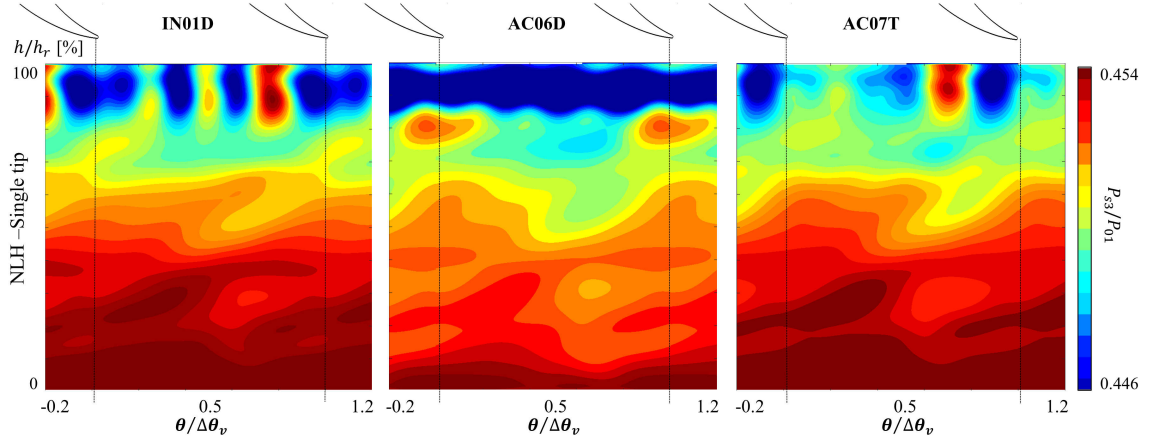


Figure 4.27: Sector-resolved time-average static pressure contours at stage outlet (single-tip NLH)

firmed by the experimental dataset, which however shows a negative offset ranging between 3-4 mbar along the blade span. The numerical predictions agree in ascribing a generally lower mean outlet static pressure to blade AC06D (-1.5mbar) with respect to IN01D and AC07T, with this discrepancy maximized in the TLV region (90% blade span). Here blade AC06D shows the largest pressure deficit among the three profiles, dropping of 9 mbar with respect to pressure level at 80% h_r (-7 mbar with respect to IN01D and AC07T). The squealer and contoured tips, conversely, show a TLV pressure drop in line with the rainbow-average profile (-2 mbar).

The agreement between NLH and RANS single-tip predictions is globally good, with minor discrepancies in the radial position of the pressure gradients in the lower 50% of the span. The RANS predictions show a more pronounced pressure minimum between 20% and 25% h_r , corresponding to the position of the LPV. In the near-tip area, the steady and unsteady predictions agree on the main tip-to-tip differences in terms of location and amplitude of the pressure minima. The full-stage NLH shows, globally, a higher degree of agreement with the single-tip NLH pressure profile of blade IN01D.

To gain a deeper insight into the tip-to-tip differences in the time-mean outlet static pressure field, single-tip NLH results are presented in Figure 4.27. In the lower 50% of the blade span, the pressure contours of blades IN01D and AC07T show negligible qualitative and quantitative differences. Blade AC06D shows a globally lower pressure level and peripheral pressure gradients which result slightly out of phase with respect to the other blade tips at mid-span. In the near-casing region, blade AC06D shows high static pressure patches appearing periodically at 85% h_r , showing a possible influence of the vane-relative position on the intensity of the UPV, while lower

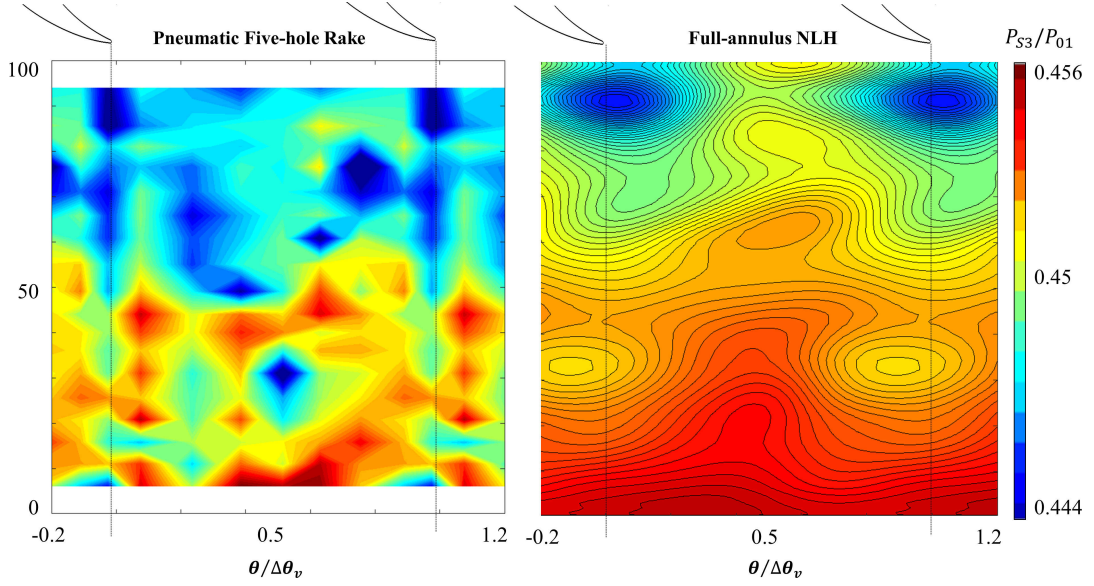


Figure 4.28: Rainbow-average time-average static pressure contours at stage outlet: pneumatic five-hole probe (left) vs. full-annulus NLH (right)

peripheral variations are seen in the other blades. A uniform region of low static pressure is visualized for tip AC06D above 90% of the rotor span, while a higher degree of vane-induced pressure modulation is shown by blades AC07T and IN01D. The measured and predicted rainbow-average static pressure contours are presented in Figure 4.28, where the experimental dataset of the pneumatic five-hole probe is compared to full-annulus NLH predictions. In reason of the low range of static pressure variations, which never exceeds 5 mbar at a given radial position, the resolution of the experimental contours appears particularly scattered. Quantitatively, however, an acceptable global match is shown between measured and predicted static pressure levels, which decrease with increasing radial coordinate. From the flow topology standpoint, the only recognizable feature is a low static pressure patch aligned with the NGV trailing edge ($\theta = 0$) in the near-casing region.

Figure 4.29a presents the sector-resolved absolute Mach number profiles at the stage outlet, with V3H probe measurements compared to single-tip NLH and RANS computations. As expected, both experimental and numerical data indicate that the near-casing Mach number peak of blade AC06D exceeds the maxima reached by blades IN01D and AC07T (by 0.08 and 0.11 respectively, according the CFD predictions). The subsequent minimum, located between 85 and 90% of the rotor span, is slightly underestimated by the RANS simulations with respect to the single-tip NLH, which are more in agreement with the experimental dataset. Below 70% of the blade span, AC06D shows a globally lower Mach number (-0.02) in comparison to blades

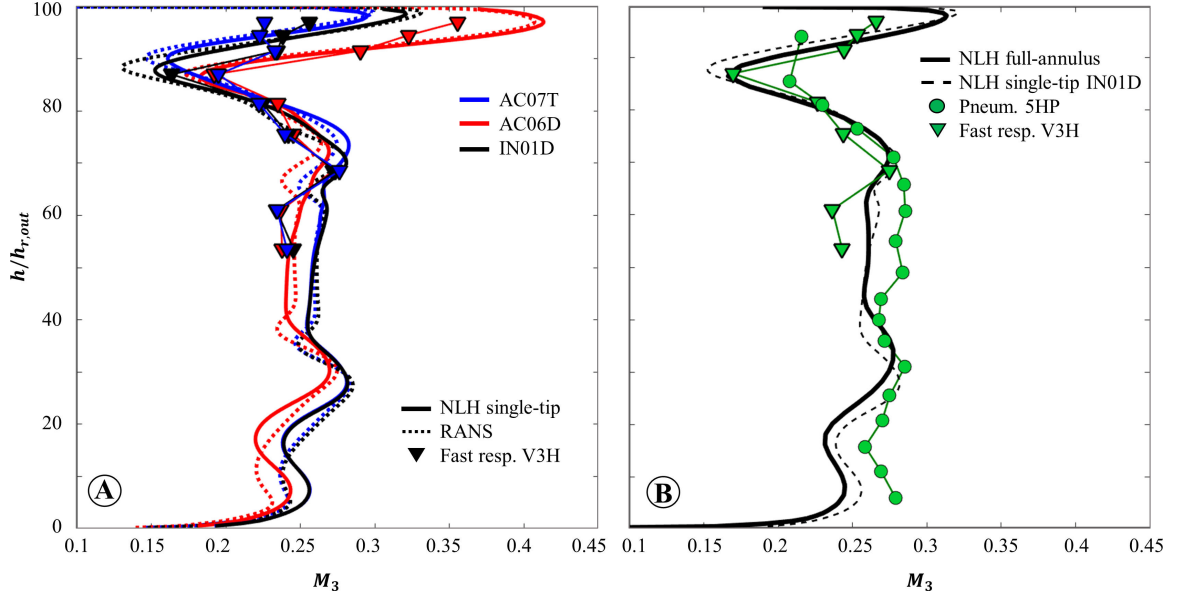


Figure 4.29: Sector- and rainbow-average time-average absolute Mach number distribution at stage outlet

IN01D and AC07T, which present no substantial difference below the location of the UPV.

The rainbow-average profiles of absolute Mach number measured at the rotor outlet by the pneumatic 5HP and the fast-response V3H probe are reported in Figure 4.29b. Additionally, the Mach distribution of blade IN01D computed by single-tip NLH is presented for reference

A higher Mach number is measured Mach in the lower 70% of the rotor span (+0.01 to +0.03) with respect to the NLH distributions. The two numerical datasets show a high degree of agreement, confirming that blade IN01D is globally representative of the rainbow-mean outlet flow field. The Mach minimum at 90% h_r is captured very well by the V3H dataset, while the pneumatic 5HP is affected by the lower spatial resolution in measuring this steep gradient.

The measured and predicted rainbow-average contours of outlet absolute Mach number are presented in Figure 4.30, where the experimental dataset of the pneumatic five-hole probe is compared to full-annulus NLH predictions. As for the time-average total pressure contours in Fig. 4.25, the topology of the vane wake signature, represented in this case by an inclined low-Mach number region, is validated by the experimental dataset. The predicted amplitude of the Mach number variation, for a given radial position, reaches a maximum of ~ 0.15 , recorded at 40% h_r , against a maximum experimental variation of 0.1.

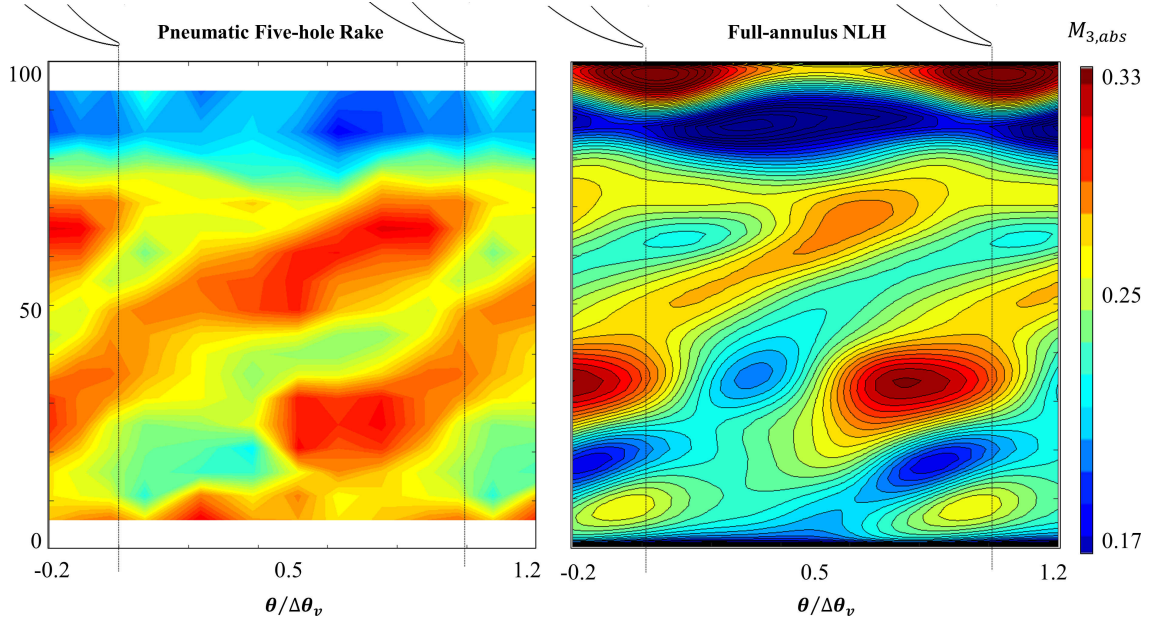


Figure 4.30: Rainbow-average time-average absolute Mach number contours at stage outlet: pneumatic five-hole probe (left) vs. full-annulus NLH (right)

Radial distributions of absolute yaw angle are presented in Figure 4.31a for the three rotor tips. Experimental profiles, recorded by the V3H probe, are compared to pitch-average profiles produced by single-tip NLH (solid lines) and steady-RANS (dashed lines) computations.

The under-turning produced by the tip leakage flow generates a maximum measured yaw angle of 10° (peripheral flow angle with respect to machine axis) in blades IN01D and AC06D at 90% of the rotor chord. A lower under-turning is produced, conversely, by blade AC07T (-15° with respect to squealer-type profiles), in agreement with the lower intensity TLV discussed in the previous sections. The NLH results slightly overestimate the peak amplitude ($+5^\circ$) at this location, showing however a better matching than the steady-RANS predictions ($+10^\circ$), and managing to capture with good accuracy the tip-to-tip differences in the near-casing region. The same offset value is detected at 70% of the rotor blade span, where a milder flow angle gradient is measured in the near-UPV region in comparison to the predicted values. In the lower half of the blade span, significant discrepancies are shown between the steady and unsteady flow predictions, with the former over-predicting the amplitude of the absolute angle variation registered between 30% and 10% of the blade span.

The time-average contours of yaw angle shown in Figure 4.32 reveal how the near-casing under-turning is nearly constant along the vane pitch for the squealer type tips, while resulting more discontinuous and of lower intensity in blade AC07T. Ad-

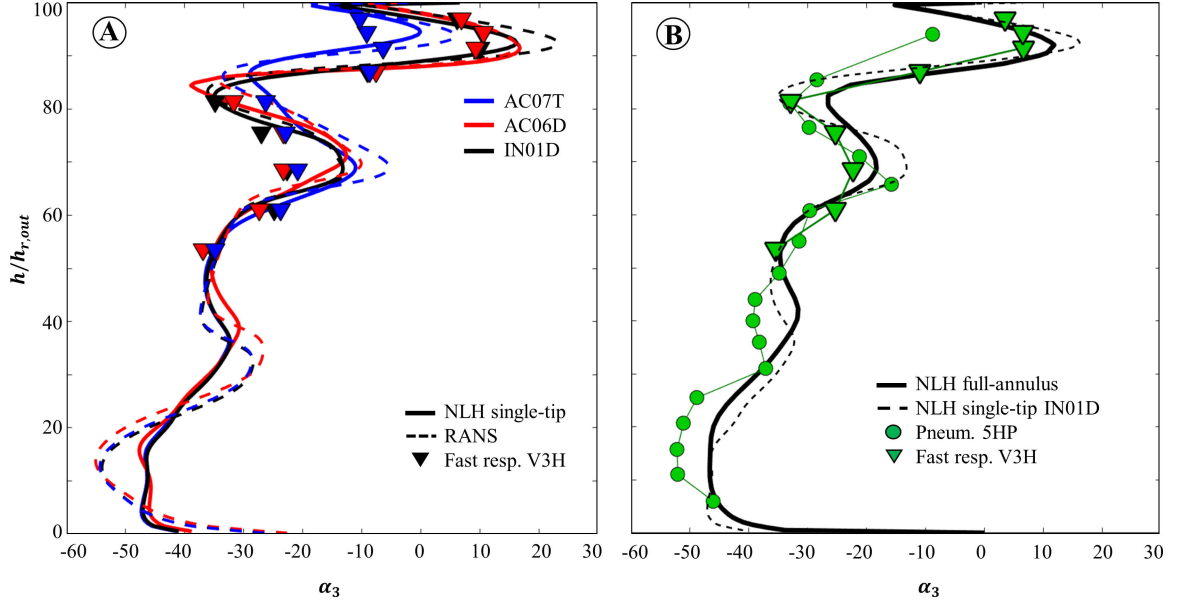


Figure 4.31: Sector- and rainbow-average time-average yaw flow angle distribution at stage outlet

ditionally, periodic yaw angle peaks are detected at 70% of the rotor blade span, with a good matching between experimental and numerical results.

Figure 4.31b shows the rainbow-average flow angle distributions measured by the V3H probe (upper 50% of the span) and pneumatic 5HP against the full-annulus NLH result. The experimental results validate the numerical predictions in the upper 50% of the rotor span, with the only major discrepancy detected in the near-casing measurement point of the 5HP, probably due to spatial resolution limitations. In the lower half of the rotor span, the measured flow angles present, in average, a negative offset of $\sim 5 - 7^\circ$ with respect to the numerical predictions. The corresponding rainbow-average flow angle contours are presented in Figure 4.33. The experimental data recorded by the pneumatic 5HP are presented on the left-hand side of the figure against full-annulus NLH predictions on the right-hand side.

The rainbow-average profiles of measured (pneumatic 5HP) and predicted (full-annulus NLH) pitch flow angle at the stage outlet are presented in Figure 4.34. The radial flow angle is shown to range between -10° near the inner endwall up to 10° at the casing. Experimental and numerical results globally agree in ascribing a quasi-linear trend to the mean pitch angle value with the radial coordinate, with the mid-span region corresponding to the near-zero angle condition. A closer look at the time average contours presented in Figure 4.35, however, reveals that significant peripheral gradients are present in this region, which do not allow to consider the mid-span flow

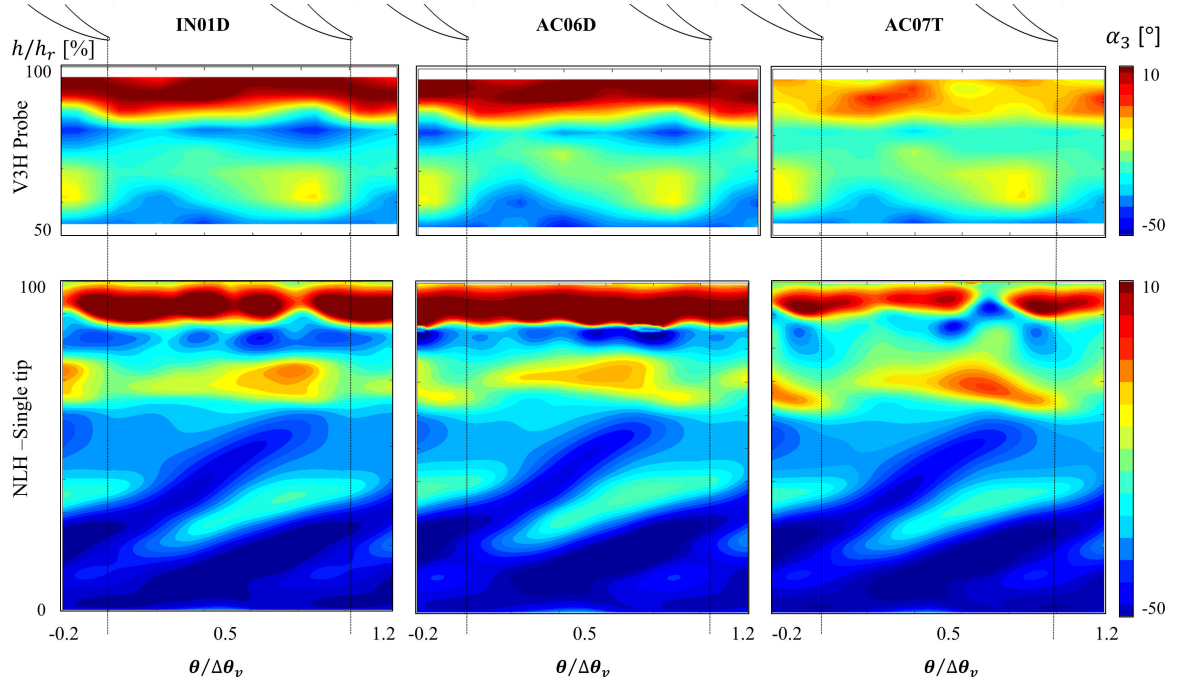


Figure 4.32: Sector-resolved time-average yaw flow angle contours at stage outlet

as completely 2D. The angle variations are, in general, over-predicted by the numerical simulations, which underestimate the level of mixing at the stage outlet.

The local deviations of the experimental profiles from the predicted ones ($\sim +2^\circ$ at 10% and 75% of the rotor span) are justified, in the time-average contours, by a lower downturning measured in the near-LPV region and in the vane-wake signature. The presence of the tip leakage flow in the near-casing region is identified by a measured local minimum followed by a flow angle peak at 85% h_r which correlates well with the numerical predictions.

Figure 4.34 additionally reports the tip-resolved profiles of average pitch angle computed by RANS simulations. The profiles of blades IN01D and AC06D globally agree with the full-annulus predictions. Consistently with what commented in Figure 4.32, however, the lower 40% of the span is home to a partial discrepancy between RANS and NLH computations. A higher flow turning ($+3^\circ$) is predicted by the steady simulations in this region. This discrepancy seems to indicate a stronger vane-dependent structure in the RANS predictions, which could be further mitigated by the vane-rotor interactions in the unsteady CFD. The experimental results, however, do not seem to fully confirm either of the two predictions, matching closely the RANS profile in the near-hub region and correlating better with the full-annulus NLH between 20% and 40% h_r .

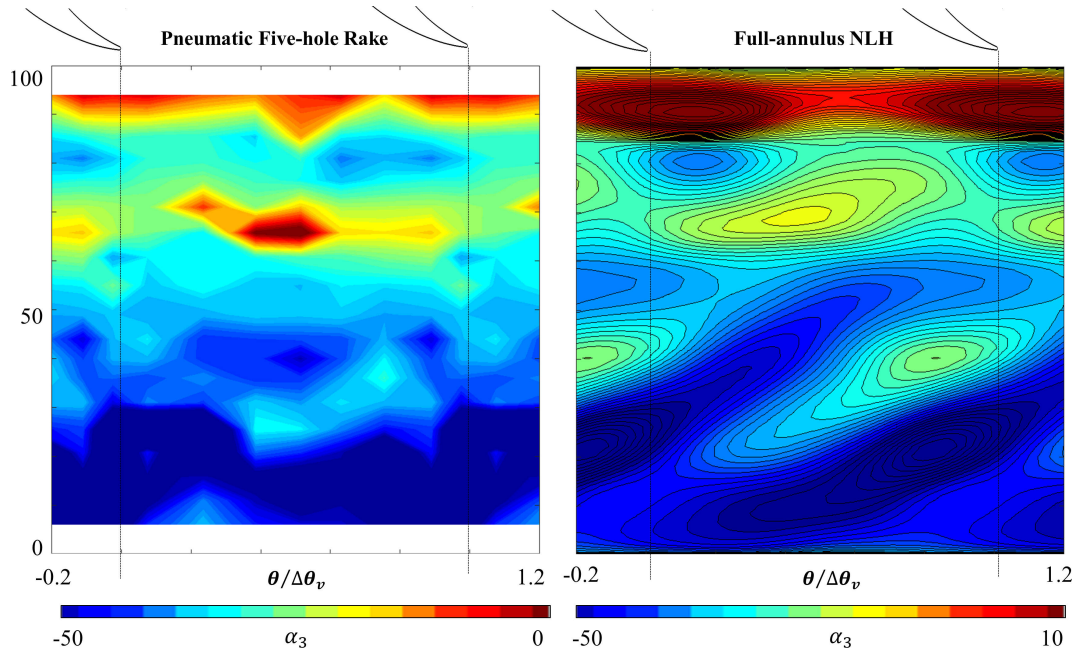


Figure 4.33: Rainbow-average time-average yaw flow angle contours at stage outlet: pneumatic five-hole probe (left) vs. full-annulus NLH (right)

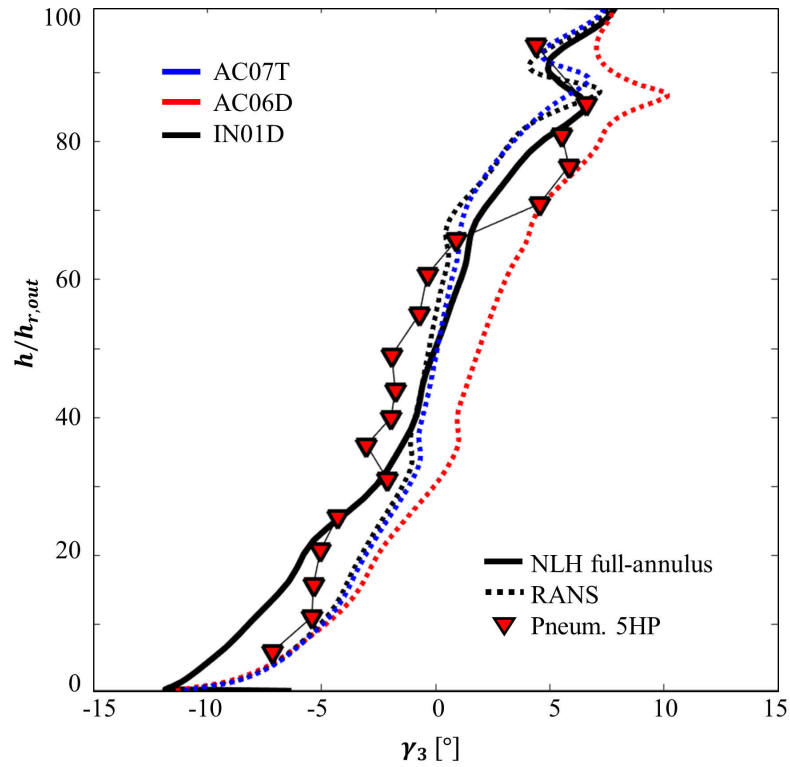


Figure 4.34: Sector- and rainbow-average time-average pitch flow angle distribution at stage outlet

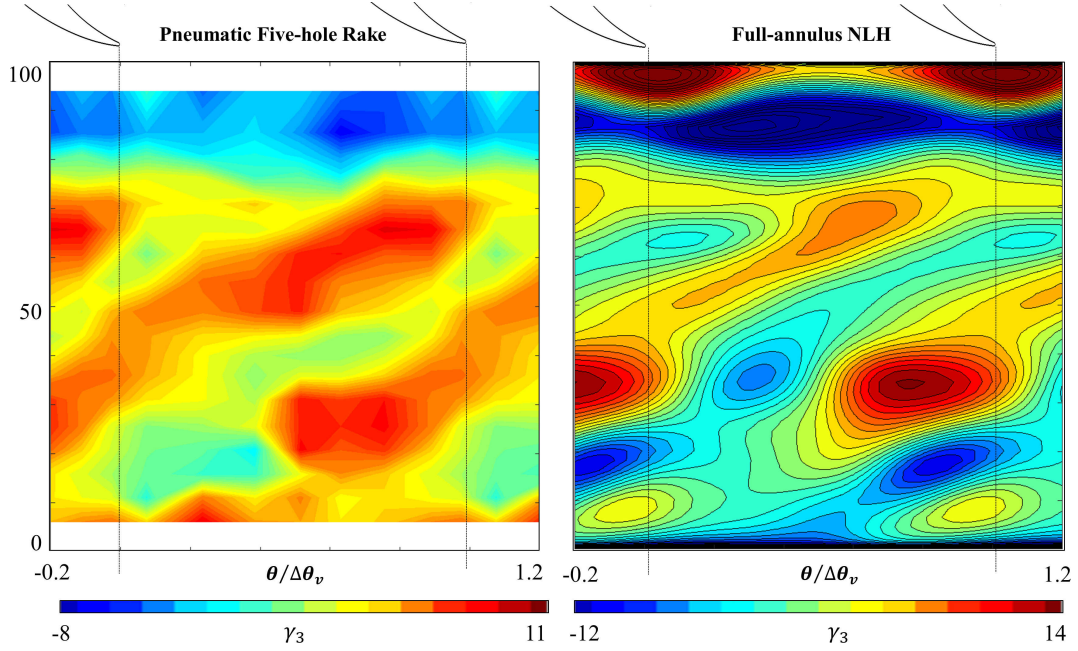


Figure 4.35: Rainbow-average time-average pitch flow angle contours at stage outlet: pneumatic five-hole probe (left) vs. full-annulus NLH (right)

Figure 4.36 reports the pitchwise-average total temperature distributions measured by K-type thermocouple rakes against the predicted temperature profiles by the iso-thermal steady RANS (solid lines) and adiabatic NLH predictions (dashed lines). In general, the experimental results show a higher mean value of total temperature with respect to the RANS predictions, with an offset ranging from 8° in the mid-span region to 15° near the casing endwall. The experimental profile is not uniformly measured over the full extent of the blade span because of manufacturing constraints of the thermocouple rake.

A higher level of agreement is shown between the experimental dataset and the NLH-predicted mean temperature value. This difference in the numerical results is due to the adiabatic wall conditions imposed in the unsteady flow predictions, compared to the iso-thermal wall conditions (300 K) imposed in the RANS computations.

The global trend and tip-to-tip differences seem, however, consistent between the steady and unsteady CFD results. For the experimental dataset, slightly more uniform along the blade span, the test-to-test variability was computed to be $\pm 5 - 6^\circ$.

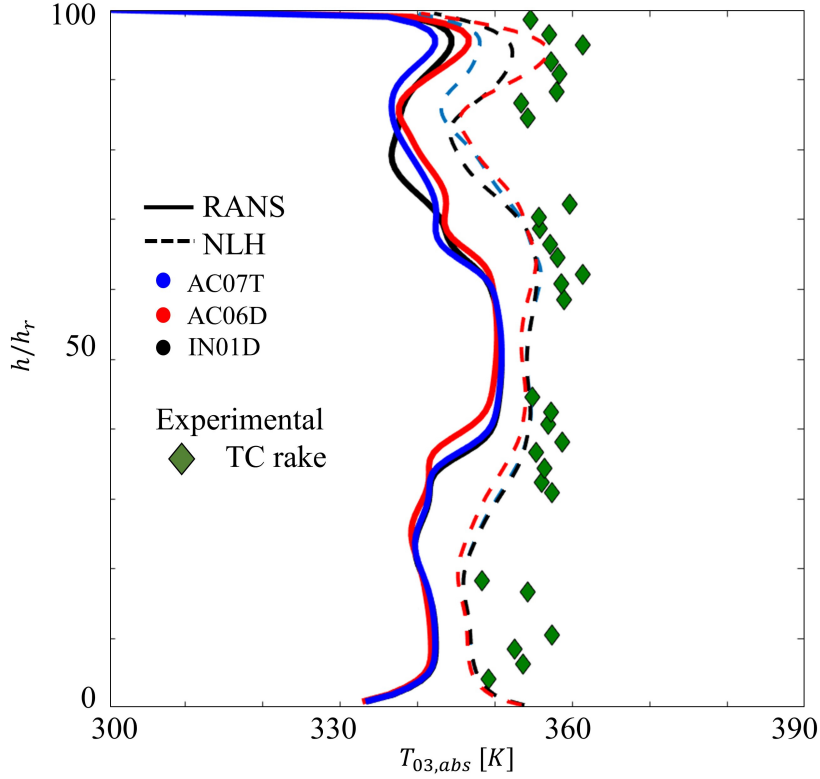


Figure 4.36: Time-average total temperature distribution at stage outlet

4.3 Unsteady rotor flow field

4.3.1 Rotor inlet

The unsteady pressure fluctuations produced by the rotor airfoils in Plane 02 are presented Figure 4.37 for the rotor casing (a) and hub endwall (b). The experimental dataset, obtained by phase-averaging measurements by fast response pressure sensors over a full vane pitch, is represented by solid lines for the three profiles. The numerical predictions, obtained from single-tip NLH computations, are represented by dotted lines. Both datasets are shown averaged in the rotor domain to isolate the rotor-induced pressure fluctuations from the potential field of the upstream vane. The horizontal axis indicates the rotor phase with reference to the airfoil leading edge ($\varphi = 0$) and with the blades traveling from right to left in the plot.

As expected, the inner endwall presents a lower mean static pressure for all the three sectors with respect to the casing distributions ($-4\%P_{01}$). In terms of AC component, an acceptable match is shown between numerical and experimental datasets. The rotor-induced pressure fluctuation results in phase between the hub and casing but the peak-to-peak amplitude is higher at the outer endwall ($3\%P_{01} \sim 30\text{mbar}$

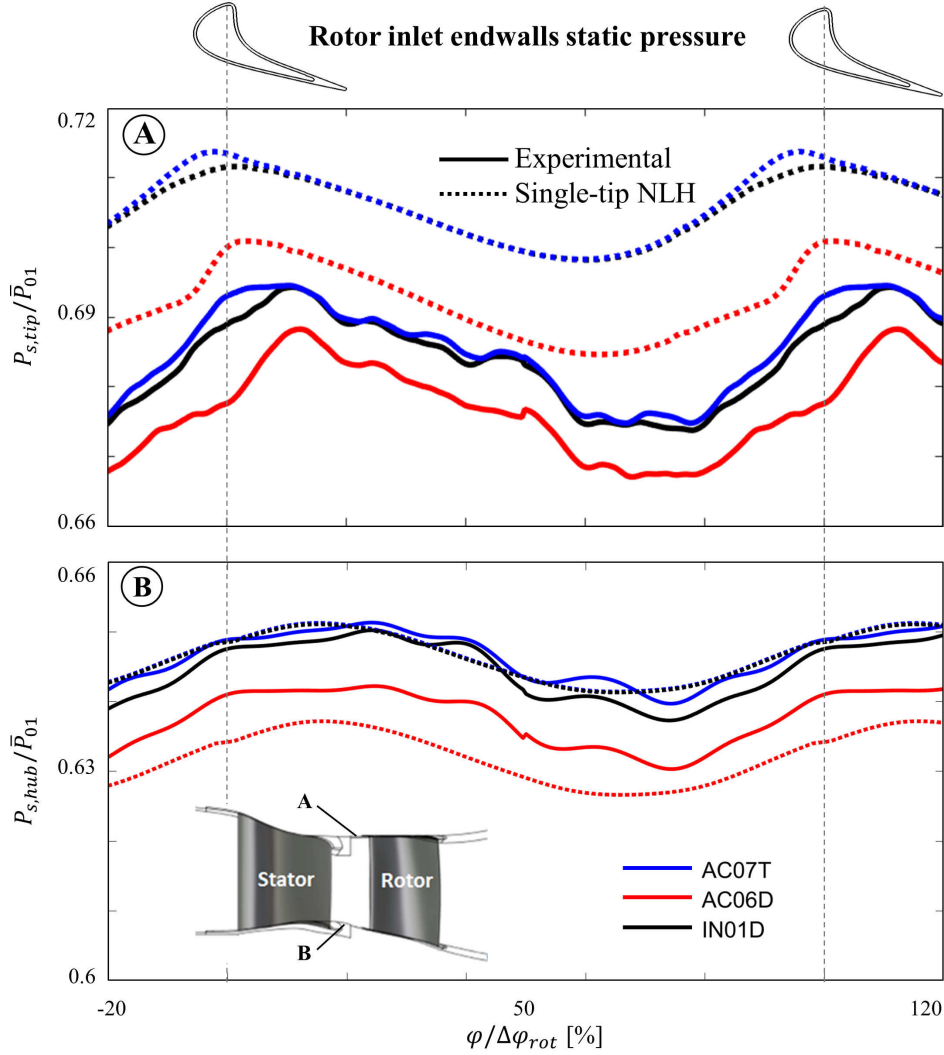


Figure 4.37: Phase-resolved casing and hub static pressure distributions at vane outlet

with respect to 20 mbar at the hub), as confirmed by the measurements. The higher fluctuation amplitude at the outer endwall can be ascribed to the abrupt acceleration experienced by the near-casing fluid as it is ingested in the tip gap from the blades front pressure side.

In absolute terms, the match between numerical and experimental results is satisfactory at the inner endwall for blades IN01D and AC07T. The measured pressure deficit of blade AC06D with respect to the other profiles is, however, higher by $0.4\%P_{01}$ with respect to the NLH results. At the outer endwall, an evident offset is present for the experimental results, which could be partially due to the calibration of the fast response pressure sensors as already commented in Figure 4.3. As for the casing, a more significant pressure deficit is measured for blade AC06D with respect to the

NLH predictions ($-0.5\% P_{01}$), explaining the slightly lower degree of reaction computed in Figure 4.10.

4.3.2 Casing pressure field

The phase-locked static pressure fluctuations measured at the rotor casing are presented as dotted lines in Figure 4.38 at three specific axial locations: 20.5%, 60.4% and 80.4% $c_{r,ax}$. The investigated locations are presented in the upper section of the figure together with the blade tip geometries.

One blade period is defined as the time one blade takes to traverse one full rotor pitch and the zero-phase position corresponds to the blade leading edge. At each axial position, the time-resolved signals of the six sensors operating along the vane pitch are synchronized to the same rotor blade phase and ensemble-averaged at identical rotor phase instants. This procedure removes the vane interaction effects and isolates the tip passage effect on the casing static pressure field. All the signals are normalized by the local time-mean value.

The sector-bounded ensemble averaging is performed over the five central channels of each sector (four in the case of blade AC07T) to ensure optimal flow periodicity. The y -axis scale is adapted in each plot to enhance the visibility of all the large and small amplitude variations over one blade cycle. The location of the rotor pressure and suction side edges is also indicated by light blue dotted lines. The plots show the corresponding numerical predictions represented by dashed red lines (RANS), green dash-dotted lines (full-annulus NLH) and blue solid lines (single-tip NLH).

The blade tip, traveling from right to left in the plots, causes a pressure fluctuation which depends on the leakage-driving aerodynamic load of the rotor airfoil. Moving through the rotor passage from the SS to the PS, a gradual increase of pressure is registered for the three tips. The steep acceleration experienced by the leakage flow inside the gap generates an abrupt pressure drop, dependent in magnitude and shape by the tip geometry.

Blades IN01D and AC06D exhibit an abrupt over-tip pressure decrease at 20.5% of the rotor axial chord (location A), indicating a low static pressure region over the squealer cavity. The over-tip pressure field results more uniform over IN01D while being modulated by the central island rim in blade AC06D. At the same axial location, the pressure increases monotonically from the SS to the PS of AC07T.

The pressure drop along the rotor channel is maximized for the three tips at 60.4% of the axial chord (location B). At this location, the flow entering the gap at the PS of AC07T experiences the highest pressure drop ($\sim 0.4P_{01}$) indicating the largest

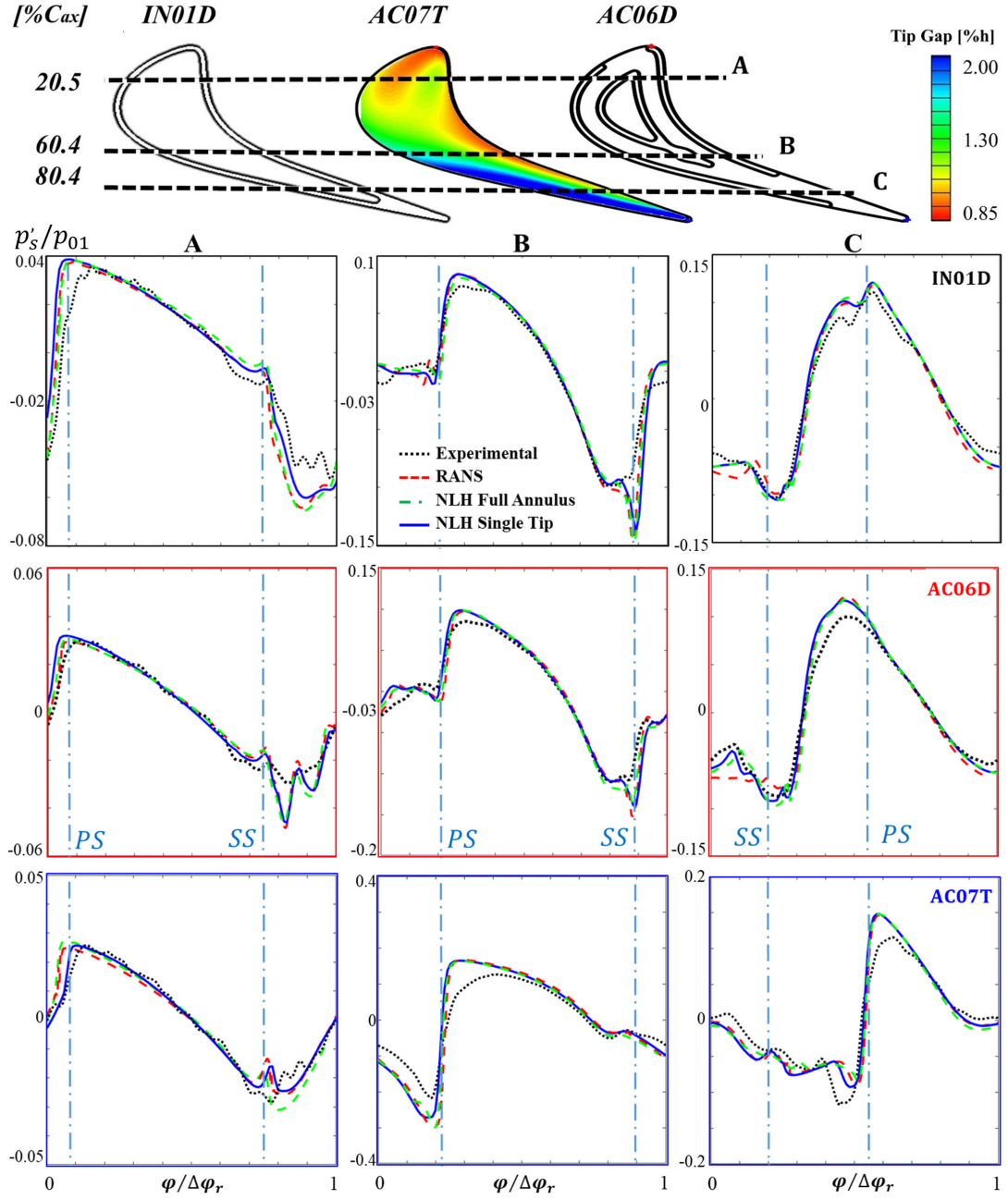


Figure 4.38: Phase-resolved casing static pressure over a rotor blade pitch: numerical and experimental results at selected axial stations

acceleration among the three tips.

Similarly, at 80.4% $c_{r,ax}$, the contoured tip shows the highest tip flow acceleration at the PS, followed by a rather constant static pressure level up the blade SS. In the squealer-type profiles, conversely, the flow expansion occurs over almost the full peripheral extension of the blade tip, reaching the minimum value over the SS rim. A good agreement is observed between experimental and numerical unsteady pres-

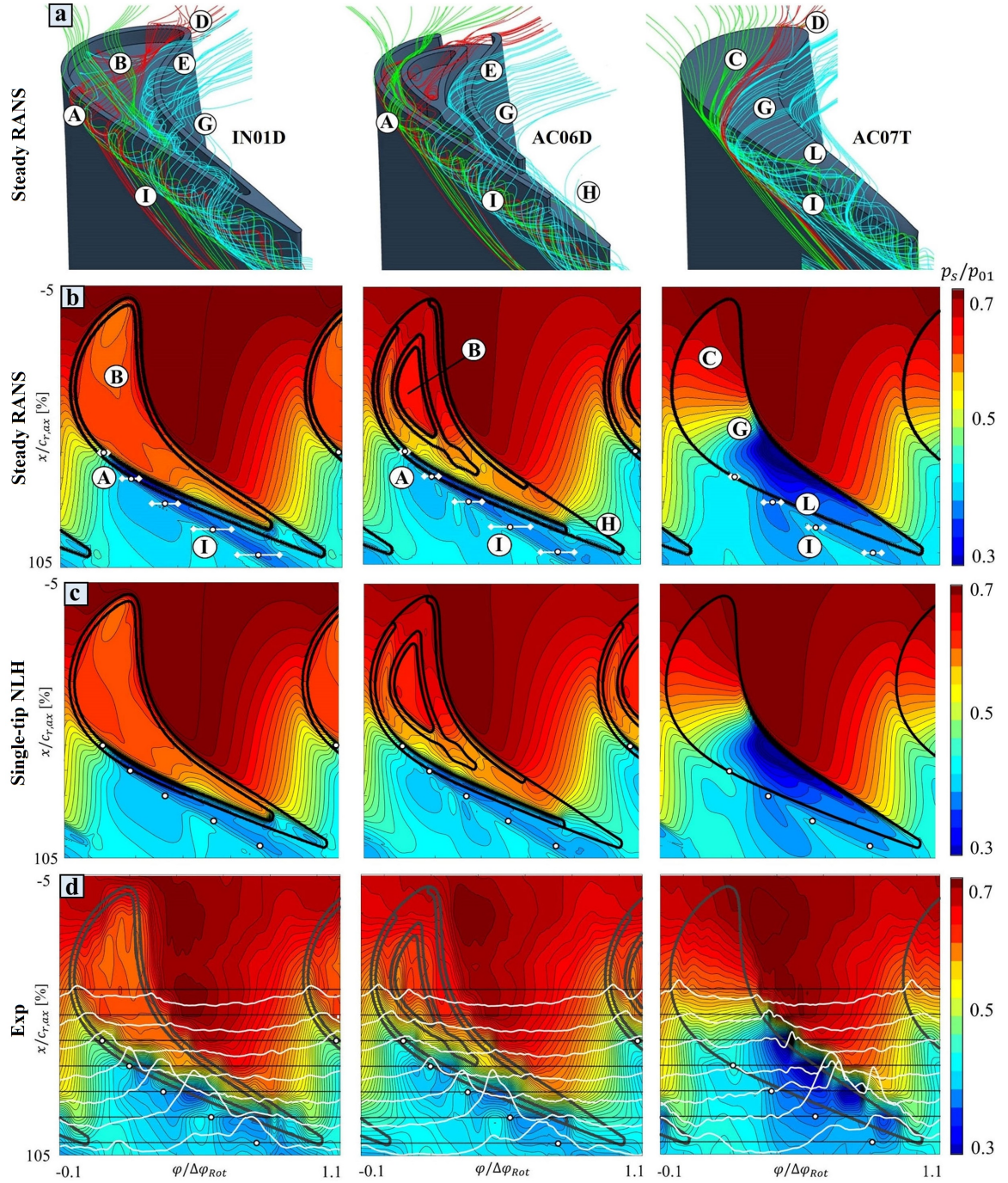


Figure 4.39: Relative velocity streamlines (a) and phase resolved casing static pressure contours: steady-RANS (b), single-tip NLH (c) and experiments (d)

sure histories at the three axial locations. The tip-to-tip flow differences are clearly captured by the unsteady CFD, especially in the tip gap region where the airfoil geometry mostly affects the flow aerodynamics.

The over-tip aerodynamic field of the three profiles is presented in Figure 4.39 by

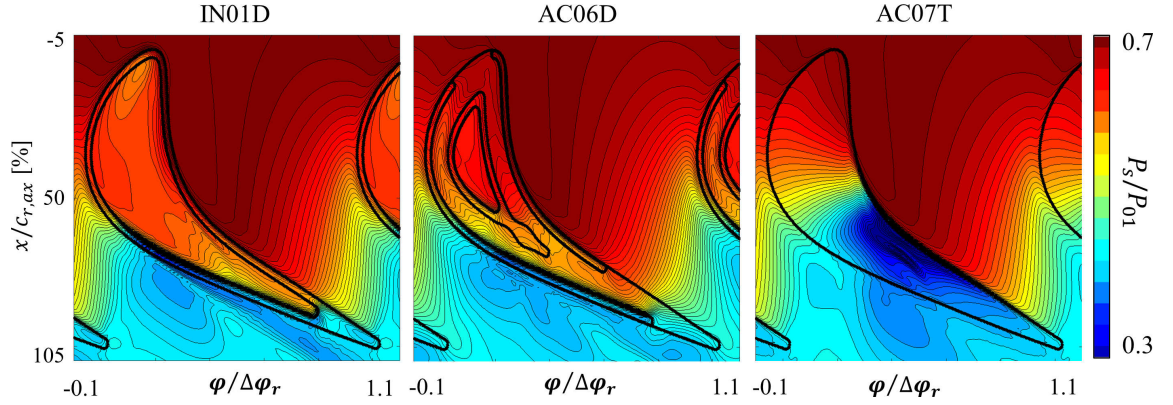


Figure 4.40: Phase resolved casing static pressure contours: full-annulus NLH computations

means of relative velocity streamlines (a) and phase resolved contours of normalized casing static pressure (b) predicted by steady-RANS computations. Time-resolved data-sets by single-tip NLH (c) and unsteady casing measurements (d) are phase-locked-averaged along the full vane pitch to obtain equivalent phase-resolved distributions in the rotor domain, ruling off any vane-dependent pressure gradients in the peripheral direction.

Qualitative time histories of experimental random unsteadiness are additionally plotted at seven distinct stations of the rotor axial chord over the experimental data-set (Figure 4.39d). Additionally, Figure 4.40 reports the phase-resolved casing static pressure contours computed from the full-annulus NLH simulations. As for the experimental results, the time-resolved pressure histories are ensemble-averaged over the five central channels of each sector (four in the case of AC07T) and reported in the rotor domain.

Globally, a very good agreement is shown between the RANS and NLH flow predictions, with no significant changes observed in the topology of the rotor-mean static pressure field. The time-resolved measurements provide a comprehensive validation of the numerical results. A slightly higher level of data scatter is observed in the measured distributions, in reason of the spatial resolution of the measurement array in axial direction (11 measurement stations between -5% and 105% $c_{r,ax}$).

The relative velocity streamlines serve to determine the impact of the tip design on the over-tip aerodynamic field. The flow entering the tip gap from the blades pressure side (PS) is represented by blue streamlines. In the case of tips IN01D and AC06D, the incoming fluid is initially diverted radially upwards and accelerated above the perimetral rim (E). In the cavity of IN01D this stream rolls up in a vortex which

follows closely the PS rim, while in AC06D the vortical structure is bounded by the rimmed PS channel up to 70% of the axial chord. The flow ingested over the front PS of blade AC07T, conversely, is shown to closely follow the tip surface, crossing the blade in pitchwise direction.

The red streamlines denote the flow entering the gap from the leading edge region. The front rim aperture of AC06D is shown to facilitate the flow ingress in the over-tip gap. This stream is partly driven over the central island and partly conveyed in the suction side (SS) rim-bounded channel, where it mixes with the flow entering the gap from the blade suction side (green streamlines).

The leading edge of tips IN01D and AC07T diverts radially the incoming flow towards the external casing (D). The ingested flow reattaches then on the squealer floor of IN01D and in the central island of AC06D, leading to regions of uniform casing static pressure (B). The shaped surface of blade AC07T, conversely, redirects and accelerates progressively the flow in axial direction leading to a decrease of shroud static pressure in the absolute frame (C).

In both squealer profiles, the over-tip stream is subject to an early SS ejection in comparison to blade AC07T. The SS rims promote the outward motion of the gap flow and increase its tangential velocity component (A). A localized low pressure patch is established in the near-SS region of the blade passage, where high velocity fluid is spilled from the over-tip region. The tip leakage vortex (TLV) trajectories, computed over the numerical dataset by means of the triple decomposition method (TDM, [89]), are indicated in the contour plots by white dots. Additionally, the change in vortex strength is quantified Figure 4.39b by horizontal bars of length proportional to the TLV circulation [80].

In blades AC06D and IN01D, from 60% $c_{r,ax}$ onwards, the flow tends to directly bypass the cavity before being discharged at the tip suction side (G). In blade AC07T, this substantial acceleration in tangential direction starts at 50% $c_{r,ax}$, where a very low pressure region is established at the casing over the tip PS surface.

The closed end of the squealer bath of tip IN01D produce a local outward ejection of the cavity stream, which is subsequently entrained by the TLV at the blade SS (I). The rear aperture (H) of blade AC06D facilitates the PS-SS fluid bypass and its subsequent entrainment in the TLV. In both squealer-like profiles, the TLV trajectory corresponds to a local rise of random unsteadiness.

The amplitude of the stochastic pressure fluctuation appears to correlate with the vortex intensity. Hence, the presence of the TLV in the near-casing region increases the non-deterministic pressure fluctuations as the vortical structure develops and grows

in intensity. The TLV trajectory marks the boundary between a near-SS low pressure region, where the high velocity leakage fluid is attached to the casing, and the passage region, where the spilled fluid is driven inwards by the vortex motion.

The absence of a rimmed structure at the PS of AC07T allows the ingested flow to follow closely the carved tip surface and accelerate constantly up to 60% of $c_{r,ax}$. At this axial location flow separation occurs over the tip surface and a recirculation region is shown to overhang the tip (L). The flow separation in the aft over-tip area is confirmed by a local increase of uncorrelated unsteadiness in the experimental results.

4.3.3 Casing heat transfer

4.3.3.1 Theoretical background.

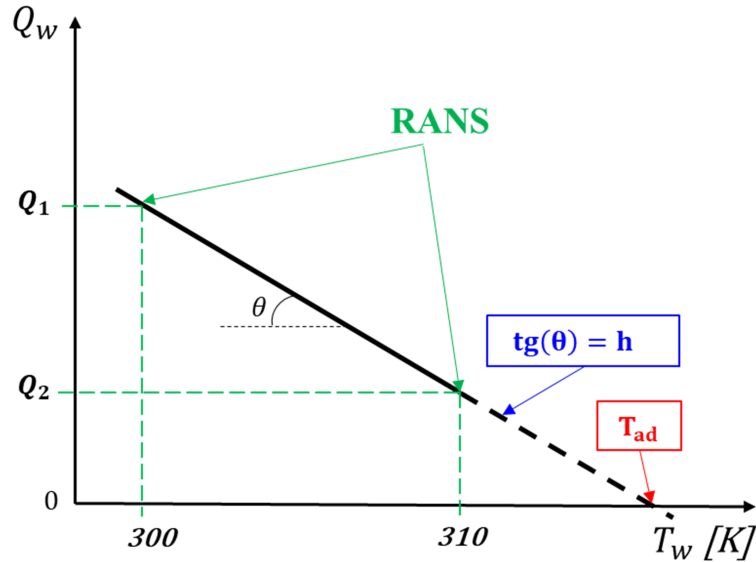


Figure 4.41: Computation of adiabatic wall temperature and heat transfer coefficient from numerical predictions at isothermal wall conditions

The thermal field of turbine rotor shrouds is characterized by periodic changes in local heat transfer rates and significant axial and circumferential flow recovery temperature variations. Due to its dependency on the thermal boundary conditions, the definition of convective heat transfer coefficient based on a constant gas temperature falls short of describing time- and space-dependent convective phenomena. An alternative approach, extensively described by Anderson and Moffat [97], consists in basing the heat flux computation on the difference between the temperature of the solid surface (T_w) and the time-resolved adiabatic wall temperature (T_{aw}) as from

Equation 4.2. The heat transfer is therefore quantified in terms of two main driving factors: the time-dependent state of the local boundary layer (h) and the thermal contribution ($T_{aw} - T_w$) .

$$q = h * (T_{aw} - T_w) \quad (4.2)$$

In the present paper, the evaluation of the local T_{aw} and h is performed by computing the fluid-to-solid heat transfer in the rotating frame at two different values of T_w (300 and 310K) as presented in Figure 4.41. By expressing the heat flux as a function of T_w , the local value of T_{aw} is evaluated by linear interpolation as the zero-heat-flux surface temperature while h represents the slope of the fitting line. As suggested by [98], the local time-accurate heat flux can be addressed as the summation of a steady and an unsteady component.

$$\bar{q} + q' = (\bar{h} + h') * (\bar{T}_{aw} + T'_{aw} - T_w) \quad (4.3)$$

By rearranging the constitutive terms of Equation 4.3, it is possible to highlight how the steady thermal load will not depend solely on the mean values of T_{aw} and h , but also on the contribution of their unsteady component. Equation 4.4 highlights how the zero-mean oscillations of the two quantities, addressed here as T'_{aw} and h' , will produce a non-null time average heat flux when out of phase.

$$\bar{q} = \bar{h} * (\bar{T}_{aw} - T_w) + \overline{h' * T'_{aw}} \quad (4.4)$$

4.3.3.2 Over-tip heat transfer field.

Figure 4.42 shows the time-resolved shroud heat transfer for the three tips in the rotor-relative frame, with the horizontal axis indicating the pitchwise location along the rotor passage. The numerical predictions (top), obtained by setting $T_w = 300K$, are shown together with the time resolved measurements, which have been averaged in the relative frame to exclude the effect of the upstream vane. Qualitative time histories of measured random unsteadiness are presented together with the computed TLV trajectories.

The measured and predicted heat transfer distributions, normalized by the heat flux of IN01D averaged between - 5 and 105% $c_{r,ax}$ over a full rotor pitch, show a high degree of agreement. A gradual decrease of heat transfer rate characterizes the passage region of the three profiles, proceeding from the blade pressure side (PS) to the suction side (SS) of the adjacent airfoil. Constant heat flux is observed over the cavity of blade IN01D and central island of blade AC06D (B), while the shaped surface of AC07T

generates a gradual decrease of shroud heat transfer from the leading edge up to the SS at 60% $c_{r,ax}$ (C).

High heat flux values characterize the stagnation point at the leading edge of the three tips (D), where the flow is locally redirected towards the external shroud. The heat flux peak, however, is not fully captured by the experimental results, possibly due to the coarser axial discretization of the measurement array. A high heat transfer stripe is shown over the full PS rim of blades IN01D and AC06D and between 25 and 90% $c_{r,ax}$ of the PS of blade AC07T, where the surface contouring emulates a squealer-like geometry. The impingement of hot fluid over the external shroud is seen to locally increase the stochastic fluctuations in this region for the three tips.

The over-tip heat flux is maximized in the zones where the flow leaking from PS

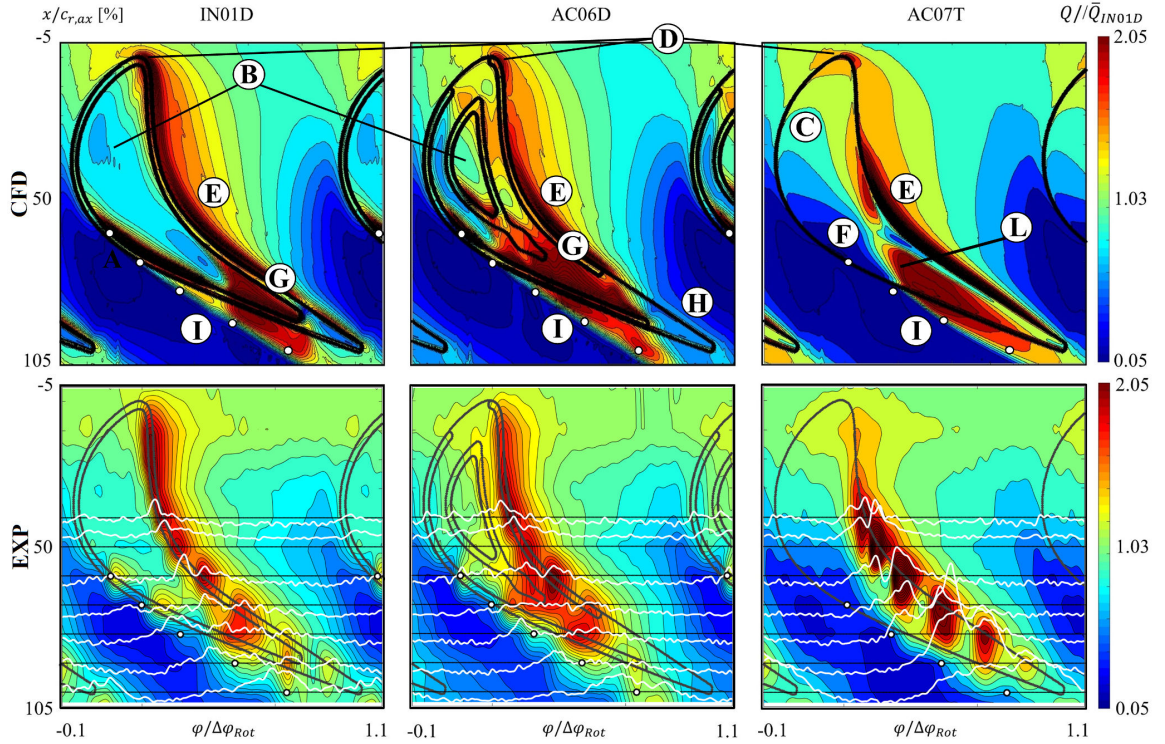


Figure 4.42: Numerical (a) and experimental (b) phase-resolved contours of normalized casing heat flux

to SS directly by-passes the squealer cavities of blades IN01D and AC06D (G). The rear rim aperture of the latter facilitates the fluid by-pass from the PS to the SS of the blade and the larger clearance is shown to significantly decrease the casing heat load (H).

A stripe of high heat flux is observed over the aft part of blade AC07T, where over-tip separation has been detected through the relative velocity streamlines (L) in

Fig. 4.39. A cold spot appears to interrupt this region at $60\%c_{r,ax}$, further developing in a narrow structure which runs alongside the PS of AC07T up to the blade trailing edge (F). This whole region is characterized by high peaks of random unsteadiness, locally interrupted in F.

In all the three tip geometries, the tip leakage vortex core (white dots) marks the boundary of the high heat flux region in proximity of the rear tip SS (I).

4.3.3.3 Adiabatic Wall Temperature.

Figure 4.43 shows the phase-resolved casing adiabatic wall temperature (a) and the evolution of the flow over six planes orthogonal to the blade camber line (b), with normalized total temperature contours completed by relative velocity surface streamlines.

The passage flow shows a similar behavior for the three tips, with the recovery tem-

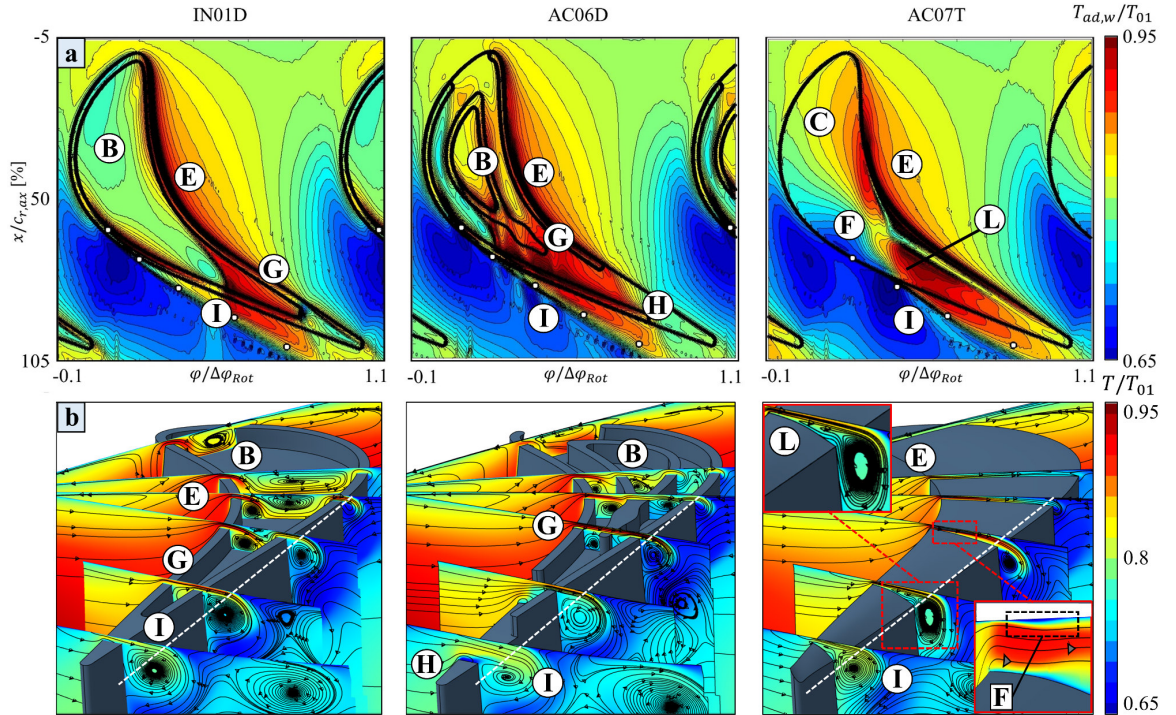


Figure 4.43: Phase resolved contours of casing T_{aw}/T_{01} (a) and orthogonal cuts with T_0/T_{01} contours and relative velocity streamlines (b)

perature exhibiting a monotone increase from the blade SS to the PS of the neighboring airfoil. High temperature regions are shown at the PS of the three profiles (E), where the flow experiences an abrupt acceleration and tangential redirection, locally exceeding the stage inlet total temperature ($T_{aw}/T_{01} > 1$). A recirculation

region exists in the squealer bath of blade IN01D, causing a drop in flow recovery temperature over the entire extent of the cavity (B) up to $70\% c_{r,ax}$.

The fluid ingested in the tip gap of blade AC06D is partly entrained in a vortical structure which develops in the PS rim-bounded channel, with values of T_{aw}/T_{01} ranging between 0.8 and 0.85. A nearly constant distribution of low T_{aw} is observed over the front part of the central island (B).

The front part of blade AC07T is characterized by a gradual decrease of adiabatic

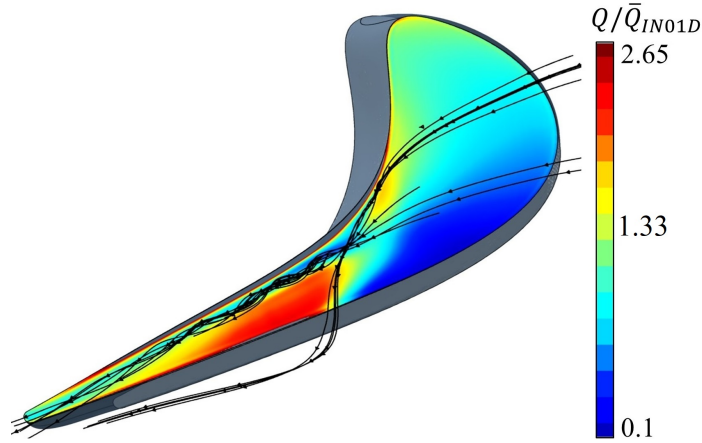


Figure 4.44: Normalized casing heat flux contours over tip AC07T with near-casing relative velocity streamlines

wall temperature (C) up to $50\% c_{r,ax}$. A region of high T_{aw} is initiated at $20\% c_{r,ax}$ at the blade PS and abruptly interrupted at $60\% c_{r,ax}$ by the presence of low temperature fluid in the near casing region (F, detail in fig. 4.43b). A better insight on the origin of this cold stream is given in Figure 4.44, showing the heat transfer distribution at the casing of blade AC07T and relative velocity streamlines in close proximity to the wall. Two distinct fluid streams are observed to enter the gap from the blade SS at 20% and $40\% c_{r,ax}$, traveling towards the blade PS and presenting lower temperature and velocity than the main gap flow. After being redirected by the over-tip pressure gradient, the two structures are discharged at the blade SS ($70\% c_{r,ax}$) and trailing edge respectively.

Blades IN01D and AC06D show a significant increase of over-tip T_{aw} at the location of direct gap flow bypass (G), which is initiated earlier in blade AC06D due to the presence of the central rim. An abrupt increase of heat transfer is observed at $70\% c_{r,ax}$ for blade AC07T, where the relative velocity of the leakage flow presents a high tangential component and the peripheral part of the TLV overhangs the tip surface (L in Figure 4.43b detail).

In the aft region of the three blades the tip leakage flow shows increased values of stagnation temperature, justifying the high heat transfer values observed in Fig. 4.42. The TLV trajectories, indicated by white dots in Fig. 4.43a, mark the boundary between regions of high and low flow temperature (I). The relative velocity streamlines show how, downstream of the vortex center, high temperature flow is driven radially inwards by the TLV and colder passage fluid moves towards the casing endwall.

4.3.3.4 Over-tip work process.

As demonstrated in fig. 4.43, rotor blade tips traveling in close proximity to the stationary shroud are responsible for unsteady work processes which can locally increase the flow stagnation temperature above the stage inlet value (T_{01}), with the leakage fluid gaining angular momentum under the action of the PS-SS pressure gradient [99], as represented in Figure 4.45. Equation 4.5 describes the increase in total temperature with respect to the rotor inlet plane (located $5\%c_{r,ax}$ upstream of the airfoil leading edge) as a function of the work produced by the rotor on the over-tip fluid.

$$T_0 - T_{r,in} = \frac{1}{c_p} \cdot U \cdot (V_t - V_{t_r,in}) \quad (4.5)$$

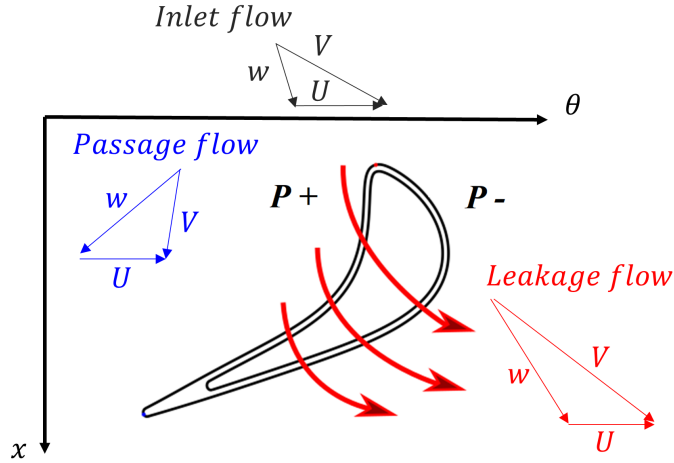


Figure 4.45: Increase in angular momentum of the over-tip flow under the action of the PS-SS pressure gradient

In the present work, the predicted tangential velocity field at the mid-gap plane is employed to model the total temperature field over the three tips. Figure 4.46 shows the phase resolved contours (a) and axial distributions (b) of mid-gap flow total temperature computed by the work exchange model (T_{mod}). The CFD-predicted

T_{aw} (solid line) and mid-gap T_0 (dashed line) are presented for comparison. The pitchwise-averaged quantities are computed over the blade tip surface, extended circumferentially by 5% of the rotor pitch in both suction and pressure side directions. A localized increase of $\bar{T}_0$, qualitatively matched by $\bar{T}_{mod}$, is observed at the early

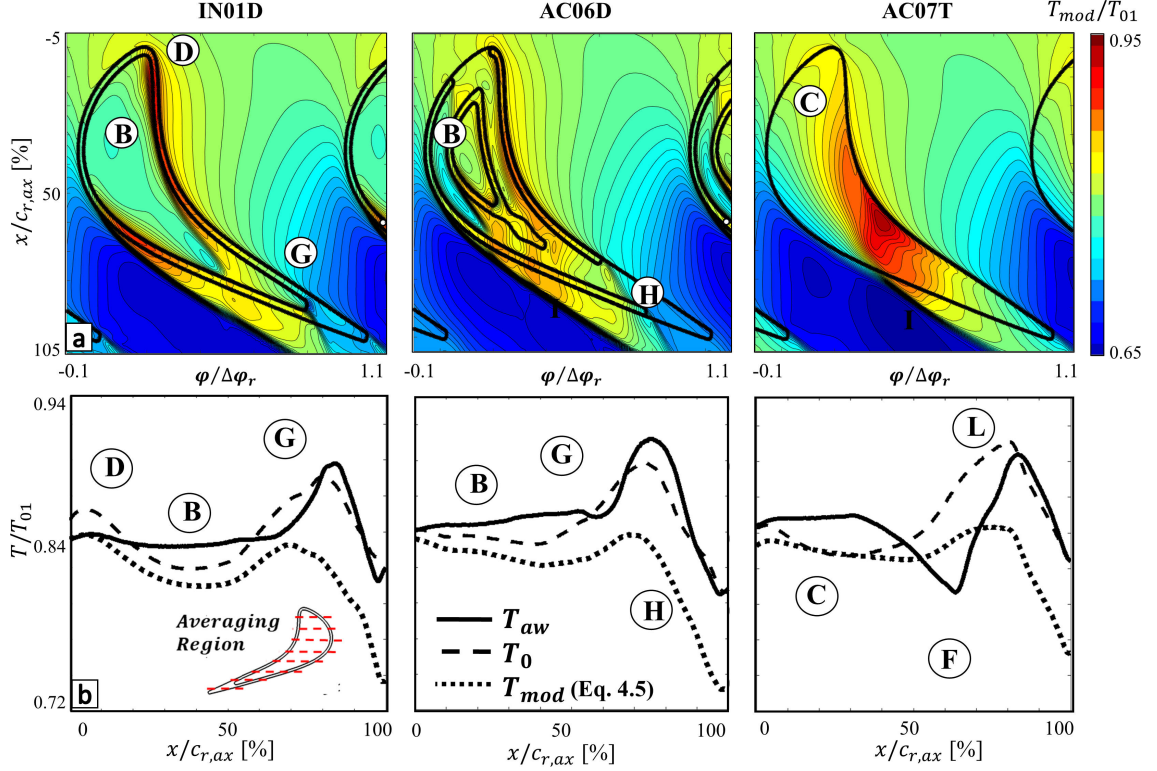


Figure 4.46: Phase-resolved contours of modeled mid-gap T/T_{01} (a) and pitch-average distributions modeled and predicted over-tip T/T_{01} (b)

10% $c_{r,ax}$ of blade IN01D (D), as the flow ingested in the gap is accelerated in tangential direction over the closed rim. Between 10 and 30% $c_{r,ax}$, the mid-gap flow temperature of IN01D gradually decreases due to the low velocity region on the bottom of the squealer bath (B) while the rimmed pattern of AC06D promotes higher flow deviation values in the same region, contributing to keep a constant level of $\bar{T}_0$ up to 50% $c_{r,ax}$. Between 20 and 50% $c_{r,ax}$, the adiabatic wall temperature reaches higher values than $\bar{T}_0$ for the three blades, indicating a greater flow turning in the near-casing region with respect to the mid-gap plane.

A steep increase of $\bar{T}_{aw}$ and $\bar{T}_0$ is shown in the second part of the axial chord for both squealer designs, motivated by the initiation of the direct bypass mechanism (G). In agreement with the aerodynamic analysis, a local peak of $\bar{T}_{aw}$ is detected at 85% $c_{r,ax}$ for blade IN01D while a more extended maximum is detected between 70

and 80% $c_{r,ax}$ for blade AC06D. Slightly higher values of adiabatic wall temperature are reached over blade AC06D with respect to IN01D (+2% T_{01}). In the case of blade AC07T, a monotonic increase of $\bar{T}_0$ is observed from 50% $c_{r,ax}$, leading to a global peak at 85% $c_{r,ax}$ of 0.9 T_{01} . Contrarily, the $\bar{T}_{aw}$ shows a steep decrease down to a global minimum of 0.78 T_{01} at 60% $c_{r,ax}$, where the near casing cold stream (F) crosses the over-tip high temperature region (L). The substantial difference between the mid-gap and casing thermal field in this region is due to over-tip flow separation detailed in Figure 4.43b.

While a good agreement is shown up to 50% $c_{r,ax}$ between $\bar{T}_0$ and $\bar{T}_{mod}$, the work exchange model fails to predict the maxima reached in the aft part of the profiles. This is explained by the fact that the model, based on the evaluation of the absolute velocity field at the mid-gap plane, is unable to account for the radial advection of higher temperature fluid in the aft part of the blade tips, where the limited thickness of the airfoil and the TLV generate a highly three-dimensional flow field with strong temperature gradients in the near casing region.

4.3.3.5 Casing convection mechanisms.

Phase-resolved contours of casing Nusselt number are presented in Figure 4.47a. The hydraulic diameter of the tip gap D_h , i.e. two times the clearance value, is considered as the characteristic length of the over-tip flow.

The convection rate over the blade passage is observed to decrease moving along the main flow path, as the absolute velocity of the flow gradually decreases. Significant peaks of Nu_{D_h} are observed at the leading edge of the three tips (D), where the fluid has been shown to impinge locally on the external shroud. As a difference from the T_{aw} distributions, the local maxima tend to propagate towards the passage at the tip PS. A rather uniform distribution of low Nusselt is detected over the squealer bath of blades IN01D and AC06D (B), where the ingested fluid tends to recirculate in proximity of the cavity floor.

High convection regions are initiated at 40% $c_{r,ax}$ at the PS of the three tips, as an effect of the entering fluid which is radially diverted by the squealer rims and by the PS of AC07T (E). The high convection region is seen to extend up to the trailing edge for blades IN01D and AC07T, while being interrupted abruptly at 70% $c_{r,ax}$ by the aft rim aperture of blade AC06D (H).

In line with the thermal distributions of Figure 4.43, a high convection region develops from the PS of AC07T (40% $c_{r,ax}$) up to the aft part of the blade SS, where over-tip separation occurs (L). This region is locally interrupted at 60% $c_{r,ax}$ by the low-velocity

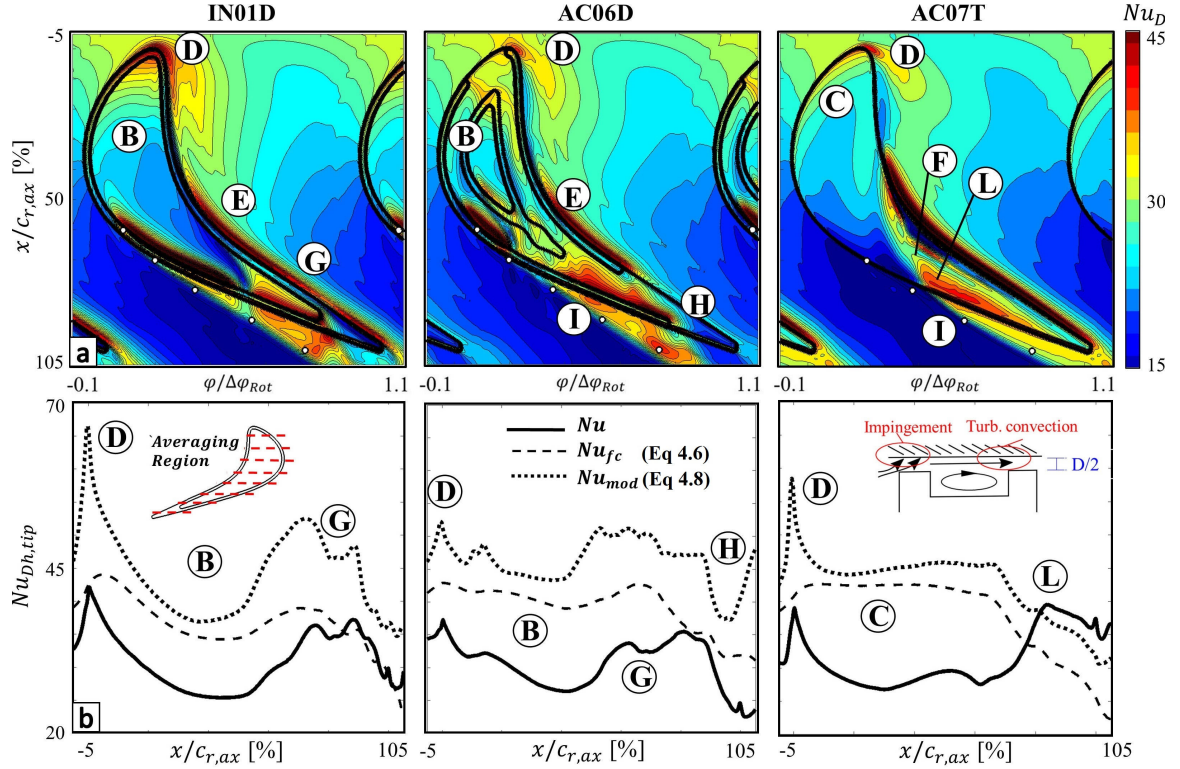


Figure 4.47: Phase-resolved numerical contours (a) and pitchwise average distributions (b) of Nu_{D_h}

flow structure (F) highlighted in Figure 4.44. The TLV core (white dots) marks the boundary between the upstream high Nusselt region, where the flow is driven towards the casing by the vortical motion, and the low-convection region over the rotor passage (I).

4.3.3.6 Convection process model.

The pitchwise-average over-tip casing Nusselt is shown by solid lines in Figure 4.47b. A low order model is introduced to represent the casing convection mechanism, considering two main contributions. The forced convection in a turbulent flow (dashed line) is evaluated by the Dittus-Boelter correlation as in Equation 4.6.

$$Nu_{fc} = 0.023 Re_{xt,D_h}^{0.8} Pr^{0.4} \quad (4.6)$$

Additionally, the local impingement heat transfer based on the radial Reynolds distribution is computed by Equation 4.7 [100].

$$Nu_{imp} = 0.181 Re_{rad,D_h}^{0.7} Pr^{0.33} \quad (4.7)$$

The CFD-predicted local mid-gap Reynolds number, based on hydraulic diameter of the tip gap D_h , is employed in the computation. The absolute velocity field is decomposed in two components, one parallel and one orthogonal to the mid-gap plane, which are employed to compute Re_{xt,D_h} and Re_{rad,D_h} respectively. The total Nusselt number (dotted line) is computed as the summation of the two contributions, as from Equation 4.8.

$$Nu_{D_h,mod} = Nu_{fc} + Nu_{imp} \quad (4.8)$$

While an acceptable qualitative agreement is detected between the two distributions in Fig. 4.47b, the model shows limitations in predicting the mean Nusselt level along the full axial extent of the profiles. Possible factors contributing to this offset are the choice of a simplified characteristic length, which should instead depend on the path followed by the leakage stream in the tip gap, together with the limited validity of these correlations in non-uniform, three dimensional flows.

A first peak of $\overline{Nu}$ exists at the LE of the three profiles (D), significantly reduced in AC06D by the presence of the front rim aperture. The signature of the central island LE is present in this tip as a local peak at 15% $c_{r,ax}$. Both peaks are caused by the local deviation of incoming fluid towards the external casing and thus mainly due to a local increase of Nu_{imp} .

The cavity regions of IN01D and AC06D produce low convection areas (B). Starting from 50% $c_{r,ax}$, both squealer tips show a steep $\overline{Nu}$ increment due to the leakage flow ejected at the SS rim and to the direct bypass mechanism (G) highlighted in Figure 4.39. The shaped surface of AC07T, conversely, produces a $\overline{Nu}$ plateau up to 60% $c_{r,ax}$ (C), locally interrupted by the low convection region present in the near casing region at mid-chord (F). A steep increase of heat transfer coefficient is registered in the last 30% $c_{r,ax}$ of this profile, where the massive over-tip separation occurs (L). This behavior, however, is not captured by the two-dimensional model.

4.3.3.7 Unsteady heat transfer contributions.

As derived in Equation 4.4, both T'_{aw} and h' contribute actively to the time-average thermal load of the rotor shroud. Figure 4.48 presents the relative impact of $\overline{h' * T'_{aw}}$ on the time average casing heat transfer, pitchwise averaged over a full rotor passage. The horizontal axis indicates the axial location normalized by the rotor axial chord. As a common feature for the three tips, an initial peak of approximately 1% of the total load is observed at the blade leading edge. This local maximum is followed by a plateau for the two squealer geometries, where the flow separates locally over

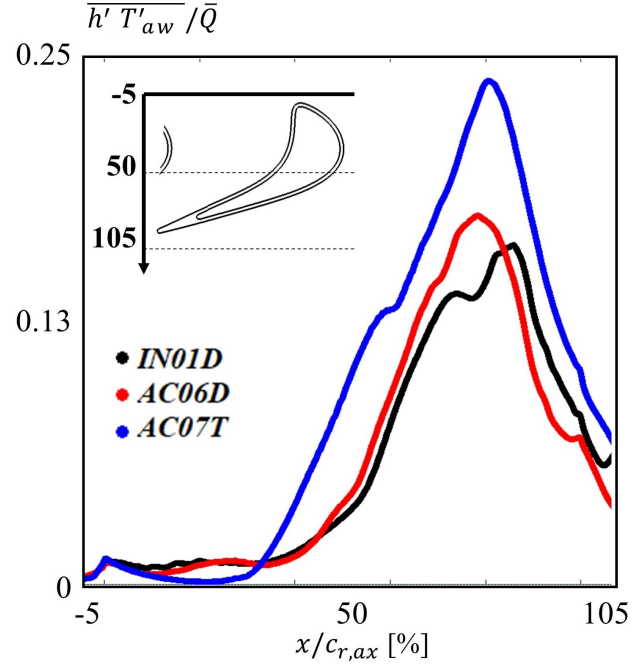


Figure 4.48: Time-mean h' and T'_{aw} contribution to casing heat flux

the perimetral rims, and by a slight decrease in the case of blade AC07T, where the contoured surface progressively accelerates the flow.

A steep increment of the unsteady contribution is observed at $35\%c_{r,ax}$ for the carved tip and at $45\%c_{r,ax}$ for IN01D and AC06D. The highest maximum is observed at $80\%c_{r,ax}$ for blade AC07T, reaching almost 25% of the total average thermal load, as a consequence of the aft over-tip flow separation. Lower peaks are registered for blades AC06D and IN01D, at 75% and $85\%c_{r,ax}$ respectively, amounting to 17% and 15% $\bar{Q}$. Thus, out-of-phase fluctuations of Nusselt and T_{aw} are present mostly in the aft part of the tips, where high unsteadiness exists in the aerodynamic and thermal field in reason of the TLV.

The unsteady heat flux contribution to the total casing heat transfer is in good qualitative agreement with what shown by Thorpe in [101]. Quantitative discrepancies in the peak values ($h'T'_{aw} \sim 75\%Q$ at $79\%c_{r,ax}$ reported by Thorpe) are present due to the different design and operating conditions of the HPT test article. This result confirms the importance of full-scale turbine testing in rotating test rigs which allow to account for the rotor-induced fluctuations of h and T_{aw} and quantify the casing heat load at engine-relevant conditions.

4.3.4 Stage outflow

4.3.4.1 Phase-resolved flow

This section presents the characterization of the phase-resolved flow field at the stage outlet plane, located 50% $c_{r,ax}$ downstream of the rotor trailing edge. All the quantities are presented in the rotor-relative domain. The horizontal axis indicates the peripheral position along the rotor pitch, with the blades traveling from right to left and the airfoil trailing edge aligned with the zero-phase coordinate. The experimental results are ensemble averaged in the rotor domain, ruling out all vane-dependent gradients in circumferential direction, and over the central passages of each sector. Four passages are considered for blades IN01D and AC06D and three passages for sector AC07T, as motivated in Section 4.7. All the quantities are represented as a variation from the average value at 53% of the blade span. The centers of the TLV, upper passage vortex (UPV) and lower passage vortex (LPV) detected by the vortex tracking algorithm on the RANS flow predictions are indicated by white dots, with the vortex boundary defined as the residual vorticity isoline corresponding to a value of $\omega = 3500$.

In Figure 4.49, time resolved contours of absolute total pressure measured by the virtual three-hole-probe and fast response pressure rakes are compared to numerical predictions by steady RANS and single-tip and full-annulus NLH. For the latter, the central channels of each individual sector are considered, in concordance to the ensemble-averaging performed over the time-resolved measurements.

The experimental results confirm the positioning of the TLV at the boundary of a pressure minimum located between 80 and 90% of the blade span (h) for the three profiles. The near-tip pressure deficit is particularly pronounced in blade AC06D, where the vorticity isoline shows the strongest TLV among the three profiles. The pitchwise position of the near-tip pressure pattern varies consistently with the TLV between the three tips: the low pressure zone is shown to remain closer to the blade suction side for tip AC07T ($\varphi = 0.9$) while traveling up to $\varphi = 0.5$ inside the rotor passage for blade AC06D.

The phase resolved pressure pattern recorded by the V3H is globally consistent with the measurements performed by the fast response pressure rake. From an absolute point of view, however, the V3H data show a milder pressure deficit in the near-TLV region, which also presents a smaller radial extent. Factors contributing to these discrepancies could be the different radial resolution in the measurement matrix of the two probes and the large unsteady flow angle variations at this location. Globally,

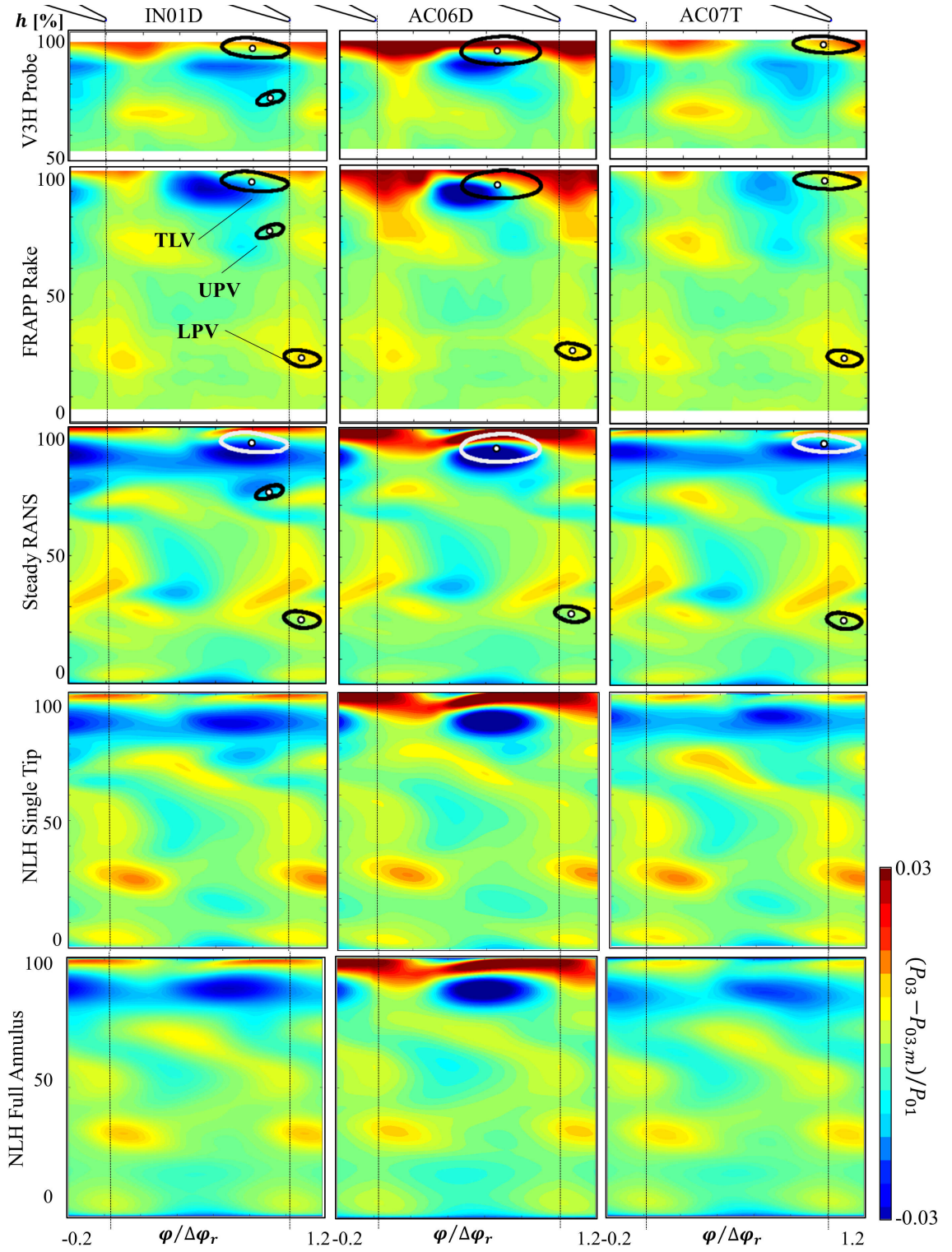


Figure 4.49: Phase-resolved P_{03}/P_{01} : virtual-three-hole probe, fast response pressure rake, Steady RANS and single-tip/full-annulus NLH results

the experimental data-set confirms the steady and unsteady flow predictions, both for the phasing and for the amplitude of the TLV-induced pressure fluctuations.

The UPV, detected only for blade IN01D, is located at 75% of the rotor blade span, at the boundary between a low and a high pressure region. For blades AC06D and AC07T, the residual vorticity of the UPV at this location is below the threshold imposed for the vortex tracking computation [102]. The phase-resolved pattern at this radial coordinate, however, appears consistent between the three tips. Both experimental and numerical results show periodic fluctuations of $0.015 P_{01}$.

No substantial tip-to-tip differences are shown in the lower 50% of the span, where the LPV is tracked at $\sim 25\%h$, in correspondence of a localized pressure peak adjacent to the rotor trailing edge phase, captured particularly well in terms of phasing and intensity by the full-annulus NLH. A small radial offset is noticeable for the LPV of blade AC06D (located at $28\%h$), consistently with a slight radial migration of the pressure peak validated by the measurements.

The RANS results, conversely, show the periodic occurrence of an inclined high-pressure region between 25% and 50% of the blade radius, which is not retrieved in the experimental pressure histories. The NLH results show, globally, a better qualitative and quantitative agreement with the measurements. Higher mixing is detected, in general, in the experimental pressure field, partially captured by the full-annulus computations.

Time resolved contours of outlet yaw angle are presented in Figure 4.50, offering a comparison between virtual three-hole-probe measurements (top), steady (center) and NLH (bottom) flow predictions. A remarkable agreement can be observed between the experimental and numerical datasets in close proximity to the casing, where the TLV appears confined in a zone of negative flow angle which varies in peripheral position according to the tip design.

The pitchwise gradient induced by the TLV presents significantly lower intensity for blade AC07T in comparison to the squealer-like profiles. Periodic flow angle fluctuations are detected between 60 and 80% h for the three tips, corresponding to the radial location of the UPV. The amplitude of these fluctuations is overpredicted by the steady RANS, especially for tip AC07T. A more mixed out distribution is observed in the NLH predictions, with the full-annulus computations offering the best match with the experimental results in terms of flow topology and fluctuation amplitude

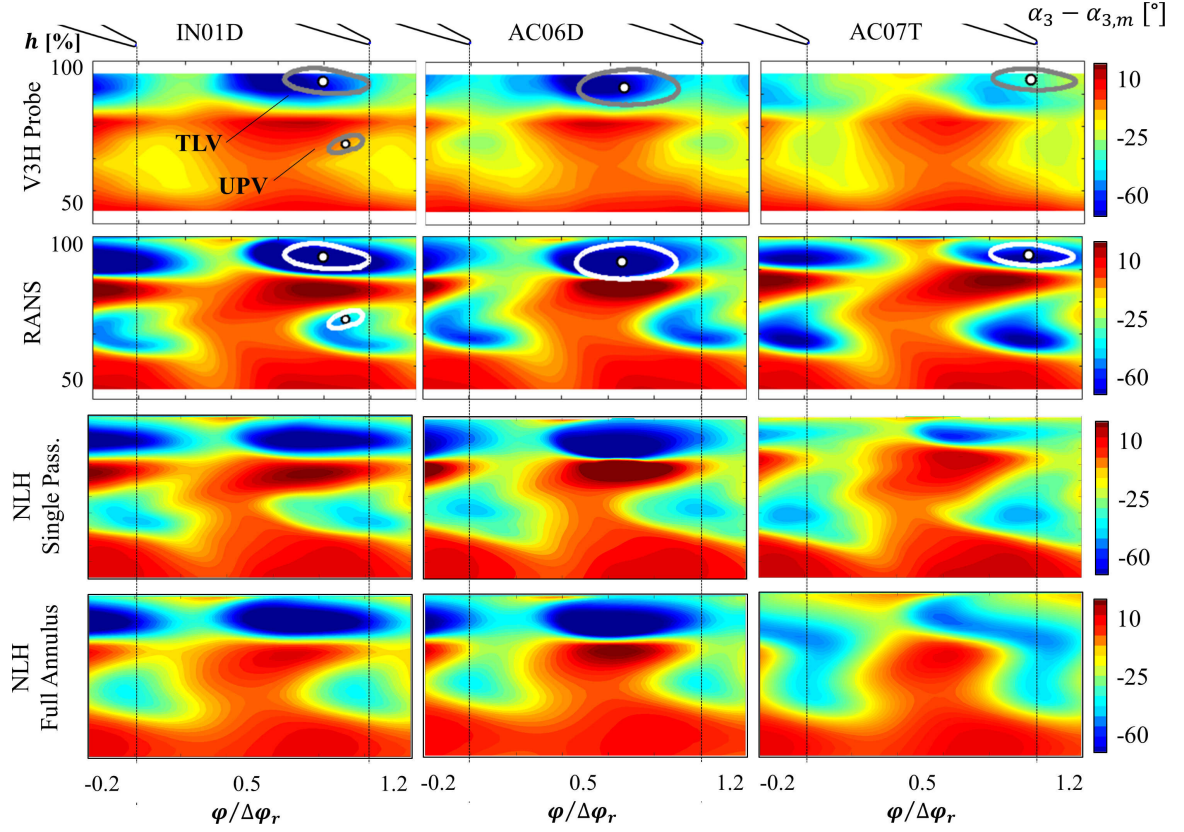


Figure 4.50: Phase-resolved outlet yaw angle: experimental and numerical results

4.3.4.2 Stage outlet static pressure

The phase resolved static pressure fluctuations at the stage outlet endwalls are visualized in Figure 4.51. The experimental pressure histories, acquired by fast response sensors and averaged over the four central channels of each sector, are compared here to numerical predictions by RANS (dotted lines) and single-tip NLH (dashed lines). Additionally, full-annulus NLH results are shown, ensemble-averaged over the same sector channels employed in the experimental data reduction. For post-processing reasons, the casing pressure fluctuations are phased with the rotor airfoils leading edge, as for all the casing contours shown in this thesis, while the hub distributions are phased with the blades trailing edge.

In general, a good agreement is detected between RANS and NLH flow predictions. The experimental dataset globally validates the fluctuation phasing and amplitude, which amounts to $1\%P_{01}$ at the hub and $2\%P_{01}$ at the casing. The measured pressure histories, however, exhibit localized discrepancies in terms of fluctuation shape and a globally higher mean pressure value at both endwalls.

In agreement with the sector-resolved time-average distributions shown in Figure 4.8,

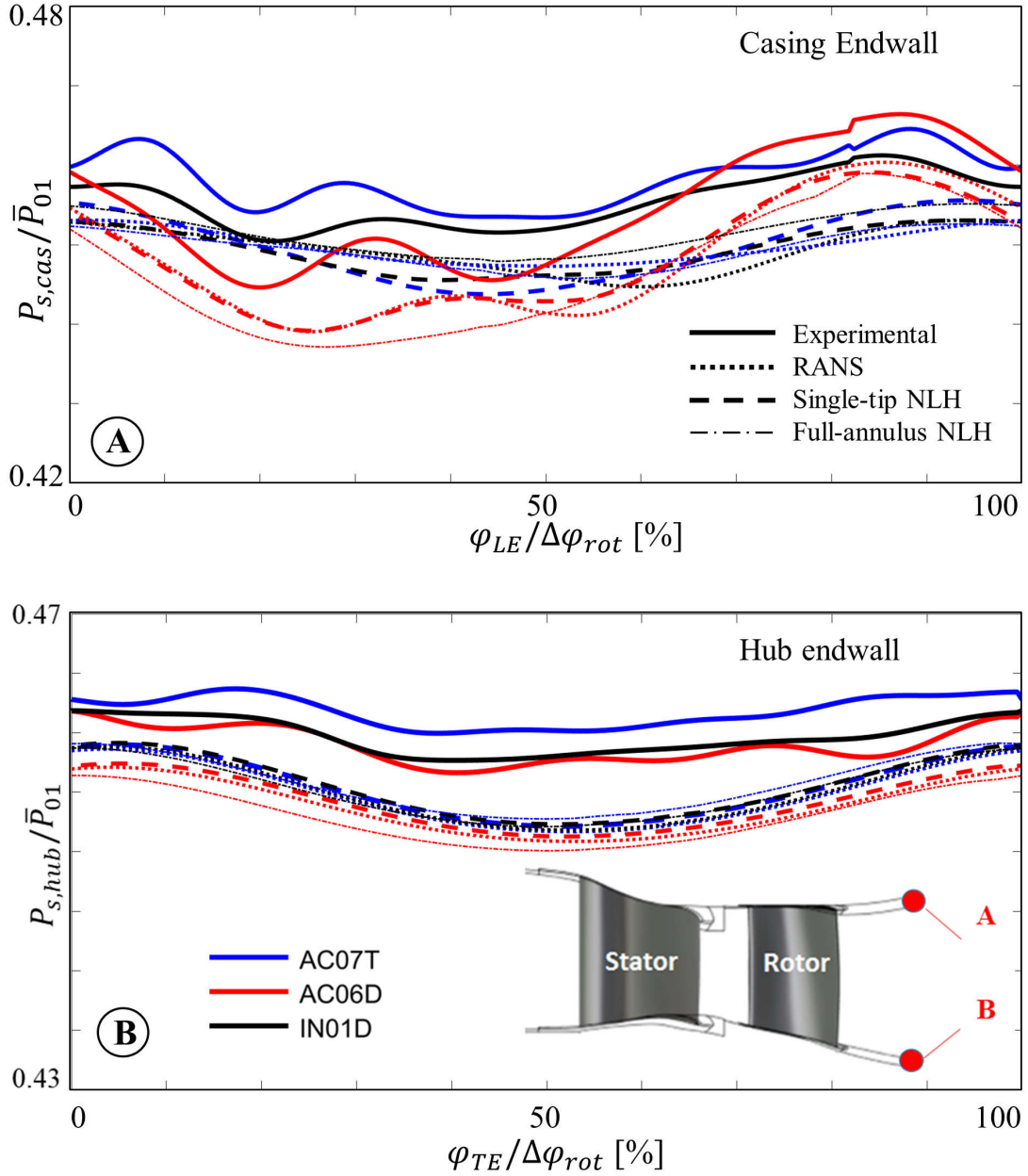


Figure 4.51: Phase-resolved pressure histories at outlet endwalls: experimental vs. numerical data

the numerical predictions at the hub endwall show a lower mean value for blade AC06D (-2 mbar) with respect to blades IN01D and AC07T. The discrepancy appears slightly more significant in the full-annulus NLH with respect to the single-tip computations. The experimental results, conversely, show a higher mean pressure level ($\sim +3$ mbar) for the carved profile with respect to the other two tip designs.

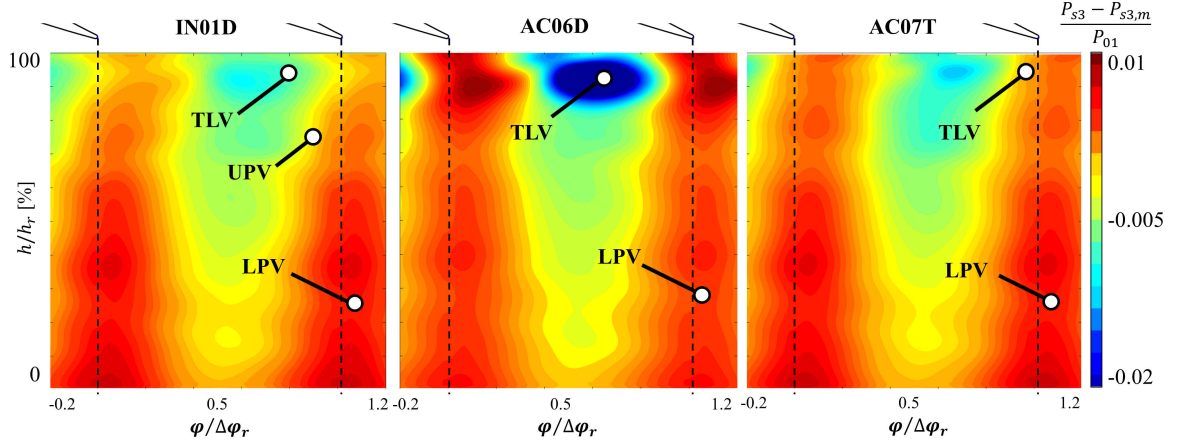


Figure 4.52: Phase-resolved static pressure contours at stage outlet endwalls (single-tip NLH)

The phase-resolved outlet static pressure field predicted by single-tip NLH computations is presented in Figure 4.52 in the rotor domain, phased with respect to the airfoil trailing edge. To highlight the potential effect of the rotor airfoils on a common base, the mean static pressure value registered at mid-span is subtracted to the predicted fluctuation of each tip design.

A low pressure region is present between 35% and 65% of the rotor pitch in the lower half of the blade span, gradually increasing in peripheral dimension with the radial coordinate. The potential field, in the lower 50% h_r is virtually unchanged for the three profiles, while substantial tip-to-tip variations are present in proximity to the outer endwall. The presence of the UPV and LPV, indicated by white dots as tracked by the vortex identification algorithm, does not seem to alter substantially the pressure field of the three tips at this distance from the rotor TE. The position of the TLV of blade AC06D, conversely, correlates well with the most significant pressure minimum within the three tips, producing the strongest near-casing fluctuation (~ 0.045 at 90% h_r) in agreement with Figure 4.51a. In blade IN01D the presence of the TLV modifies only slightly the potential field in the near-casing region, while this effect is almost negligible for blade AC07T, in good agreement with the lower strength shown by the TLV in Figure 4.49.

4.3.4.3 Relative flow quantities

The derivation of the relative flow quantities from the experimental dataset is fundamental in order to gain understanding of the rotor flow-physics and determine the

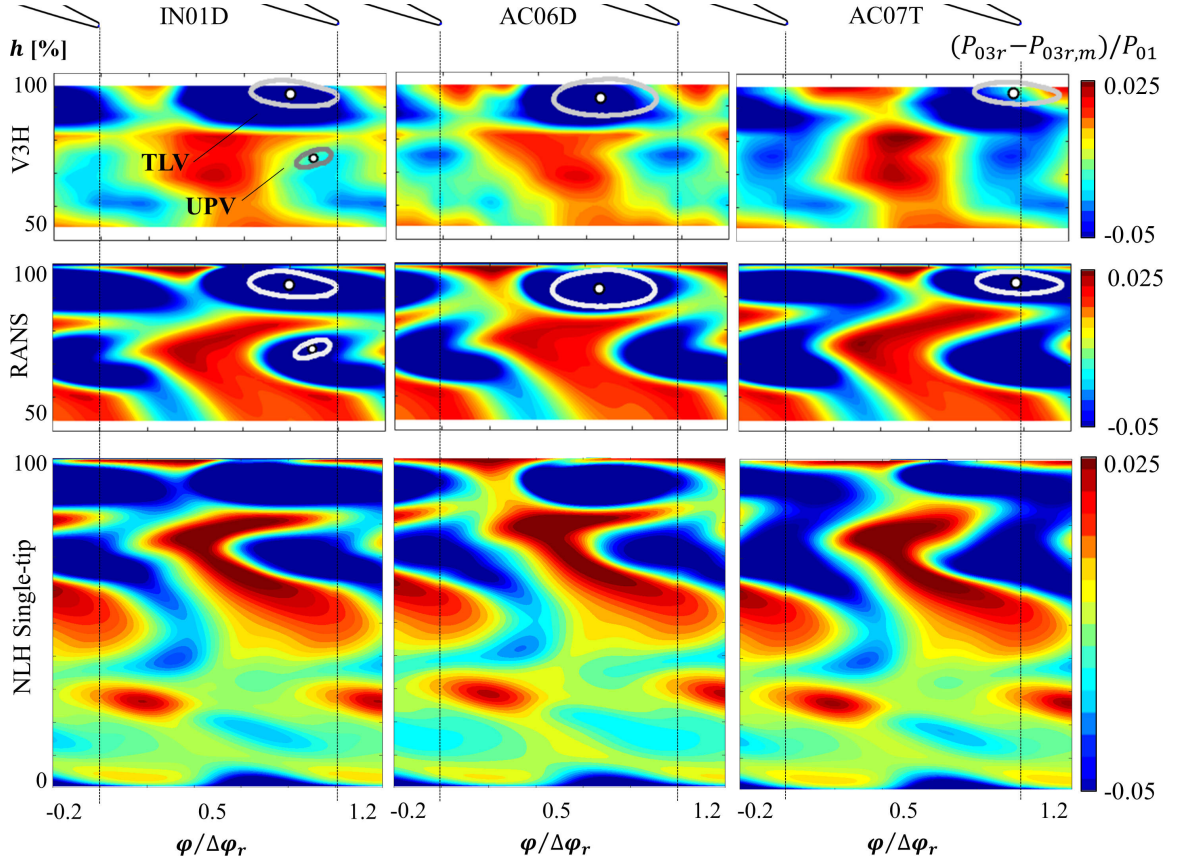


Figure 4.53: Phase-resolved outlet relative total pressure: experiments vs. CFD

validity of the numerical predictions. To this aim, high frequency, high spatial resolution measurements by the virtual three-hole-probe were interrogated in the upper 50% of the rotor span.

The absolute outlet Mach number was first determined from the unsteady total and static pressure distributions measured by the V3H probe as from Equation 4.9.

$$M_{03,exp} = \sqrt{\frac{2}{\gamma - 1} \left(\frac{p_{03,exp}}{p_{s3,exp}} \right)^{\frac{\gamma - 1}{\gamma}} - 1} \quad (4.9)$$

The radial profile of steady total temperature, measured by means of thermocouple rakes, was employed for the computation of the static temperature at the rotor outlet as from Equation 4.10. The numerical predictions were interrogated to estimate the impact of such assumption: a maximum static temperature fluctuation of 6% was detected at constant radial position, which was considered to have a negligible impact with respect to the measurement uncertainty.

$$T_{s3,exp} = \frac{T_{03}}{1 + \frac{\gamma - 1}{2} M_{03,exp}^2} \quad (4.10)$$

The absolute flow velocity at the outlet of the stage was thus computed as from Equation 4.11.

$$v_{3,exp} = M_{03,exp} \sqrt{\gamma R T_{s3,exp}} \quad (4.11)$$

Time-resolved experimental histories of flow yaw angle (axial-circumferential plane) were combined to numerically-predicted pitch angle (axial-radial plane) distributions to assess the unsteady flow direction downstream of the rotor. From the outlet velocity triangles, the relative velocity and Mach number were computed. The outlet relative total pressure distribution was therefore calculated as from Equation 4.12.

$$P_{03r,exp} = P_{s3,exp} \left(1 + \frac{\gamma - 1}{2} M_{03r,exp}^2\right)^{\frac{\gamma}{\gamma - 1}} \quad (4.12)$$

The comparison between measured and predicted distributions of outlet relative total pressure is presented in Figure 4.53. Large pitchwise gradients are shown in the near-casing region, where a low pressure patch is shown in correspondence of the TLV position. The tip-to-tip difference in terms of peripheral and radial location of the minima are overall validated by the experimental contours and consistent between the steady and unsteady CFD computations. Flow gradients of comparable intensity are visualized in circumferential direction between 60 and 80% of the blade span in the numerical contours, representing the footprint of the upper passage vortex in terms of relative pressure loss.

In order to quantify the error introduced by neglecting the outlet temperature fluctuation in the experimental methodology, the same procedure has been employed to compute P_{03r} from the RANS dataset, considering the predicted pitch-averaged total temperature profile in the computation. The resulting contours of computed relative total pressure ($P_{03r,tpa}$) are displayed in the top part of Figure 4.54. A high level of agreement is demonstrated with the extracted P_{03r} distributions for the three profiles. In the bottom part of the Figure 4.54, the quantitative difference between extracted and computed datasets is given. The regions of maximum discrepancy correspond to the temperature fluctuation peaks induced in the near casing region by the TLV and, between 60% and 80% of the blade span, by the UPV. The pressure difference is, however, within ± 2.5 mbar in the near-casing area and within ± 4 mbar in the UPV region, always below $\sim 5\%$ of the amplitude of the local deterministic pressure

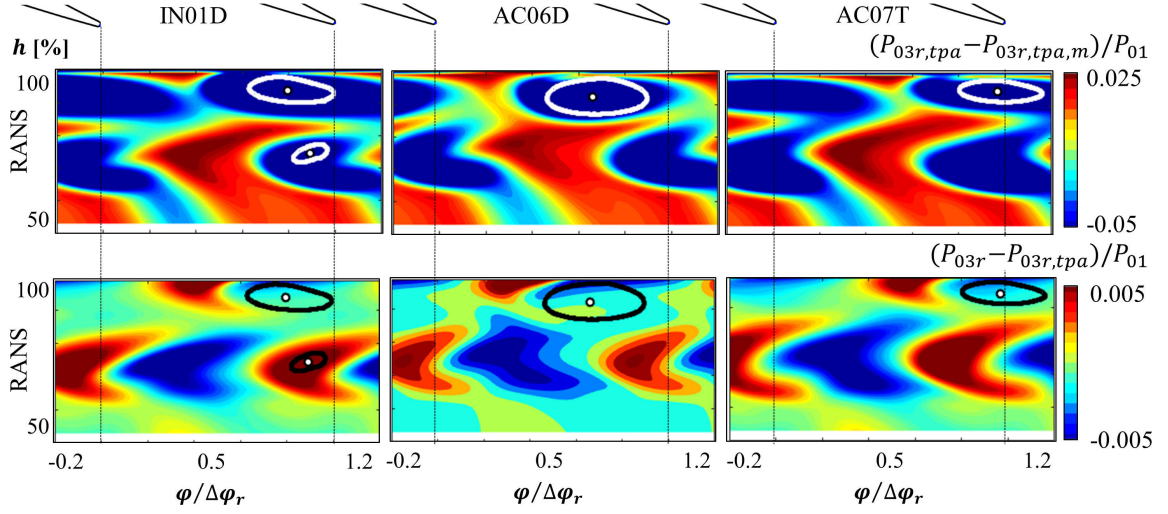


Figure 4.54: P_{03r} computed from CFD dataset using $\bar{T}_{03}$ to mimic experimental procedure (top) and quantitative comparison with CFD-extracted P_{03r} (bottom)

fluctuation.

4.4 Rotor flow structures

4.4.1 Vortex identification

In this section, the RANS predictions are exploited in combination with the vortex tracking technique to analyze the over-tip flow structures of the three tip profiles. Figure 4.55 presents, from left to right:

- Vectors of radially integrated mass-flow through the tip gap perimeter. The suction side is divided into regions characterizing the different tip flow regimes. The extension of each region is provided in terms of the normalized suction side arc length.
- Contours of tip heat transfer normalized by the area-average value of blade IN01D, combined with vector-lines of the tip surface viscous stress (surface streamlines),
- Contours of relative total pressure normalized by the turbine inlet total pressure at mid-span at several axial cutting planes, combined with tracked vortex boundaries (red line: clockwise, green line: counter-clockwise),

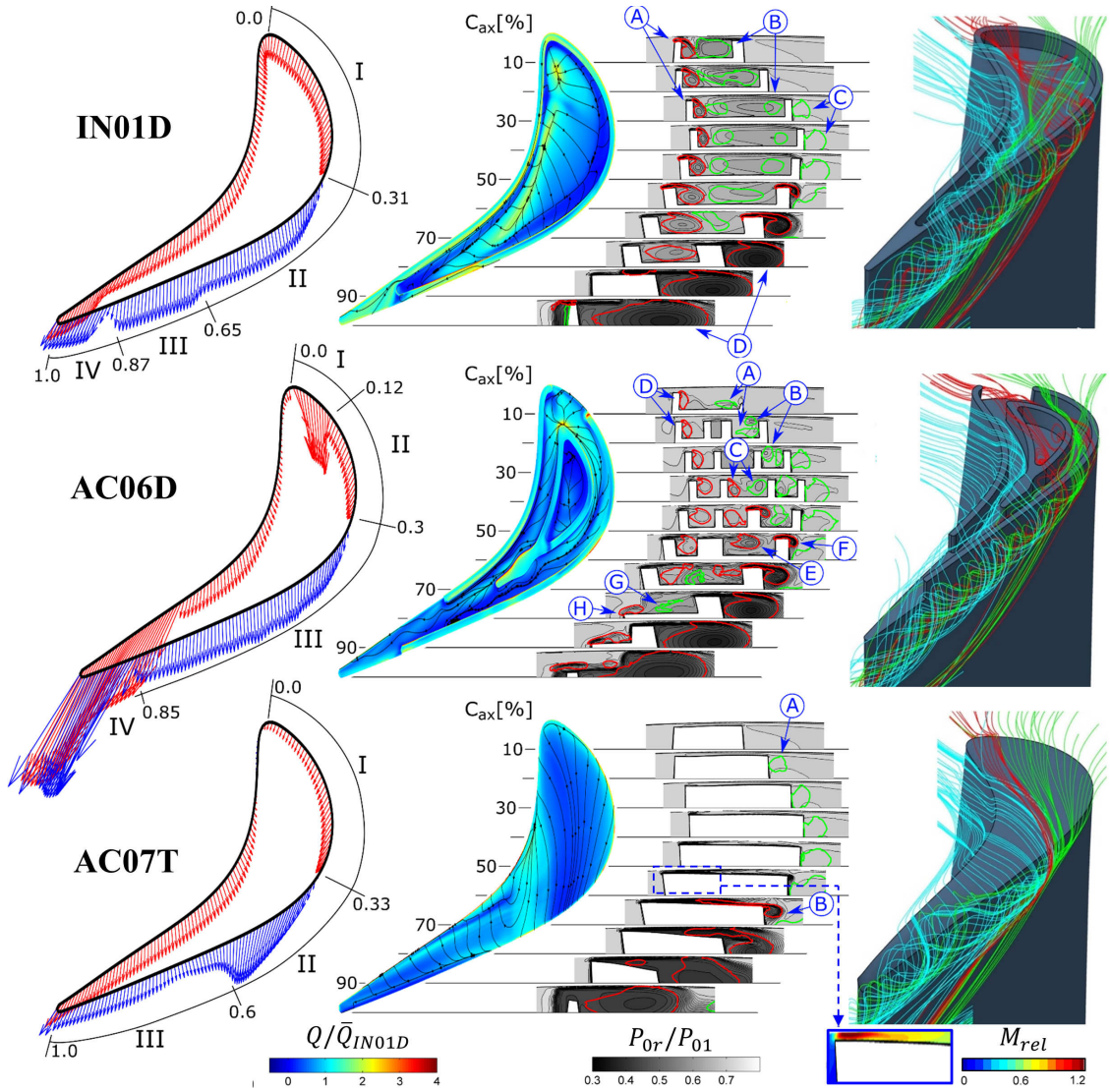


Figure 4.55: Left to right: tip leakage mass-flow vectors; tip heat flux with surface streamlines; rel. total pressure cuts with vortex boundaries; rel. velocity streamlines

- Relative velocity streamlines characterized by the tip flow ingestion location (green line: suction side, red line: leading edge, blue line: pressure side)

In blade IN01D, the flow is observed to enter from both the pressure side (PS) and the suction side (SS, Zone I), creating a region of high heat transfer upon impingement onto the cavity floor around $15\%c_{r,ax}$. The vortex tracking technique reveals the creation of a counter rotating vortex pair (markers *A* and *B*), confirmed by the reattachment line shown by the tip surface streamlines on the cavity floor. Vortex *B* is observed to stretch peripherally as the cavity becomes wider, generating a wide central part dominated by shear. This explains why the TDM detects two co-rotating

vortex cores instead of a single compact structure entraining the ingested endwall fluid. Additionally, the scraping of the casing boundary layer is identified as a counter-clockwise vortical structure, initiating the upper passage vortex (marker C) starting from $30\%c_{r,ax}$.

The impingement of the flow entering via the PS gap between 25% and $70\%c_{r,ax}$ generates a chordwise strip of high heat transfer on the pressure side of the squealer cavity. The flow remains attached towards the SS rim, it travels up the rim and then exits the gap suction side (Zone II). This regime of squealer operation is called "double-seal", as the flow has to overcome a double contraction caused by the pressure and suction side rims [103]. The flow ejected from the suction side rolls up into the tip leakage vortex (marker D).

The transition from a double to a single seal regime registers a significant increase in leakage mass flow of about 25% (Zone III). As the cavity becomes narrower and is completely occupied by vortex A , the PS-ingested fluid is directly accelerated over the two rims and by-passed to the blade SS. The end of the cavity is marked by a drop of spillage mass flow at 87% SS arc length. This is caused by the fluid entrained in vortex A that leaves the cavity by bursting radially upwards, impinging onto the outer casing wall and acting as a fluidic sealing of the tip gap. The mass flow rises again in the aft suction side, Zone IV, due to the reduced sealing effectiveness of the flat tip section in the aft part of the rotor airfoil. Here, the flow exits the gap more aligned with the suction side edge thanks to the exiting cavity jet that reduces the leakage driving pressure gradient.

As already discussed in Figure 4.39, the front rim opening of blade AC06D allows extra mass flow to be ingested in the over-tip region (Zone I), as identified by the red streamlines. Two co-rotating vortices are generated in the SS channel: vortex A is formed by the fluid ingested in the gap through the LE rim opening, while vortex B is sustained by the fluid entrained from the SS external rim (Zone II), indicated by green streamlines. Both vortices continue along the SS channel and merge at about $40\%c_{r,ax}$. As for the squealer cavity of IN01D, a vortex pair (marker C) with two counter-rotating branches develops in the central island at $40\%c_{r,ax}$ cut. The vortex signature is visible on the chord-wise reattachment line created on the inner cavity floor.

As the inner rims merge, the outlet mass-flow increases gradually throughout the rear part of the SS (Zone III). The PS channel flow is characterized by a single chord-wise vortex (marker D) entraining the flow entering the external PS rim (blue streamlines). The vortical structure travels downstream the PS channel and progressively overcomes

the inner side rims of the central cavity to finally emerge into the rotor passage starting from $60\%c_{r,ax}$. At each rim cross, a clockwise vortex is generated (markers E and F). At the meeting of the PS and SS channels, the interaction of the co-rotating vortices (marker D and E) forms a counter-rotating vortex (marker G).

The aft opening of the PS rim causes a sudden increase of mass flow swallowed into the tip gap. A local flow separation is generated over the tip surface, forming a clockwise vortex (marker H) which progressively grows in the over-tip region up to the blade TE. A significant increase in leakage mass-flow is detected in Zone IV, corresponding to the aft-SS rim aperture.

The tip gap flow of the carved blade AC07T differs considerably from the squealer-

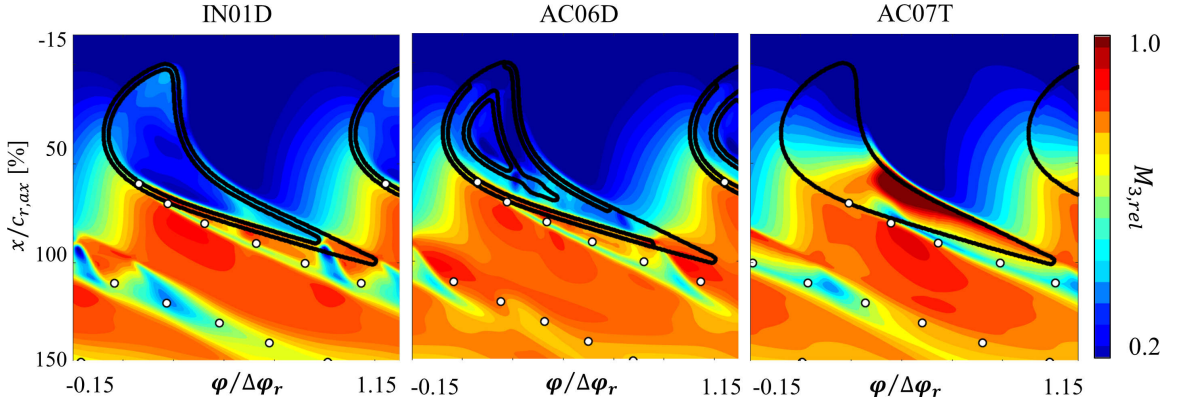


Figure 4.56: Phase-resolved contours of mid-gap relative Mach number

type profiles. Despite a weak profiling imitating a squealer design, the shape of blade AC07T resembles rather a flat tip in the frontal part of the rotor.

The flow is seen to cross the blade width in nearly parallel streamlines from the pressure side towards the suction side. Due to the tight running clearance, only a small separation region is created in proximity of the pressure side edge, highlighted by the thin reattachment line in proximity of the blade LE. The early formation of a scraping vortex (marker A) is captured on the pressure side of the blade, caused by the low-momentum fluid entrained from the casing boundary layer due to the reduced running tip clearance.

The leakage mass flow of blade AC07T gradually grows in Zone II until 60% of the suction arc length. At this point, the tip leakage mass flow rapidly drops because of the blockage generated by the over-tip flow separation detected in Figure 4.39. The separation region extends towards the SS and becomes connected with the tip leakage vortex (marker B) as illustrated by the vortex boundaries.

From $50\%c_{r,ax}$ onward, the fluid entering the over-tip gap reaches sonic conditions

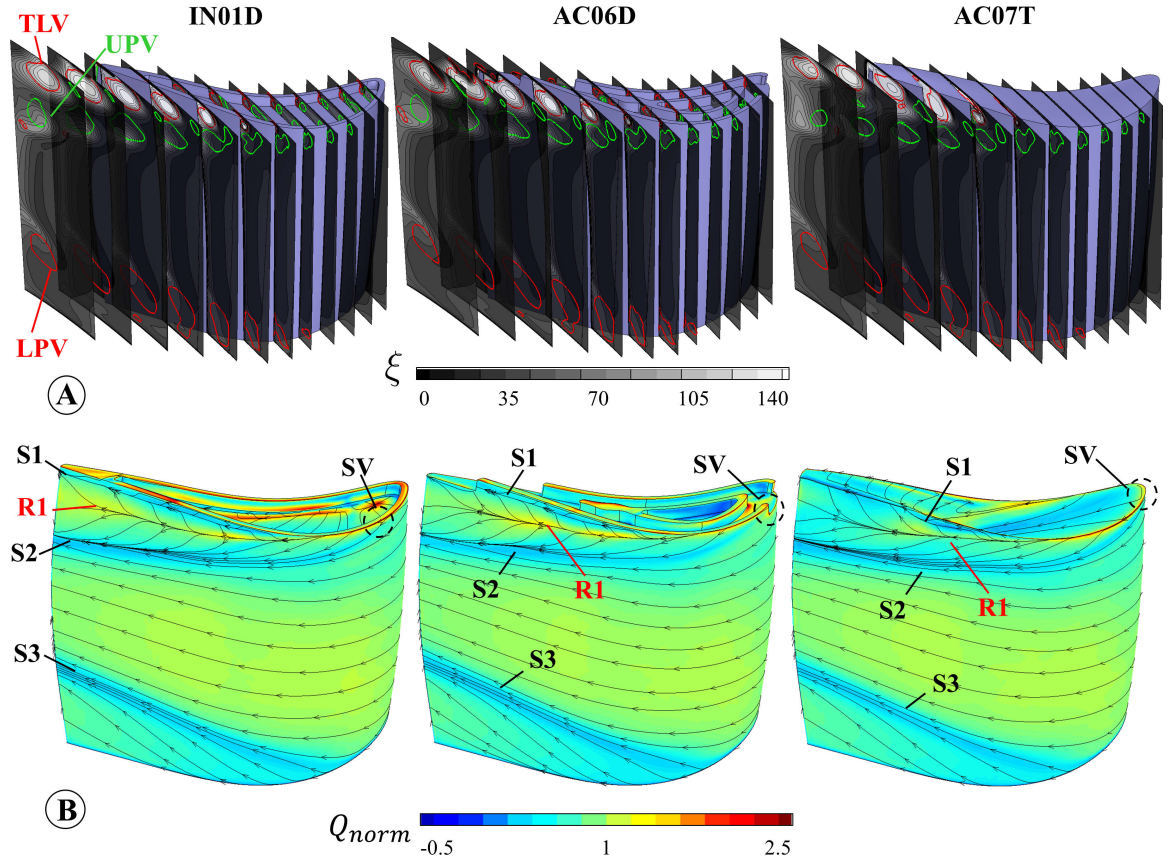


Figure 4.57: Vortex boundaries by TDM (red: clockwise, green: counter-clockwise), rotor pressure loss contours (a); heat flux contours, surface streamlines (b)

near the pressure side entrance (detail in Figure 4.55) and is further accelerated to supersonic speeds by the divergent passage. Due to the small width-to-gap ratio in the aft part, the opening passage acts as a choked knife-edge, effectively reducing the mass flow in Zone III.

The relative Mach number contours at the mid-gap plane of the three profiles are presented in Figure 4.56. Blade AC07T is the only tip design that accelerates the leakage stream to sonic conditions, while the central cavities of IN01D and AC06D show comparatively lower flow velocities ($M_r \sim 0.3 - 0.4$). The tracked trajectories of the TLV, show how the core of the vortical structure correlates well with a low-Mach region which propagates from the blades SS. In blade AC06D, the by-pass jet leaving the tip from the rear-SS aperture interacts with the tip leakage vortex and is observed to enhance the mixing downstream of the rotor TE.

The evolution of the vortices as captured by the TDM method is shown in Figure 4.57a in a series of axial planes along the rotor chord. Additionally, the rotor pressure loss coefficient, defined in the rotating frame of reference as from Equation 4.13, is

presented on the same planes in form of contour plots.

$$\xi(\mathbf{z}) = \frac{\bar{p}_{02b,r} - p_{0,r}(\mathbf{z})}{\bar{p}_{0,r}(\mathbf{z}) - \bar{p}_s(\mathbf{z})} \quad (4.13)$$

The most significant vortical structures in the rotor flow field are the tip leakage vortex (TLV) and the upper (UPV) and lower (LPV) passage vortices.

The UPV, rotating in the counter-clockwise direction (green boundary), is initiated at $20\%c_{r,ax}$ for blade AC07T and at $30\% c_{r,ax}$ for the squealer-type geometries. The early initiation of the UPV in blade AC07T is caused by the stronger scraping of the boundary layer associated with a tighter tip clearance. Conversely, the generation of the TLV, rotating in clockwise direction (red boundary), is delayed for the contoured tip $70\%c_{r,ax}$ with respect to blades IN01D and AC06D ($60\%c_{r,ax}$). The interaction between the UPV and TLV is clearly visible. The TLV pushes the UPV away from the suction side surface, promoting the generation of a small additional vortex at $80\%c_{r,ax}$ (co-rotating with the UPV), which however dissipates rapidly with distance. The LPV is first detected at $50\%c_{r,ax}$ for all the three tips and shows little or no influence of the tip design. The relative total pressure loss cores, denoted by the lighter regions, correlate well with the vortex locations.

Figure 4.57b shows the predicted heat flux contours over the three rotor airfoils completed by surface streamlines. Reattachment and separation lines are here denoted with markers R and S respectively. The TLV extends between the reattachment line R1 and the separation line S1, in good agreement with the findings presented in [104]. In the case of blade AC07T, the separation line S1 is only visible over a short distance of the suction side between 40% and 70% of rotor axial chord, merging then with the tip SS edge in the aft part of the blade, where a major over-tip flow separation (eventually merging with the TLV) has been detected.

The UPV of blade AC07T forms earlier than for the squealer-type geometries, as indicated by the SV mark in the figure. The UPV occupies the region between R1 and separation line S2 whereas the LPV is located below the separation line S3. The suction side heat transfer shows the signature of the vortex interaction: stripes of high heat transfer are seen along the impingement lines and bands of lower magnitude along the lift-off lines. The baseline squealer blade exhibits the largest impact of the vortex system on the heat transfer, with more intense peaks generated at the vortex separation and reattachment regions. Conversely, the heat flux peak correlated to the reattachment line R1 is the lowest for blade AC07T, whose TLV shows the lower intensity and cross-section among the three blades.

The generation and development of the TLV, UPV and LPV is presented schematically in Figure 4.58 for the three tip profiles, while Figure 4.59 shows the evolution of the vortex tracking parameters along the rotor axial chord. As previously discussed in

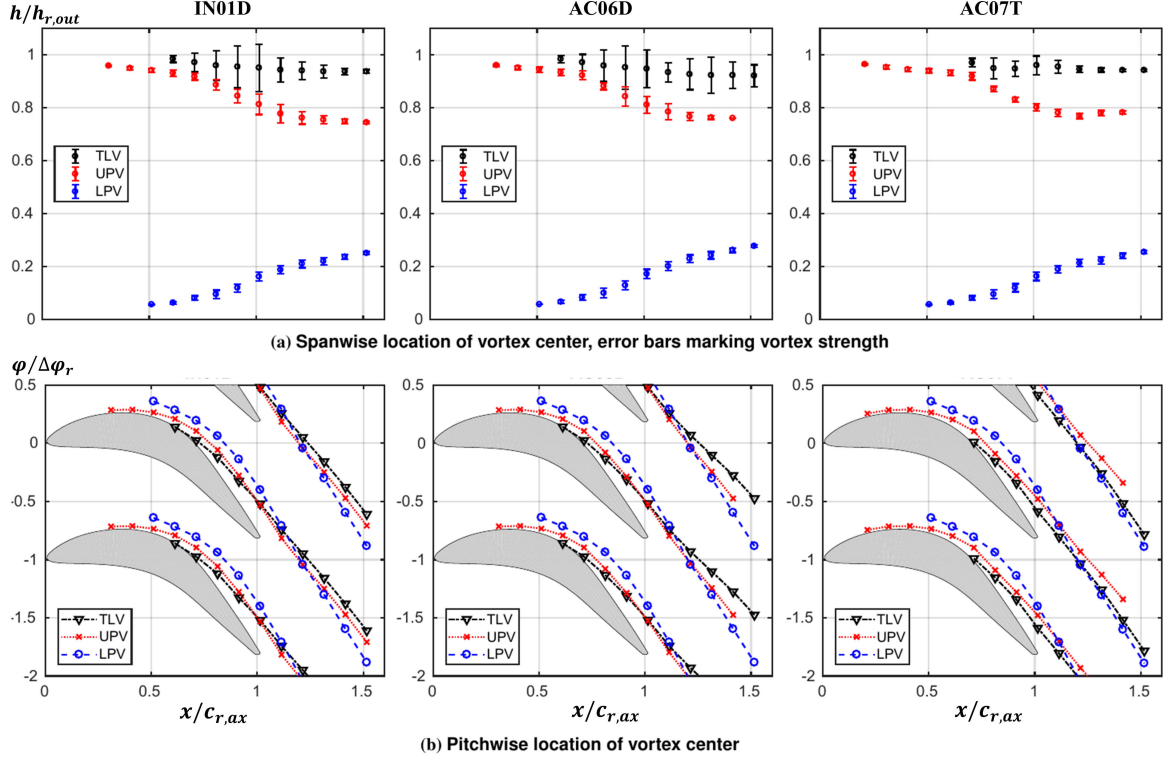


Figure 4.58: Vortex tracking per blade geometry: spanwise (a) and pitchwise (b) position of the vortex center for the three blade tips

Fig.4.57, the UPV exhibits an earlier formation in the contoured blade tip ($20\%c_{r,ax}$ for blade AC07T and at $30\%c_{r,ax}$ for the squealer-type geometries), whereas its trajectory is weakly influenced by the tip design. The UPV moves radially inwards with the axial distance, reaching 75% of the blade span at the rotor outlet plane. The UPV of blade AC07T penetrates less into the blade channel ($+5\%$ of the blade span), but moves further peripherally into the blade passage ($+10-15\%$ of the rotor pitch). The intensity of the UPV is instead more sensitive to the blade design, as shown in Figure 4.59-c. The UPV of the squealer-type blades grows rapidly starting from $80-85\%c_{r,ax}$ whereas the UPV associated with the carved tip maintains a relatively low intensity with axial distance. Downstream of the rotor, the UPV of blade AC06D and IN01D loses quickly strength. At $150\%c_{r,ax}$ the UPV of blade AC06D and AC07T has eventually mixed-out, while that of blade IN01D has almost disappeared.

The LPV, arising at $50\%c_{r,ax}$, presents a similar trajectory for the three tip profiles, apart from a slight increase in vortex strength and spanwise position in blade AC06D,

in concordance with the total pressure contours shown in Figure 4.49. The intensity of this vortical structure is, globally, weaker with respect to the UPV, which however shows a faster mixing process downstream of the rotor trailing edge.

The tip leakage flow is ejected from the tip SS at $60\text{-}70\%c_{r,ax}$, where it initiates

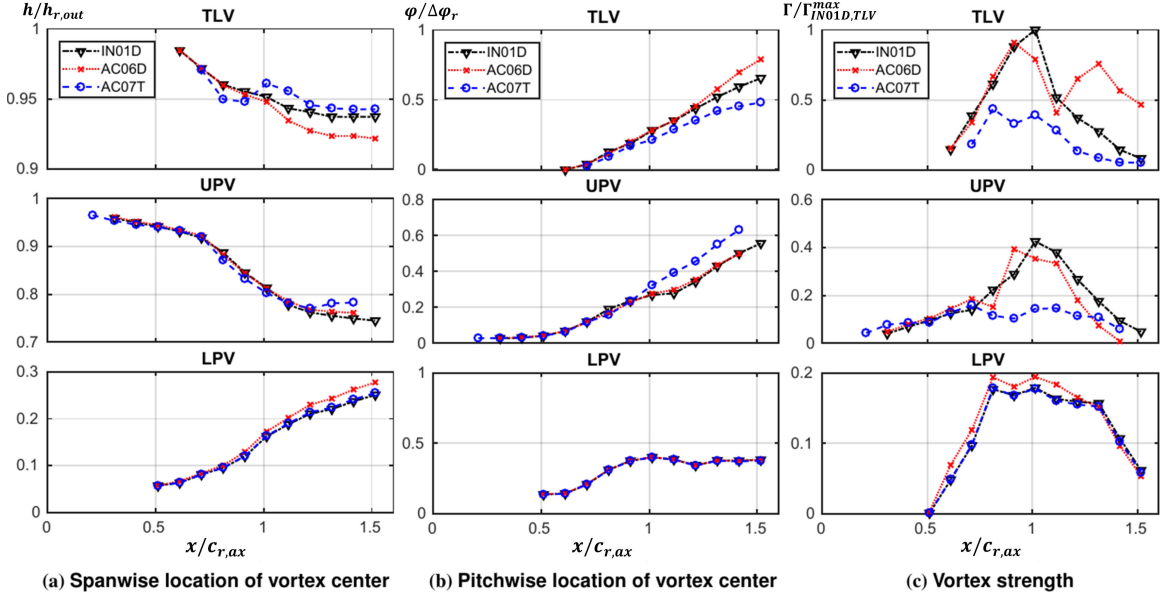


Figure 4.59: Vortex tracking per vortex parameters: spanwise position (a), and pitchwise position with respect to the blade suction side (b) and vortex strength (c)

the TLV forming a counter-rotating pair with the UPV. The TLV supplies energy to the UPV and both vortices grow while traveling towards the rotor trailing edge. The maximum strength of the TLV reaches about two times that of the UPV and five times that of the LPV (in proximity of the rotor trailing edge). The TLV associated with the contoured tip is generally two or three times weaker than that of the squealer-type geometries along the entire vortex path. The two squealer geometries show a similar evolution of the TLV upstream of the opening of the AC06D rim at $90\%c_{r,ax}$. Downstream of this point, the TLV of blade IN01D continues to grow up to the rotor TE, where the TLV of the three tips start to dissipate.

Downstream of the rotor, the TLV of blade AC06D penetrates deeper into the blade channel and moves the furthest into the blade passage with respect to the TLV of blade IN01D and AC07T. The TLV of the contoured tip, conversely, is the one that travel less both in terms of radial and peripheral direction among the three tips. The TLV of blade AC06D reaches its peak intensity where the suction side rim opens and shortly after starts dissipating. However, after the initial dissipation, the TLV exhibits a large rise in intensity between 110% and $130\%c_{r,ax}$. The growth of the TLV

downstream of the rotor is likely caused by the inviscid interaction with the strong tip leakage jet bursting into the rotor passage through the full open tip of blade AC06D. A graphical representation of the proposed TLV-tip jet interaction is illustrated in Figure 4.60. This schematic, drawn to scale using the measured TLV trajectory and

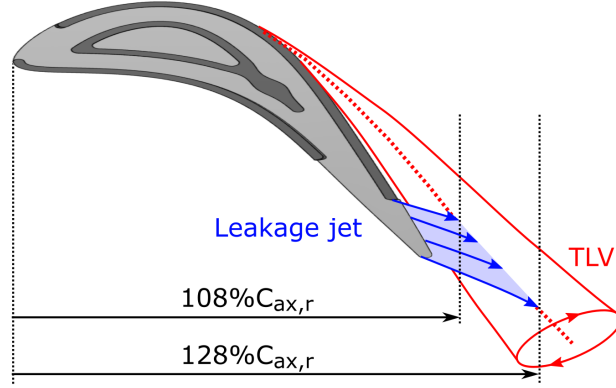


Figure 4.60: Interaction of the tip leakage jet with the TLV. The position of the TLV and the jet velocity vectors are drawn to scale with the measured values

the tip jet speed and direction, supports the hypothesis that the high-speed leakage jet reinforces the TLV core when the two flow structures interact at about 10%-30% $c_{r,ax}$ downstream of the rotor TE. The effect of the tip jet-TLV interaction is also visible in the TLV trajectory, which is shown to deviate substantially towards the rotor passage at 110-115% $c_{r,ax}$.

4.4.2 Flow random unsteadiness

The experimental flow random unsteadiness (σ), computed on the phase-lock averaged measurements, is presented in this section and correlated with the vortical structures tracked for the three tip profiles. The random unsteadiness is a quantification of the fluctuations in the flow which are not correlated with the blade passing frequency and, hence, expected to be amplified in proximity of the vortices location. Figure 4.61 shows experimental contours of $\sigma(P)$ obtained at each rotor phase position from the time-resolved pressure measurements at the over-tip rotor casing (top) and outlet plane (bottom). White dots indicate the trajectories of the tip leakage vortex (casing and outlet planes) and upper and lower passage vortex (outlet plane) computed by the TDM.

The casing plot is extended up to 150% $c_{r,ax}$ to demonstrate the correct phasing of the tracked TLV at Plane 03. The vortex boundaries are additionally indicated at the

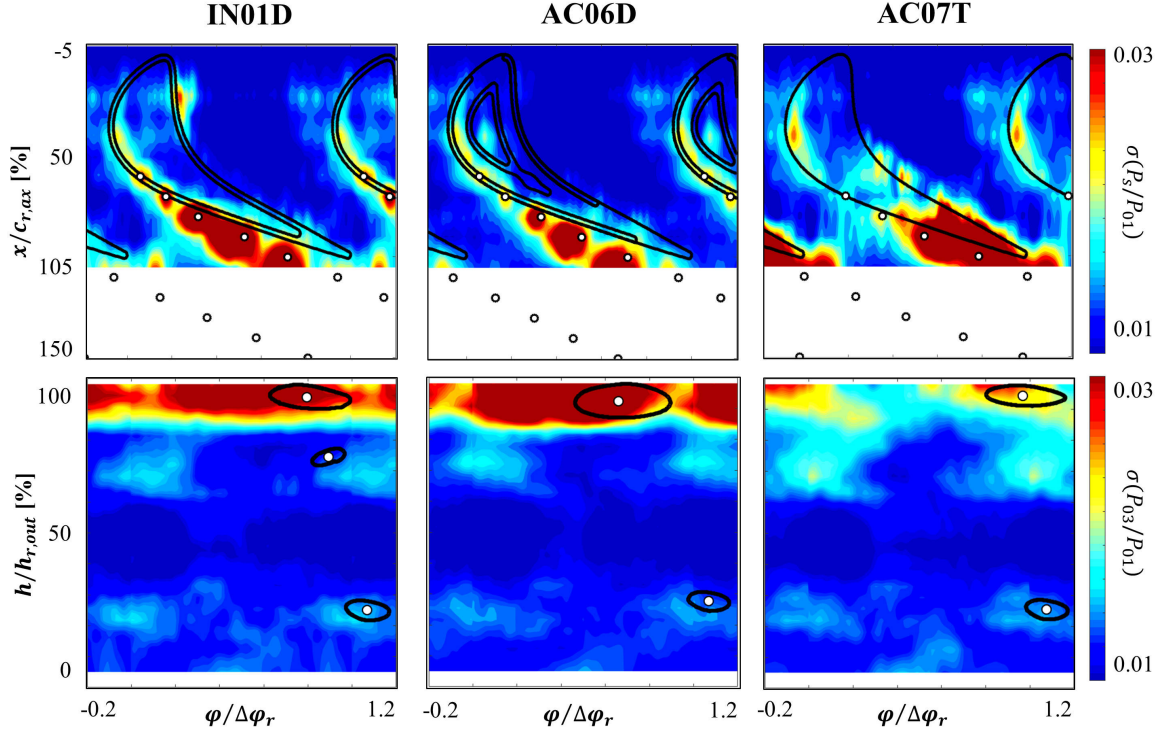


Figure 4.61: Experimental phase-resolved random unsteadiness: rotor casing static pressure (top) and stage outlet total pressure field (bottom)

rotor outlet by solid lines. In general, the pressure field shows low random unsteadiness levels at the over-passage casing and between 30 and 50% of the blade span at the rotor outlet, with fluctuation levels not exceeding 1% of P_{01} for the three profiles. Tips IN01D and AC06D show significant stochastic fluctuations in correspondence of the TLV trajectory at the rotor casing, resulting in a near-tip region at the rotor outlet (85 to 100% of the blade span) with unsteadiness levels up to 5% of P_{01} in correspondence of the TLV location. Minor $\sigma(P_s)$ peaks are visualized over the SS rim of the squealer type profiles between 30 and 70% of the rotor axial chord, indicating an early ejection of tip flow towards the blade SS anticipating the formation of the TLV.

The heat transfer RMS measured at the rotor casing is shown, for comparison, in Figure 4.62. Contrarily to the pressure field, high mean $\sigma(Q)$ is shown over the rotor passage of the three tips, with stochastic fluctuations levels close to 30% of the average heat flux over blade IN01D. The maxima of $\sigma(Q)$ reach 50% Q_{IN01D} over the TLV path of the squealer-type profiles. Stochastic fluctuation peaks are visualized over the PS rim of blades IN01D and AC06D, where the flow entering the gap is locally diverted towards the turbine casing.

A localized peak in $\sigma(P_s)$ is shown at the SS of the contoured tip between 30 and 45% of the rotor axial chord, only partially correlated with a slight fluctuation increase in the heat transfer contours. This fluctuation is indeed motivated by cold flow entering the gap at the blade suction side, as shown in Figure 4.39, and being subsequently re-ejected in the near-SS passage. Despite the tight operative tip clearance of blade AC07T and the relatively low intensity of the TLV produced by this blade, the over-tip separation occurring in the aft tip region generates large random fluctuations of pressure and heat flux at the rotor casing, exceeding in extension and intensity the TLV of the squealer tips. Downstream of the rotor trailing edge, however, the highly turbulent flow is quickly mixed out. As a result, at the outlet plane the near-tip $\sigma(P_{03})$ of blade AC07T is the lowest among the three profiles.

The presence of the UPV and LPV at the stage outlet is highlighted by periodic

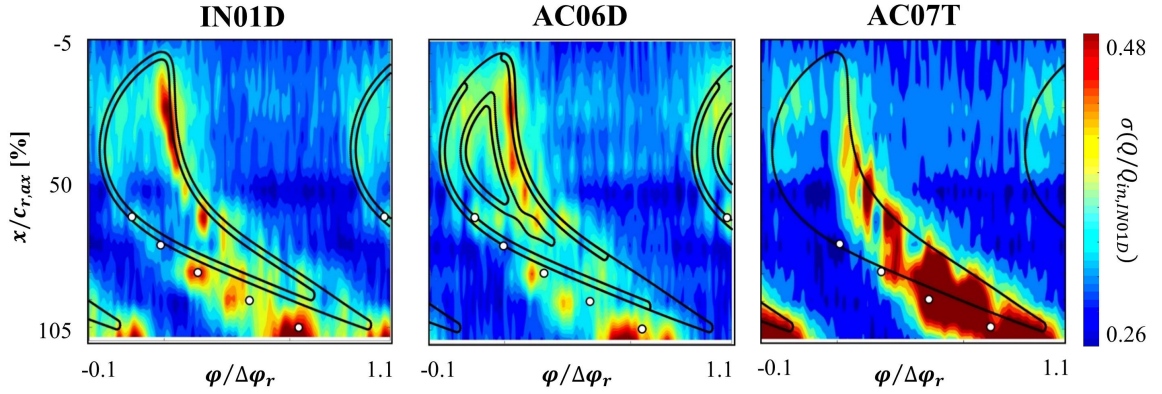


Figure 4.62: Experimental phase-resolved random unsteadiness: rotor casing heat transfer field

peaks of $\sigma(P_{03})$ traveling at 25 and 75% of the blade height. At this axial location, however, the position of the UPV is only available for blade IN01D, as the residual vorticity of this structure was below the TDM threshold for the remaining tips. A slight peripheral shift ($\sim 5 - 10\%$ of the rotor pitch) is observed here between the experimental $\sigma(P_{03})$ peak at $75\%h$ and the numerical UPV core.

4.5 Blade rows interactions

4.5.1 Vane flow contribution

4.5.1.1 Rotor Casing

Figure 4.63 presents the distributions of maximum pressure variation ($\Delta P_s/P_{01}$, intended as peak-to-peak amplitude) on the ensemble-averaged pressure signals ob-

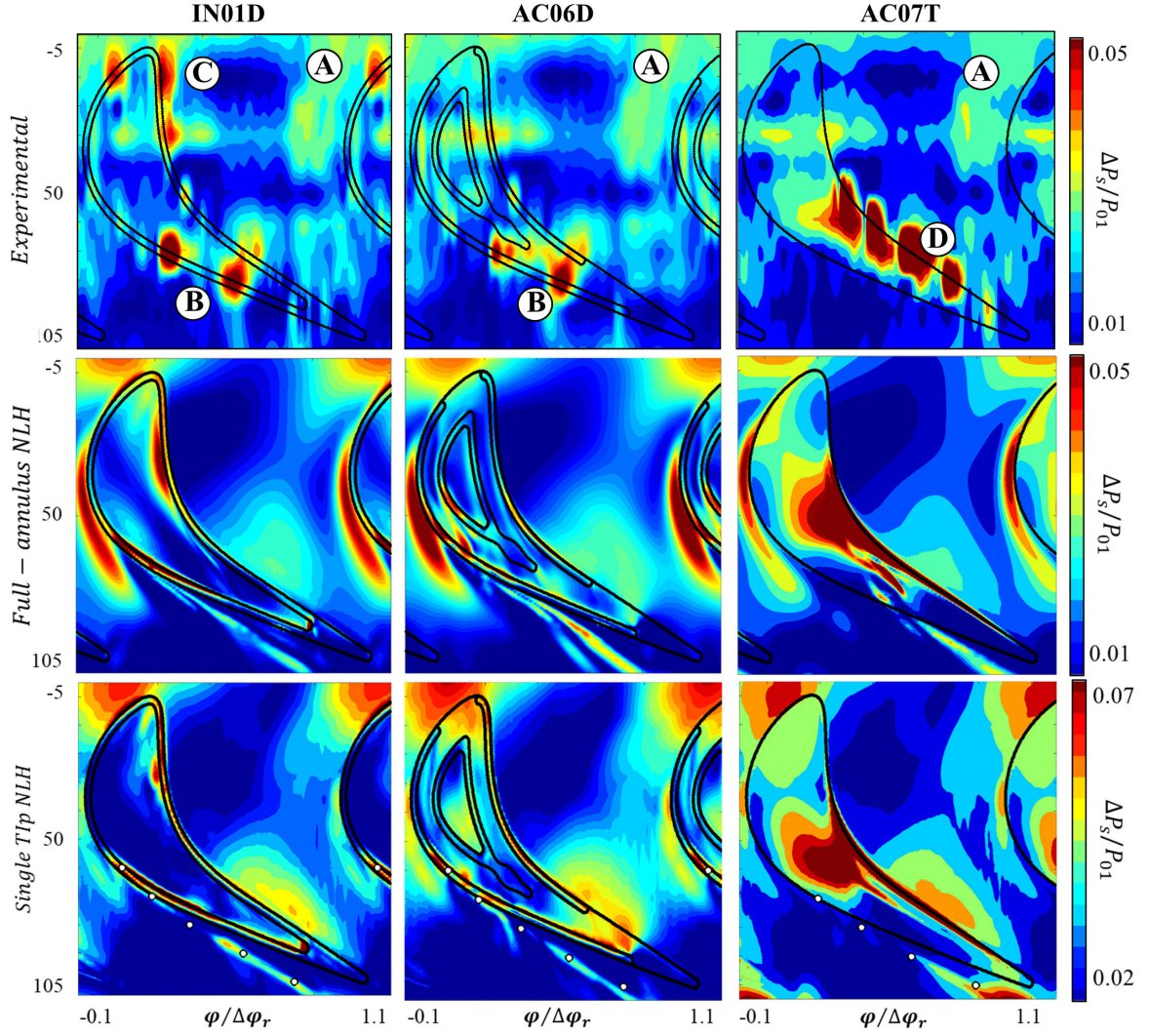


Figure 4.63: Phase resolved casing static pressure variability along vane pitch: experimental and numerical results

tained from the circumferential average of the six casing sensors (top). Single-tip (bottom) and full-annulus (center) NLH predictions are processed accordingly to retrieve the vane-induced pressure variation as the tips travel along a full NGV pitch. The contours represent the maximum variation of casing static pressure registered in the rotor domain as the airfoils travel along a full vane pitch and are instrumental in quantifying the effects of the vane-rotor interaction on the over-tip pressure field. Numerical and experimental results confirm that the vane clocking has negligible impact on the passage flow in proximity of the tips front pressure side ($\Delta P < 2\%$ of P_{01}). The effect of the blade row interaction is strong ($\Delta P \sim 4\text{--}5\%$ of P_{01}) in the inter-row gap and in the early 35% of the rotor axial chord of the rotor passage adjacent to the blade suction side (marker A) due to the NGV potential field. A local

peak of pressure variability, matched by the FA-NLH results, is visualized over the front pressure side rim of blade IN01D (marker C), where the vane-relative position modulates the ingestion of fluid in the over-tip area.

The vane interaction is at its highest ($\Delta P > 6\%P_{01}$) in the aft over-tip region of blades IN01D and AC06D, between 65% and 80% of the rotor axial chord (marker B). At these axial stations, the deterministic pressure fluctuation in the rotor frame amounts to 22-25% P_{01} , as shown in Figure 4.38. The reason for this increased over-tip flow variability is the vane-driven pressure variation at the pressure side of the blades, particularly evident in the NLH predictions, which modulates the PS-SS flow by-pass. The tracked TLV core, indicated by white dots in the single-tip NLH results, correlates with a local increase of flow variability, revealing a slight influence of the vane-relative position on the vortex locus.

Large stator-dependent interactions ($\Delta P > 8\%P_{01}$) are observed on a wide portion of the over-tip gap of blade AC07T, between 50% and 80% of the rotor axial chord (marker D), where the over-tip flow reaches sonic condition (as confirmed by Fig. 4.56). This flow variability maximum is confirmed by both single-tip and full-annulus NLH predictions.

The effect of the vane-rotor interaction on the heat transfer field is presented in

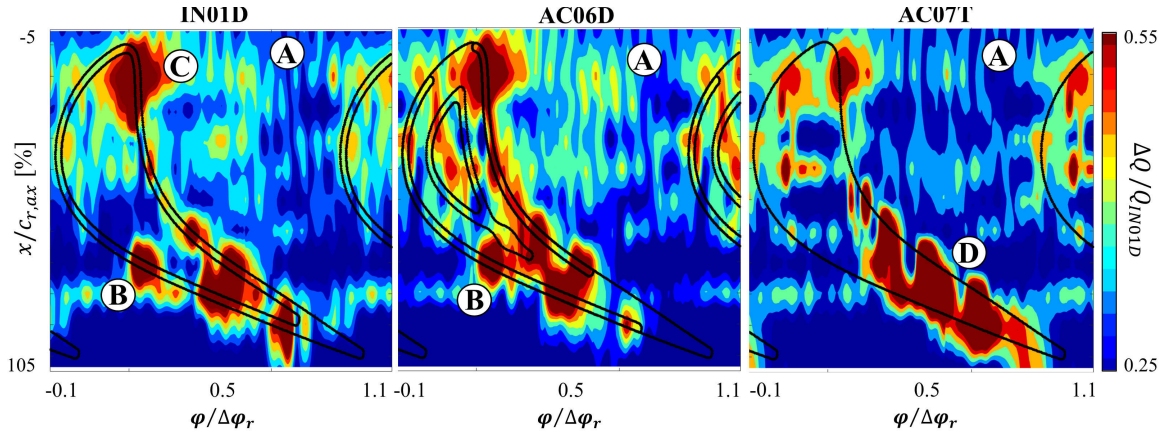


Figure 4.64: Phase-resolved casing heat flux variability along vane pitch

Figure 4.64 as ΔQ normalized by the average heat flux computed over a full vane pitch between -5% and 105% $c_{r,ax}$ of blade IN01D.

The measurements show that the vane pitch position affects significantly the early pressure side of all three blades (marker C), where the vane potential field modulates the flow ingestion inside the tip gap, with local periodic variations higher than 55% Q_{IN01D} . As demonstrated in Figure 4.43, this region is characterized by a local

peak of recovery temperature motivated by the work process undergone by the over-tip flow. Hence, this peak in ΔQ is to be ascribed to a vane-periodic change in flow turning, which in turn modulates the local adiabatic wall temperature.

The largest heat transfer variations with vane indexing are found on the over-tip gap of the three blades in the same region where the vane-induced pressure variations were also at their highest, that is the direct by-pass area for the squealer profiles (marker B) and the rear over-tip surface of blade AC07T (marker D). These regions correlate well with high total temperature zones highlighted in Figure 4.43, indicating a higher vane-dependent effect on the flow turning than on the casing convection mechanism.

4.5.1.2 Stage outlet

Figure 4.65 presents the distributions of maximum vane-dependent pressure variation ($\Delta P_{03}/P_{01}$) on the ensemble-averaged pressure histories at the stage outlet. The experimental dataset (top), recorded by the fast-response pressure probes, is presented in comparison to flow variability predictions by the full-annulus (center) and single-tip NLH computations.

These plots indicate the vane impact on the outlet pressure field as a function of the rotor blade position. The near-casing region, which in Figure 4.49 shows TLV-correlated fluctuations up to $\pm 3 - 4\%P_{01}$ ($\sim 60 - 80\text{mbar}$ peak-to-peak amplitude), appears particularly affected by high flow-variability spots. The vane interaction reaches peaks of $\sim 50\text{mbar}$ for blade IN01D (marker A) and $\sim 60\text{mbar}$ for blade AC06D (marker B) in correspondence of the TLV position. This indicates a significant modulation of the TLV strength by the vane-relative position, which affects the blade tip loading (as shown in Fig. 4.63) and, thus, the leakage mass flow. The single-tip NLH predictions seem to capture better the ΔP maxima, while the a higher amount of mixing is shown in the full-annulus NLH results, probably due to the coarser grid adopted to mesh the rotor domain.

Blade AC07T shows, conversely, a lower near-casing flow variability in the experimental dataset, in concordance with a weaker TLV with respect to the other blades. The single-tip NLH predictions, in this case, tend to overestimate the vane interaction, in concordance with an overestimation of the TLV pressure fluctuation amplitude reported in Figure 4.49.

Lower levels of flow variability, up to $\sim 3\%P_{01}$, are detected in two regions of the core flow, between 55% and 65% (marker D) and between 20% and 50% of the rotor blade span (marker E). In both cases, the flow measurements show a considerably milder variability with respect to the NLH predictions. The single-tip computations,

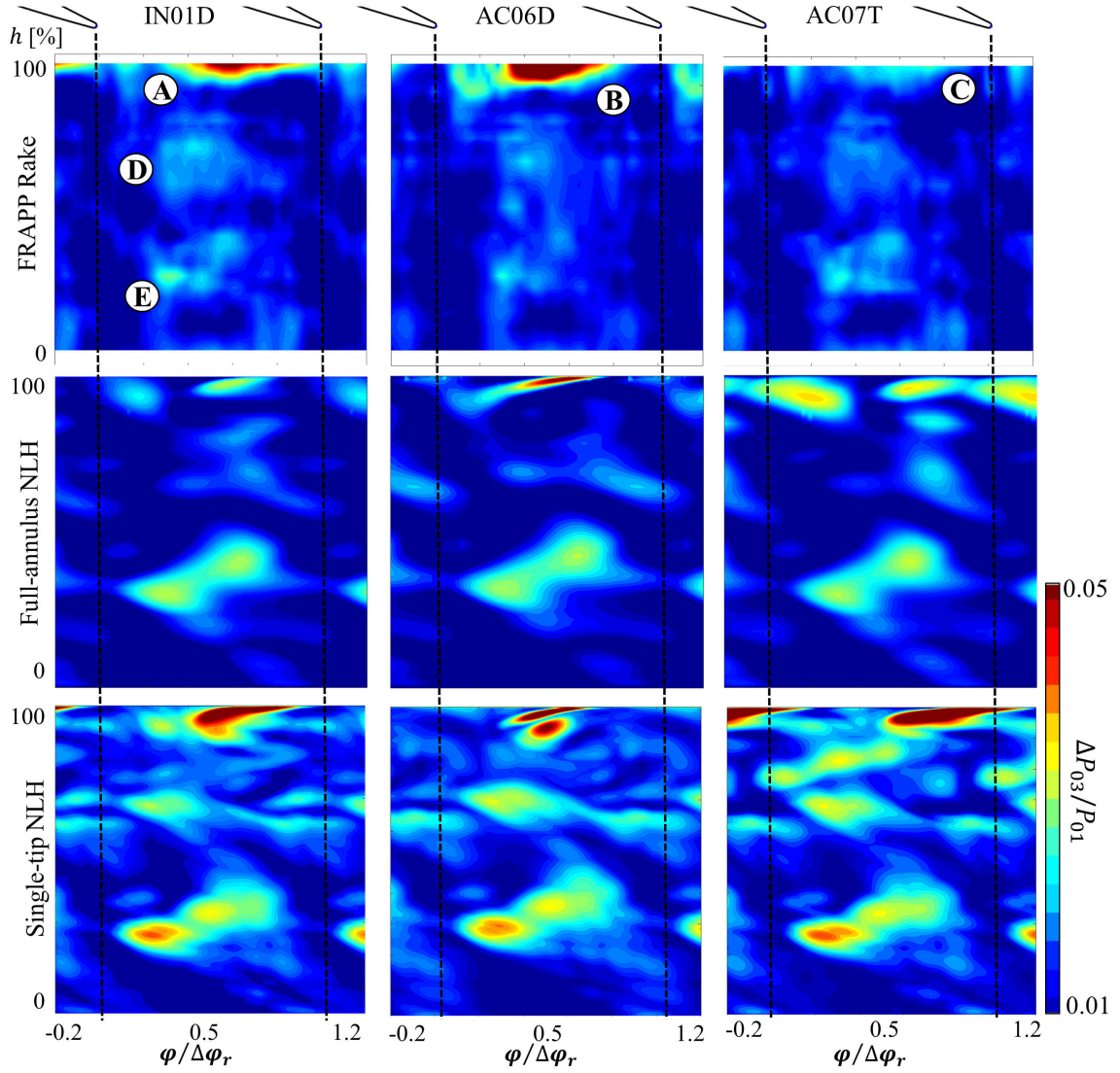


Figure 4.65: Phase-resolved outlet total pressure variability along vane pitch: experimental vs. numerical data

in particular, tend to overestimate the flow variation amplitude by more than 50% of the measured value, with peaks of $\pm 4.5\%P_{01}$ reported at $20\text{--}25\%h_r$.

4.5.2 Modal analysis

A clear identification of the main sources of flow unsteadiness within the turbine stage is critical in order to gain a full understanding of the time-varying flow field and of the interactions between different blade rows. The flow analysis performed in the rotating and stationary domain allowed to separate the rotor-mean phase-resolved flow field from the time-average vane contribution.

In this section, the study pursues a more detailed analysis of the blade rows interactions impact on the correlated flow unsteadiness in the numerical and experimental results. While instantaneous flow representations can only give a visual indication of the degree of periodicity of different flow structures developing in the stage, a flow study in the frequency domain is pursued here to obtain a quantitative assessment of the spatial and temporal modes of the HPT flow [105].

4.5.2.1 Modal decomposition theory

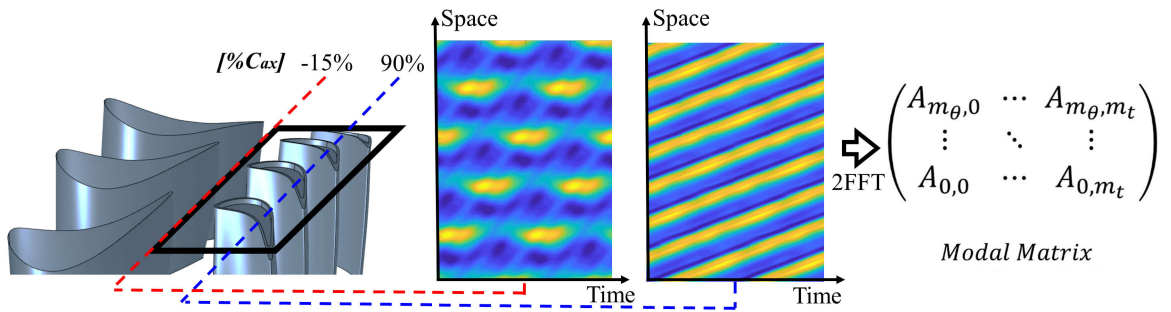


Figure 4.66: Modal decomposition of casing static pressure field

The modal analysis is a data reduction procedure which allows to decompose the correlated unsteadiness of the turbine flow and assess the modulation mechanisms which generate the periodic flow structures that exist at the different investigation locations.

In the example presented in Figure 4.66, the modal decomposition is performed over the rotor casing static pressure represented in the space-time domain at two axial stations along the rotor axial chord. The vertical axis refers to the circumferential position normalized by the vane pitch. The horizontal axis represents the temporal coordinate normalized by the rotor blade passing period. The flow evolution in both the spatial and temporal domains is therefore resolved. Vane-dependent periodicities, which do not vary their position in time, appear here as horizontal flow structures. Rotor-dependent periodicities present instead a slope which is proportional to their velocity of propagation and, therefore, to the rotor speed.

In the case exemplified in Figure 4.66, the static pressure field at $-15\%c_{r,ax}$ is highly dependent on the vane-relative position, modulated by the potential field of the upstream NGV. A much stronger rotor periodicity is shown by the same quantity at $90\% c_{r,ax}$, where the rotor blade tips, periodically traveling in close proximity to the turbine casing, drive the static pressure fluctuation pattern.

A double Discrete Fourier Transform (Matlab FFT algorithm) is applied to these

distributions in order to obtain the modal matrix, where each element represents a specific spatial-temporal mode [106]. Those periodicities which are stationary in the vane frame of reference will be addressed as vane modes. Conversely, the flow structures which show a steady behavior in the rotor relative frame represent the rotor modes. The total energy content of the rotor- and vane-dependent periodicities can be computed by knowing their position in the modal matrix, as proposed in [107]. All the remaining modes will be non-stationary in both reference frames (vane and rotor-relative domain) and will be addressed as vane-rotor interactions.

An example of phase-time representation of the NLH-predicted relative total pres-

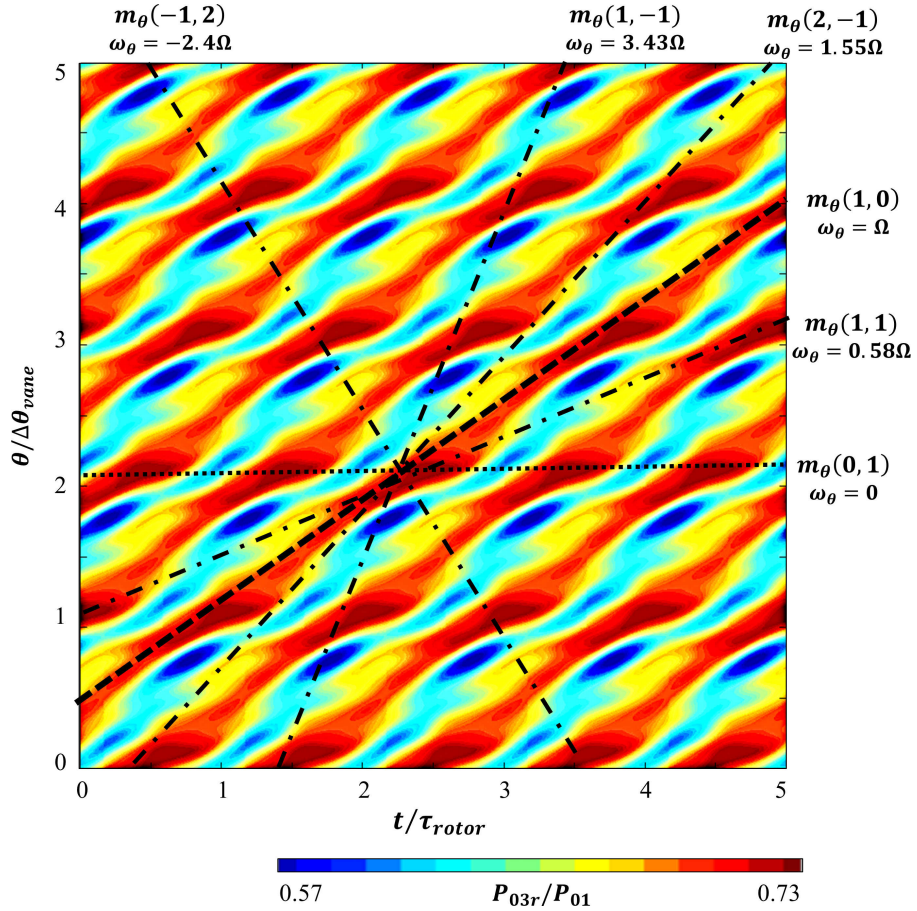


Figure 4.67: Phase-time representation of NLH-predicted P_{03r} at 75% h_r

sure at 75% h_r of the outlet plane of blade IN01D is presented in Figure 4.67. The non-uniform, periodic flow field is first considered as a superposition of infinite circumferential rotating structures, defined as angular modes (m_{θ}). Each angular mode presents a number of lobes (periodic flow structures) equal to a linear combination of the number of vanes (N_V) and rotor blades (N_R) of the HP stage, as from Equa-

tion 4.14:

$$m_\theta(n, k) = n \cdot N_R + k \cdot N_V \quad (4.14)$$

where n and k are integer numbers. The angular position of these circumferential patterns changes in time according to a propagation equation, allowing to define a rotational speed ω_{m_θ} characterizing each spinning mode.

$$\omega_{m_\theta}(n, k) = \Omega \cdot n \cdot N_R / m_\theta \quad (4.15)$$

Co-rotating and counter-rotating modes can therefore be defined with respect to the rotation direction of the rotor blades. These structures will present, respectively, positive and negative inclination in the phase-time representation of Figure 4.67.

A transformation of this signal from the spatial-temporal frame to the frequency domain allows to decompose the flow unsteadiness into fundamental modes. After removing the mean value from the time- and space-resolved signal, a discrete Fourier transform is applied in azimuthal direction (i.e. along θ_{vane}) at each instant in time. This analysis is performed considering a signal sampling frequency equal to the number of samples per rotor revolution, allowing the identification of the azimuthal modes m_θ corresponding to the definition given in Equation 4.14. The energy spectrum resulting from the spatial FFT applied to the phase-time pattern in Figure 4.67 is given on the left-hand-side of Figure 4.68.

The spatial modes corresponding to the first harmonics of the blade passing fre-

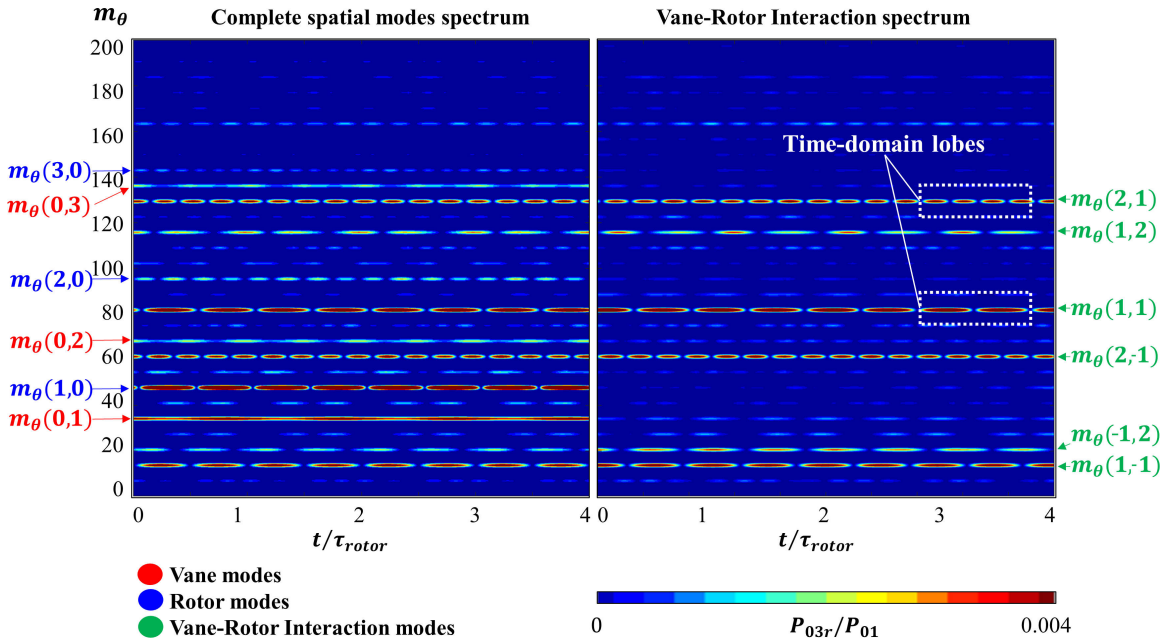


Figure 4.68: Spatial modes spectra of NLH-predicted P_{03r} at 75% h_r : full modes (left) vs. vane-rotor interaction spectrum (right)

quency, i.e. $m_\theta(1, 0)=48$ and $m_\theta(2, 0)=96$, are clearly visible in the plot. As expected, $m_\theta(2, 0)$ presents a double number of lobes per rotor period with respect to $m_\theta(1, 0)$. The stationary influence of the stator-related flow structures, on the other hand, is visible at $m_\theta(0, 1)=34$, $m_\theta(0, 2)=68$ and $m_\theta(0, 4)=136$, with the third harmonic presenting a smaller amplitude compared to the others.

To compute the modal matrix for this flow pattern, a Fast Fourier Transform is applied in the time (i.e. horizontal) direction of the spectrogram. This allows to further decompose each spatial mode m_θ in temporal modes m_t . It is now possible to define a relationship between spatial and temporal modes (Equation 4.16).

$$m_t = m_\theta - k \cdot N_V \quad (4.16)$$

An analysis of the flow in the spatial- and temporal-frequency domain is therefore possible, with each particular flow periodicity represented in the modal matrix (Figure 4.66, right) by specific coordinates (m_θ, m_t) . For each of these periodic modes, a rotational speed is defined as from Equation 4.17[106].

$$\omega_{m_\theta, m_t} = \Omega \cdot m_t / m_\theta \quad (4.17)$$

The modes which uniquely derive from the Vane-Rotor Interactions (VRI) can therefore be isolated by subtracting from the modal matrix all those components which are steady in either the stator or rotor frame of reference. Following the definition given in Equation 4.17, these modes will present rotational speed equal to zero and Ω respectively, i.e. $m_t = 0$ and $m_\theta = m_t$.

The remaining angular modes will therefore only represent the blade rows interactions. An inverse time-FFT can be applied to the modal matrix to visualize only the VRI modes in the spatial modes spectrum. The result is shown on the right-hand-side of Figure 4.68, which highlights the main VRI modes present in the unsteady pattern of P_{03r} at 75% h_r . The vane-rotor interaction periodicities can be retrieved in Figure 4.67. The total energy content of the vane-rotor interactions can be calculated by applying Equation 4.18 to the "polished" VRI modal matrix.

$$E_{SRI} = \sum_{m_\theta, m_t} A_{m_t, m_\theta}^2 \quad (4.18)$$

In this case, A_{m_t, m_θ} represents the amplitude of the corresponding mode. The radial distributions of modal energy can now be computed by iterating this procedure at different spanwise positions for different categories of modes.

4.5.2.2 Over-tip Pressure Modes

Figure 4.69 shows the instantaneous contours of time- and space-resolved static pressure at the over-tip rotor casing. Numerical predictions by single-tip NLH computations are compared against unsteady casing pressure measurements for the three tips. The contours are visualized in the vane frame of reference, with the horizontal axis indicating the peripheral position normalized by the vane pitch and phased with the vane trailing edge ($\theta = 0$). The position of the rotor airfoils leading edge relative to the NGV pitch is indicated in green in the plot to aid the potential field interactions analysis. These plots are instrumental in quantifying the degree of passage-to-passage repeatability of the rotor outflow against the influence of the vane potential field.

The highest degree of pressure field variability between adjacent rotor passages is observed in the over-channel region between -5% and 25-30% $c_{r,ax}$, confirming the variability shown in Figure 4.63 in the rotor domain. The experimental dataset confirms that a high pressure region arises in the near-pressure side region of the three tips as they cross the NGV trailing edge, as indicated by marker A (rotor tip LE at 16% of the vane pitch). The numerical predictions indicate that the over-pressure is partly attenuated by the open rim of blade AC06D, while resulting more evident in tips IN01D and AC07T.

As the rotor blade travels further, reaching 47% of the vane pitch, the upstream over-pressure zone gradually detaches from the high pressure fluid pocket near the blade pressure side, as indicated by marker B. This process continues as the rotor blade reaches 75% of the vane pitch, where also the experimental results show two distinct high-pressure zones: one over the near-PS channel at 55% $c_{r,ax}$ and one entrained by the following rotor blade, whose leading edge is now aligned with the vane TE ($\theta_{LE}/\theta_v = 0.05$). The latter justifies the high level of ΔP_s visualized at the front SS of the three blades in Figure 4.63.

The over-tip pressure field shows a significantly higher degree of rotor passage-to-passage periodicity. The most evident vane-dependent pressure variation is visualized over tips IN01D and AC07T when the rotor LE is at 16% of the vane pitch. The experimental results show, at this stage, the vane-induced over-pressure region (marker A) partially overhanging the near-LE tip surface and slightly increasing the over-tip mean pressure level up to 50% of the rotor chord. In the numerical dataset, the cavity of tip IN01D shows slightly higher and more uniform pressure level at this location. The pressure gradually decreases in the near-PS cavity region as the blade travels to 47% and 75% of the vane pitch. Over blade AC07T, as the high-pressure fluid from the vane migrates from the LE towards the late PS (marker A to B/C), the over-tip

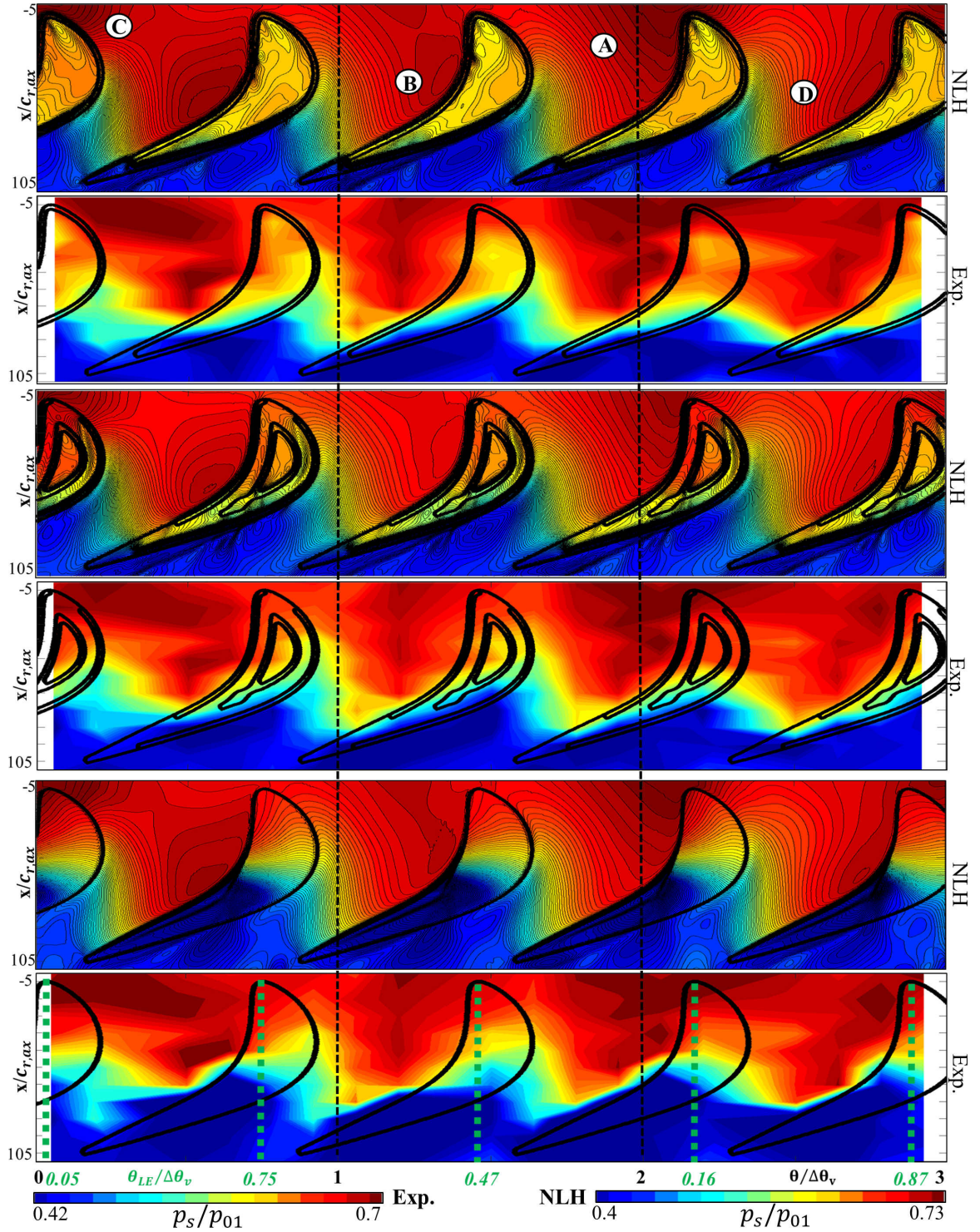


Figure 4.69: Instantaneous casing static pressure in the vane frame

acceleration appears slightly delayed (over-tip $\min(P_s)$ shifted downstream).

Figure 4.70 represents the modal decomposition applied to the static pressure field at the rotor casing. The datasets presented in Figure 4.69 are considered for this

analysis: experimental measurements (symbols) and full-resolution numerical results from single-tip NLH computations (solid lines). The unsteady pressure histories are exploited in their integrity in the vane frame of reference, meaning that vane- and rotor- induced flow gradients are preserved in the space-time domain. Additionally, the spatial resolution of the modal analysis performed over the CFD is downgraded (dotted lines) to reproduce the spatial discretization of the experimental measurement matrix (6 peripheral positions / vane pitch, 11 axial locations between -5% and 105% $c_{r,ax}$) and allow a relevant comparison with the experimental spectra.

In the left chart (Fig. 4.70a) the total energy of each profile is presented. Low energy levels are observed in the front part of the three tips, up to 40% $c_{r,ax}$. At this location, the steep acceleration experienced by the gap flow over blade AC07T generates high P_s fluctuations. The corresponding energy peak is located at 60% $c_{r,ax}$ and exceeds by more than 150% the peaks experienced at 70% $c_{r,ax}$ by the two squealer-like geometries. This large difference is motivated by the significantly lower flow velocities reached over blades IN01D and AC06D. The modal decomposition performed on the CFD data with downgraded spatial resolution shows a good agreement with the experimental results, in terms of both peaks magnitude and location. This result indicates the sensitivity of the modal decomposition to the spatial and temporal data discretization. The "ranking" in terms of energy content of the different profiles, however, results unaffected by the resolution change.

Figure 4.70(b) presents the relative contribution of the different modes to the total

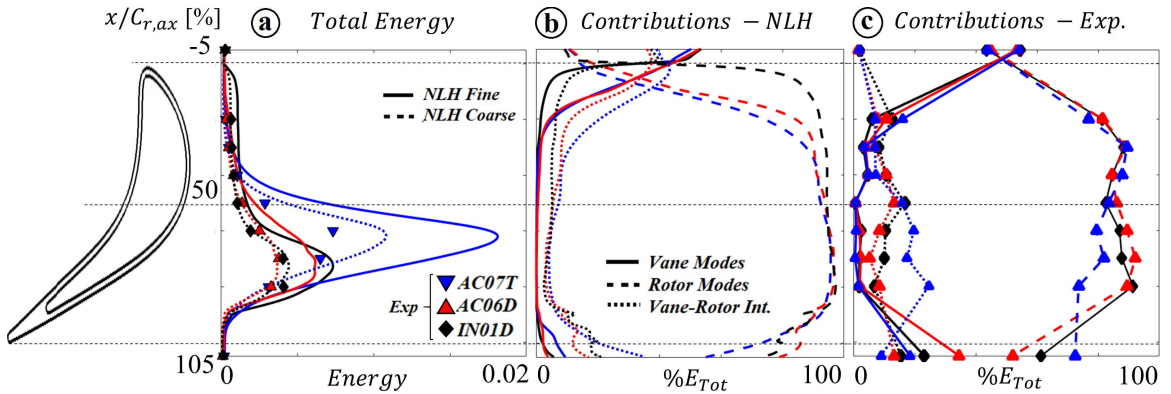


Figure 4.70: Modal decomposition of over-tip casing static pressure field: total energy (a), numerical predictions (b) and experiments (c) relative energy spectra

energy content in the full-resolution NLH study, while Figure 4.70(c) reports the relative energy in the experimental case.

The vane influence results significant only upstream of the rotor LE. From 0 to 20%

$c_{r,ax}$ the vane and vane-rotor modes decrease abruptly for blade IN01D, characterized by a continuous leading edge rim. A more gradual decay is observed for the open rim geometry (AC06D) and carved profile (AC07T), for which the rotor modes still constitute only 40% of the total energy content at $10\%c_{r,ax}$. This trend indicates a higher influence of the vane potential field in the front part of these tips, in reason of the absence of a tight clearance LE rim to break this dependence.

After $30\%c_{r,ax}$, the over-tip pressure field of the three profiles is highly periodic with the rotor motion, with the VRI accounting only for 10% of the total energy content. The rotor-periodic fluctuations decrease at $85\%c_{r,ax}$, with the carved profile showing the most significant rise of vane modes. The over-tip spectrum is supported by the experimental results in Fig. 4.70(c), while more significant discrepancies are found at the first and last axial stations, respectively -5% and 105% $c_{r,ax}$. The experimental VRI appear dumped in favor of rotor-only modes at the upstream location and of vane periodicities downstream. These inconsistencies might be due to the smaller magnitude of the pressure fluctuations outside the gap, which affect the experimental resolution of minor flow periodicities ($0.5\%P_{01}$).

Full-resolution NLH and experimental results show, globally, a high level of agreement in terms of relative energy content of the casing pressure field, demonstrating a lower sensitivity of the relative energy spectra to the spatial resolution of the measurements. This result suggests that a "qualitative" assessment of the unsteadiness content of the flow can be pursued even with a rather coarse spatial measurement grid, while the quantitative evaluation of the modal energy requires a high resolution in both the time- and space-domain.

4.5.2.3 Outlet Pressure Modes

Figure 4.71 shows the instantaneous outlet total pressure contours predicted by single-tip NLH computations. The contours are visualized in the stationary frame of reference, with the horizontal axis indicating the peripheral position normalized by the vane pitch and phased with the vane trailing edge. The instantaneous position of the rotor airfoils is additionally visualized by dotted lines corresponding to the blades trailing edge. These plots are instrumental in quantifying the degree of modulation of the rotor outflow by the upstream vane.

The near-casing area registers different degrees of periodicity between the three tip designs (marker A). Blade AC06D, characterized by the strongest tip leakage vortex,

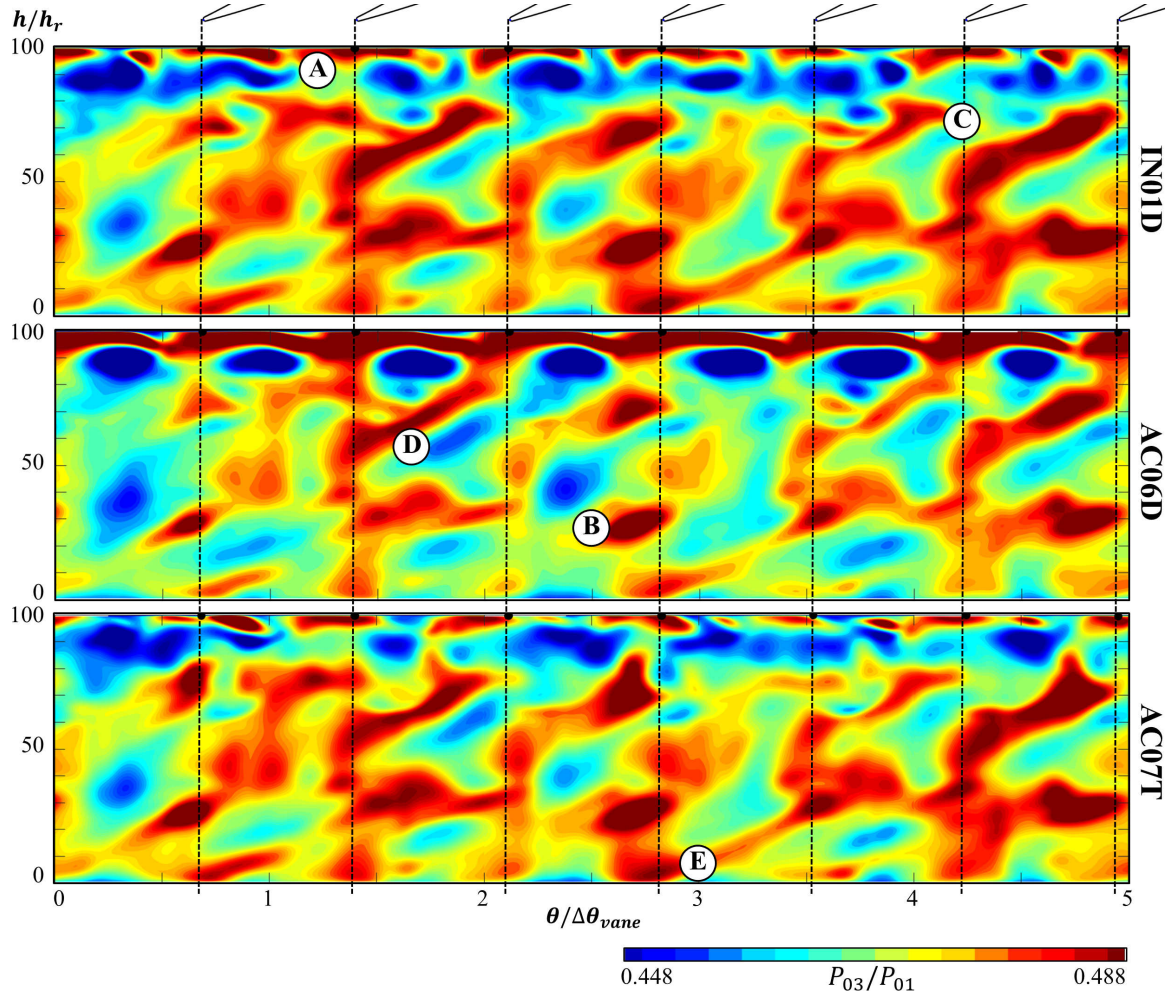


Figure 4.71: Instantaneous contours of outlet total pressure in the vane frame (NLH)

shows a highly repeatable pressure minimum at 90% of the rotor span, only partially affected in its peripheral extension by the vane-relative position. The outflow of AC07T, conversely, shows a high degree of vane modulation in the same area, with the TLV pressure deficit changing significantly in shape and dimension as a function of the upstream vane position.

The flow region between 70 and 80% h_r , where the UPV is located, is indicated in the plots by marker C. A lower passage-to-passage repeatability is registered in this region for all the profiles, indicating an increased degree of vane-dependent flow modulation. Likewise, low levels of rotor periodicity are detected in the LPV region for the three profiles (20-25% h_r), where a vane-periodic pressure minimum appears to be occasionally replaced by a pressure peak (marker B).

A rotor-periodic region of low total pressure, leaned towards the sense of rotation of the airfoils, indicates presence of the rotor wake between 35 and 70% of the rotor

span (marker D).

The modal decomposition is applied to the time-resolved P_{03} field comparing experimental and numerical results. A typical phase-time dataset generated to perform the modal analysis is exemplified in Figure 4.72, which shows the NLH-predicted (left) and experimental (right) total pressure diagram at 75.5% h_r for blade IN01D. The unsteady flow pattern is observed to correlate well between the predicted and measured dataset, with the main periodicity propagating at the rotational speed of the rotor indicated by the dashed line. Similarly to what shown in Figure 4.67, a time-steady modulation by the vane (indicated by the horizontal dotted line) and further VRI patterns can be retrieved in this diagram. A detailed analysis of this specific plot, however, goes beyond the purpose of the example and will not be pursued.

The plot in Figure 4.73 represents the modal decomposition applied to the time- and

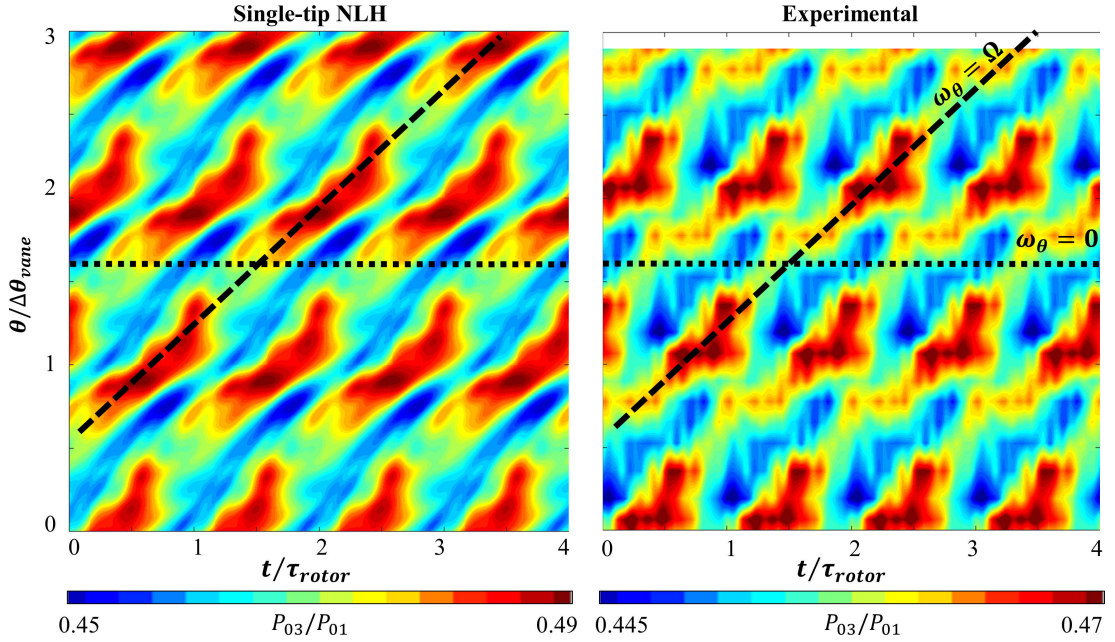


Figure 4.72: Phase-time contours of stage outlet total pressure at 75.5% h_r : NLH predictions vs. measurements

space-resolved total pressure field at the rotor outlet. Two sets of data are considered for this analysis: experimental measurements from the FRAPP rakes in Plane 03 and unsteady flow predictions by single-tip NLH computations. The unsteady datasets are exploited in their integrity in the vane frame of reference, with the vane- and rotor-induced flow variations preserved in the space-time domain.

The left section of the plot presents the radial distribution of total energy content (solid line for CFD predictions, symbols for measurement data points). The energy

breakdown is represented in terms of percentual relative mode contribution by the vane-, rotor- and VRI-modes (left to right) at each radial location of the three tip designs.

The total energy peak is located, for all the tips, in the near-casing area in reason of the strong periodic pressure fluctuations due to the TLV. A very good correlation is shown between experimental and numerical results (marker A). The simulations are shown to properly catch the tip-to-tip differences, particularly significant in this area, in terms of total energy content. Blades IN01D and AC07T show a rather similar behavior, with the modal energy peak located at $\sim 95 - 97\%h_r$. A local maximum is shown by AC06D at 90% of the blade span, in reason of the propagation of the TLV at lower radial position in this profile. The peak exceeds by more than 100% the maxima associated to the other tips, consistently with the experimental results, and the energy breakdown confirms that it is almost entirely due to rotor-periodic fluctuations ($\sim 95\%$ of the energy content).

The vane modes contribution is slightly higher in this area ($h > 80\%h_r$) for tips IN01D and AC07T, for which, however, the NLH are shown to over-predict the role of VRI modes (marker C). This behavior confirms what highlighted by markers A and C in Figure 4.65, that is a higher vane clocking effect predicted in the near casing area with respect to the experiments.

At about $75\% h_r$, a less pronounced energy peak is present in the three profiles in correspondence of the UPV location. The experimental results confirm the predicted location and magnitude of the local maximum for blade AC06D and AC07T, while showing a slightly lower radial position ($-5\%h_r$) in IN01D. The presence of the rotor wake at mid-span is highlighted in the NLH predictions by a predominance of rotor-periodic modes (marker D), while a higher level of VRI is observed in the measurements.

The position of the lower passage vortex (LPV) is denoted by an energy peak in the lower half of the blade span, at about $20-22\% h_r$ for blades IN01D and AC07T and at $24-25\% h_r$ for AC06D (marker B). The vane modulation of the LPV structure, already discussed in Figure 4.65, is here confirmed by the high vane-modes relative energy. The measurements mark a considerably lower peak of absolute energy, due to the enhanced mixing levels observed in Figure 4.49 at this location. Interestingly, however, the measured relative energy content validates the vane-periodicity level shown by the numerical results ($E_v = \sim 50\%E_{tot}$).

In close proximity to the inner endwall, marker E shows a significant discrepancy

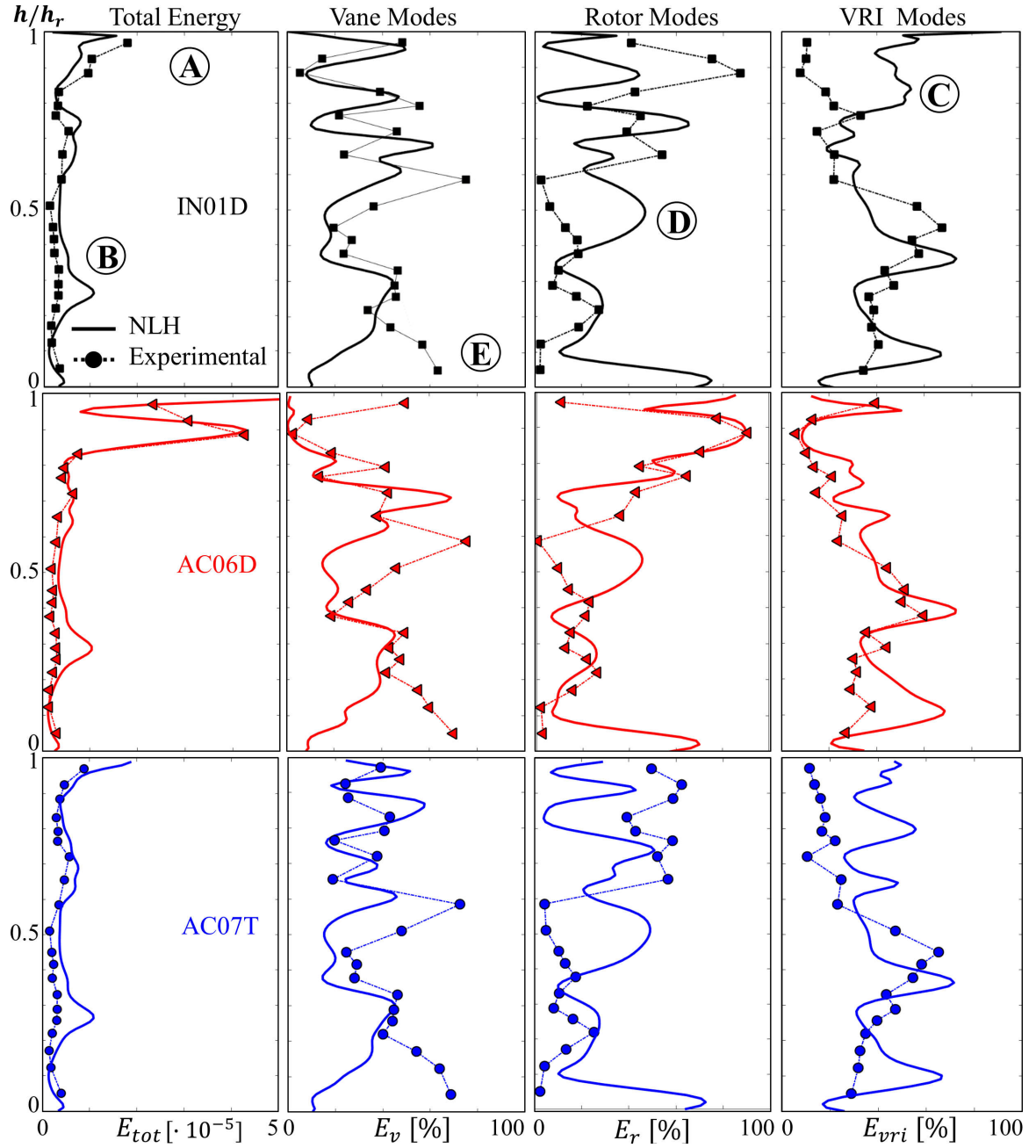


Figure 4.73: Modal decomposition of measured and NLH-predicted outlet total pressure field: total energy, vane-, rotor- and VRI-modes contributions (left to right)

between the predicted predominance of rotor-periodic fluctuations and the measurements, which show a higher influence of the vane clocking on the outlet pressure field.

4.5.3 Modal analysis of the entropy field

In this section, the time- and space-resolved entropy field at the mid-gap plane and rotor outlet is addressed. The modal decomposition analysis is applied to generate a breakdown of the modal contributions and locate the origin of the high-entropy structures which propagate through the HPT stage.

4.5.3.1 Mid-gap entropy field

Figure 4.74 shows the instantaneous entropy distributions at mid-gap for the three tips, i.e. at 99.31% h_r for the squealer profiles and 99.58% h_r for the carved tip, extracted from the single-tip NLH results. The contours are computed in the vane frame, between -15 and 150% of the rotor axial chord, taking as reference the vane trailing edge. The entropy levels are referred to standard conditions of temperature and pressure and are instrumental in detecting the flow regions where major loss generation mechanisms are present.

Periodic spots of high entropy, caused by the incoming vane wakes, are observed to impact on the rotor blades leading edge as they travel from left to right in the carpet plots. The slightly higher values exhibited by the wakes impacting the AC07T tip are justified by the different spanwise position of this mid-gap plane.

Inside the rotor passage, a gradual increase of entropy is registered from the suction towards the pressure side of the three tips. The intensity peak in the rotor passage is reached at the blade PS, around 50% $c_{r,ax}$, and presents a partial dependence in both magnitude and shape from the vane-relative position.

The most significant impact of the tip geometry is registered in the over-tip region. Relatively low values of entropy are registered in the front part of the tips surface. Proceeding towards 60% $c_{r,ax}$, the increase is very gradual in the central cavity of IN01D (A) and in the inner squealer of AC06D. In both cases, the vane-relative position is seen to qualitatively modify the over-tip entropy pattern, while not substantially affecting the maximum values achieved. The pressure side rim of both profiles generates localized high-entropy streams which do not travel further inside the cavity. High-entropy fluid is channeled in between the internal and external rims

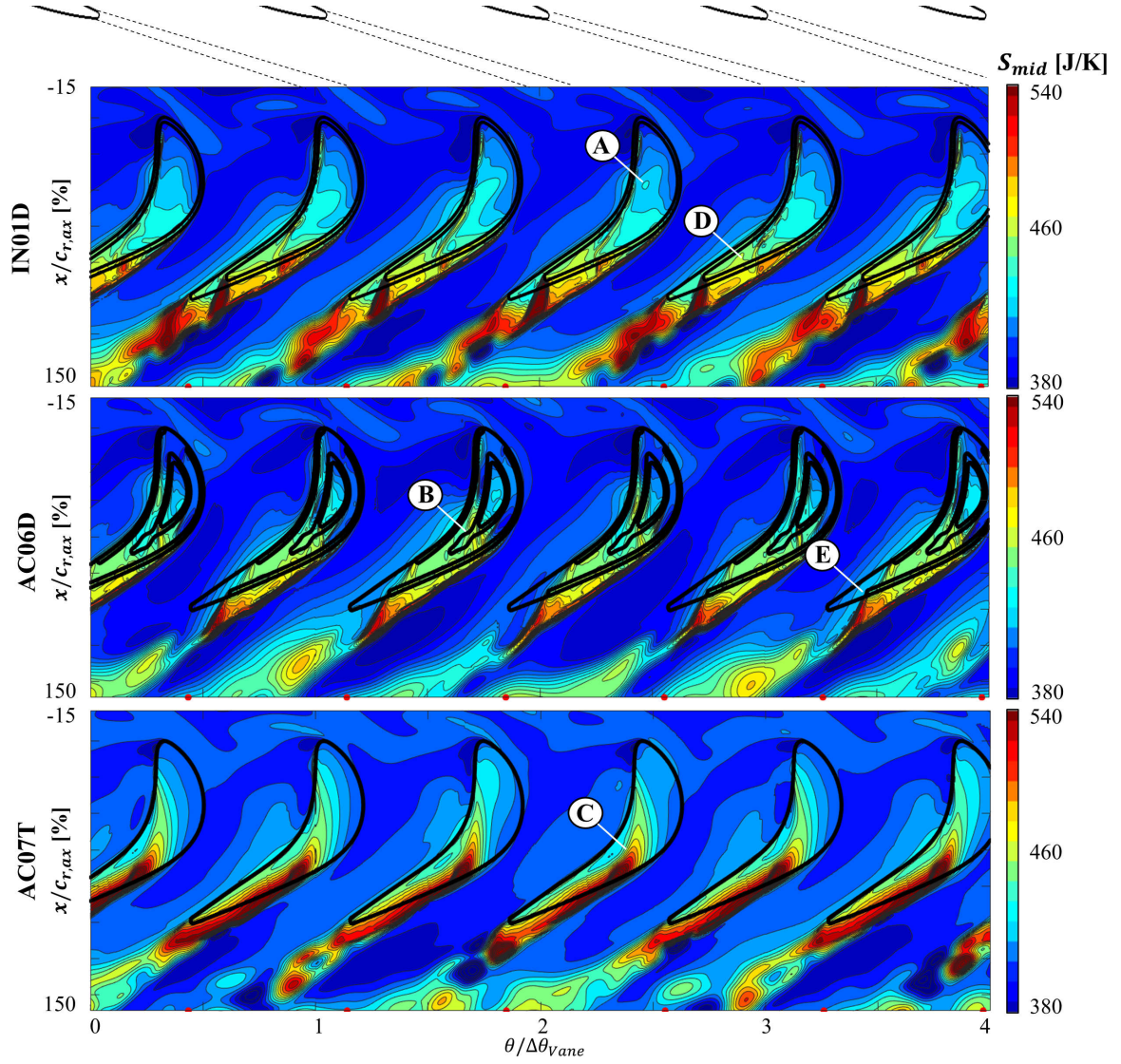


Figure 4.74: Instantaneous contours of mid-gap entropy in the vane frame (NLH)

of AC06D (B), eventually climbing over the central island single-rim end at $60\%c_{r,ax}$. The entropy gradient is much steeper on the AC07T profile, which exhibits over-tip maxima that significantly exceed the squealer profiles (C). As for the casing static pressure analysis, the highest level of vane modulation is in the first $50\% c_{r,ax}$, while an increased blade-to-blade repeatability is visualized in the aft part of the blades. The TLV is characterized by high levels of loss for all the three profiles. The stream is injected in the rotor passage from the blade SS, at an axial position which depends on the tip design: $50\text{-}55\%c_{r,ax}$ for the squealer geometries, $60\text{-}65\%c_{r,ax}$ for the carved tip. This delayed emission of high-entropy flow at the suction side of AC07T is in good agreement with what demonstrated by the vortex tracking results in Figure 4.58.

Major differences are observed in the flow pattern over the aft part of the blades.

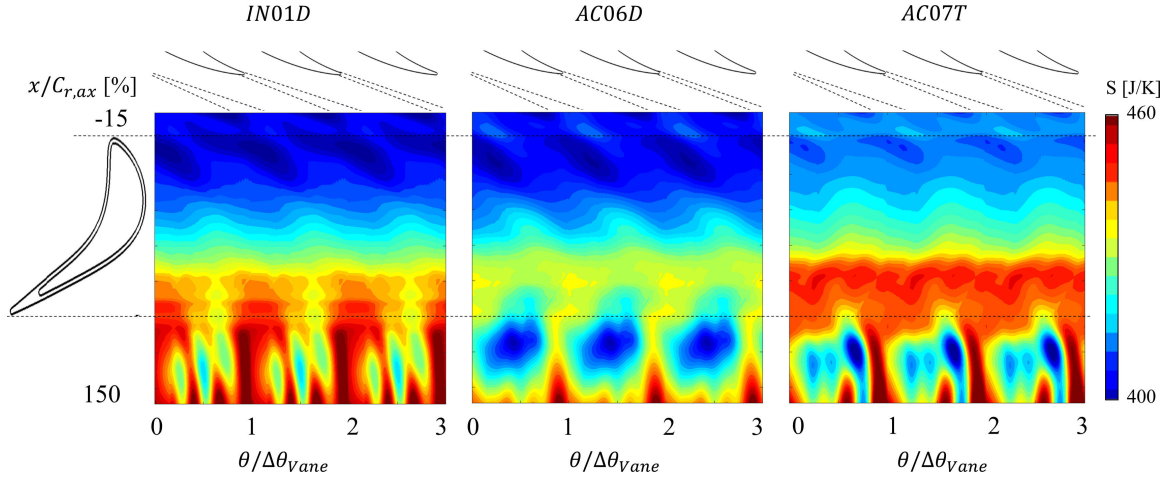


Figure 4.75: Contours of time-average mid-gap entropy in the vane frame for (left to right) IN01D, AC06D, AC07T

The closed end of the squealer cavity (D) produces an upturned high-entropy stream which does not vary significantly in intensity with the vane-relative position and interacts with the tip leakage stream. The open-end rim of AC06D produces a strong inviscid interaction between the low-entropy by-pass jet and the tip leakage vortex (E), highly rotor-periodic between 90 and 105% $c_{r,ax}$. The interaction becomes more vane-dependent between 105 and 130% $c_{r,ax}$, where the flow is not anymore guided by the rotor airfoils. The carved tip generates a more intense, localized, rotor-periodic leakage stream in the over-tip region. Downstream of the TE, the entropy pattern of AC07T appears increasingly variable as the unguided flow leaves the rotor passage. The time-average mid-gap entropy field of the three profiles is shown in Figure 4.75. The three blades show a common pattern between -15 and 55% $c_{r,ax}$, with a slightly higher mean entropy registered over AC07T, probably due to the smaller clearance value. The steep acceleration and subsequent over-tip flow separation over tip AC07T generates a uniform pattern of high entropy between 60 and 100% $c_{r,ax}$. The aft part of blade IN01D, characterized by the closed rim end, exhibits higher entropy levels than AC06D. The higher clearance value and by-pass jet of the latter contribute to limit the mean entropy level between 90 and 100% $c_{r,ax}$.

Downstream of the rotor TE (100% $c_{r,ax}$) the circumferential entropy gradients become more significant, with high entropy regions largely modulated by the vane-relative position. This pattern marks the increased influence of the upstream NGV over the entropy generated in the TLV of the three profiles.

Between 110% and 150% $c_{r,ax}$, blade AC07T exhibits periodic entropy peaks between 90% and 100% $\Delta\theta_{vane}$. In blade AC06D, vane-periodic low-entropy regions are present between 110 and 130% $c_{r,ax}$, due to the variable blade tip loading modulating the jet-leakage interaction. The low entropy level downstream of this tip is justified by the lower radial position of the TLV demonstrated in Figure 4.58.

Figure 4.76 shows the energy spectra obtained for the time-resolved mid-gap entropy fields of the three blades. The analysis is performed between -5 and 150% of the axial chord, in order to document the mixing process experienced by the tip leakage flow downstream of the rotor trailing edge.

The three blades show a monotonous increase of energy content as the gap flow

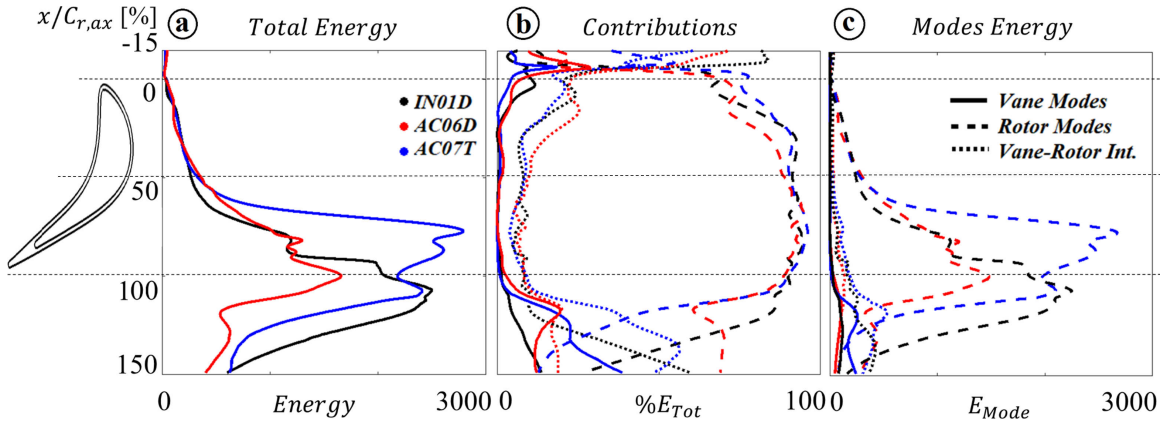


Figure 4.76: Modal decomposition of mid-gap entropy field: total energy (a), mode contributions (b) and mode absolute energy spectra (c)

develops over the front over-tip region. As for the pressure field, the AC07T profile exhibits the highest and earliest energy peak (70% $c_{r,ax}$), due to the the high velocity fluid separating over the tip surface. The squealer tips, on the other hand, show an increasing trend up to 100% of the axial chord.

At 90% $c_{r,ax}$, blade IN01D exhibits a steep rise of rotor modes due to the upward emission of fluid from the cavity closed end, reaching the energy content of AC07T at the trailing edge station. The open-end rim of blade AC06D attenuates the entropy fluctuations in the near-casing plane, resulting in a 25% lower energy content at the tip TE.

From the airfoil TE to 150% $c_{r,ax}$, IN01D shows a monotonous increase of vane-rotor interactions, but keeps the most rotor-periodic behavior among the three tips. For AC07T, a rapid increase of vane and vane-rotor interaction modes is visualized. Blade AC06D shows a local maximum of vane-relative energy around 120% $c_{r,ax}$, suggesting

a dependence of the jet-leakage interaction (Figure 4.60) on the vane-induced variation of rotor tip load. Unlike blades IN01D and AC07T, the outflow of blade AC06D shows a subsequent increase in rotor-periodicity up to $150\%c_{r,ax}$.

4.5.3.2 Rotor outlet entropy field

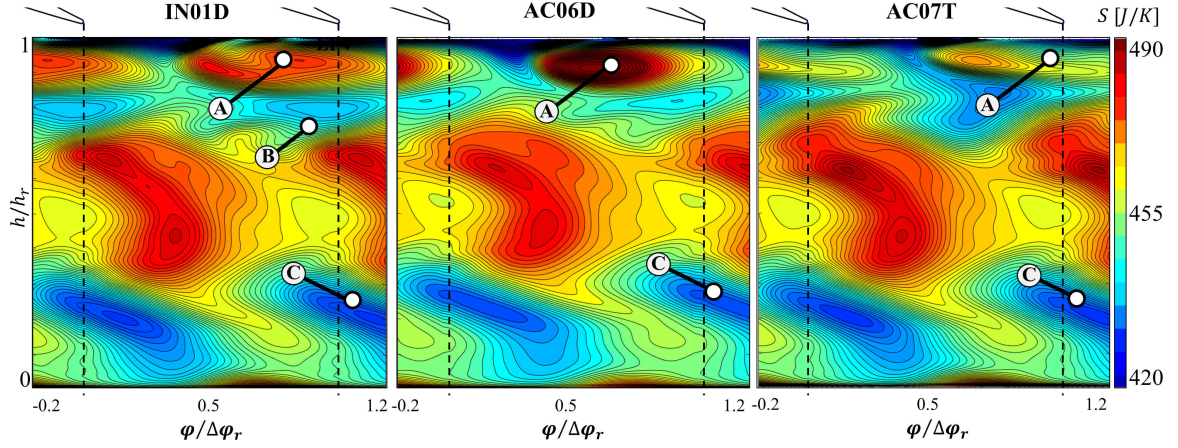


Figure 4.77: Contours of phase-resolved outlet entropy in the rotor-relative frame

The phase-resolved outlet entropy field predicted by single-tip NLH is shown in Figure 4.77, averaged in the rotor relative frame of reference. The horizontal axis indicates the position along the rotor passage, proceeding from the blade PS ($\varphi = 0$) to the SS ($\varphi = 1$). The RANS-tracked positions of the TLV, UPV and LPV are indicated by white dots.

From 0 to 50% of h_r the entropy field exhibits a globally comparable topology between the different blades. Between 30 and 70% h_r , a high entropy region slightly leaned towards the blade pressure side corresponds to the high-loss fluid of the rotor wake. Blade AC07T shows the wake with largest circumferential extension in the rotor domain. The position of the LPV (marker C) is denoted by a lower entropy level.

The most evident tip-to-tip flow differences are detected between 80 and 100% h_r , where a high loss region indicates the presence of the TLV (marker A). The maximum entropy levels are observed for blade AC06D, while a weaker, more scattered entropy pattern is detected for IN01D. The carved profile (AC07T) exhibits the weakest and most stretched out TLV loss region. The presence of the UPV is marked by a low-entropy flow structure (marker B in blade IN01D) in between the rotor wake and the TLV regions.

The time-average outlet entropy field is shown in Fig. 4.78, represented in the vane

frame of reference. The horizontal axis indicates the TE-phased position along the NGV passage, proceeding from the suction side ($\theta = 0$) to the pressure side ($\theta = 1$). The circumferential gradients in this plot depend exclusively on the presence of the upstream vane. The rotor, traveling from left to right in the plots, only contributes to the radial entropy variations.

In the near hub region of the three tips, at 20% h_r , a low entropy region with strong

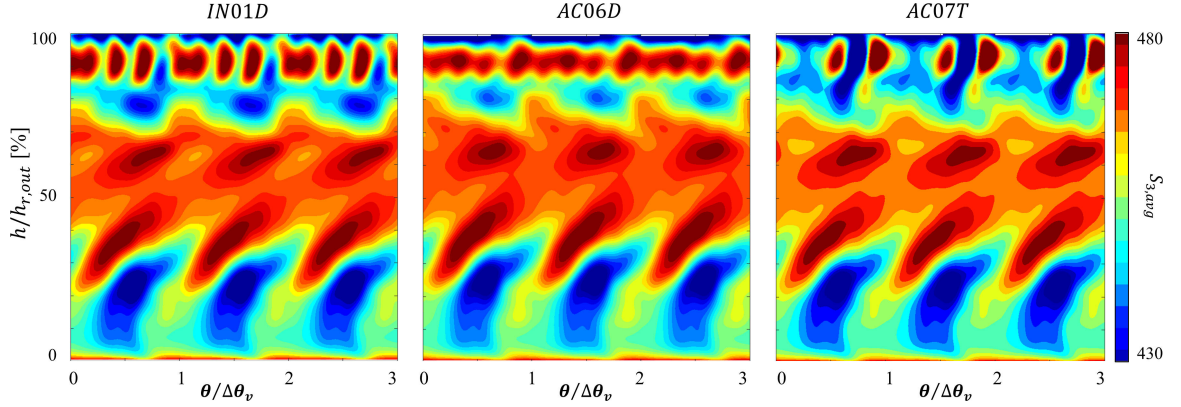


Figure 4.78: Contours of time-average outlet entropy in the vane frame of reference

vane-periodic modulation is observed. From 25 to 50% h_r high entropy avenues, leaned towards the sense of rotation of the rotor blades, appear at each vane passage. These structures represent the vane wake avenues and result significantly repeatable between the different tips.

Substantial tip-to-tip differences are observed in the upper 50% of the blade span. At 70-75% h_r , at the radial location of the UPV, the two squealer profiles show a vane-periodic pattern of high- and low-entropy regions. In the near tip region (90-59% h_r), blade AC07T exhibits the strongest vane-periodic variation of entropy among the three tips, followed by blade IN01D. This periodicity marks the vane dependency of the TLV. Blade AC06D presents the most uniform loss distribution at this height, in concordance with a highly rotor-periodic TLV shown in Figure 4.77.

The modal decomposition of the outlet entropy field is shown in Figure 4.79. The total energy distribution (Figure 4.79a) indicates the energy peak in correspondence of the TLV, UPV and LPV for the three tips. The magnitude of the maxima depends on the intensity and on the level of localization of the periodic flow structures.

Blade AC07T shows the highest TLV entropy fluctuation among the three tips, mostly due to a high level of vane modulation, as shown in Figure 4.79b. The two squealer tips reach the same energy maxima in the tip region, but with different radial distributions. The fluctuation peak of blade AC06D results at lower radial position with

respect to blade IN01D, as a consequence of the lower radial location of the TLV core, highly rotor periodic.

At 75% of the blade span, a second energy peak is detected for the three profiles, corresponding to the position of the UPV. In the two squealer profiles, this flow region is shown to be very sensitive to the vane-relative position, while the highest degree of rotor-periodicity is observed for blade AC07T, where the UPV generates the highest peak of absolute total energy.

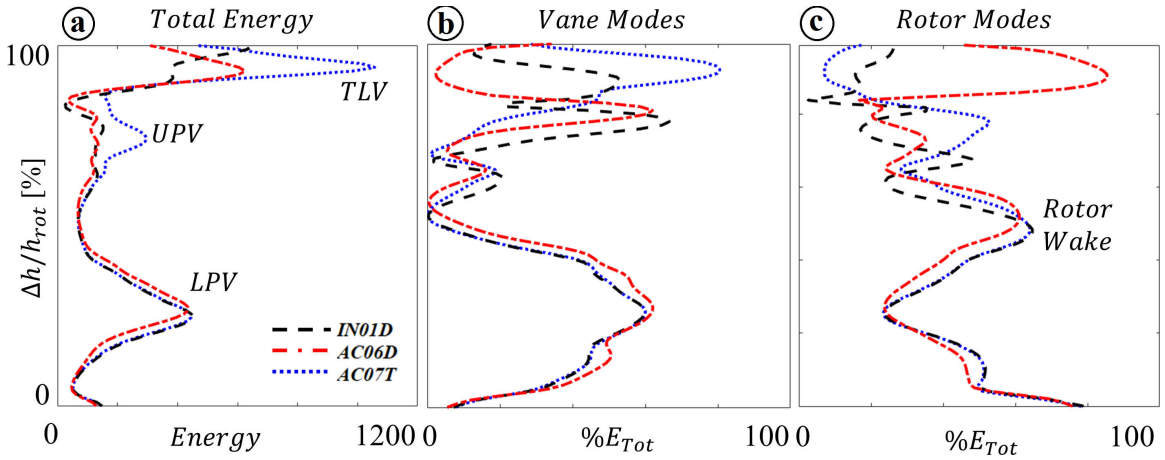


Figure 4.79: Modal decomposition of rotor outlet entropy field: total energy (a), vane (b) and rotor (c) modes

Between 40 and 60% h_r , where the high-entropy rotor wake fluid is detected in the rotor-mean field, a high contribution of rotor modes is observed. These results support the findings of Lengani et al. [108], who performed a modal analysis of the total pressure field immediately downstream of an unshrouded HP turbine stage. Rotor-only modes were seen to dominate the unsteadiness in the tip region (as for blade AC06D) and at mid-span (as for the three tips in the present study).

Moving further downwards along the blade span, the tip-to-tip differences tend to attenuate, with a peak appearing around 22-23% h_r for blades AC07T and IN01D and at 24-25% h_r for AC06D, corresponding to the radial location of the LPV.

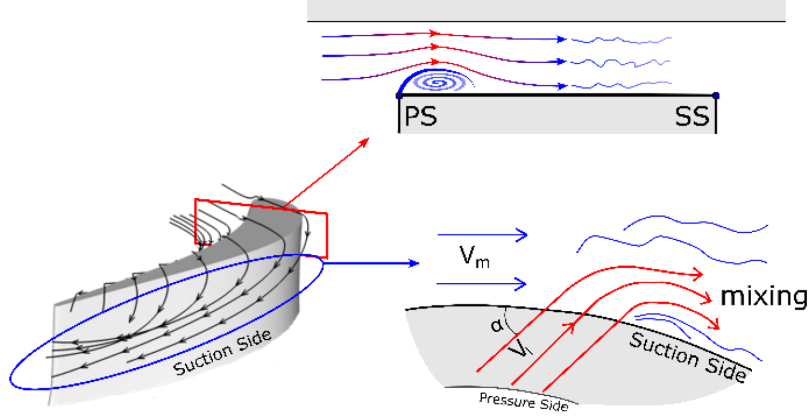


Figure 4.80: Tip leakage loss generation inside the tip gap and during mixing with the passage flow [8]

4.6 Tip Performance

4.6.1 Tip Leakage Loss Modelling

This section focuses on the quantitative evaluation of the performance of the three rotor tips. The entropy production generated by the tip leakage flow of every tip is characterized in terms of two main contributions: the entropy generated in the over-tip region and the entropy generated by the mixing of the leakage stream with the main channel flow at the blade SS [80].

Figure 4.80 presents a schematic of the two loss generation mechanisms over a flat tip profile. The over-tip loss contribution is generated when the PS fluid is ingested in the rotor tip gap and therefore depends on the flow topology developing over the different tip geometries. The mixing contribution, conversely, is generated by the tip leakage flow which leaves the gap at the blade SS and is entrained by the passage flow field at a different speed and direction.

The loss generated in the over-tip region is computed through an entropy balance operated in the control volume bounded by the rotor tip surface (RT), the tip gap perimeter extruded up to the external casing (TGP) and the over-tip casing itself (OT). The entropy budget in the tip gap is decomposed in (1) irreversible entropy creation ($\dot{S}_{gap}^{irrev}$) and (2) reversible entropy change due to gas-to-wall heat flux ($\dot{S}_{gap}^q$) as from Equation 4.19.

$$\dot{S}_{gap} = \dot{S}_{gap}^{irrev} + \dot{S}_{gap}^q \quad (4.19)$$

The irreversible entropy generation can therefore be computed by simply subtract-

ing the heat flux contribution from the integrated entropy flux:

$$\dot{S}_{gap}^{irrev} = \int_{TGP} \left(c_p \ln \frac{T_s}{T_{s,ref}} - R_{fluid} \frac{p_s}{p_{s,ref}} \right) d\dot{m} - \int_{RT+OC} \frac{\dot{q}}{T_{s,wall}} dA \quad (4.20)$$

The loss generation due to the tip leakage flow mixing with the channel stream is computed following the model proposed by the Denton [11]. Three terms are considered to contribute to the irreversible entropy change $\dot{S}_{mixing}^{irrev}$:

- leakage mass-flow: $\dot{S}_{massflow} = \dot{m}_l$
- mixing flow speed: $\dot{S}_{Mach} = c_p(\gamma - 1) \cdot M_m^2$
- streamwise momentum deficit: $\dot{S}_\alpha = 1 - v_l/v_m \cdot \cos\alpha$

$$\dot{S}_{mixing} = \dot{S}_{massflow} \cdot \dot{S}_{Mach} \cdot \dot{S}_\alpha \quad (4.21)$$

The leakage flow quantities in Equation 4.21 ($\dot{m}_l$, $\vec{v}_l$) are extracted from the TGP cells and radially integrated.

For the evaluation of M_m and v_m , Coull [109] proposes to use the isentropic values derived from the SS static pressure at 90% of the rotor span. A similar approach is adopted here with the difference that the pressure distributions are sampled at 80% h_r in order to avoid the significant drop in static pressure produced by the tip flow ejection at the blade SS.

The total tip leakage loss is computed as the combination of the losses inside the gap and those due to the mixing:

$$\dot{S}_{leakage} = \dot{S}_{gap}^{irrev} + \dot{S}_{mixing} \quad (4.22)$$

The entropy change is converted into an equivalent efficiency drop [11] according to:

$$\Delta\eta = -\frac{T_{03}\dot{S}_{leakage}}{\dot{m}c_p(T_{02b} - T_{03})} \quad (4.23)$$

4.6.2 Tip Loss Evaluation

The mixing loss contributions, defined for Equation 4.21, are presented in Figure 4.81 together with the corresponding total mixing loss for the three tip designs and the Flat tip at design clearance (FlatD). All values are normalized by the corresponding maximum value computed for FlatD. The distributions are presented as a function of the normalized airfoil SS length.

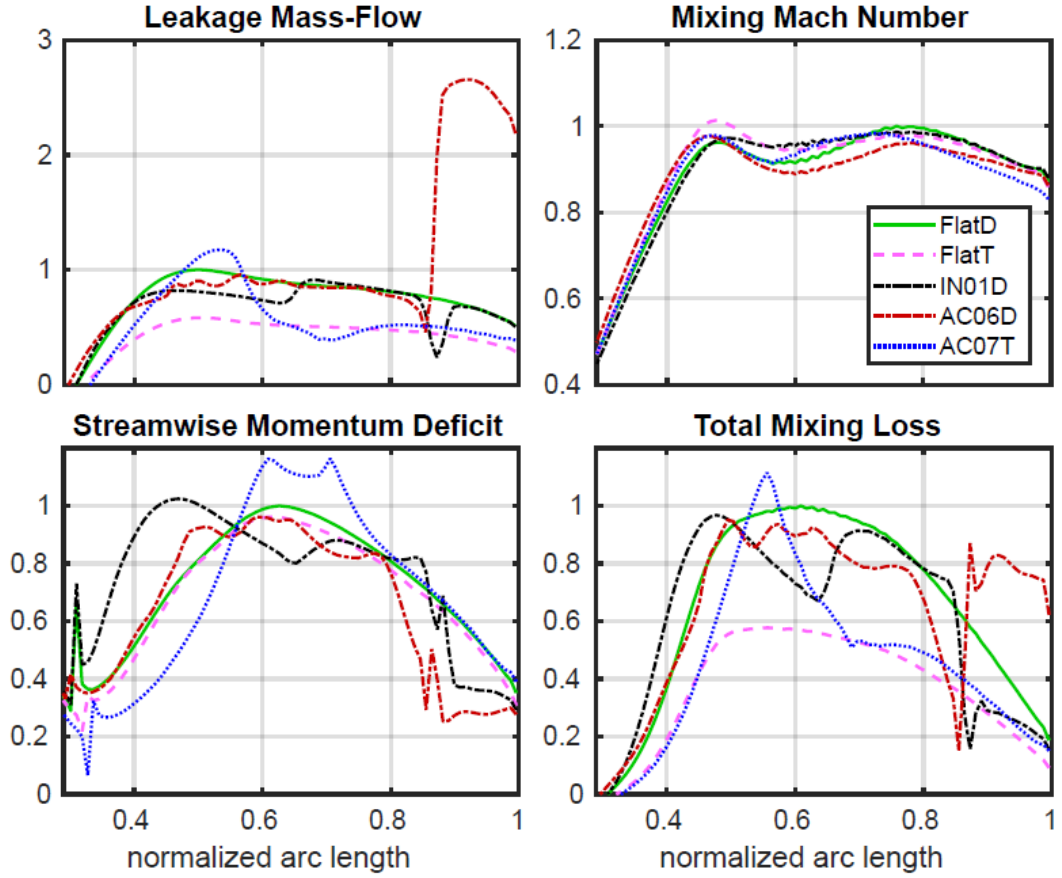


Figure 4.81: Tip leakage mixing loss contributions over the blade SS length

The highest total loss in the early 50% SS length is shown by blade IN01D, boosted by a large streamwise momentum deficit indicating a larger peripheral component of the leakage flow ejected in the front part of this tip. Blade AC06D, conversely, is shown to generate similar mixing losses to those of the FlatD tip in the aft SS, despite a slight increase in streamwise momentum deficit loss compensated by a local reduction of spilled mass-flow at 50% SS length. In the front part, the contoured geometry of tip AC07T promotes a better alignment between the leakage fluid and the main channel stream, as testified by the lower momentum loss contribution. At 50% of the SS length the tip flow of blade IN01D gains streamwise momentum, progressively aligning to the tip perimeter, while the leakage mass-flow decreases thanks to the wider squealer cavity, contributing to reduce the total loss up to 65% SS length. At this location, the double-rim regime leaves place to the direct by-pass mechanism, as shown in Figure 4.39a, producing a local increase in both contributions. The total loss is observed to drop abruptly at 87% SS length, where the fluid traveling inside the squealer bath is diverted radially upwards by the closed cavity end. The

cavity flow impinging on the external casing forms a fluidic curtain which actively limits the leakage massflow, as testified by Figure 4.55, and contributes to further align the spilled flow with the channel stream. The latter effect contributes to contain the total mixing loss below the level shown by FlatD up to the airfoil trailing edge.

The multiple rim geometry of blade AC06D contributes, between 50% and 80% SS

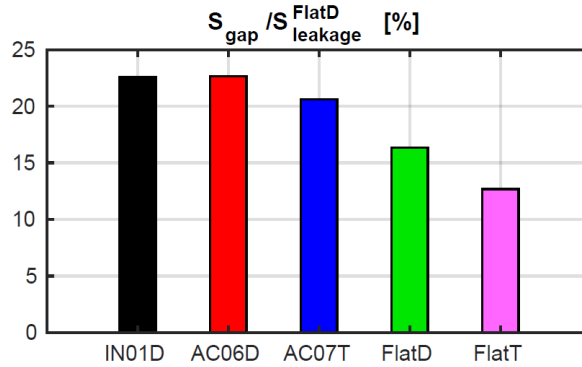


Figure 4.82: Entropy generation in the tip gap for the three profiles

length, to generate a globally better alignment of the leakage flow with the streamwise direction with respect to FlatD. The mixing Mach number, additionally, is observed to be significantly lower for this profile. In the aft part of AC06D, however, the leakage mass-flow is enhanced dramatically by the rear-rim aperture, reaching values more than three times higher with respect to the flat tip. The total loss in the rear part of this blade, as a consequence, is more than two times higher than for IN01D, only partially limited by a better alignment of the leakage flow with the free-stream direction.

In blade AC07T, after an early loss mitigation obtained by a better flow mixing angle, the leakage stream misalignment with the passage flow increases between 50% and 60% of the SS length together with the leakage mass flow. These two contributions boost the total mixing loss of AC07T above the other tip profiles at 55% of the SS length. From this point onward, the tight clearance and the rounded knife-edge pressure side accelerate the leakage flow to choking conditions, blocking the spilled mass-flow over the remaining SS length. In the aft part of the blade (70% to 100% SS length), the total mixing losses are further reduced through a gradual decrease of the mixing Mach numbers.

Figure 4.82 presents the integral value of entropy generation inside the tip gap normalized by the total tip leakage loss of the Flat tip case at design tip clearance. The

gap entropy, obtained directly from the CFD calculations, corresponds to about one fifth of the total tip leakage loss where the mixing losses represents the largest contributor. Despite the significant changes in tip design and the large variations in tip wetted area, the tip-to-tip discrepancy in terms of total gap loss of the three blades is less than 3% of the total value.

The entropy generated by the tip leakage flow is computed as the difference between the total entropy production of each individual tip profile and the entropy production computed for a zero tip gap case along the rotor axial chord. For both cases, the wall heat transfer contribution is subtracted from the computed entropy budget. The resulting entropy evolutions are presented in Figure 4.83.

The integral values of tip leakage loss computed by the tip loss model of Denton are additionally plotted two axial chords downstream of the rotor trailing edge. This location is chosen in order to provide a direct comparison with the mixed-out loss predicted by the CFD calculations.

In the front part of the blades, the entropy increase generated by the tip leakage is extremely limited. Significant tip-to-tip differences, however, are shown in the first 50% of the axial chord, with the squealer-type profiles showing higher loss production rates than the smoothly contoured tip. At 50-55% $c_{r,ax}$, the loss generation starts increasing linearly with the axial coordinate, reaching about 50% of the total mixed out value at the rotor trailing edge. Consequently, half of the total entropy production related to the tip leakage flow occurs downstream of the rotor row, promoted by the mixing process and by the interaction between the TLV and UPV. of each profile. The tip loss model is shown to capture correctly the performance ranking between the different blade tip designs, despite a global over-prediction of the absolute values of the mixed out loss.

A relevant consideration can be made about the location where losses are evaluated. In the numerical study performed for the design phase of the three rotor tips, the aerodynamic performance was evaluated at an axial plane located 50% $c_{r,ax}$ downstream the rotor TE. In a real engine environment, where a multi-row configuration is likely to be adopted, the single stage performance is measured at the inlet of the following row rather than at an hypothetical mixed-out state of the turbine flow. Hence, the different rates of entropy generation shown by the three profiles demonstrate the necessity to account for the machine design when evaluating the performance of the optimized tip seals. In the present analysis, the turbine flow generated by blades IN01D and AC07T can be considered mixed-out at 170% $c_{r,ax}$ while the interaction between the TLV and the tip leakage jet from the SS open rim of AC06D delays the

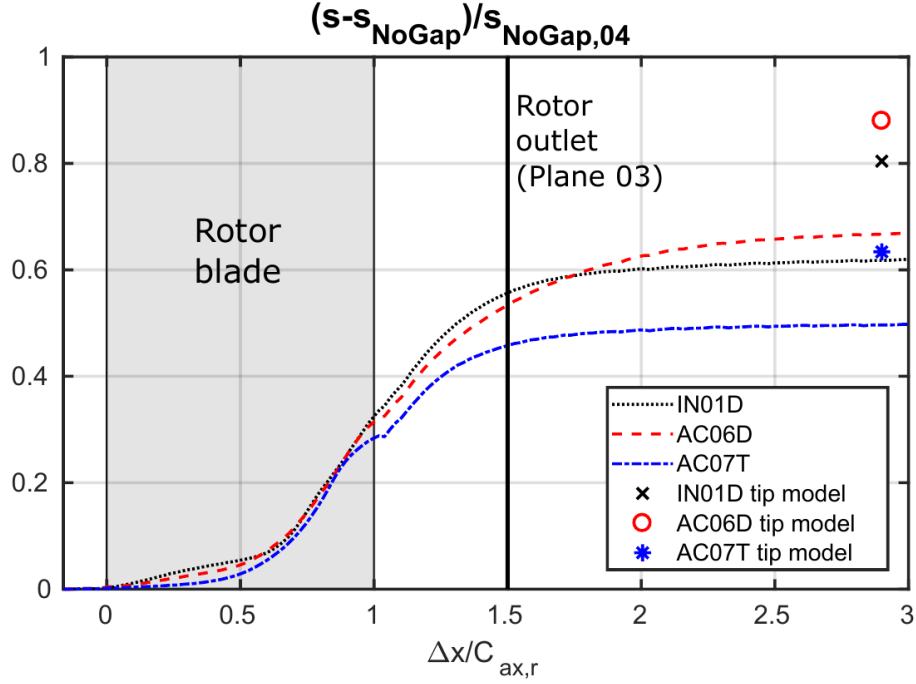


Figure 4.83: Entropy generation due to tip leakage flows (CFD) compared to predictions of the tip loss model

conclusion of the mixing process up to 250% $c_{r,ax}$. Both optimized tips, however, outperform blade IN01D at 150% $c_{r,ax}$, while the mixed-out loss of AC06D is the largest at mixed-out conditions.

4.6.3 Tip performance analysis

The aerodynamic performance of the three tip profiles is quantified in terms of rotor efficiency η . The rotor efficiency is evaluated from the shaft power and isentropic enthalpy drop between the inlet and exit planes:

$$\eta = \frac{\omega_{rot} \cdot Torque}{\dot{m} c_p T_{02b} \left[1 - \left(\frac{p_{03}}{p_{02b}} \right)^{\frac{\gamma-1}{\gamma}} \right]} \quad (4.24)$$

The results of the performance analysis performed on the full-stage RANS flow results are presented in Figure 4.84, in comparison to the numerical predictions of the optimization stage and to the tip leakage loss model.

The leakage loss model captures well the relative efficiency increase of blades AC07T and FlatT, although the absolute magnitude of the increment is slightly over-estimated. The larger loss predicted by the model for blade AC06D is partly due to the location where the efficiency is evaluated in the numerical study. As presented in

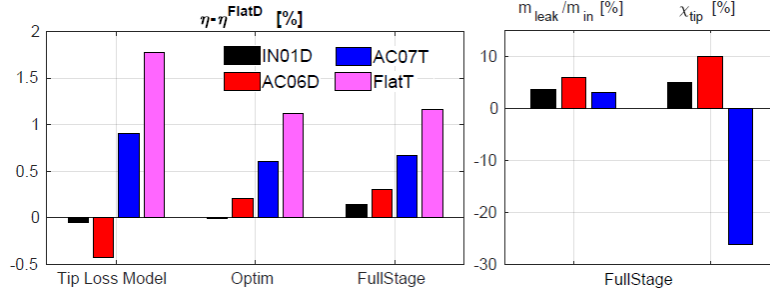


Figure 4.84: Aerodynamic performance parameters

Figure 4.83, the flow mixing process at the outlet of blade AC06D is delayed by the TLV-jet interaction and can be considered concluded at $250\% c_{r,ax}$. The CFD loss evaluation is performed here at $150\% c_{r,ax}$, where AC06D benefits from the delayed flow mixing.

A tip efficiency evaluation is performed in order to weight the relative gain/loss of performance of a certain tip design against the actual available margin for tip loss reduction. The maximum margin is in fact the difference in efficiency between the case of an ideal turbine rotor running with no tip clearance and a reference tip geometry operated at a relevant tip clearance condition. Hence, the tip efficiency parameter is introduced as from Equation 4.25.

$$\chi_{tip}(g) = \frac{\eta(g) - \eta_{ref}(g)}{\eta_{NoGap} - \eta_{ref}(g)}, \quad (4.25)$$

The tip efficiency (χ_{tip}) ranking is presented on the right-hand side of Figure 4.84. The reference case is updated to be consistent with the nominal tip clearance value of the tip under investigation (FlatD for IN01D and AC06D, FlatT for AC07T). Overall, the relatively low values of tip efficiency shown by the three tip profiles suggest that there is still significant margin for improvement in terms of blade tip sealing effectiveness. Blade AC06D is observed to reach the highest tip efficiency $\chi_{tip} \sim 10\%$ despite having the largest fraction of tip leakage mass flow. The simple squealer tip, on the other hand, shows a smaller increase in efficiency ($\sim 5\%$). Blade AC07T shows a negative tip efficiency ($\chi_{tip} = -26.15\%$), meaning that the aerodynamic performance of the contoured surface is rather poor in comparison to a flat tip operated at equal clearance value.

The thermal performance of the blade tip geometries is gauged by the integral tip heat load, which drives the total amount of coolant mass-flow necessary to maintain a desired blade metal temperature. While only the tip surface was considered during the optimization phase, the final performance analysis accounts for the heat transfer

budget onto both the rotor and casing surfaces.

Figure 4.85 shows the characterization of the average heat flux and integral heat load of the three profiles. Small heat flux discrepancies are detected on the pressure side ($\sim 1.6\%$) and suction side ($\sim 3.3\%$) of the rotor tips under investigation. Higher variations are shown, conversely, in the rotor shroud region directly exposed to the tip surface (indicated as over-tip in Figure 4.85). The lower mean heat flux is shown, in this area, by tip IN01D in reason of the large squealer bath which channels the leakage fluid and actively decreases the casing convection (as discussed in Figure 4.47). The complex pattern of blade AC06D, conversely, promotes frequent over-rim accelerations of the tip flow and causes an early establishment of the direct by-pass mechanism. Due to the enhanced casing convection and higher $T_{ad,w}$ in the over-tip region (discussed in Figure 4.43), the mean heat flux is the highest among the three tip cases (+21% higher than for blade IN01D).

The contoured surface of blade AC07T reduces the local heat flux in the front part

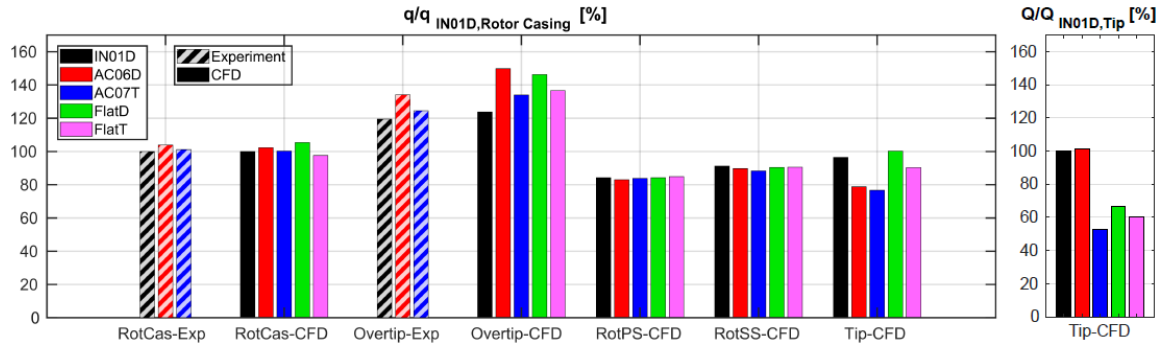


Figure 4.85: Heat flux performance parameters

of the airfoil. The supersonic tip-flow conditions and the over-tip flow separation in the last 40% of the axial chord greatly enhance the casing heat load in the aft part of the profile, placing the contoured tip in between the two squealer-type profiles in terms of average heat flux.

While the cavity bath of blade IN01D allows to greatly reduce the casing heat load, the hot fluid traveling on the cavity floor strongly affects the average heat flux of this rotor tip. The rimmed pattern of blade AC06D, conversely, limits the reattachment of the hot fluid on the tip surface, decreasing the heat flux to almost the same level of blade AC07T (-20% with respect to IN01D).

Blade AC06D reaches the maximum area-integrated heat load among the three tip profiles (Figure 4.85-right), due the large wetted surface of the multi-cavity design. The comparatively smaller tip surface of blade AC07T and the flow separation in the

aft part of the blade, which pushes the hot leakage fluid towards the outer casing, generate an integral thermal load on the tip which is 50% lower than in the squealer-type profiles.

4.7 Rainbow rotor performance

This section deals with the effectiveness of the rainbow rotor approach for the investigation of multiple blade geometries, focusing on the flow periodicity established at the measurement locations by the individual sectors of the rotor disk.

The analysis focuses on the casing static pressure and heat transfer and on the outlet pressure field. The experimental time-resolved signals are ensemble-averaged over each individual sector, retrieving the sector-mean flow field and evaluating the impact of the neighboring sectors on the peripheral blade passages. Full-stage, full-annulus NLH computations are interrogated in support to the experimental results, allowing to validate the sector-resolved flow physics and assess the appropriate selection of blade passages for the ensemble-averaging of the measured flow quantities.

4.7.1 Experimental flow variability

The high-frequency pressure and heat transfer signals measured at the over-tip casing of the three profiles are ensemble averaged over the central passages of each sector (five passages for IN01D and AC06D, four passages for AC07T). The level of flow periodicity of each individual sector is here assessed by computing the variation of the measured flow quantity over the selected blade passages.

The axial distributions of time-average static pressure variability are presented over

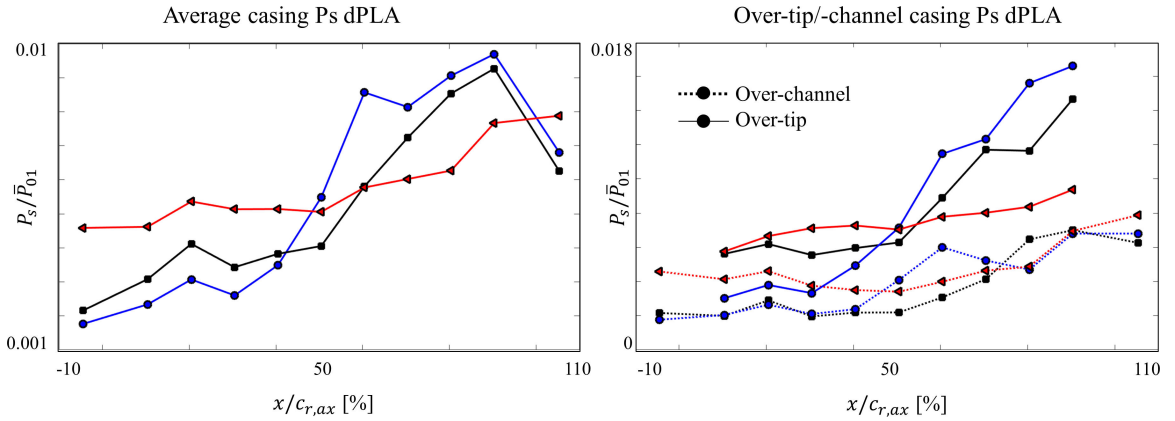


Figure 4.86: Axial distributions of casing static pressure passage-to-passive variability

the full extent of the rotor axial chord in Figure 4.86. On the left-hand side, the dataset has been pitch-averaged over the full rotor passage and is therefore representative of the mean pressure variability detected at the rotor casing. On the right-hand side, the averaging is performed separately in the over-tip and over-channel regions of the three profiles.

Blades IN01D and AC07T present a high flow repeatability in the first 50% of the axial chord, while the open leading-edge rim and the complex pattern of blade AC06D are shown to cause higher tip-to-tip differences already in the front part of the tip (0.5% P_{01}). In the aft part of the rotor airfoil, however, blades IN01D and AC07T present a steeper increase in pitch-average variability which leads them to overcome blade AC06D in the total variability budget. The increasing trend continues up to the rotor trailing edge, with the global value never exceeding, however, 1% of the stage inlet total pressure.

The mean passage-to-passage variation registered at the rotor casing in the time-

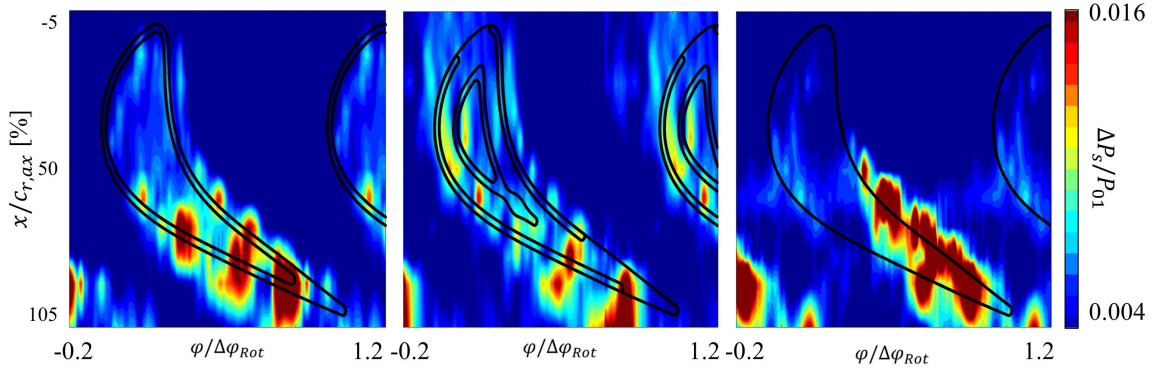


Figure 4.87: Maximum experimental channel-to-channel variation of casing static pressure for sectors IN01D, AC06D and AC07T (left to right)

resolved static pressure measurements is presented in Figure 4.87. The variation is computed as the difference between maximum and minimum P_s/P_{01} values recorded at each rotor phase location, averaged between the six sensors along the vane pitch. The global variability in the central passages of the three sectors is below 0.5% P_{01} with localized peaks of 2.5% P_{01} in close proximity to the TLV region (aft blade SS). The increased over-tip flow variability in the front part of AC06D is mainly located at the SS channel between the internal island and the peripheral rim. High flow variations are detected in the aft over-tip region of the contoured tip (AC07T), where the over-tip flow separation has been highlighted in Figure 4.38. These localized peaks of pressure variability, however, represent less than 7% of the maximum pressure fluctuation registered at this specific axial position. The variability globally stands below 0.15% of the fluctuation value in the over-passage regions.

Full-annulus NLH flow predictions have been interrogated to assess the passage-to-passage flow variability within the same blade channels considered in the experimental study. The numerical results, presented in Figure 4.88 show significantly lower flow variations with respect to the experimental dataset.

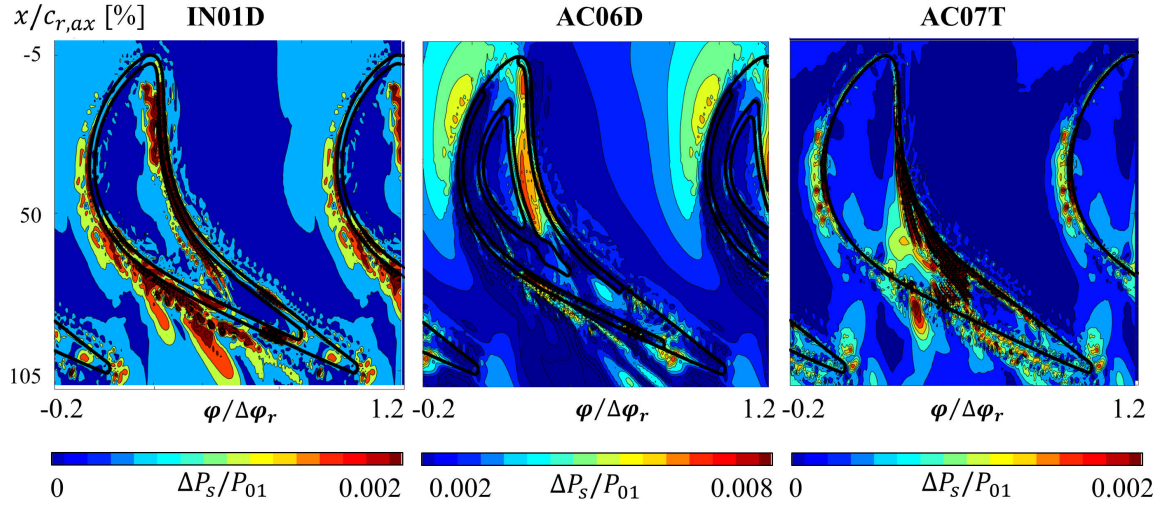


Figure 4.88: Predicted channel-to-channel variation of casing static pressure (full-annulus NLH)

The over-channel flow repeatability is almost everywhere within $0.1\%P_{01}$. Blade IN01D shows higher pressure variations along the perimeteral rim, with peaks up to $0.4\%P_{01}$ in the front part of the PS. Blade AC06D shows the highest over-tip variability, particularly evident along the PS channel ($0.8\%P_{01}$). The near-TLV region is confirmed to have, globally, a lower level of flow repeatability. The mid-chord PS and aft part of blade AC07T show significantly lower variability with respect to the experimental results.

In conclusion, a higher sector flow repeatability is shown in the NLH predictions, with variability peaks located along the front-PS rim of the squealer profiles and at the SS rim of blade IN01D. The aft over-tip flow separation of blade AC07T does not produce the same high variability pattern as in the experimental results. As the NLH computations allow to only model the flow unsteadiness correlated with the blade passing event, this result may suggest that part of the measured variability, rather than being caused by the multi-sector rotor setup, is due to tip-to-tip geometrical discrepancies or to stochastic flow contributions.

A good level of passage-to-passage repeatability is displayed by the shroud heat flux signals, shown in Figure 4.89. The passage-to-passage variations are, in this case, globally below 10% of the mean casing heat load of blade IN01D. This level of variability corresponds to less than 5% of the measured fluctuation amplitude and stands well below the average level of uncorrelated unsteadiness detected in the measurements. The less repeatable regions are in close proximity to the TLV region and on the aft surface of blade AC07T, showing a good correlation with the static pressure

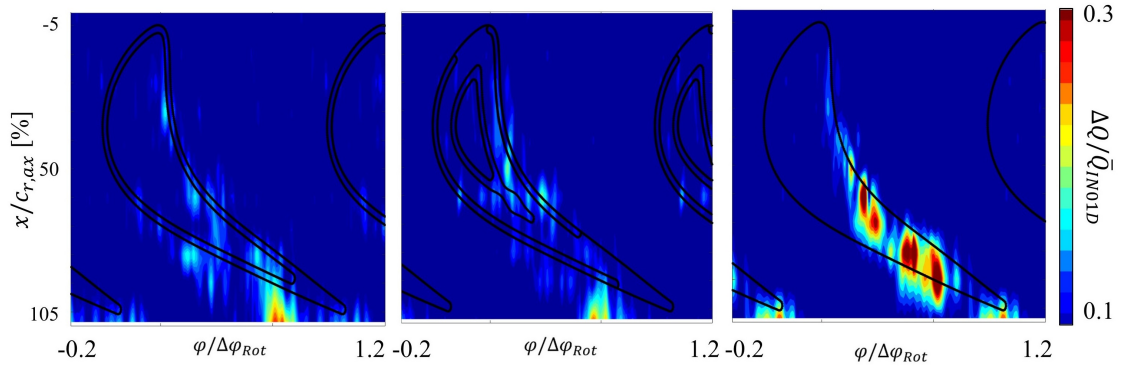


Figure 4.89: Maximum experimental channel-to-channel variation of casing heat flux for sectors IN01D, AC06D and AC07T (left to right)

analysis.

Figure 4.90 displays the pitchwise-average heat flux variability levels for the three

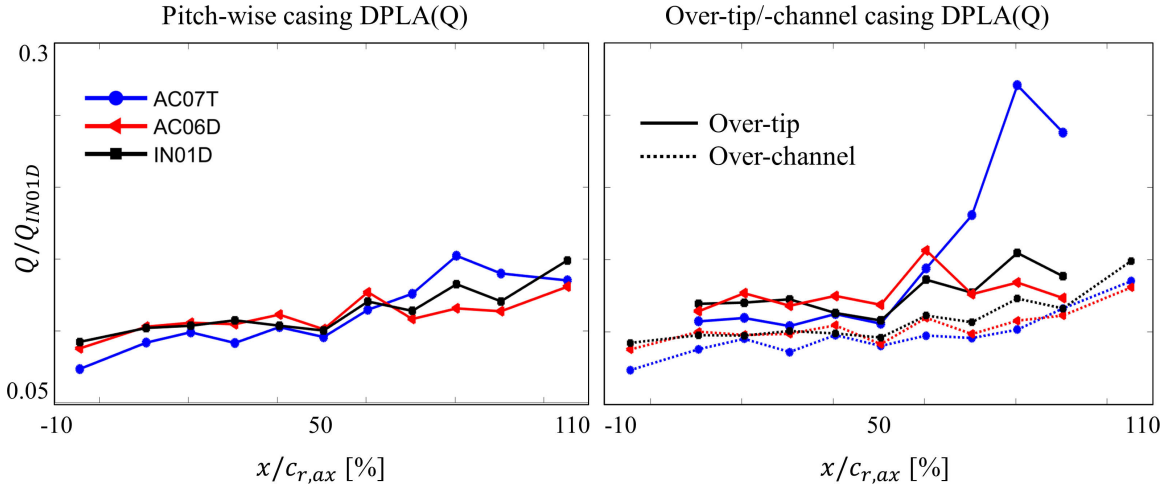


Figure 4.90: Axial distributions of casing heat flux passage-to-passage variability

profiles. As for the pressure signals, the heat flux variability has been averaged over the full rotor passage (left) and by separating the over-tip and over-channel regions (right). The passage-to-passage heat flux variations show a globally increasing trend, starting from an overall average of $3 - 5\% Q_{IN01D}$ at the rotor inlet and reaching a level of about $9 - 10\% Q_{IN01D}$ at the rotor trailing edge. The over-channel distributions show an overall uniform trend up to $70\% c_{r,ax}$, where a slight increase is registered in reason of the presence of the TLV.

In the over-tip region, conversely, a steep increase is registered for blade AC07T starting from 50% of the rotor axial chord. The global peak, reached at $80\% c_{r,ax}$ amounts to nearly $30\% Q_{IN01D}$. As already highlighted in Figure 4.89, this increased flow variability corresponds to the massive over-tip flow separation in the aft part of blade

AC07T, which introduces a high stochastic content in the heat flux signals. Such high level of non-correlated unsteadiness, in this case, is not completely ruled out by the phase-lock averaging, and persists as a source of flow variability.

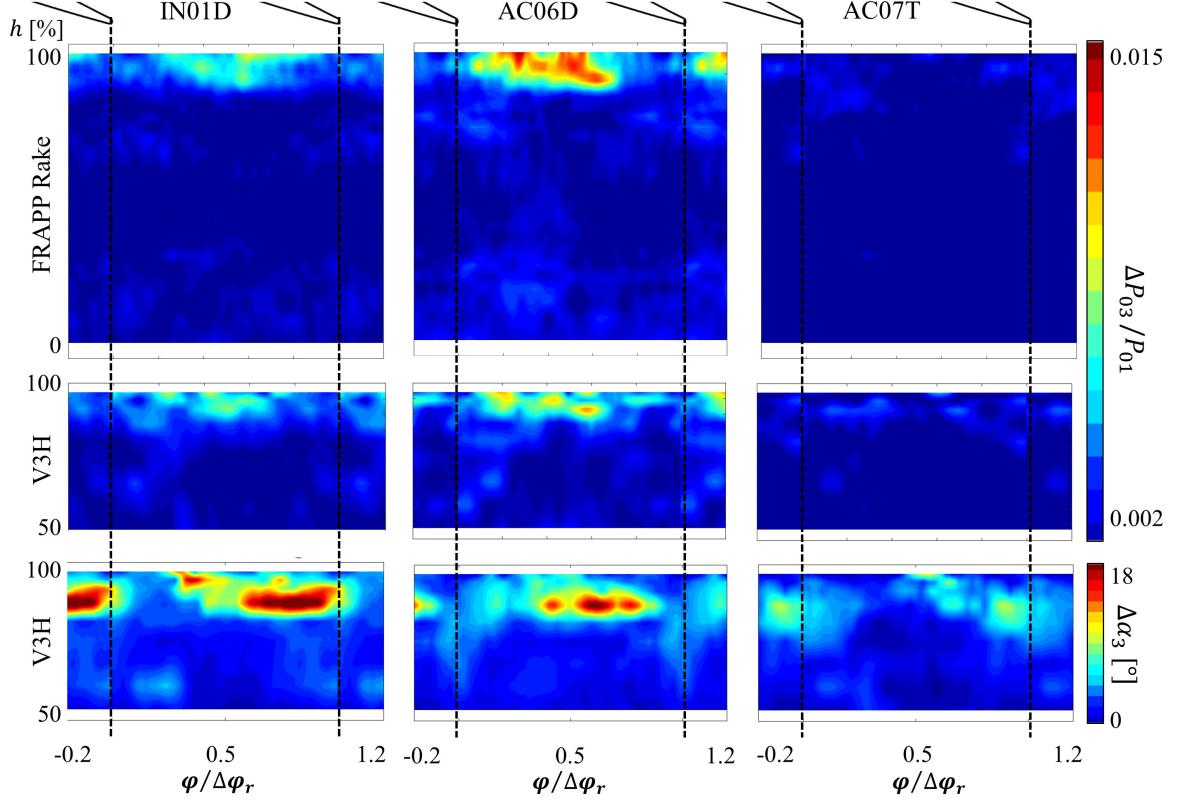


Figure 4.91: Maximum channel-to-channel variation of outlet total pressure measured by FRAPP rake (top) and V3H probe (center) and variation of yaw flow angle by V3H probe (bottom) for sectors IN01D, AC06D and AC07T (left to right)

The contour plots in Figure 4.91 show the sector-mean passage-to-passage variation registered at the stage outlet for the total pressure measurements performed by FRAPP rakes (top row) and V3H probe (central row). Additionally, the flow angle variability measured by the V3H probe is reported in the bottom row. Consistently with the data reduction applied for the flow analysis, four central rotor blade channels are considered in the ensemble-averaging sectors IN01D and AC06D and three blade channels are considered for tip AC07T.

Overall, the passage-to-passage variation of total pressure remains below 0.3% of P_{01} for both measurement techniques, that is below 5% of the peak-to-peak fluctuations of the mean passage signal (as presented in Figure 4.49). The maximum passage-to-passage variation, amounting to 1.5% P_{01} , is reached in the upper 10% of the rotor

blade span in blades AC06D and IN01D, which present a stronger TLV. Significantly higher passage periodicity is shown in the same region for blade AC07T, due to the reduced operative clearance and to the weaker TLV. Significant agreement is shown between the two sets of data, providing an important cross-validation of the sector-wise P_{03} variability, which is generally below $\sim \pm 1.5 \text{ mbar}$ and does not vary between different measurement systems.

The bottom row shows the variation of absolute yaw flow angle recorded by the virtual-three-hole probe. The passage-to-passage differences do not exceed $\sim \pm 1 - 1.5^\circ$ between 50% and 80% of the blade span. Localized peaks of $\sim \pm 7^\circ$ are shown in the near-casing region of blades IN01D and AC06D, at a good level of correlation with the position of the maximum variability detected in the pressure measurements. The variability is limited to a maximum of $\pm 3^\circ$ in blade AC07T, in consonance with a weaker TLV.

As discussed for the casing measurements, the flow variability peaks at Plane 03 might be partly due to the residual stochastic component of the TLV in the phase-lock averaged flow, in which case they do not indicate a significant variability in the flow topology of the different passages (as will be further discussed in Section 4.7.2). A further, weaker, contribution might come from mild tip-to-tip geometry discrepancies within sectors due to manufacturing tolerances.

4.7.2 Sector Periodicity

While the measured passage-to-passage variability allows to identify the regions with lowest flow periodicity within each rotor blade tip, a deeper insight into the sector flow physics is necessary in order to motivate the measured flow variability.

The contour plots in Figure 4.92 and Figure 4.93 show the full-sector static pressure field established by the three tip profiles in the rotor domain. The numerical predictions, generated by full-annulus NLH computations, are compared to the sector-average static pressure field measured during a short duration test. The experimental data have been phase-lock averaged over 25 rotor revolutions and further ensemble-averaged among the sensors sharing the same axial position to cancel out the vane-induced peripheral flow gradients. The resulting pitch-averaged PLA signals have been directly employed to reconstruct the phase-resolved flow field over each sector of the rainbow rotor.

In the present analysis, the sector blades are enumerated in increasing order from left to right, with zero-phase reference being the blade tip leading edge. The rotor channels are denoted by the numbers of the adjacent blades defining them (i.e. channel

1-2 indicating the first channel of each individual sector, channel 2-3 the second and so on).

In the attempt to provide a more precise quantification of the possible sector-wise

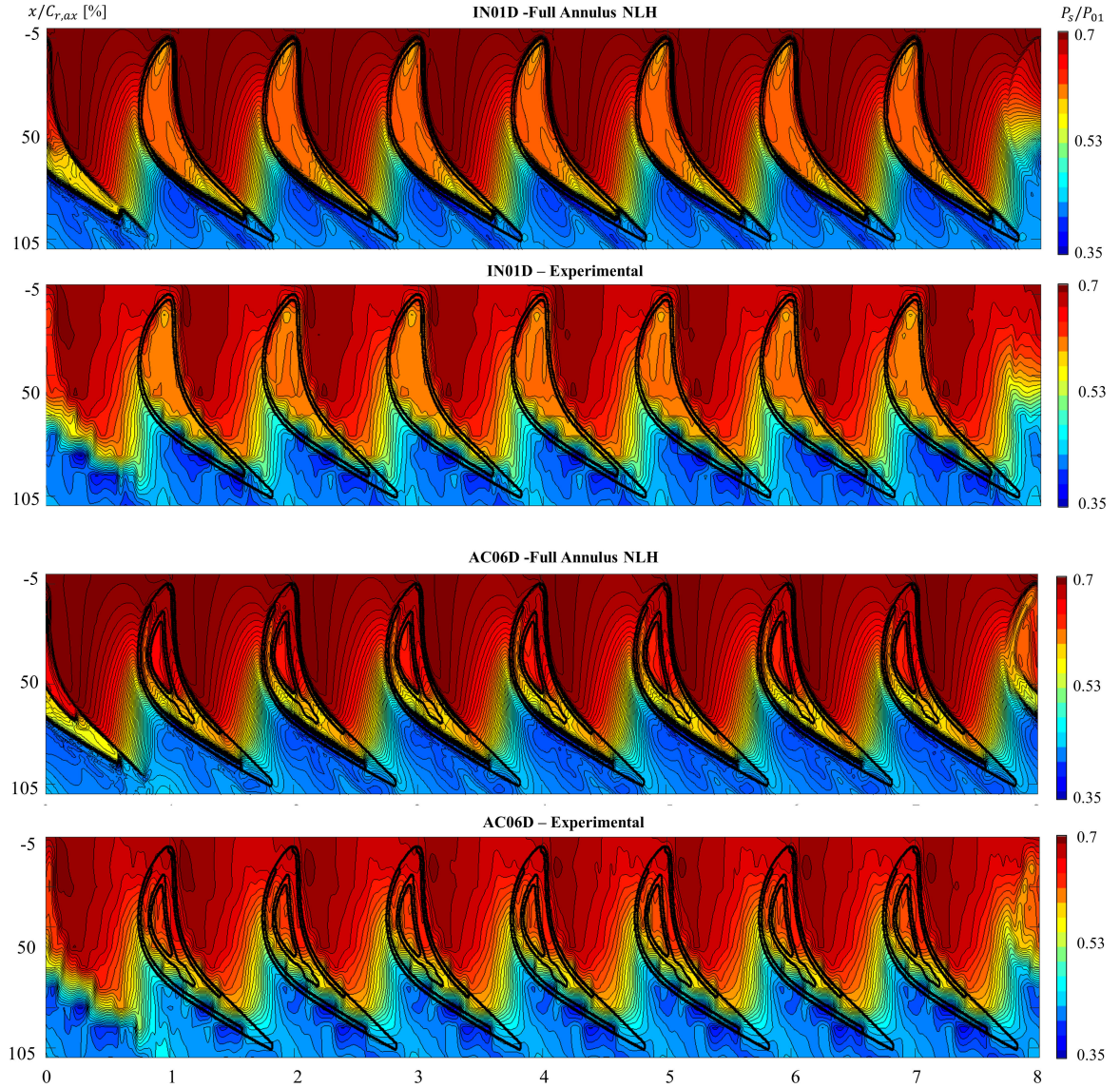


Figure 4.92: Full-sector casing static pressure field: tips IN01D and AC06D

flow variations, the pressure fluctuations at 20%, 50% and 80% of the rotor axial chord are extracted and presented concurrently in Figure 4.94. This helps to put into perspective the blade-to-blade variations in each single sector and demonstrate how the sector-wise flow variations appear globally negligible with respect to the sector-to-sector differences in flow topology and fluctuation intensity.

For an enhanced readability, Figure 4.95 and 4.96 present the time histories plotted with individual ranges tailored to highlight the measured and predicted sector-wise

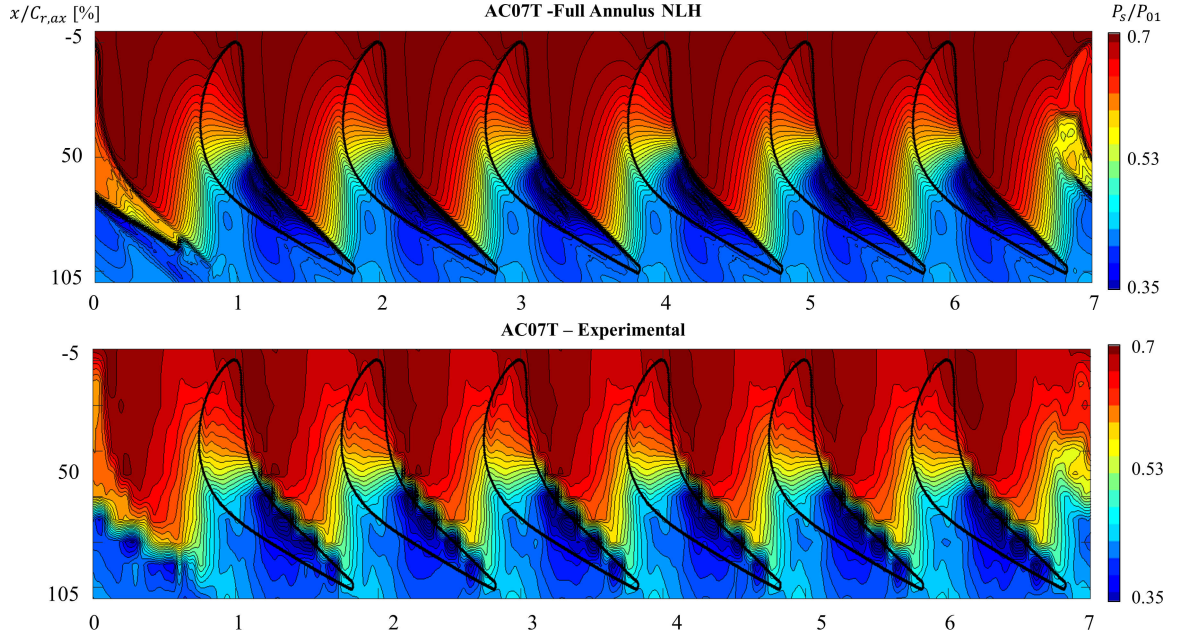


Figure 4.93: Full-sector casing static pressure field: tip AC07T

flow variability.

Numerical and experimental results show a high level of flow repeatability in the over-channel region of sectors IN01D, with the progressive flow expansion pattern which appears unchanged between channel 1-2 and 5-6. A slight positive offset ($\sim 2\%P_{01}$) is shown in the CFD results (red line in Figures 4.95 and 4.96), consistent with the analysis performed in Fig. 4.12. The smoother numerical contours, in this case, are due to the slightly coarser grid employed to mesh the rotor domain in the full-annulus computation.

Mild flow variations are measured over the squealer cavity of blade IN01D, in concordance with the analysis carried out for Figure 4.87. The cavity flow of blades 1, 3 and 7 appears to produce a lower pressure distribution in the contours, in accordance with a lower mean ($\sim -5\text{mbar}$ offset) level shown at 20% of the rotor axial chord in Figure 4.95 (marker A). This decrease, however, is not encountered in the numerical predictions. A similar pattern is visualized at 50% of the rotor axial chord, with blades 4,5 and 6 showing a slight positive offset ($\sim +4\text{mbar}$) in the over-tip area (marker B).

A higher rate of channel-to-channel pressure variation is visualized in the near-SS and near-PS regions in the first 20% $c_{r,ax}$ of blade AC06D, partly matched by the numerical predictions, which however do not modify significantly the topology of the flow field. A closer look at the fluctuation patterns in Figure 4.95 reveals a slight vari-

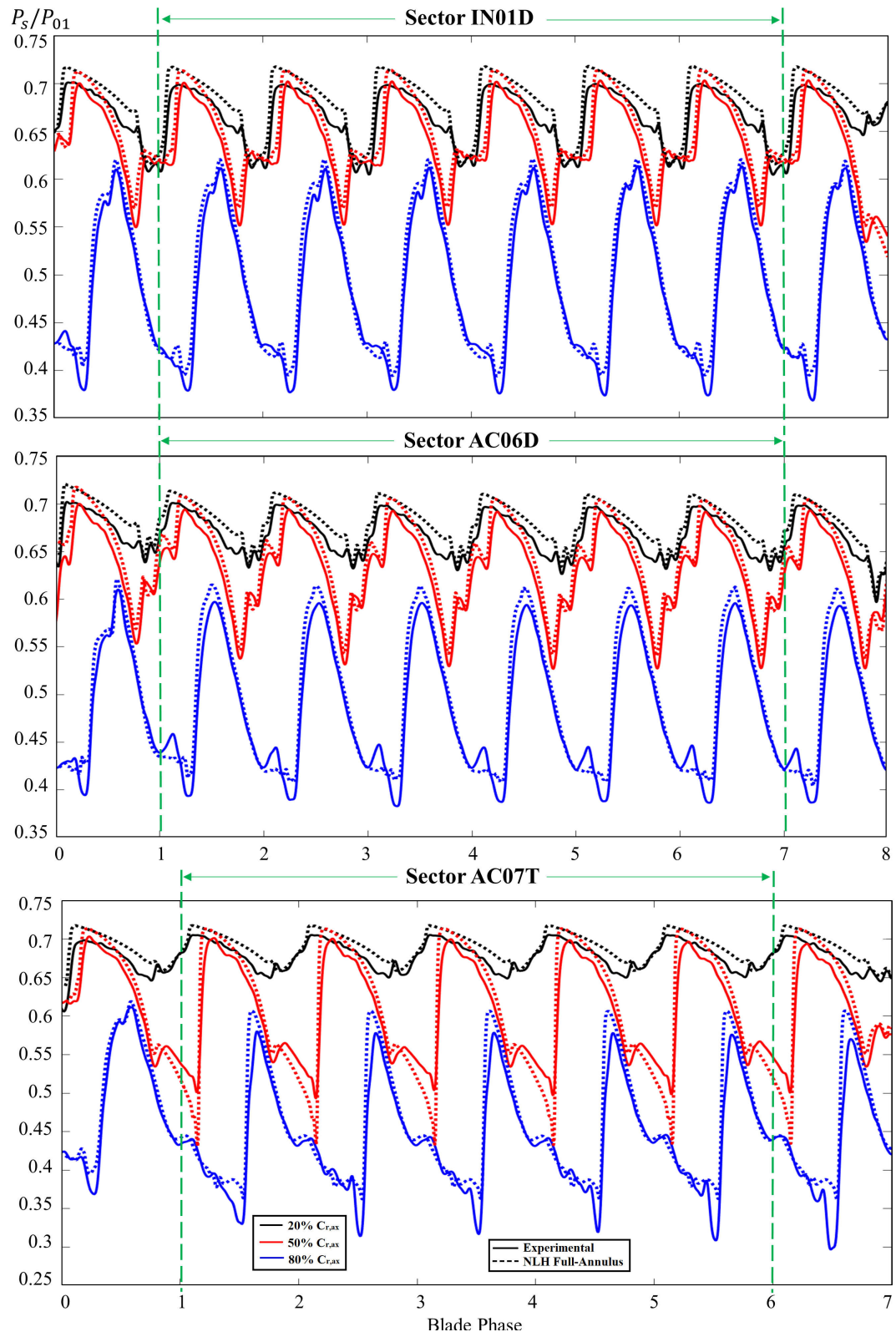


Figure 4.94: Casing pressure fluctuations at selected axial stations

ation in the over-tip pressure fluctuation at 20% $c_{r,ax}$ (marker C), with a maximum difference of ~ 6 mbar between airfoils 5 and 6. This variation is presumably due to manufacturing tolerances, given the absence of relevant trends developing along the sector or similar discrepancies in the numerical dataset.

For a better visualization of the sector-wise trends, the maximum and minimum pressure value registered in each blade passage are monitored in Figure 4.97. Solid lines indicate the experimental results while dashed lines indicate the full-annulus NLH results. All values are reported as the variation from the first blade of each sector.

At 20% $c_{r,ax}$, the experimental pressure minimum of blade AC06D is seen to decrease monotonically ($-0.8\%P_{01}$) between blade 1 and 7, with blade number 5 being confirmed as an outlier from this trend. In the NLH predictions, the decrease is mostly located between blades 1 and 2, while a uniform pressure minimum is detected between blade 2 and 7.

At 50% $c_{r,ax}$, the numerical computations match well a global decrease of the pressure minimum of $2.5\%P_{01}$ along the sector (marker D in Figure 4.95). The highest drop is registered between the SS of blade 01, exposed to the channel shared with the neighboring sector, and the SS of blade 02 ($-1.5\%P_{01}$).

At 80% $c_{r,ax}$ of the two squealer-type profiles, the measured variations tend to show less significant trends along the sectors. Figure 4.95 shows that, at 80% $c_{r,ax}$, the low-pressure regions in the near-tip SS of blade IN01D present a mild qualitative difference from the NLH predictions (marker E). The tip-to-tip discrepancies (not present in the NLH results) confirm the lower flow repeatability measured in the near-TLV zones in Figure 4.87.

The pressure maxima at the PS of blade IN01D (marker F in Figure 4.95) present a globally increasing trend at 80% $c_{r,ax}$, partially matched by the NLH result. Figure 4.97 indicates that the global increase with respect to blade 01 does not exceed $0.3\%P_{01}$ at any of the axial locations under investigation.

In sector AC06D, the decrease of PS pressure peak between blade 01 and blade 06 ($\sim 0.4\text{--}0.5\%P_{01}$) is matched quantitatively by the NLH predictions at 20% and 80% of the axial chord, while slightly underestimated at mid-chord. The high level of matching with the experimental results, however, confirms that the sector-wise pressure decrease at both PS and SS of AC06D is due to the rainbow configuration and is not driven by experimental assembly tolerances.

This analysis serves to validate the choice of blades 2 to 6 for the casing pressure ensemble averaging in the squealer profiles. For tip IN01D the maximum variation

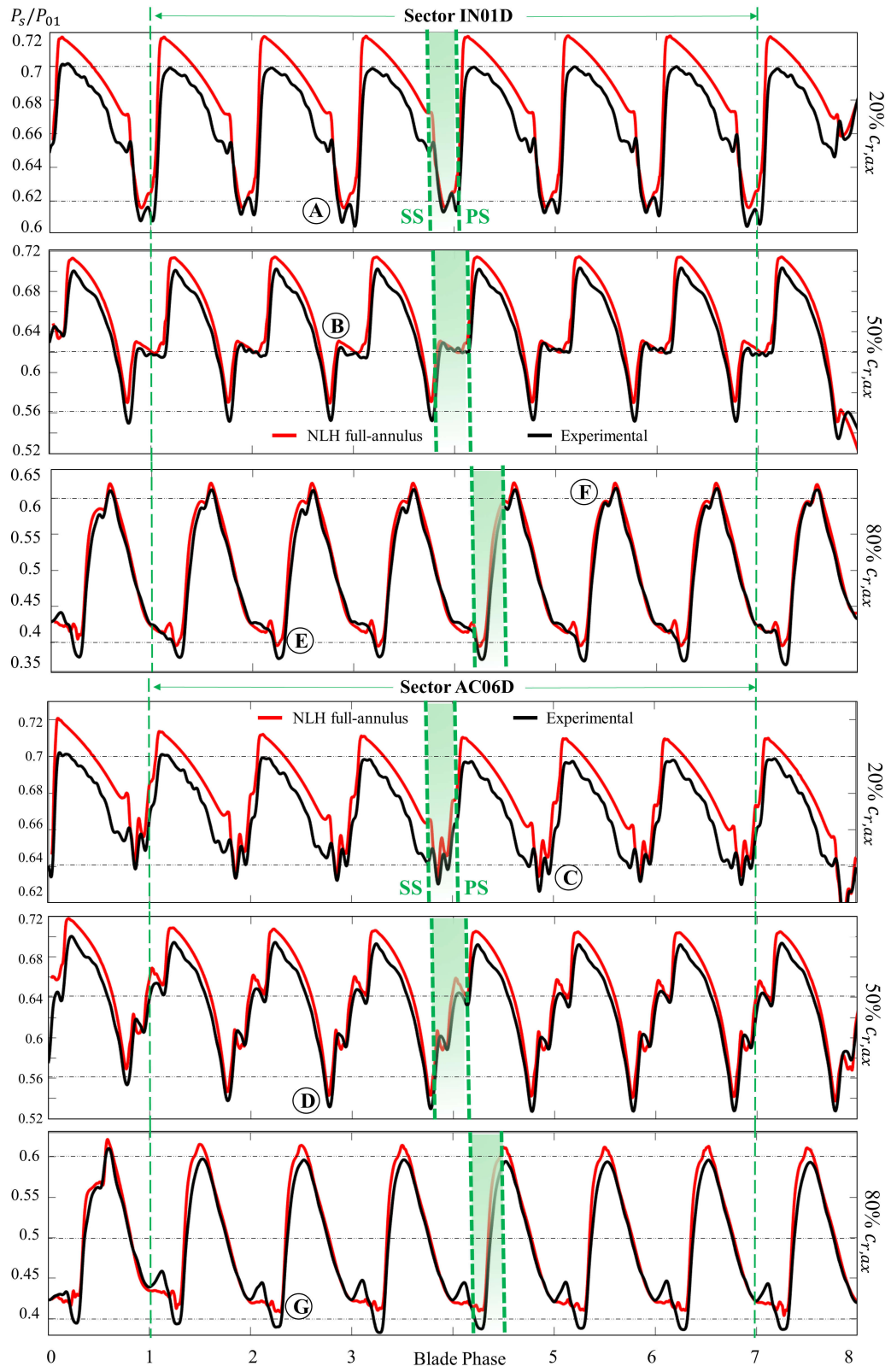


Figure 4.95: Casing pressure fluctuations: sec. IN01D and AC06D

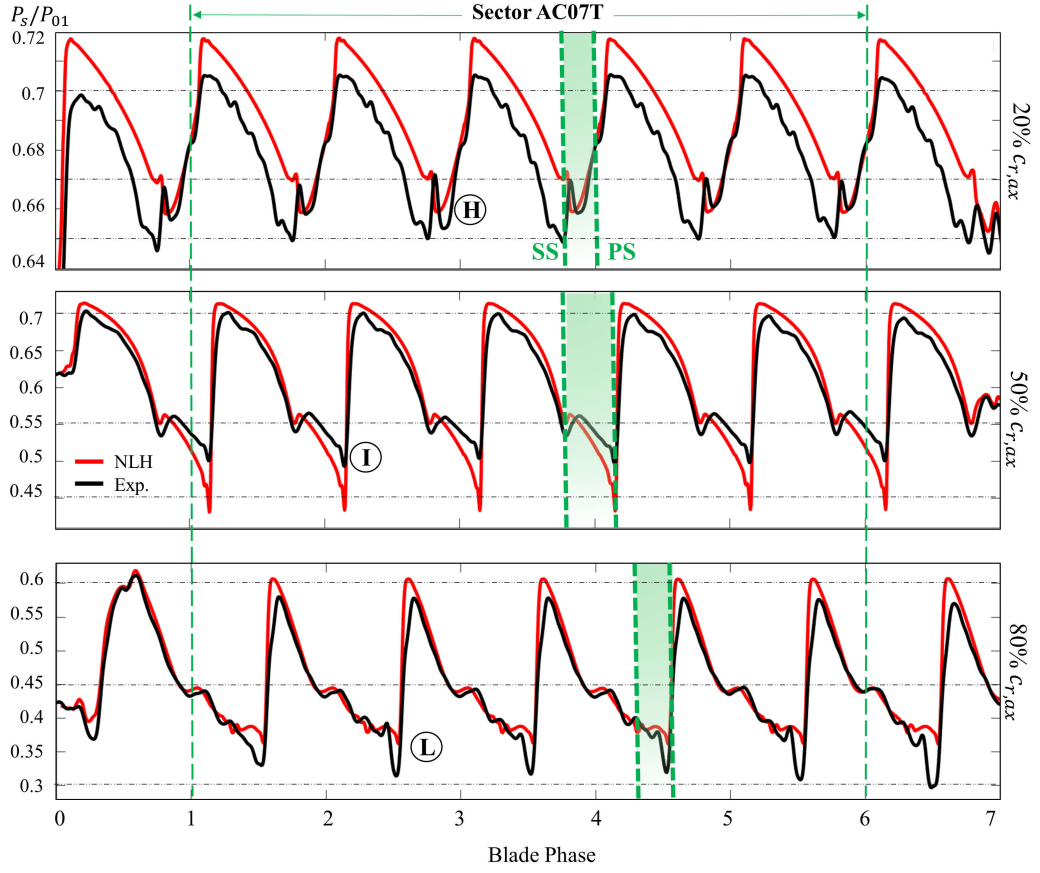


Figure 4.96: Casing pressure fluctuations: sector AC07T

in this averaging domain is registered at the SS of blade 3, at 20% $c_{r,ax}$, equal to -4 mbar with respect to the sector-average value. This discrepancy is believed to be due to a manufacturing tolerance of the blade, even if no significant clearance variation was measured at the front SS of this tip with respect to the neighboring blades, as reported in Figure 2.10. For blade AC06D, the highest pressure variation is usually registered between blade 01 and blade 02, while within the ensemble-averaging domain δP_s is lower than $1\%P_{01}$ at the blade SS and $0.3\%P_{01}$ at the blade PS.

The pressure field established over sector AC07T presents, as expected, substantial differences with respect to the squealer-type profiles.

The over-channel pressure field, shown in Figure 4.93, shows a good repeatability along the sector, providing a quantitative validation of the predicted flow expansion pattern. The pressure history extracted at 20% $c_{r,ax}$, presented in Figure 4.96, shows the near-PS global maximum varying between $70.44\%P_{01}$ (blade 6) and $70.52\%P_{01}$ (blade 1), that is $\sim 1\text{ mbar}$ sector-wise variation. This variation is seen to increase to $\sim 7\text{ mbar}$ at 50% $c_{r,ax}$ and $\sim 10\text{ mbar}$ at 80% $c_{r,ax}$.

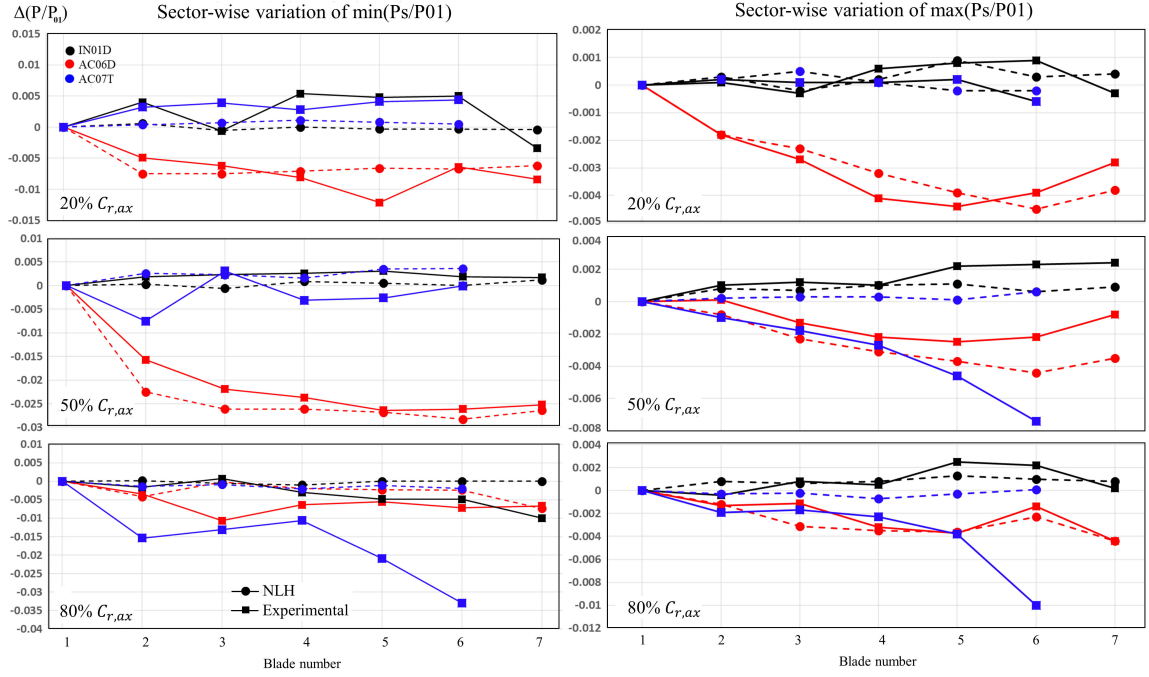


Figure 4.97: Sector-wise variation of rotor passage P_s/P_{01} maxima (left) and minima (right) for the three sectors: experiments vs. FA-NLH

Figure 4.97 shows that the trend of the PS pressure maximum is monotonically decreasing at these axial positions, as a difference from the NLH predictions. The decrease results maximized in the aft SS of the blade, at 80% $c_{r,ax}$, where blade 06 exhibits a pressure minimum lower by 3% P_{01} (~ 30 mbar) than blade 01. Comparing these trends with the measured clearance values presented in Figure 2.10, a high correlation is found between the static pressure decrease and the progressive increase of clearance between blade 1 and 6. The outlier, in this case, is represented by blade 02, which shows the highest SS pressure decrease at 80% $c_{r,ax}$, driven by a higher gap in the front part of the tip as reported from the dial gauge measurements.

In the over-tip region, mild flow variations are shown at the different axial locations in Figure 4.96. At 20% $c_{r,ax}$, a small discrepancy in fluctuation shape is detected over blade 3 (marker H). The absence of similar anomalies in other tips, together with no confirmation in the CFD dataset, suggests that this discrepancy is due to shape tolerances of the tip, not noticeable in Figure 2.10.

From the pressure contours in Figure 4.93, the numerical dataset is shown to predict a slightly steeper acceleration of the over-tip flow field. As a result, at 50% $c_{r,ax}$ the numerical pressure minimum slightly underestimates the measured value ($\sim 50 - 55$ mbar), as indicated by marker I.

In the aft part of tip AC07T, the NLH computations are shown not to capture

completely the absolute value of the SS pressure minimum, over-predicting it by $\sim 40\text{mbar}$. The measured flow variability is intensified in the over-tip region, in particular near the blade SS (marker L). While the measured sector-wise pressure minimum varies by $0.03\% P_{01}$ between blade 1 and blade 6, Figure 4.97 demonstrates that the choice to limit the ensemble averaging to blades 02 to 05 reduces the pressure variability to $\sim 5\text{mbar}$ ($0.005\%P_{01}$).

The phase-resolved outlet total pressure field established by the three sectors under investigation is presented in Figure 4.98. The numerical contours are generated by full-annulus NLH computations while the experimental contours have been recorded by FRAPP rakes. The datasets are plotted in the rotor domain, so that no vane-dependent circumferential flow gradients are present. The rotor blades are traveling from right to left in the plots and the horizontal axis shows the rotor passage phased with the airfoils trailing edge.

In the present plots, the airfoils of the investigated sectors correspond to the phase positions from 1 to 7 for the squealer-type profiles and from 1 to 6 for tip AC07T. With the measurement plane located at a distance of $50\%c_{r,ax}$ from the rotor TE, the signature of the sector flow is visible between passages 1-2 and 7-8. To aid the quantification of the sector-wise flow differences and the interpretation of the results, the pressure histories at 45% , 73.5% and $88.5\% h_r$ are extracted and presented in Figure 4.99. The sector-wise variation of pressure minimum and maximum for each rotor passage is monitored in Figure 4.100.

The outlet pressure field of blade IN01D shows a high degree of sector repeatability in the lower half of the blade span. At $45\% h_r$, the experimental and numerical results agree in showing an overall stable pressure level between passages 1-2 and 7-8 (marker C in Figure 4.99). Figure 4.100 confirms a variability below ± 1 mbar between blade 01 and 07 at this radial station.

At $45\% h_r$, blade AC06D shows a more significant sector-wise decrease of mean total pressure (~ 6 mbar), measured between channel 1-2 and channel 7-8 (marker E). The NLH predictions match within ± 1 mbar the experimental minima and maxima variation in Figure 4.100.

The sector flow variability increases with increasing radius in both squealer-type profiles. At $73.5\% h_r$ the predicted flow field in Figure 4.98 shows a higher level of mixing with respect to the single-tip simulations, probably due to the coarser mesh employed in the FA-NLH. At this location, the UPV-induced fluctuation measured in sector IN01D amounts to $\sim 13\text{mbar}$ against a prediction of $\sim 16\text{mbar}$ and is maintained stable along the central passages of the channel (2-3 to 6-7). Interestingly,

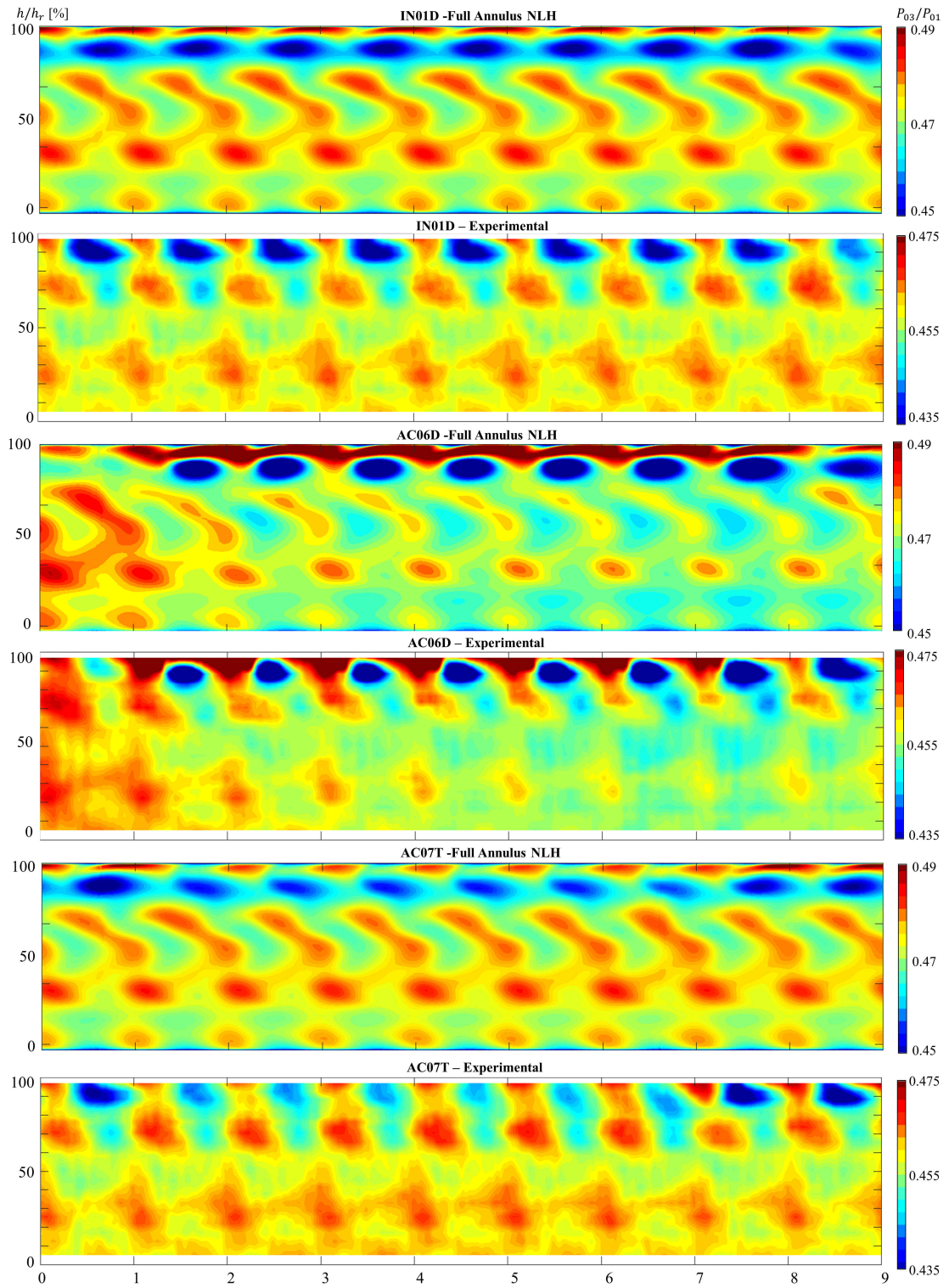


Figure 4.98: Full-sector outlet total pressure field

the peripheral blade passages are predicted to have a lower fluctuation amplitude (-2mbar) with respect to the central channels, while a slight amplification is verified

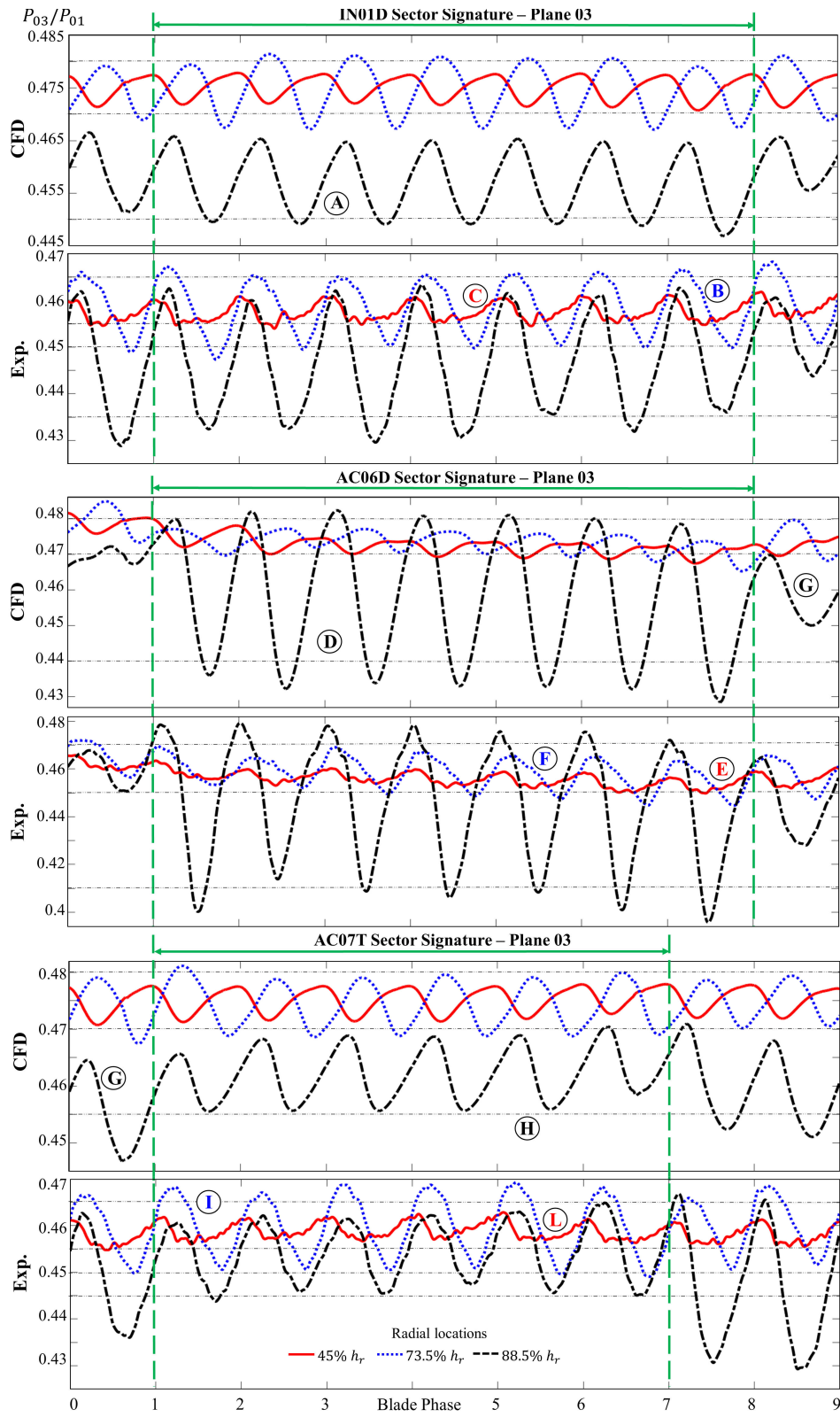


Figure 4.99: Total pressure fluctuation at selected radial positions

in the experimental dataset (marker B) which might suggest a slight reinforcement of the UPV. This difference is noticeable in Figure 4.100, where a discrepancy of $\sim 0.2\%P_{01}$ is found in the mean pressure variation with respect to blade 01 for both the minimum and maximum values at $73.5\% h_r$.

In blade AC06D the mean sector pressure level at $73.5\% h_r$ shows a global decrease of 8-9 mbar in the experimental dataset, against a NLH-predicted sector-wise decrement of 4 mbar (marker F).

In the near-casing area, at $88.5\%h_r$, the TLV-induced fluctuation presents a more significant variability in the experimental dataset. The fluctuation amplitude in blade IN01D is ~ 25 mbar in channel 5-6 and ~ 35 mbar in channel 3-4, while the numerical predictions show a constant amplitude of 16-16.5 mbar between blades 1 and 6 (marker A). This variability is confirmed by the trends of measured minima and maxima in Figure 4.100.

An even stronger amplitude variation is measured in the TLV-fluctuations of tip AC06D, with a tip-to-tip variation up to ~ 12 mbar between passage 1-2 and 2-3. This flow pattern, however, is not retrieved in the numerical simulations, which only predict a slight variation in passage 2-3 (marker D). In Figure 4.100, the pressure minimum is observed to grow by almost 15 mbar ($0.015\% P_{01}$) between blade 01 and 02. The predictions show a globally decreasing trend through the sector (-7 mbar sector-wise pressure drop), qualitatively matching the experiments between blade 3 and 7. The predicted maximum is seen to first increase by $0.2\% P_{01}$ in blades 01 to 03 and then decrease by $0.4\% P_{01}$ between blades 03 and 07, while the experimental results show a globally decreasing maximum for a total drop of 0.6% across the sector. The measured P_{03} contour of blade AC07T, shown in Figure 4.98, presents near-casing high pressure patches established in between the TLV-induced pressure minima, which extend radially inwards and connect to the high pressure structures at $70\% h_r$. This behavior is not retrieved in the predicted flow field. In reason of this, the pressure fluctuations measured at $73.5\% h_r$ present a nearly double amplitude (~ 18 mbar) with respect to the predicted ones (marker I, Figure 4.99). Figure 4.100 shows that the measured minima and maxima at this location are repeatable within $0.2\% P_{01}$ and validate well the sector-wise variation predicted by the NLH. A high quantitative agreement is found at $45\% h_r$, where the pressure fluctuations amplitude is $\sim 4.5 - 5$ mbar in both datasets (marker L).

Sector AC07T shows a higher mean pressure level in the near-casing region with respect to the other tips. This is confirmed by the pressure history at $88.5\% h_r$ visualized in Figure 4.99, where AC07T is shown to generate a mean pressure higher

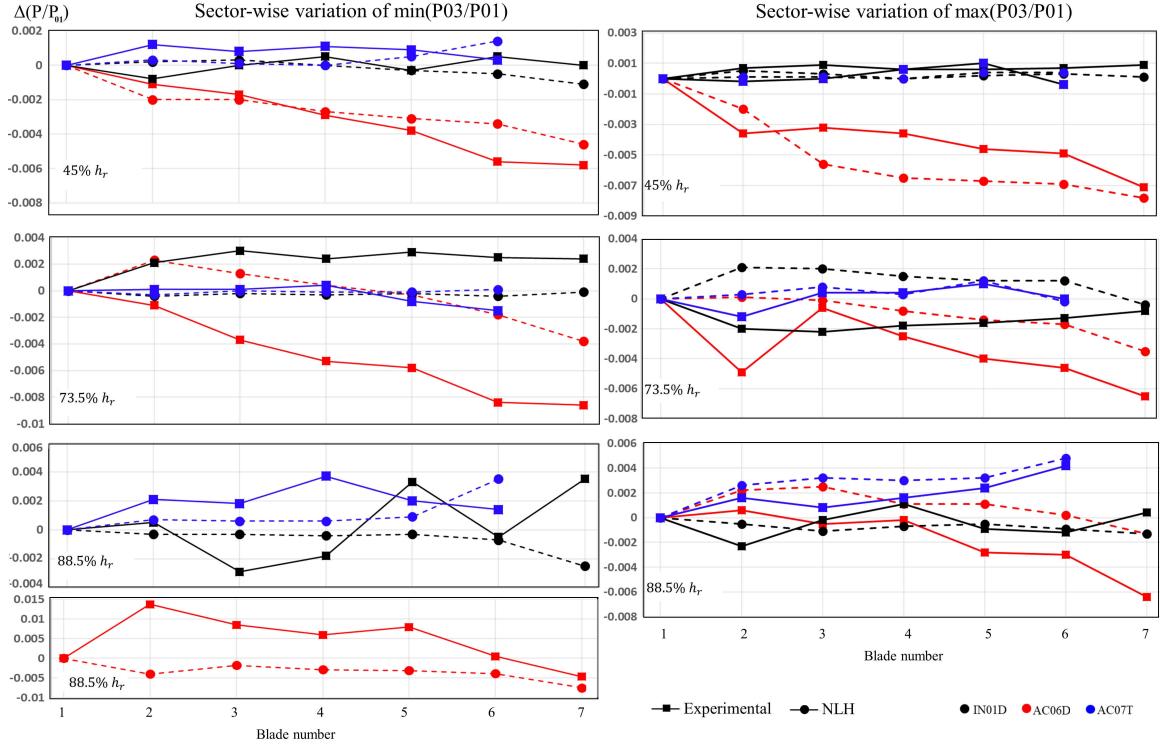


Figure 4.100: Sector-wise variation of rotor passage P_{03}/P_{01} maxima (left) and minima (right) for the three sectors: experiments vs. FA-NLH

by 5 mbar with respect to the neighboring sector IN01D (marker G). At 88.5% h_r , the predicted fluctuation amplitude and mean value are remarkably uniform in the central channels of the sector (marker H), varying within 1 mbar over channels 2-3 to 5-6. The measured fluctuation amplitude is generally higher than predicted, with an average peak-to-peak value of $\sim 1.6\%P_{01}$ against a predicted $\sim 1.3\%P_{01}$. This demonstrates a slightly higher level of flow mixing in proximity to the TLV for the full-annulus simulations, as indicated by the pressure contours in Figure 4.98.

The only inconsistency measured within the central passages of AC07T at 88.5% h_r is the slightly higher pressure minimum of blade 4 (+2mbar offset) observed in Figure 4.100. The pressure maxima evolution along the sector shows a consistent quantitative trend in the measured and predicted datasets, both showing a global increase of 4 mbar between blade 1 and blade 6.

In conclusion, this study motivates the selection of four passages for the sector-averaging of IN01D and AC06D (blades 3-6) and of three passages in the case of tip AC07T (blades 3-5). The choice targeted to minimize the impact of the sector flow variability while keeping a statistically relevant number of blade passing events per short-duration test ($4 \times 25 = 100$ events for squealer-type profiles, $3 \times 25 = 75$ events

for AC07T). In the following section, the effect of the averaging domain on the rotor-mean P_{03} field is illustrated.

4.7.3 Sector-averaging study

A parametric study is conducted in order to assess the impact of the channels selection on the sector-average flow field evaluation. The analysis is limited to the outlet pressure field, which has shown the highest rate of blade-to-blade variability. Full-annulus NLH computations and experimental results are compared.

The predicted P_{03}/P_{01} field is presented in Figure 4.101 in the rotor-relative frame. The sector-wise ensemble average is computed over different selection of rotor blade channels. The averaging domain includes (top to bottom) one, three and five central channels for the squealer-like tips and one, two and four central channels for the carved profile. The time-resolved experimental dataset is ensemble-averaged over the same channel sets and presented in Figure 4.102.

In agreement with the flow repeatability study, the choice of different sets of rotor channels within the sectors does not generate substantial modifications in the rotor-mean flow topology. In the near-casing region, the pressure pattern generated by the TLV shows little or negligible variations between 1- and 3-passage averages.

A more evident effect is, on the other hand, the higher mixing detected at 70% of the rotor span for increasing number of averaging channels in blade AC06D (marker A) and IN01D (marker B). The pressure peak at this location results indeed slightly attenuated in the 5-passage average of the squealer-type profiles, and in a minor way for blade AC07T (marker C). The agreement of experimental and numerical results over this effect confirms its correlation with the rainbow setup.

In the lower 50% of the blade span, the only evident effect of the averaging domain is the amplitude of the pressure peak at $30\%h_r$, indicated by marker D in blade AC06D and marker E in blade IN01D. The increased mixing at this location is particularly evident in the experimental contours of both squealer-like tips and represents probably the most significant effect of the channel selection retrieved in this study.

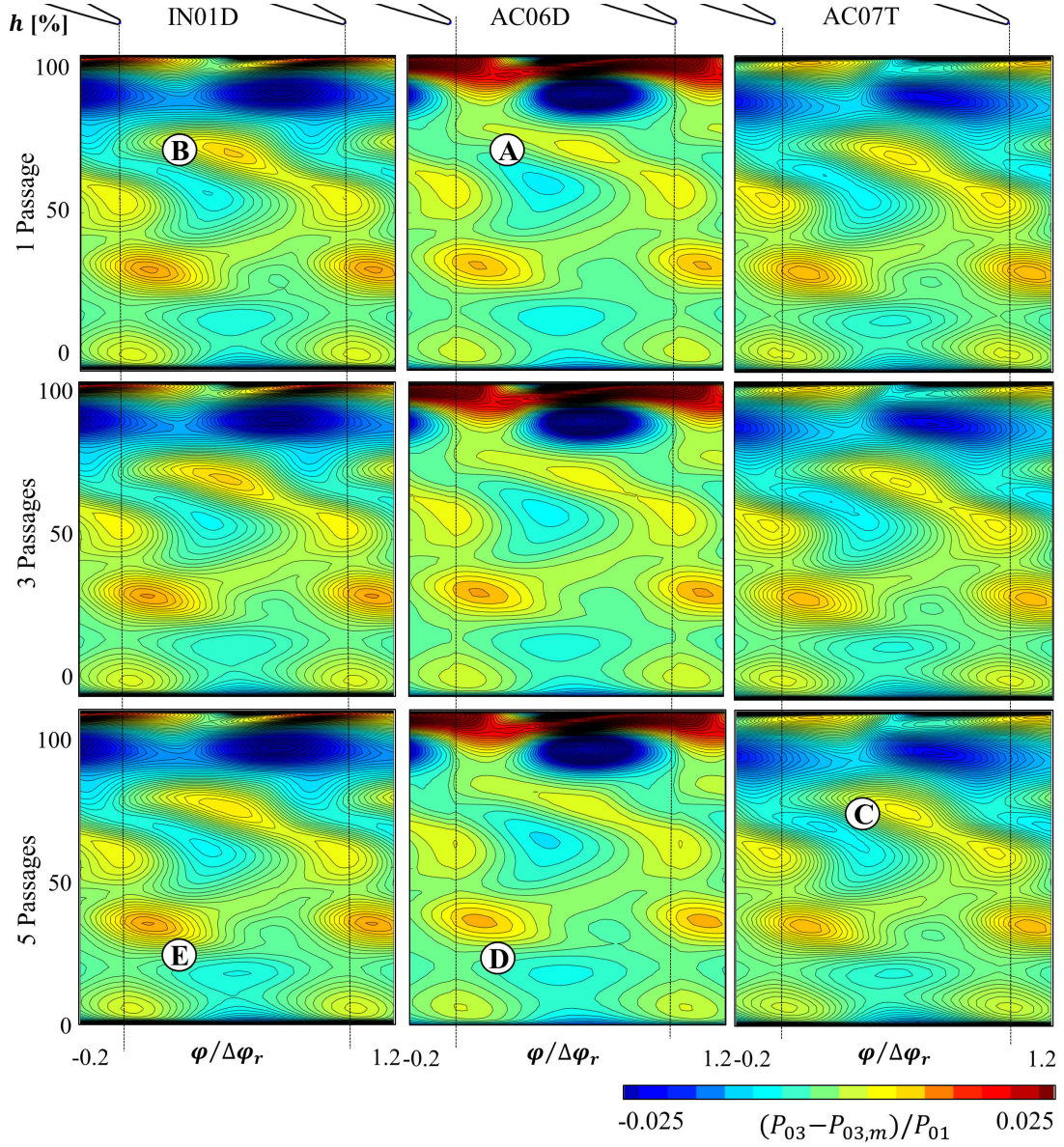


Figure 4.101: Outlet total pressure contours for different averaging domains, full-annulus NLH

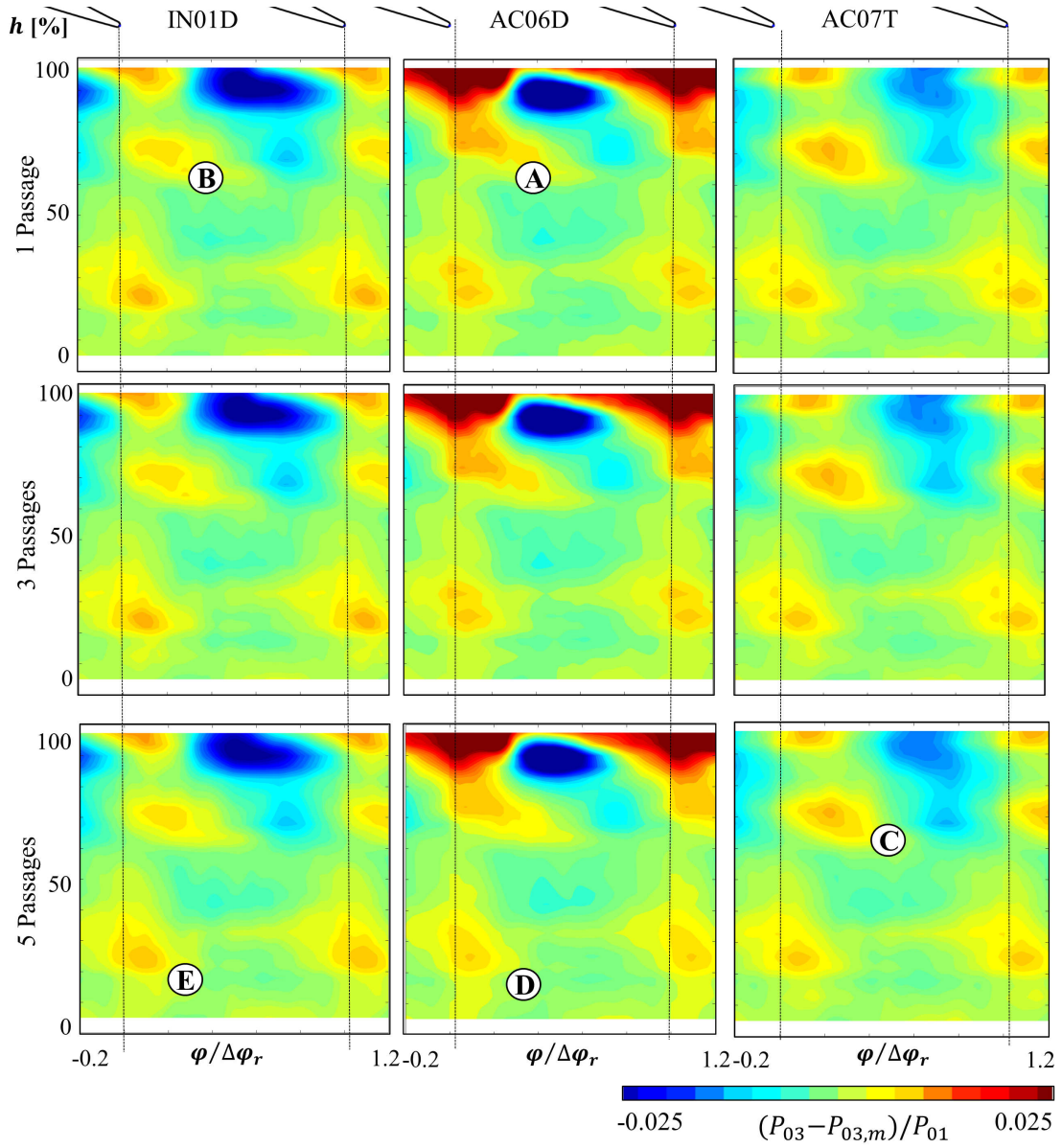


Figure 4.102: Outlet total pressure contours for different averaging domains, experiments

Chapter 5

Conclusions

This thesis presented a complete numerical and experimental investigation on the time-resolved flow field of a high-pressure unshrouded turbine adopting three different blade tip profiles. The aerothermal performance of two advanced tip geometries was compared against a conventional squealer tip. The experimental investigation was carried out at engine-representative conditions in the high-speed turbine facility of the von Karman Institute.

The first part of the treaty focused on the methodologies employed to characterized the HPT flow field:

- The experimental setup installed in the short-duration high-speed turbine test rig was described in detail. The test article consists of a full-stage HPT operated in rainbow-rotor configuration to allow the simultaneous testing of multiple blade tip geometries. The multi-sector rotor setup was presented together with the methodology that allowed to determine the correct running tip clearance for the rotor airfoils.
- The test section is heavily instrumented with high-frequency instrumentation to monitor the unsteady aerodynamic and thermal field of the turbine. Fast response heat transfer and pressure measurement arrays are flush-mounted in the over-tip section of the rotor casing, allowing to determine the aerothermal impact of the individual profiles. Additional low- and high-frequency pressure measurements are carried out at the turbine endwalls at selected axial stations (vane/rotor outlet) to help determine the sector-resolved pressure ratio established by the rainbow configuration. A set of slow- and fast-response aerothermal probes are traversed at the 50% $c_{r,ax}$ downstream of the rotor trailing edge to monitor the rainbow- and sector-averaged outlet flow field.

- Advanced data treatment techniques were employed to convert the high-frequency measurements into comprehensive sector-resolved information. Specific data-processing procedures were implemented to identify the blade-periodic and random fluctuations generated by each blade tip sector. The statistical convergence of the phase-lock averaging over the short-duration tests was demonstrated for the time-resolved datasets of casing static pressure and heat transfer.
- Multiple numerical studies, performed with test-calibrated boundary conditions, were interrogated to gain a detailed insight of the stage flow physics. Full-stage RANS computations, with the mixing-plane adopted at the vane-rotor interface, were compared to full-stage unsteady NLH computations to assess the impact of blade rows interactions on the stage flow. Full-stage, full-annulus NLH computations were exploited to model the rainbow-rotor setup and assess the flow periodicity established within each sector.

The aerothermal characterization of the HPT stage mounting the three different rotor tip geometries was then presented:

- The HPT flow conditions were monitored at each test during the experimental campaign. The total pressure and total temperature profiles at the stage inlet showed a high degree of uniformity along the rotor annulus, with circumferential variations within the measurement repeatability ($\sim \pm 1.5 \text{ mbar}$ and $\pm 5 \text{ K}$ respectively). The sector-resolved degree of reaction was evaluated finding a good agreement between experimental and numerical results. Blade AC06D showed a slightly lower degree of reaction (-2%) with respect to the other profiles, in reason of a larger effective tip clearance and a consequent mass-flow increase of $\sim 1\%$. The experimental operative point, established by setting the total stage mass flow to match the correct pressure ratio, was demonstrated in good agreement with the numerical study, where the pressure ratio of each individual profile is set as boundary condition.
- The over-tip aerodynamic field was observed to vary significantly with blade tip geometry. The time-resolved pressure dataset served to validate the steady and unsteady flow predictions in the rotor domain, confirming the impact of the distinct tip geometry features on the over-tip pressure fluctuations. The signature of the complex rim pattern of blade AC06D was clearly captured by the experimental and numerical datasets. The highest pressure fluctuations, as high as 45% of the time-mean level, have been measured at the mid axial

chord of blade AC07T, where the contoured surface accelerates the leakage flow to sonic conditions. A vortex tracking technique was employed to monitor the propagation of vortical structures in the cavities of the squealer-type profiles and in the aft surface of blade AC07T. The TLV generation was substantially delayed by the contoured surface and the tight clearance of AC07T. An inviscid jet-vortex interaction was highlighted at in the aft part of blade AC06D, significantly increasing the strength of the TLV, but highly modulated by the upstream NGV.

- The rotor casing unsteady heat flux was analyzed in detail for the three tips. All blade geometries showed a large increase in over-tip heat transfer between 60% and 80% of the rotor axial chord. A critical comparison with the aerodynamic dataset demonstrated that this was due to the establishment of the direct by-pass mechanism in the tip leakage over the squealer-type profiles and to the establishment of the large over-tip flow separation in the aft part of blade AC07T. Heat transfer peak-to-peak variations about 3.5 times the time-mean value were measured as the foot-print of the blade PS for all the three blades along a large fraction of the rotor axial chord. The TLV trajectory was shown to mark the boundary between high-heat flux regions, near the blade SS, and low-heat flux regions, where the hot leakage flow is driven radially inwards by the vortical motion.
- The analysis of the casing adiabatic wall temperature revealed the presence of significant work exchange mechanisms in the over-tip region, which locally raised the total temperature of the gap flow above the stage inlet total temperature. The proposed gas temperature model, based on work-exchange mechanisms in the mid-gap plane, correlated well with the temperature evolution in the front part of the blades but did not capture the local increase in the aft part of the tips, mainly generated by 3D flow structures.

A reduced order model was developed to evaluate the heat transfer mechanism at the over-tip casing. The impingement heat flux component allowed to account for local 3D flow features driven by the tip design, with a maximum contribution of 35% of the total shroud heat flux over the LE rim of blade IN01D. The model showed a good qualitative agreement with the mean Nusselt distribution over the squealer tips, while it was not able to capture the over-tip flow separation of blade AC07T.

The fluctuating components of h and T_{aw} contributed to up to 25% of the global

time-average shroud thermal load, with peak values between 75% and 85% $c_{r,ax}$ for the three tips. The maximum contribution was detected over AC07T, in correspondence of the over-tip flow separation in the aft part of the blade.

- The analysis of the unsteady outlet flow field was pursued by combining measurements of total pressure, static pressure, flow angles, Mach number and total temperature. The comparison with RANS and NLH results allowed to validate the impact of the tip design on the outlet flow field and the measurements were demonstrated to successfully capture the tip-to-tip differences in terms of time resolved total pressure and flow angle contours. The outlet relative total pressure was computed by employing the time-average total temperature distribution measured by thermocouple rakes, demonstrating the error committed by neglecting the impact of the unsteady temperature fluctuations. The unsteady measurements located the tracked TLV and UPV in regions of low relative total pressure. Where the residual vorticity of the UPV was below the tracking threshold (AC06D and AC07T), the vortex footprint in terms of relative pressure deficit was still clearly visible.
- The stochastic component recorded in the casing static pressure and heat transfer measurements marked the TLV trajectory in the squealer-type profiles. The predicted vortex intensity correlated well with the peaks in stochastic fluctuations of pressure (up to 5% P_{01}) and heat flux (50% $\overline{Q}_{IN01D}$). In the case of blade AC07T, the over-tip flow separation in the aft part of the blade, connected to the TLV structure, showed the highest stochastic contribution in both pressure and heat transfer measurements. The tracked position of the UPV and LPV corresponded to increased levels of uncorrelated outlet pressure fluctuations.
- A modal analysis was performed over the time- and space-resolved flow at the rotor casing and at the stage outlet. The predominance of rotor-only modes was highlighted in the pressure fluctuations in the over-tip region. The carved profile, operated at a tighter clearance, showed an energy peak in the region of maximum acceleration of the tip flow (60% $c_{r,ax}$), exceeding by 200% the peaks registered over the squealer-like profiles 10% $c_{r,ax}$ downstream. The absolute energy content was shown to be sensitive to the spatial discretization of the target flow field, while the relative modes contributions were not substantially affected by the spatial resolution change. The modal decomposition performed on CFD and experimental data with the same spatial resolution generated congruent results. In the time-resolved outlet entropy field, the highest

energy content was registered in the near-tip area. The high-entropy fluid in the near-casing region of AC07T strongly modulated by vane modes, while the rear-rim aperture of blade AC06D promoted a rotor-periodic entropy pattern. The tip geometry did not affect the energy spectrum in the lower half of the blade span. In agreement with previous studies, the rotor wake was shown to be mainly rotor periodic, while the near-hub flow was significantly influenced by the vane-relative position.

- The entropy loss generated inside the tip gap was estimated as about 20% of the overall tip loss, nearly independent to the tip configuration. The main process affecting the tip efficiency was demonstrated to be the mixing of the leakage stream with the main passage flow. While the rate of entropy generation by tip flows is maximized between 50% and 150% $c_{r,ax}$, the largest tip-to-tip variations were detected downstream of the rotor trailing edge, making the classification of the tip aerodynamic performance sensitive to the axial location considered. The tip performance ranking in terms of rotor efficiency gain confirmed the results expected from the original numerical optimization. Blades IN01D and AC06D outperform the flat tip operated at equivalent clearance, improving the rotor efficiency by 0.15% and by 0.3% respectively. The contoured tip, instead, reduces the rotor efficiency by 0.50% in comparison to a flat tip operated at tight clearance. Blade AC07T shows however the best thermal performance among the three tips, managing to reduce the tip heat flux by nearly 20% compared to the baseline squealer.
- The impact of the rainbow rotor setup on the sector flow periodicity was analyzed experimentally and allowed to validate the simultaneous multi-tip testing strategy. The casing and outlet pressure field were generally very repeatable (below 0.3% P_{01}) in the rotor passage and up to 80% of the blade span. The maximum variation, up to 1.5 – 2% P_{01} , was found in the TLV region of tip AC06D, well correlated with random unsteadiness peaks. Similarly, a high level of periodicity was detected in the outlet flow angle contours (below 2 – 3°) up to 80% of the rotor span, against local maxima of 14° in close proximity to the TLV of blade AC06D. The increased flow variability in correspondence of the vortex core could be motivated by the presence of higher stochastic flow residuals.

The sector flow analysis performed on the full-annulus NLH simulations validated the choice to perform the ensemble-averaging over 5 central channels for

the casing measurements (four for AC07T) and over 4 channels for the outlet measurements (three for AC07T). The pressure minima shown the highest variability at the flow investigation regions: a maximum sector-wise variation of $\sim 1\%P_{01}$ was detected at 50% $c_{r,ax}$ over the SS of tip AC06D. Full annulus NLH predictions confirmed this variation as a direct effect of the rainbow-rotor setup. At the outlet, the maximum passage-to-passage variation within the selected channels was in the near-casing region of AC06D, equal to $\sim 0.8\%P_{01}$.

In conclusion, it is the author's opinion that this research provides a solid basis for future challenges in this field:

- Instrumented airfoils could be employed in order to gain a more detailed insight in the unsteady rotor flow field. Fast-response pressure sensors and thin-film arrays would allow to quantify the impact of blade rows interactions on the pressure and heat transfer field in the rotating frame of reference.
- The modal analysis performed in this thesis has demonstrated that a finer measurement matrix should be adopted at the over-tip rotor casing to fully resolve the impact of blade-rows interactions on the unsteady rotor pressure field.
- Optical techniques could be implemented in the annular cascade to characterize the complex turbine flow by means of full-field measurements. Quartz windows, flush-mounted at the over-tip rotor casing, would allow optical access for high-speed infrared thermography for the characterization of the tip surface temperature. An endoscope system might be employed to introduce a laser plane inside the test section and perform Particle Image Velocimetry to characterize the seeded turbine flow. The main challenge, in this case, would be the synchronization of the image acquisition system with the blade passing event and the minimization of spurious light reflections inside the test section.
- Multi-hole fast response pressure probes would represent a critical improvement in terms of accuracy and time expense with respect to virtual-three hole probes. The miniaturization of the probe heads and careful design of the cavity-line systems connecting the sensors to the flow would be critical in order to achieve high-bandwidth measurements in conditions of low flow intrusivity.

- From the numerical standpoint, full-annulus NLH computations should be performed by considering the presence of the downstream sonic throat (hence, imposing the stage mass flow) to fully mimic the behavior of the experimental turbine stage operated with rainbow rotor. This would provide further insight into the role of the short-duration test rig in setting the HPT test conditions in terms of pressure ratio and mass-flow established in each individual rotor sector.
- A further aerothermal analysis should be performed for the three tip designs at off-design flow conditions to assess the robustness of the tips performance for variable flow regimes. The carved profile (AC07T), in particular, could be tested at higher levels of aerodynamic load in order to fully exploit the desensitization of the leakage mass-flow at transonic flow conditions.
- The rotor efficiency increase achieved by AC06D demonstrates the potential of complex rimmed tip seals in the reduction of losses linked to the tip leakage phenomenon. The aerodynamic performance is, however, particularly sensitive to the method and location where the evaluation is performed. A future program should investigate how the flow mixing and the presence of downstream components impact the evaluation of the single turbine stage performance as well as the efficiency of the overall machine.

Appendix A

Measurement Setup

The measurement setup adopted for the experimental test campaign is detailed in the figure below, together with the estimated running tip clearance along the rotor annulus commented in Figure 2.14.

Plane 01:

- Reference probes (T_{01} , p_{01}): DAO103-A, DAO103-B
- Pneumatic total pressure rake (P_{01}): DAO109
- Thermocouple rakes (T_{01}): DAO106, TAT33, TAT35
- Hot-wire rakes (Tu_{01}): HW-Rake A, HW-Rake B
- Endwalls static pressure ($P_{s1,tip}$, $P_{s1,hub}$): pneumatic taps

Plane 02:

- Endwalls static pressure ($P_{s2,hub}$): pneumatic taps, Kulite sensors

Rotor casing:

- Static pressure array (P_{cas}): 33 x pneumatic taps, 33 x Kulite sensors
- Heat transfer array (Q_{cas}): thin film sensors
- Tip clearance inserts: capacitive sensors, wear gauges

Plane 03:

- Total pressure rakes (P_{03}): FRAPP rakes (A/B), pneumatic Kiel rake
- Directional probes (α_{03} , γ_{03}): pneumatic 5HP rakes

- Virtual three hole probe (α_{03}): V3H MS-A/B/C/E/F, V3H Kul-A
- Endwalls static pressure ($P_{s3,tip}$, $P_{s3,hub}$): pneumatic taps, Kulite sensors

Table A.1 lists the main aerothermal instrumentation employed in the measurement campaign. The measurement matrix is defined as total number of points over the blade span or along the rotor axial chord (Rad./Ax.), and over the full vane pitch (Circ.). Additionally, the table reports the frequency response of each measurement system (FR), the total error with 95% confidence level (Err 95% = systematic + random) and, where relevant, the precision error (Prec.).

Quant.	Inst.	Rad./Ax.	Circ.	FR	Err. 95%	Prec.
p_{01}	Ref. probe - pneumatic	1	2	50 Hz	$\pm 1mbar$	-
T_{01}	Ref. probe - thermocouple	1	2	20 Hz	$\pm 1K$	-
p_{01}	Pneumatic rake	60	2	50 Hz	$\pm 1mbar$	-
T_{01}	Thermocouple rake	80	5	20 Hz	$\pm 1K$	-
Tu_{01}	Hot wire rake	80	2	20 kHz	$\pm 3\%$	-
p_{s1}	Endwalls pneum. taps	1	5	50 Hz	$\pm 1mbar$	-
p_{s2}	Endwalls pneum. taps	1	9-12	50 Hz	$\pm 1mbar$	-
p_{s2}	Endwalls fast resp. sens.	1	6	100 kHz	$\pm 5mbar$	$\pm 0.1mbar$
$p_{s,cas}$	Casing fast resp. sens.	11	9	100 kHz	$\pm 5mbar$	$\pm 0.1mbar$
$p_{s,cas}$	Casing pneumatic taps	11	9	50 Hz	$\pm 1mbar$	-
Q_{cas}	Casing thin films	11	9	100 kHz	$\pm 6\%$	$\pm 2\%$
p_{03}	Pneumatic rake	40	9	50 Hz	$\pm 1mbar$	-
T_{03}	Thermocouple rake	40	9	20 Hz	$\pm 1K$	-
p_{03}	Fast response p. rake	40	9	80 kHz	$\pm 5mbar$	$\pm 0.1mbar$
p_{03}	Ref. probe - pneumatic	1	2	50 Hz	$\pm 1mbar$	-
T_{03}	Ref. probe - thermocouple	1	2	20 Hz	$\pm 1K$	-

Table A.1: Aerothermal instrumentation summary

One further consideration can be made about the miniaturized pressure probes employed in virtual three-hole-probe mode. The use of calibration surfaces for the computation of static pressure, total pressure, Mach number and yaw angle does imply the necessity of a Monte Carlo Analysis to assess the final measurement error on each of these flow quantities. It is the author's opinion that such uncertainty quantification would provide further soundness to the present results and the study should be developed in future work.

Appendix B

List of Publications

B.1 Published papers

- Cernat, Bogdan C., Paty, M., De Maesschalck, C., and Lavagnoli, S., 2019. Experimental and numerical investigation of optimized blade tip shapes Part I: Turbine rainbow rotor testing and numerical methods. *Journal of Turbomachinery*, 141(1)
- Paty, M., Cernat, Bogdan C., De Maesschalck, C., and Lavagnoli, S., 2019. Experimental and numerical investigation of optimized blade tip shapes Part II: Tip flow analysis and loss mechanisms. *Journal of Turbomachinery*, 141(1)
- Cernat, Bogdan C., and Lavagnoli, S. , 2019 "Characterization of the Unsteady Aerodynamics of Optimized Turbine Blade Tips through Modal Decomposition Analysis." *International Journal of Turbomachinery, Propulsion and Power* 4.2: 12.
- Cernat, Bogdan C., and Lavagnoli, S., 2020 "Experimental investigation of tip design effects on the unsteady aerodynamics and heat transfer of a high speed turbine", *Proceedings of ASME Turbo Expo: Turbomachinery Technical Conference and Exposition*, GT2020-15159 , Accepted for publication in the *Journal of Turbomachinery*.

B.2 Conference proceedings

- Cernat, Bogdan C., Paty,M. and Lavagnoli, S., 2018, "On the Unsteady Aerodynamics of a High-Speed Turbine With Squealer Blade Tips." 2018 Joint Propulsion Conference.

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