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**EXPERIMENTAL INVESTIGATION OF SECONDARY FLOW CONTROL
DEVICES IN TURBINE BLADE CASCADES FOR AERO-ENGINE
APPLICATIONS**

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Doctor of Philosophy

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Abstract

The present thesis focuses on the experimental investigation of the performance of flow control devices designed to mitigate the strong secondary flows that arise in modern, compact aero-engine turbine blade cascades. Specifically, the study investigates the effectiveness of suction side shelf-like fences, innovative endwall ridging designs and their combined use, together with squealer and squealer–winglet tip configurations under different coolant mass flow ratios, in mitigating secondary vortex structures. To address this challenge, the analysis is conducted on two representative turbine blade cascades developed for lightweight, high specific thrust aero-engine applications: a nozzle guide vane (NGV) cascade characterized by long chord, low aspect ratio vanes operating in a highly diffusive channel environment, and a highly loaded unshrouded low pressure turbine (LPT) rotor cascade featuring tip clearance. Due to their geometrical characteristics, both cascades exhibit intense vortex structures—a strong passage vortex in the NGV and an extended tip leakage vortex in the unshrouded LPT rotor. These vortical structures generate secondary losses comparable to profile losses, hence providing an excellent framework in which mitigation devices are required and their effectiveness can be evaluated under strongly three-dimensional flow conditions. The performances of fences and ridges in attenuating the passage vortex in the NGV cascade, and of squealer and squealer–winglet tip configurations combined with coolant ejection in reducing the tip leakage vortex in the LPT cascade, are assessed through dedicated experimental campaigns in the large-scale, low-speed wind tunnels of the Aerodynamics and Turbomachinery Laboratory at the University of Genova. Flow measurements in tangential planes were performed both upstream and downstream of the baseline (without secondary flow devices applied on) and controlled configurations of both cascades. Total pressure losses were evaluated using a Kiel probe, while exit velocity and flow angle distributions were captured with a five-hole probe. This experimental approach provides a comprehensive overview of how the devices act in reducing secondary flows and, consequently, influence the cascade efficiency with respect to the baseline cases. Different operating conditions were also investigated, including Reynolds number variations for both cascades, as well as tip clearance and coolant mass flow ratio variations for the unshrouded LPT rotor, enabling a detailed analysis of leakage flow behavior. The experimental results show that, compared to the baseline configurations, fences limit the penetration of the passage vortex in the endwall normal direction, while ridges effectively suppress the cross-passage motion induced by the vortex. An optimized ridging design yields further reductions in secondary losses, and consistent benefits are also observed under combined fence–ridge application. These passive control strategies enhance flow uniformity at the turbine cascade exit, generating a favorable carry-over effect and a consequent improvement in stage efficiency. For tip vortex mitigation, squealer cavity reduces losses nearly proportionally to tip gap size with respect to a flat tip configuration, while squealer–winglet configu-

ration provides additional reductions compared to both flat and squealer tips. Coolant ejection at low mass flow rates produces negligible effects on losses and downstream flow patterns, whereas higher injection rates confine the leakage vortex closer to the endwall but increase the associated losses. These findings are subsequently incorporated into a refined tip-loss model to generalize the observed trends. The experimental database also supports validation activities for both low- and high-fidelity simulations performed by Morfo Design. The simulations successfully reproduce the main features of losses and flow deviations, with LES providing deeper physical insight. In particular, the analyses clarify the mechanisms through which fences and ridges mitigate passage vortex effects inside the vane channel, and those by which squealer and squealer–winglet tips weaken the tip leakage vortex and modulate the influence of coolant ejection. Overall, the results demonstrate the capability of these flow control devices to effectively suppress secondary flow structures and improve turbine efficiency, highlighting their potential for integration in next-generation compact aero-engines.

Preface

This thesis presents an experimental investigation of different secondary flow control and mitigation devices aimed at counteracting the strong passage vortex that develops at the hub of a low aspect ratio Nozzle Guide Vane inserted in a diffusive channel and the tip leakage vortex that forms at the tip of an unshrouded LPT row of modern aero-engines. Chapter 1 provides an introduction and outlines the general framework of the PhD project, including an overview of the techniques reported in the literature for secondary loss reduction and those experimentally assessed in this work. Chapter 2 describes the flow phenomenology within a turbine blade cascade, with particular attention to secondary flows and the associated loss generation mechanisms. The influence of geometrical and flow parameters is discussed, and the main secondary loss correlations available in the literature are reported. Chapter 3 presents a comprehensive literature review of active and passive systems for secondary flow control and mitigation. Passive devices such as blade fences and endwall ridges to weaken the passage vortex, as well as squealer and squealer–winglet tip geometries to suppress the tip leakage vortex, are discussed. Active flow control approaches, including coolant injection from the blade tip to enhance the aero-thermal performance, are also reviewed. Chapter 4 describes the experimental campaigns carried out in the wind tunnels of the Aerodynamics and Turbomachinery Laboratory of the University of Genova. Details of the experimental setup, measurement techniques and data analysis procedures are provided, along with a thorough description of the investigated cascades and the geometrical features of the applied devices. Finally, Chapter 5 presents the results of the work. The capability of fences and ridges to mitigate the strong passage vortex in the low aspect ratio, diffusive NGV cascade is discussed, together with the effectiveness of the squealer and squealer–winglet tip configurations, also in combination with coolant injection from the tip, in contrasting the tip leakage vortex forming in the tip region of the unshrouded LPT cascade investigated. Additional insight is provided by dedicated RANS and LES simulations carried out by Morfo Design on both cascades. Chapter 6 concludes the thesis with a brief summary of the investigated control devices and their effectiveness in mitigating secondary flows in turbine cascades. The appendix provides a detailed analysis of the error propagation for the assessment of the main variable uncertainties, alongside a brief overview of the numerical setup adopted for the RANS and the LES simulations conducted on both cascades.

Contents

Abstract	I
1 Introduction	1
2 Turbine Secondary Flows and Losses	6
2.1 Secondary Vortices	8
2.1.1 Passage Vortex	9
2.1.2 Horseshoe Vortex	10
2.1.3 Minor Secondary Vortices	15
2.1.4 Tip Vortex	17
2.2 Effect of Geometrical and Flow Parameters on Secondary Losses	19
2.2.1 Blade Loading Effect	19
2.2.2 Aspect Ratio Effect	22
2.2.3 Tip Clearance Effect	23
2.2.4 Reynolds Number Effect	24
2.2.5 Other Geometrical and Flow Parameter Effects on Secondary Losses	26
2.3 Secondary Loss Correlations	28
3 Secondary Flow Control Devices	32
3.1 Active Systems	32
3.1.1 Flow Momentum Injection	33
3.1.2 Boundary Layer Suction	35
3.1.3 Streamwise Vorticity Generation	36
3.1.4 Excitation of Natural Instabilities	39
3.1.5 Other Active Techniques	39
3.2 Passive Devices	41
3.2.1 Structural Modifications	42
3.2.2 Endwall Contouring	42
3.2.3 Endwall Fences	45
3.2.4 Endwall Splitter	47
3.2.5 Endwall Ridging	48
3.2.6 Blade Fillet	51

3.2.7	Blade LE-bulb	53
3.2.8	Blade Leaning and Re-Staggering	54
3.2.9	Blade Snaking	55
3.2.10	Blade Grooves	56
3.2.11	Blade Fences	56
3.2.12	Squealer Tip	59
3.2.13	Winglet Tip	64
3.2.14	Squealer-Winglet Tip	66
3.2.15	Other Tip Solutions	69
3.2.16	Summary of Secondary Flow Control Devices	71
4	Experimental Campaigns: Setup and Cascade Configurations	75
4.1	Test Rigs	76
4.1.1	Genova Laboratory Test Section	79
4.1.1.1	NGV Configurations	82
4.1.1.2	Fences	83
4.1.1.3	Conventional Endwall Ridging	85
4.1.1.4	Fences and Conventional Endwall Ridging	85
4.1.1.5	Optimized Endwall Ridging	86
4.1.1.6	NGV Test Matrix	90
4.1.2	Savona Laboratory Test Section	90
4.1.2.1	Unshrouded LPT Configurations	94
4.1.2.2	Squealer Tip	94
4.1.2.3	Squealer-Winglet Tip	94
4.1.2.4	Unshrouded LPT Test Matrix	96
4.2	Measurement Techniques and Acquisition Chains	98
4.2.1	Hot-Wire Anemometer with Constant Temperature Anemometer Module	98
4.2.2	Static Pressure Taps with Digital Sensor Array	100
4.2.3	Kiel and five-hole pressure probes with differential transducers . .	102
4.3	Data Analysis	106
4.3.1	Flow Turbulence Characterization	106
4.3.2	Blade Loading Distributions	107
4.3.3	Aerodynamic Loss Evaluation	108
4.3.4	Velocity Field Reconstruction	110
5	Results	112
5.1	Aggressive NGV Results	113
5.1.1	Detailed Experimental Characterization of the Inlet Boundary Con- ditions	113
5.1.2	Baseline NGV	117

CONTENTS

5.1.3	NGV with secondary flow control devices applied	121
5.1.4	Further analysis with RANS and LES	129
5.2	Unshrouded LPT Results	138
5.2.1	Characterization of the Inlet Boundary Conditions	138
5.2.2	Flat Tip	139
5.2.2.1	No Gap Characterization	139
5.2.2.2	Gap Variation Characterization	142
5.2.3	LPT with secondary flow control devices	145
5.2.3.1	Squealer Tip	145
5.2.3.2	Squealer-Winglet Tip	151
5.2.3.3	Comparison of Different Tip Leakage Flow Control De- vices	157
5.2.4	Tip Loss Correlation	165
5.2.5	Further analysis with RANS and LES	168
6	Conclusions	182
	Appendix 1	185
	Appendix 2	189

Nomenclature

(s, n, z) Streamwise, Normal to Streamwise, Spanwise Coordinate System $[(m, m, m)]$

(x, y, z) Axial, Pitchwise, Spanwise Coordinate System $[(m, m, m)]$

α Flow Yaw Angle $[\circ]$

α_s Flow Yaw Angle Between Probe Head Axis and Cascade Reference system $[\circ]$

β Pitch Angle, Blade Angle $[\circ]$

δ Boundary Layer Thickness $[m]$

δ^* Boundary Layer Displacement Thickness $[m]$

$\dot{m}_0$ Inlet Mass Flow Rate $[kg/s]$

$\dot{m}_c$ Coolant Mass Flow Rate $[kg/s]$

ε Turbulence Dissipation Rate $[m^2/s^3]$

η Efficiency $[-]$

γ Stagger Angle $[\circ]$

κ Meanline Radius Ratio $[-]$

λ Ainley and Mathieson Parameter $[-]$

ν Kinematic Viscosity $[m^2/s]$

ω Rate of Dissipation per Unit Turbulent Kinetic Energy $[1/s]$

$\omega_{\square}$ Loss Coefficient $[-]$

ρ Density $[kg/m^3]$

σ Solidity $[-]$

NOMENCLATURE

τ	Tip Gap Size [m]
θ	Flow Turning [$^\circ$]
ϕ	Flow Coefficient [-]
ζ	Enthalpy loss coefficient [-]
A	Area [m^2]
A, B, C, D, E	Coefficients of $Q(z)$ [-]
a_0, a_1, a_2	Loss Correlation Coefficients[-]
A_n	Annulus Area [m^2]
AR	Aspect Ratio [-]
AVR	Axial Velocity Ratio [-]
B	Generic Coefficient [-]
B, n	Hot-Wire Empirical Calibration Coefficients [$V^2/(m/s)$], [-]
C	Blade Chord [m]
C_α	Exit Flow Yaw Angle Coefficient [-]
c_D	Discharge Coefficient [-]
C_f	Skin Friction Coefficient [-]
C_L	Lift Coefficient [-]
c_p	Pressure Coefficient [-]
$c_{cooling}$	Cooling Coefficient in Loss Correlation [-]
C_{pt}	Total Pressure Loss Coefficient [-]
$CP_{32}, CP_{54}, CP_t, CP_{din}$	Pressure Coefficients for Five-Hole Probe Calibration [-]
CR	Convergence Ratio [-]
d	Turbulence Grid Bar Diameter [m]
D_1, D_2	Measuring Planes

NOMENCLATURE

E	Wire Voltage [V]
$E(\omega)$	Loss Model Error [-]
E_0	Supply Voltage at Zero Flow Velocity [V]
F_p	Pressure Force [N]
$FT(\cdot)$	Fourier Transform
h	Blade Height [m], Enthalpy [J]
$I.D., O.D$	Inlet-Outlet Turbine Diameters [m]
K	Turbulence Grid Geometric Parameter [-]
k	Turbulent Kinetic Energy [m^2/s^2]
K_E, K_G	Constants Related to Blade Loading Distributions [-]
$k_{winglet}$	Winglet Coefficient in Loss Correlation [-]
L	Duct Length [m]
L_0	Integral Length Scale [m]
Ma	Mach Number [-]
p	Static Pressure [Pa]
P_1, P_2, P_3, P_4	Points for Turbulence Measurements (indicated by *)
p_t	Total Pressure [Pa]
PS	Power Spectrum
$Q(z)$	Boundary Condition Equation
$R(t)$	Auto-Correlation Function
R_A, R_B	Curvature Radius of Particles A and B [m]
Re	Reynolds Number [-]
s	Blade Pitch [m]
t	Time [s]

NOMENCLATURE

T_0	Integral Time Scale [s]
Tu	Freestream Turbulence Intensity [-]
v	Flow velocity [m/s]
X	Generic Signal Acquired
x_t	Streamwise Coordinate Downstream of the Turbulence Grid [m]
Y	Losses in Literature [-]
Z	Ainley and Mathieson Simil Zweifel Number [-]
Z_{TE}	Passage Vortex Width at Trailing Edge [-]
Z_w	Zweifel Number [-]
$Z_{w_{eff}}$	Effective Zweifel Number at the Tip [-]
*	Position of P_1, P_2, P_3, P_4 for Tu Measurements
2D	Two-Dimensional
3D	Three-Dimensional
$\cdot^*$	Complex Conjugate
$\cdot^{(MA)}$	Mass-Averaged
$\cdot_1$	Inlet
$\cdot_2$	Outlet
$\cdot_{ad}$	Made non-dimensional
$\cdot_c$	Coolant
$\cdot_{inv}$	Inviscid Flow
$\cdot_{is}$	Isoentropic
$\cdot_{leak}$	Leakage
$\cdot_{max}$	Maximum Value of Cases: Baseline, $Re_x = 400k$ (NGV); Flat, $Re_x = 200k$, $\tau/C_x = 1\%$ (LPT)
$\cdot_{mid}$	Midspan reference

NOMENCLATURE

$\cdot_{ng}$	No-gap reference
$\cdot_n$	Normal to Streamwise
$\cdot_{pl}$	Plenum
$\cdot_{PS}$	Pressure Side
$\cdot_p$	Profile
$\cdot_{ref}$	Maximum Pitchwise and Spanwise Averaged Value of Cases: Baseline, $Re_x = 400k$ (NGV); Flat, $Re_x = 200k$, $\tau/C_x = 0\%$ (LPT)
$\cdot_{rms}$	Root Mean Square
$\cdot_{SS}$	Suction Side
$\cdot_s$	Secondary, Streamwise
$\cdot_{tip}$	Tip
$\cdot_t$	Total
$\cdot_x$	Axial
$\cdot_y$	Tangential
$\cdot_z$	Spanwise
$\bar{\cdot}$	Pitchwise Average, Averaged Value
$\bar{\cdot}_{max}$	Maximum Pitchwise Averaged Value of Cases: Baseline, $Re_x = 400k$ (NGV); Flat, $Re_x = 200k$, $\tau/C_x = 0\%$ (LPT)
$\bar{\bar{\cdot}}$	Pitchwise and Spanwise Average
A, B	Particle Names
ANN	Artificial Neural Network
BC	Boundary Conditions
BL	Boundary Layer
CAD	Computer Aided Design
CFD	Computational Fluid Dynamics

NOMENCLATURE

CTA	Constant Temperature Anemometer module
DBD	Dielectric Barrier Discharge
DSA	Digital Sensor Array
EWB	Endwall Ridging
HPC	High Performance Computing
HPT	High Pressure Turbine
HSV	Horseshoe Vortex
HWA	Hot-Wire Anemometer
ITD	Intermediate Turbine Diffuser
LE	Leading Edge
LES	Large Eddy Simulation
LPT	Low Pressure Turbine
NGV	Nozzle Guide Vane
PS	Pressure Side
PV	Passage Vortex
RANS	Reynolds-Averaged Navier-Stokes equations
SKE	Secondary Kinetic Energy
SS	Suction Side
TE	Trailing Edge
TET	Trailing Edge Thickness
TLV	Tip Leakage Vortex
TRL	Technology Readiness Level
VG	Vortex Generator
VGJ	Vortex Generator Jet

Chapter 1

Introduction

The present thesis focuses on the experimental investigation of secondary flow control and mitigation devices aimed at reducing the intense secondary vortices that develop in compact, high specific thrust aero-engine configurations. To achieve overall weight reduction in these new-generation engines, the mid pressure turbine stage is often omitted, with the nozzle guide vane (NGV) integrated in the intermediate turbine diffuser (ITD), thereby directly connecting the high pressure turbine (HPT) to the low pressure turbine (LPT). Moreover, while low pressure turbine blades in conventional architectures are typically shrouded to minimize leakage losses, in these configurations the LPT is instead equipped with fully unshrouded blades to meet the requirements of lightweight and compact aero-engines.

Specifically, the NGV design follows the integrated concept described by Raab *et al.* [1], in which the intermediate turbine diffuser is merged with the structural struts of the S-shaped duct. These vanes are realized as long chord, limited turning blades that simultaneously fulfill structural and aerodynamic functions, featuring a low aspect ratio (AR) that enables the guidance of the flow from the smaller HPT radius to the larger LPT radius while imparting the required swirl to the downstream stage. To achieve the necessary pressure recovery, the NGVs are placed within short and aggressive diffuser geometries—well beyond the classical limits identified by Sovran and Klomp [2]—resulting in highly diffusive channels, particularly near the tip region. Under these conditions, the aggressive diffuser not only promotes flow separation but also intensifies the development of pronounced secondary flows and particularly strong passage vortices (PVs). These vortical structures are responsible for secondary losses comparable in magnitude to profile losses, as highlighted by Wegener *et al.* [3], and for significant yaw angle distortions at the cascade exit. Such distortions can further deteriorate the aerodynamic performance of the downstream cascade [4]. At the same time, the tip clearance of unshrouded turbine blades—necessary to prevent rubbing between the rotating blade tips and the stationary casing—induces leakage flows across the gap. The pressure difference between the blade pressure side (PS) and suction side (SS) drives this flow, which, upon emerging near

the suction side, interacts with the main passage flow and generates a strong tip leakage vortex (TLV). This secondary flow not only alters the local thermal field at the blade tip—reducing blade lifetime and increasing the risk of mechanical failure in this critical region—but also significantly contributes to overall aerodynamic losses. Although particularly severe in low aspect ratio HPT blades, tip leakage effects remain notable even in high aspect ratio LPT configurations, as shown by Lakshminarayana [5], who estimated that tip leakage losses can account for up to 40% of total blade losses.

Therefore, to prevent the formation of secondary flows, various flow control strategies can be employed depending on the operating conditions and design constraints. Active flow control systems suppress the generation of secondary vortices primarily a priori by exploiting engine energy sources [6], whereas structural modifications and passive devices, applied to the blade or endwall surfaces, aim to alleviate or mitigate secondary vortex development without any external energy input [7].

Since active devices are sensitive to inlet flow unsteadiness and channel curvature, and may not always lead to an improvement in duct efficiency [8], passive approaches are more widely adopted. These typically involve either structural modifications of the channel hosting the vanes or blades, or the application of dedicated flow control devices. Over the years, a variety of optimized meridional shapes, especially developed for intermediate turbine diffusers, have been proposed and experimentally tested, as documented by Gräsel *et al.* [9], with the aim of reducing boundary layer (BL) separation and mitigating secondary flow structures. Another widely recognized structural modification strategy for weakening the passage vortex in intermediate turbine diffusers is the endwall contouring introduced by Deich *et al.* [10]. This approach involves shaping the casing to produce non-axisymmetric endwalls in both streamwise and pitchwise directions, with the aim of reducing secondary flows by lowering secondary kinetic energy and minimizing exit angle deviations. However, due to their geometric complexity, these solutions often underperform under off-design conditions and may interfere with endwall cooling, posing challenges in design, manufacturing, and maintenance. For this reason, the use of dedicated devices provides an alternative, offering increased design flexibility without requiring major modifications to the meridional channel or endwalls, and is therefore among the most promising solutions to maintain low losses and uniform flow at the NGV exit [11]. Regarding endwalls, Prümper [12] proposed the concept of endwall fences—small barriers placed on the endwall within the blade passage to block cross-flow and inhibit the migration of the passage vortex. Derived from endwall fences, endwall mounted splitters consist of high aspect ratio vanes positioned in the middle of the inter-blade channel, aimed at preventing secondary vortex propagation from the pressure side to the suction side of an adjacent blade [13]. However, widespread implementation remains limited by cooling requirements. A more advanced passive approach to secondary flow control, derived from the multi-splitter concept of Russo *et al.* [14], is the endwall ridging (EWR), which consists of grooved structures aligned with the streamwise flow, inspired by the functional

principle of riblets. These microstructures are designed to manipulate near wall boundary layer flow, reducing skin friction by altering turbulent vortices and thereby minimizing energy dissipation [15]. Originally applied in external aerodynamic applications, such as aircraft nacelles and wings, and motivated by the economic benefits of riblets discussed by Lynch and Klinge [16], Siegel *et al.* [17] investigated the feasibility of larger scale engineered grooved surfaces (ridged surfaces) to mitigate secondary vortices in internal aerodynamic applications. Thus, further studies have explored ridges in various turbomachinery contexts: Lietmeyer *et al.* [18] examined their application on compressor blades, while Khader and Sayma [19] analyzed their impact on radial turbines. In particular, Miao *et al.* [20] investigated endwall ridges in axial turbines, demonstrating their ability to weaken secondary flow structures and reduce passage vortex lift-up. Nonetheless, these studies emphasized the need for further design optimization to fully exploit the potential of endwall ridging. Regarding blade surface devices, fences are passive, shelf-like structures designed to limit secondary vortex penetration along the blade height. Initially studied by Kawai [21] and later refined by Rubeschini *et al.* [22] and Giovannini *et al.* [23], fences enhance flow uniformity at the NGV exit, improving downstream rotor interaction and overall stage efficiency. Despite their simplicity, fences are recognized as easy to manufacture and install, making them a practical solution for mitigating secondary flow penetration.

To reduce losses associated with tip leakage flow, various passive and active control systems have been adopted over the past decades. Vortex generators mounted on the casing wall [24], as well as their active counterparts [25], can be used to influence tip flow, but the most widely adopted solutions involve tip modifications such as squealers and winglets as well as coolant injection from the tip. Bindon and Morphis [26] initially investigated alternative tip shapes, replacing the standard flat tip with a radiused pressure edge to channel the radial leakage flow, reducing the gap separation bubble and improving overall blade performance. Subsequently, Nho *et al.* [27] tested eleven different tip configurations and found that full squealer tips—characterized by raised rims along the blade tip—most effectively reduce clearance mass flow and can improve cascade efficiency. Moreover, Mischo *et al.* [28] numerically demonstrated that recessed cavity tips also reduce the thermal load at the blade tip. More recently, Jeong *et al.* [29] analyzed several plain winglet tip designs, characterized by tangential extensions of the blade edge, observing that winglets are able to mitigate leakage vortex strength by lowering the pressure gradient between the blade PS and SS. Finally, Jiang *et al.* [30] assessed novel tip configurations combining squealer and winglet concepts. Regarding active flow control, coolant injection into the tip clearance is one of the most widely adopted techniques. The cooling flow can be introduced either from the casing, as reported by Mischo *et al.* [31], or more commonly directly from the blade tip surface. In the latter case, the coolant effectively shields the blade tip from high thermal loads by limiting heat transfer between the mainstream flow and the blade [32]. Additionally, Chen *et al.* [33] demonstrated that

the injected coolant can generate a blockage effect in flat tip gaps, thereby weakening the tip leakage vortex. Other studies analyzed the impact of the number [34] and the position of the injection coolant holes [35] on mitigating the tip leakage vortex. Finally, Sakaoglu and Kahveci [36] numerically evaluated twelve different cooling configurations for flat and squealer tips, highlighting that cooled squealer tips with four-seven holes positioned along the mean camber line (aligned with the separation line) achieved the best combination of thermal protection and aerodynamic performance.

The present thesis aims to analyze the mechanisms through which secondary flow control strategies can mitigate passage and tip leakage vortices. To address this challenge, two large scale turbine blade cascades designed for lightweight and high specific thrust aero-engine applications are experimentally investigated. These include a representative nozzle guide vane cascade, characterized by long chord, low aspect ratio vanes integrated within the diffusive intermediate turbine duct of a modern aero-engine, and a highly loaded unshrouded low pressure turbine rotor cascade operating with tip clearance. The low aspect ratio and divergent meridional channel of the NGV promote the formation of a strong passage vortex, while the relatively mild aerodynamic loading at the blade leading edge induces a weaker horseshoe vortex. Conversely, the unshrouded LPT rotor blades generate a pronounced tip leakage vortex accompanied by substantial associated losses. These distinct secondary flow phenomena motivate the exploration of multiple innovative control strategies:

- To mitigate the passage vortex developing within the NGV channel, a two shelf-like fence arrangement on the suction side of the vane and endwall integrated ridged surfaces, inspired by trapezoidal grooves aligned with the local flow direction, are implemented. Specifically, the NGV cascade is first tested in its baseline configuration without any control devices. Subsequently, tests are conducted on the vane equipped with suction side fences, followed by the vane with conventional endwall ridges. The combined effect of fences and conventional ridges is then evaluated, and finally, an optimized ridged layout specifically tailored to the present high diffusion NGV cascade is tested.
- To mitigate the tip vortex developing on the unshrouded highly loaded LPT rotor blade, various squealer and squealer–winglet tip configurations are tested. The flat tip blade is first examined as a reference case. This is then modified to incorporate an optimized cavity squealer with equal rims on both pressure and suction sides, featuring four coolant holes positioned along the mean camber line beneath the separation line to assess the effects of different coolant mass flow ratios on the leakage flow. Subsequently, the tip is redesigned to host a squealer–winglet configuration, maintaining the same number, size, and location of coolant holes as the cavity squealer.

The effectiveness of these devices in mitigating the passage vortex in the NGV cas-

cade and the tip vortex in the LPT cascade is evaluated through extensive experimental campaigns conducted in the large scale test sections of the low speed wind tunnels at the Aerodynamics and Turbomachinery Laboratory of the University of Genova. Flow measurements in radial–tangential planes, both upstream and downstream of the cascades, are performed using a Kiel probe to quantify total pressure losses, while velocity and flow angle distributions at the cascade exit are captured with a five-hole probe. This experimental methodology provides a comprehensive assessment of the ability of these devices to reduce secondary flows and their consequent impact on cascade efficiency under varying operating conditions. Indeed, tests are conducted at two Reynolds numbers, representative of engine cruise and take-off/landing conditions, while for the LPT rotor, the effects of different tip configurations and coolant injection on the tip leakage flow are additionally assessed for various tip gaps. The effects of these parameters on the tip leakage vortex core, in terms of both losses and flow angles, are then used to calibrate a new tip loss correlation capable of generalizing the loss response to parameter variations. Furthermore, the experimental data provide a benchmark for both low-fidelity (RANS) and high-fidelity (LES) simulations performed by Morfo Design. The numerical analyses successfully reproduce the main experimental trends in losses and flow deviations, with LES offering more detailed insights in the underlying flow physics. In particular, the simulations highlight how fences and ridges mitigate the passage vortex within the NGV channel, while squealer and squealer–winglet tips weaken the tip leakage vortex and modulate the effect of coolant injection. Overall, the results demonstrate that these control devices can effectively manage secondary flow structures and enhance turbine efficiency, highlighting their potential for integration in next-generation aero-engine designs.

Chapter 2

Turbine Secondary Flows and Losses

The flow through a turbine stage experiences an entropy increase due to irreversibility, leading to an energy loss compared to the ideal behavior. The mechanisms responsible for loss generation in turbines have been extensively investigated by Denton [37], who identified viscous effects in boundary layers, mixing in free shear layers, heat transfer between regions at different temperatures (such as between mainstream and coolant flows), and non-equilibrium processes like shock waves as the primary contributors. Excluding the contribution of thermal exchanges and neglecting shock-related losses, which are generally not significant in medium–low pressure turbines for direct-driven high specific thrust aero-engines, attention can be concentrated on losses arising from viscous and mixing effects. These can be broadly divided into profile losses and secondary losses. Profile losses originate from the development of boundary layers along the pressure and suction sides of the blades, where in the decelerating region between the blade passage throat and trailing edge (TE), the boundary layer may undergo significant thickening or even separation. In addition, the mixing of the boundary layers from both sides with each other and with the surrounding mainstream at the trailing edge produces a quasi-two-dimensional wake, perceived as a velocity and therefore a total pressure deficit, ultimately altering the inlet conditions and thereby the performance of the downstream cascade. Figure 2.1 shows the schematic view of profile losses proposed by Nsonga *et al.* [38], illustrating how the boundary layers developing on the rotor blade surfaces merge to form a wake that impinges on the pressure side of the downstream stator vanes as a negative jet structure, as described by Hodson and Dawes [39].

Secondary losses, by contrast, include endwall losses [40], which originate from the boundary layers developing over the hub and tip walls. Their interaction with the mainstream deflected by the blades generates complex vortical structures that extract kinetic energy from the flow and reduce its total pressure. The secondary vortex system is shown in Figure 2.2, following the model proposed by Benner and Sjölander [41]. Due to their inherently three-dimensional nature, these vortices significantly distort the exit flow conditions, deteriorating the inlet conditions of the downstream cascade and ultimately re-

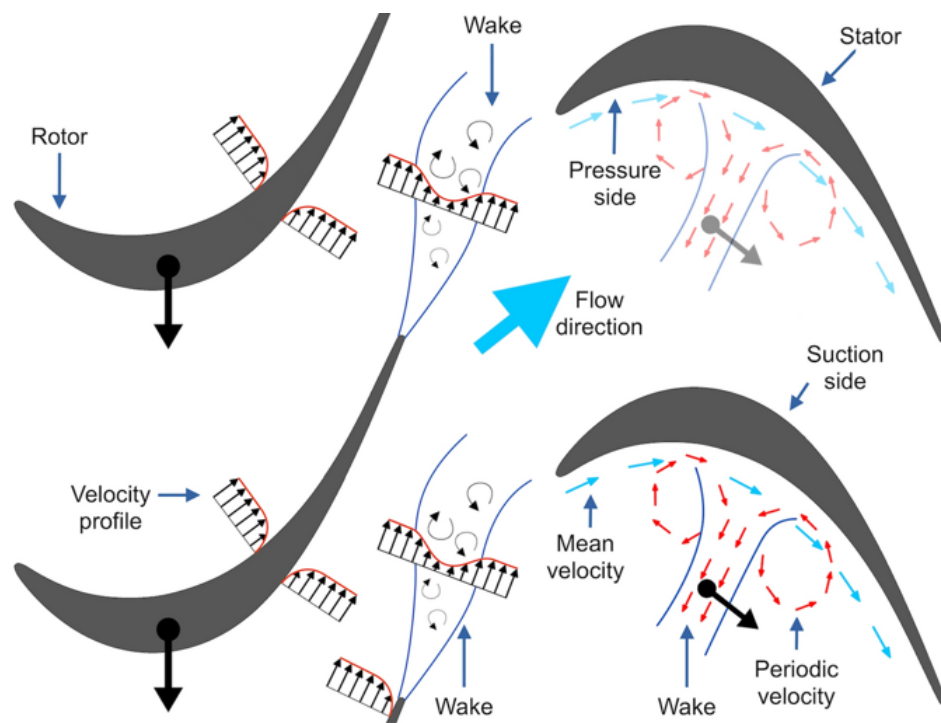


Figure 2.1: Schematic representation of the mechanism of profile loss generation proposed by Nsonga *et al.* [38].

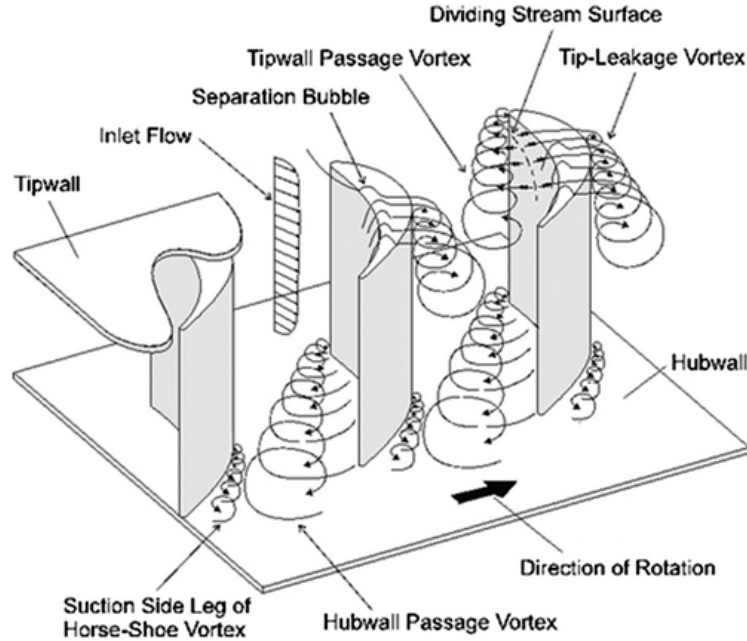


Figure 2.2: Representation of secondary loss mechanism generation in an axial turbine row proposed by Benner and Sjölander [41].

ducing the overall turbomachinery efficiency.

It follows that turbine losses ultimately depend on the contributions generated within individual turbine blade cascades, which are themselves strongly influenced by blade geometry, channel design, and flow characteristics.

Secondary losses are particularly relevant in the cascades investigated in this thesis, where the combination of medium-to-low aspect ratio, a diffusive meridional channel, and the use of unshrouded blades can lead to secondary losses exceeding those of profile origin. Therefore, the aerodynamic performance analysis presented in this thesis necessarily includes a detailed investigation of secondary flow structures.

2.1 Secondary Vortices

The scientific literature provides numerous theoretical and experimental contributions on the analysis of the three-dimensional vortical structures that originate near the endwalls within the blade channel. Among the most influential are the works of Langston [42] and Sieverding [43], who extensively investigated the secondary vortex systems arising in turbine cascades and proposed models to describe them. The formation of secondary flows is generally attributed to a momentum deficit, which in turbine cascades can be mainly traced to the boundary layer developing along the endwall surfaces, and subsequently

interacting with the main flow deflected by the blades.

By observing Figure 2.2 again, the main vortical structures developing within a turbine cascade can be identified as the passage vortex, the horseshoe vortex, and the tip leakage vortex. Additional minor secondary vortices, such as corner and trailing edge vortices, can also be recognized in the literature.

2.1.1 Passage Vortex

The passage vortex is the first vortical structure to be investigated, dating back to the pioneering works of Hansen *et al.* [44] and Smith Jr [45] in the early 1950s, and later analyzed through the analytical theories and experimental studies of Horlock and Lakshminarayana [46], as it represents the largest and most intense vortex forming within a blade cascade. Subsequently, Sieverding [43] provided a more detailed description, showing that this vortex originates from the interaction between the boundary layers developing along the hub and casing walls and the main flow deflected by the blades. As fluid particles constituting the non-uniform radial velocity profile enter a blade passage, they are subjected to a tangential deviation caused by the pressure gradient between the pressure side of one blade and the suction side of the adjacent blade. This deviation, which depends on the particle velocity and thus on its radial location, gives rise to the passage vortex.

To gain a deeper understanding of the mechanism responsible for the formation of the passage vortex, it is useful to consider the model proposed by Lakshminarayana [5], illustrated on the right side of Figure 2.3, and to examine the trajectories of two fluid particles entering the blade passage: particle A, located at the edge of the boundary layer with a velocity equal to the freestream, and particle B, located within the boundary layer. In the blade-to-blade plane (top left of Figure 2.3), particle A is subjected to a resultant force F_p due to the pressure difference between the pressure side of one blade and the suction side of the adjacent blade. According to Newton's third law, an apparent centrifugal force acts in opposition, directly proportional to the square of the velocity V_A and inversely proportional to the curvature radius R_A , allowing particle A to follow the undisturbed flow through the passage along the red trajectory. Assuming static pressure is conserved within the boundary layer, F_p remains constant across the passage height. In contrast, for particle B (bottom left of Figure 2.3), the centrifugal force decreases due to the reduction in velocity V_B caused by viscous effects. For the streamline to remain in equilibrium, the centrifugal force and pressure gradient force must balance; therefore, the decrease in velocity leads to a smaller curvature radius R_B , forcing the streamline to deviate toward the blade suction side (green trajectory). Consequently, the streamline of particle B is more deflected than that of particle A, generating a flow crossing from the pressure side toward the suction side near the endwall (endwall cross-flow). Due to the pressure balance and flow continuity near the endwall, the streamlines cannot impact the suction side directly but are instead forced to recirculate, generating a helical vortex that moves axially and

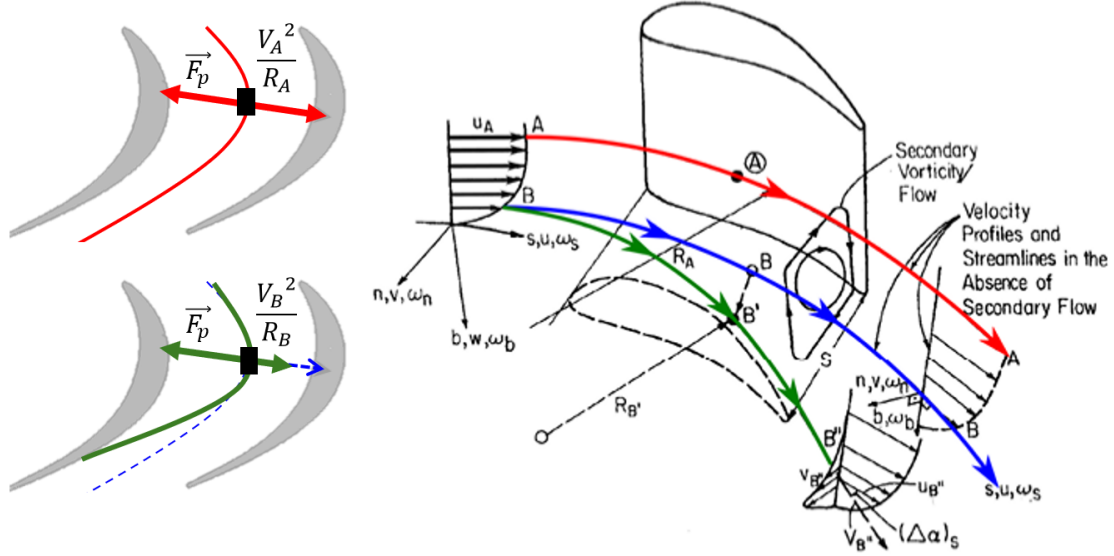


Figure 2.3: Representation of the passage vortex formation mechanism: on the left, trajectories of the particles A and B in the blade-to-blade plane; on the right, Lakshminarayana's model [5].

is deflected from pressure to suction side by the pressure gradient, as shown in Figure 2.4 (see also the smoke flow visualization of the passage vortex along the rear part of the blade suction surface, shown later in Figure 2.7.c.3-4). In each blade passage, two passage vortices form: one near the lower endwall and a corresponding one near the upper endwall, with opposite rotation directions as visible in Figure 2.2. Moreover, although the passage vortex decays in strength through mixing, up to one-third of the total losses can still occur downstream of the trailing edge as a result of the continuous dissipation of the secondary flow kinetic energy, as observed by Moore and Adhye [47].

It is also important to note that, although multiple turns are often illustrated to depict the vortical structure of the passage vortex, Langston [42] and Sieverding and Van Den Bosche [48] observed that, within the blade passage, the PV completes only one or at most two full revolutions around its centerline before convecting downstream, in agreement with the Rankine vortex model (as can be seen later in Figure 2.7.b).

2.1.2 Horseshoe Vortex

Another significant vortical structure forming in a turbine blade row is the horseshoe vortex. It is smaller than the passage vortex and can interact with it. Its intensity and size depend on the blade leading edge radius [5], the thickness of the incoming boundary layer, and the flow incidence angle [43]. The formation of this vortex is strongly influenced by

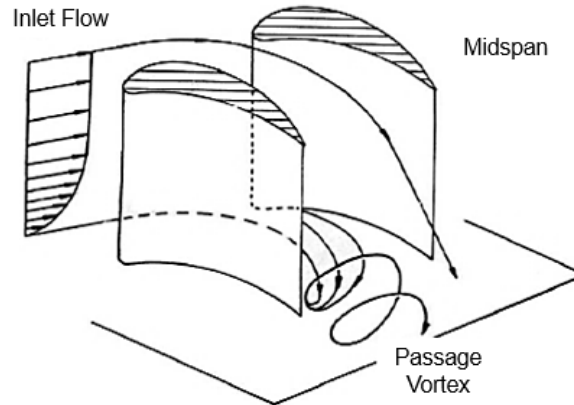


Figure 2.4: Schematic representation of the passage vortex developing inside a turbine blade channel.

the boundary layers developing on the hub and casing surfaces, which impact the blade leading edge, as can be observed in Figure 2.5.

The formation mechanism is particularly affected by the pressure gradient in the incoming flow caused by the boundary layer, where the total pressure p_t decreases from its freestream maximum to the wall minimum due to the no-slip condition. This pressure gradient drives the fluid from the center of the passage toward the endwall, initiating the vortical structure. Upon encountering the blade leading edge, the horseshoe vortex splits into two counter-rotating legs: one along the pressure side and one along the suction side of the blade, forming the characteristic horseshoe shape. Within this structure, a saddle point can be identified as the stagnation point between the counter-rotating legs, representing the origin of the recirculation, while the endwall separation line marks where the flow detaches from the endwall surface, feeding the vortex. A visualization of these key flow features is presented later in Figure 2.7.a [49]. Then these HSV legs interact with the passage vortex. The pressure side leg rotates in the same sense as the passage vortex, amplifying it; the two structures, once merged, tend to shift toward the suction side of the adjacent blade due to the pressure gradient imposed by the blade. This is why, in the literature, the term ‘passage vortex’ often already refers to this merged structure. The suction side leg, in contrast, rotates opposite to the passage vortex and is therefore often referred to as the “counter vortex.” This leg continues along the suction side, remaining close to the endwall, until it reaches the region where the passage vortex interacts with the blade suction surface. Here, the suction side leg of the horseshoe vortex meets the passage vortex. The first experiments of Langston [42] suggested that the suction side leg of the horseshoe vortex remains confined in the suction surface endwall corner and, due to its sense of rotation opposite to the passage vortex, tends to weaken toward the

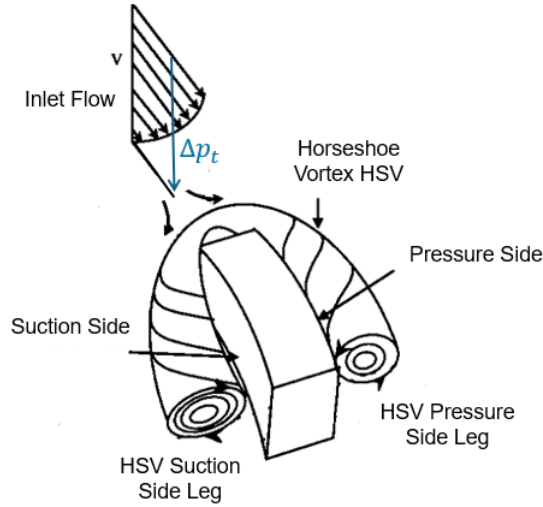


Figure 2.5: Illustration of the horseshoe vortex formation mechanism at the blade leading edge.

passage exit, being dissipated by the passage vortex. This interpretation is reflected in his secondary flow model, shown in the top part of Figure 2.6. Kawai *et al.* [50], on the other hand, observed that while the pressure side leg of the HSV entrains the endwall cross-flow to form and develop the passage vortex, the suction side leg climbs along the blade suction surface, over the passage vortex, before eventually being dissipated by it, confirming the mass transfer contour analyses of Goldstein and Spores [51]. Conversely, using colored smoke streakline visualization, Sieverding and Van Den Bosche [48] (see also Figure 2.7) experimentally observed that the pressure side leg of the horseshoe vortex follows a smooth trajectory through the passage without noticeable rotation, while the suction side leg wraps around it. This different HSV behavior has been attributed to variations in the boundary layer along the endwall, which under certain conditions can cause the horseshoe vortex to be deflected away from the endwall and begin spiraling around the outer edge of the passage vortex, as well represented by the secondary flow model proposed by Sharma and Butler [52], illustrated in the bottom part of Figure 2.6. Although its rotation is often emphasized, it is important to note that the HSV performs at most one or two cycles. This behavior was later confirmed by the smoke visualization experiments reported in Figure 2.7.c conducted by Wang *et al.* [53]. Meanwhile, the passage vortex continues to advance almost undisturbed toward the trailing edge, remaining close to the blade suction surface. If the counter vortex is sufficiently strong, it may still be observed at the passage exit; otherwise, it is completely dissipated through interaction with the passage vortex [5]. It is also worth noting that low momentum fluid from the inlet boundary layer near the endwall does not enter the horseshoe vortex but flows along the blade suction surface above the

passage vortex, remaining there up to the trailing edge (see bottom of Figure 2.6).

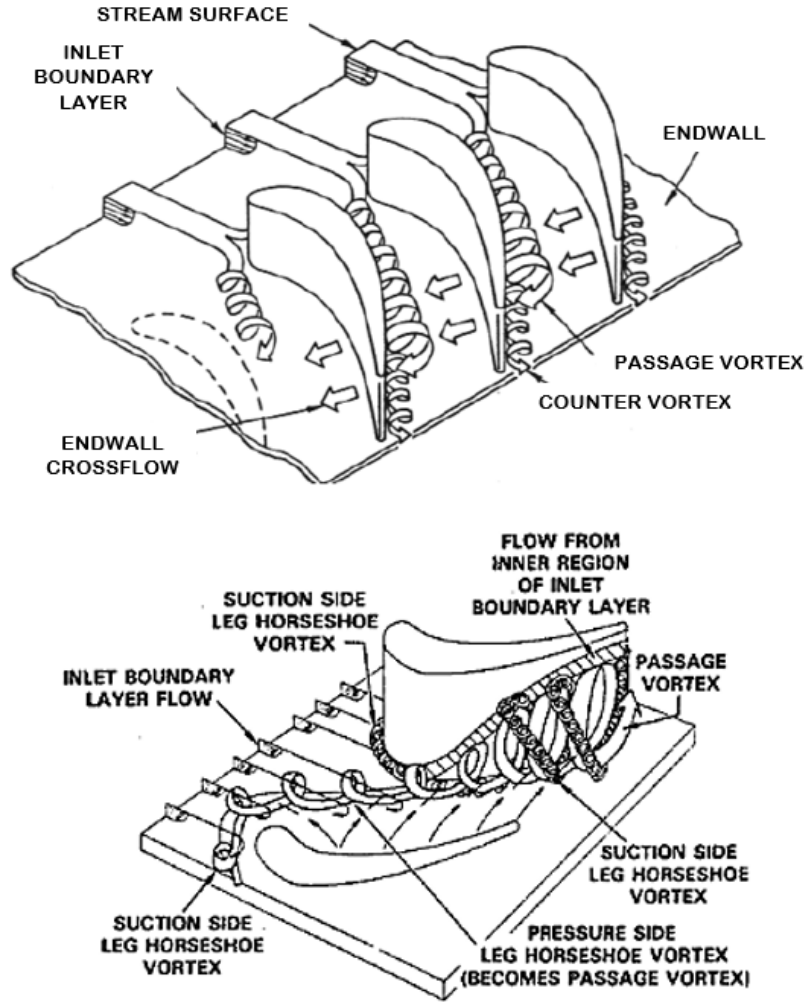


Figure 2.6: Secondary flow models from (top) Langston [42] and (bottom) Sharma and Butler [52], showing the suction side leg of the horseshoe vortex respectively remaining along the blade suction surface or spiraling around the passage vortex.

For completeness, Figure 2.7 gathers the most effective visualizations of smoke, oil, and bubble streaklines that have contributed to understand the behavior and main characteristics of the interaction between the passage and horseshoe vortices, since their existence cannot be considered independent [48]. In Figure 2.7.a, visualizations by Gaugler and Russell [49] are shown, obtained through (a.1) photographs of oil traces and (a.2) bubble streaklines, where the saddle point, endwall separation line, and endwall cross-flow can be clearly identified. Figure 2.7.b shows colored smoke visualizations by Sieverding

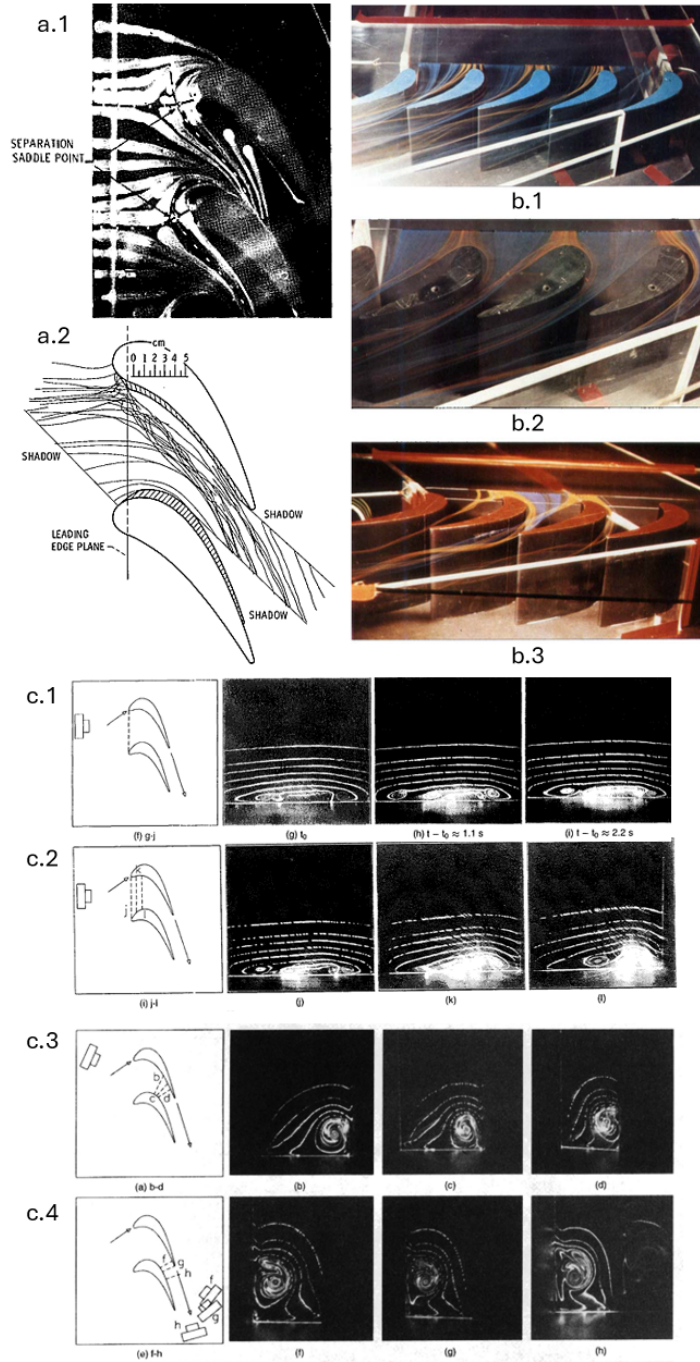


Figure 2.7: Vortex structure visualization in a turbine cascade using (a) oil traces and bubble streaklines [49]; (b) colored smoke [48]; and (c) smoke-wire patterns [53].

and Van Den Bosche [48] of the interaction between the passage vortex and horseshoe vortex, with smoke injected in a nozzle vane outside the boundary layer (b.1), inside the boundary layer (b.2), and inside the rotor blade boundary layer (b.3). Figure 2.7.c presents smoke wire pattern photographs by Wang *et al.* [53], illustrating the formation of both HSV legs near the leading edge at different time instants separated by 1.1 seconds (c.1) and at different axial positions (c.2), as well as the lifting of the passage vortex (PV+HSV) near the trailing edge at various axial positions within the blade passage (c.3) and downstream of the blade (c.4).

2.1.3 Minor Secondary Vortices

In addition to the previously described vortical structures, other secondary vortices are present within a turbine blade cascade. Although smaller in intensity, these vortices can contribute to cascade losses because, being co-rotating, they may sum and produce non-negligible losses. Among these structures are corner vortices and the trailing edge vortex, as represented in the model of Hawthorne [54] in Figure 2.8.

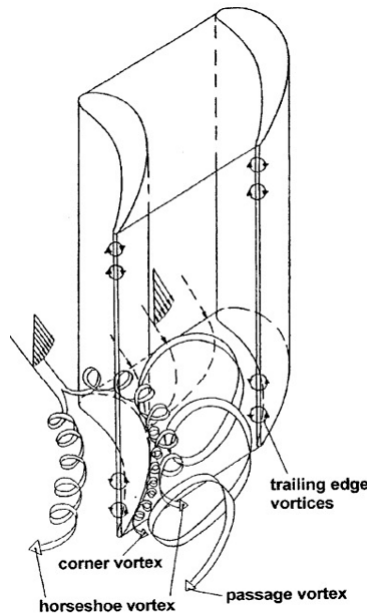


Figure 2.8: Model by Hawthorne [54] illustrating the interaction between the suction side corner vortex and the trailing edge vortex.

Corner vortices developing in a turbine cascade are illustrated in the models by Wang *et al.* [53], reported in Figure 2.9, and Goldstein and Spores [51]. These vortices originate at the junction between the blade and the endwall and can occur at the leading edge, on the suction side, or on the pressure side. They are induced by the presence of a nearby intense

vortex. In fact, under high aerodynamic loading the strong intensity of the horseshoe and passage vortices can generate shear stresses that induce the formation of a counter-rotating secondary vortex. Being local phenomena, corner vortices tend to remain confined to the blade–endwall junction region. Among these, leading edge corner vortices are located near the junction between the blade and the endwall and are induced by the rotation of the horseshoe vortex. Suction side corner vortices originate just downstream of the region where the passage vortex lifts off from the endwall due to its interaction with the blade pressure side. Pressure side corner vortices are generated by the downward velocity component of the flow near the blade–endwall junction, approximately at one-third of the blade chord.

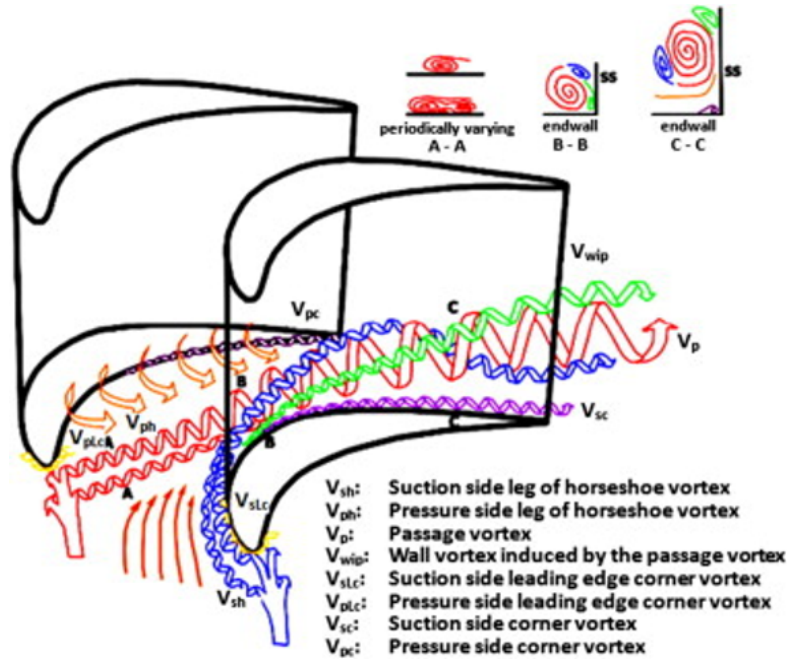


Figure 2.9: Schematic model by Wang *et al.* [53] of the main corner vortices inside a turbine stage.

The trailing edge vortex, instead, is represented in the model by Sauer *et al.* [55] in Figure 2.10 and forms due to the interaction between the wake and the passage vortex, resulting in a counter-rotating structure relative to the passage vortex. These vortices detach periodically from the trailing edge, giving rise to a phenomenon known as vortex shedding, often associated with the formation of coherent flow structures, such as von Karman vortices, which induce pressure oscillations and unsteady forces on the blade profile.

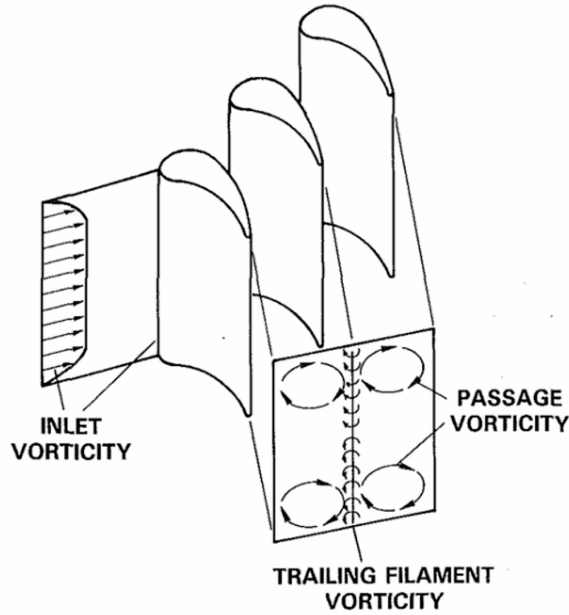


Figure 2.10: Trailing edge vortex model by Sauer *et al.* [55].

2.1.4 Tip Vortex

The clearance gap required to avoid rubbing between stator and rotor components generates the tip leakage vortex, a secondary vortex of major importance that can cause losses even larger than those of the passage vortex [5]. The first flow visualizations of the blade tip region were performed by Allen and Kofskey [56] on a low speed rotating turbine rig. Later, Denton [37] described in detail the leakage flow mechanisms for both shrouded and unshrouded blades. In shrouded configurations, as shown in Figure 2.11, the leakage flow passes through the seal from upstream to downstream, acquiring meridional and swirl velocity components. It then forms a jet that mixes within the gap before discharging into the mainstream, increasing irreversibility and strongly distorting the inlet conditions of the downstream row.

Instead, in unshrouded blades, the pressure gradient imposed by the blade drives the leakage at the blade tip also from the pressure side to the suction side of the blade. Bindon [57] used smoke visualization to observe this phenomenon, as shown in Figure 2.12, also noting that a separation region develops due to the clearance cross-flow. Sjolander and Amrud [58] demonstrated that the reattachment of this bubble depends on blade loading as well as on blade thickness and axial chord. Because of the constriction at the blade tip induced by the separation bubble, downstream of the bubble maximum height, the leakage evolves into a jet. For thin blades, this jet quickly penetrates the passage and mixes with the main flow, causing strong mixing losses. For thick blades, mixing first

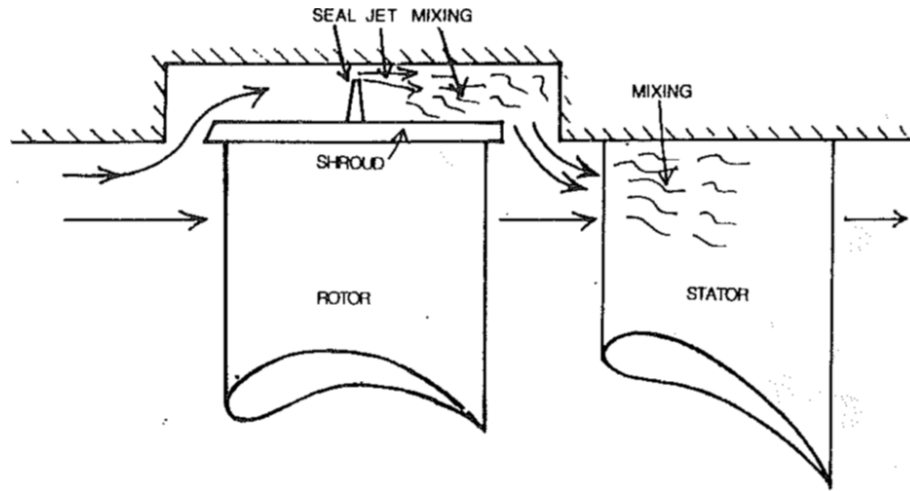


Figure 2.11: Leakage flow model proposed by Denton [37] for shrouded blades.

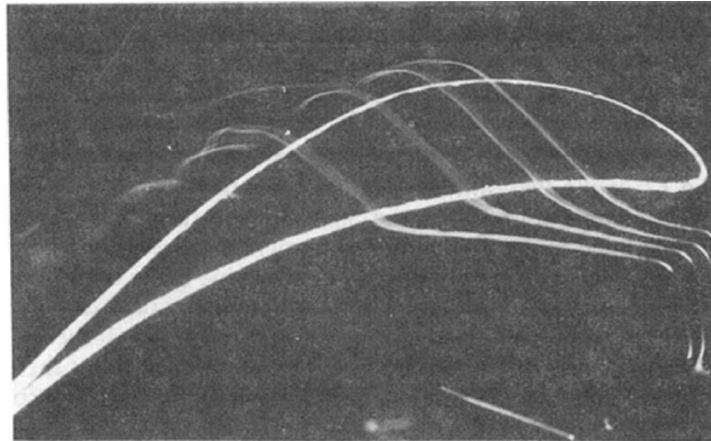


Figure 2.12: Smoke visualization of tip leakage flow by Bindon [57].

occurs inside the gap—just downstream of the maximum height of the bubble—then the jet interacts with the suction side and merges with the main flow, leading to further loss generation. These two leakage flow behaviors are summarized in the schematic representation by Denton [37] in Figure 2.13. Moreover, observing Figure 2.2, it is also evident that the TLV interacts with the passage vortex, as reported by Yamamoto [59]. Being counter-rotating, the TLV pushes the passage vortex toward the casing, leading to high aerodynamic losses concentrated at the blade tip. In case of a strong tip leakage vortex, it can also be responsible for the complete dissipation of the passage vortex.

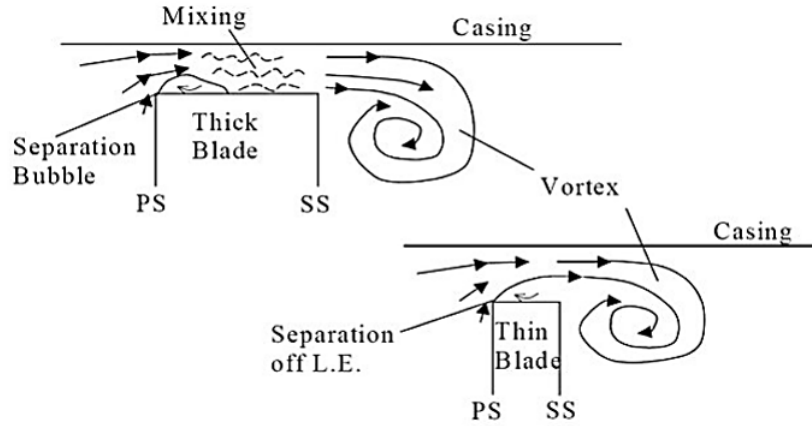


Figure 2.13: Tip leakage vortex for thin and thick unshrouded blades according to the model of Denton [37].

2.2 Effect of Geometrical and Flow Parameters on Secondary Losses

Secondary vortical structures in turbine cascades, including the tip leakage vortex in unshrouded blades, are strongly influenced by both geometric and flow parameters. Key geometric characteristics—such as blade loading, aspect ratio, and tip clearance when present—together with flow conditions, most notably the Reynolds number, govern the onset and development of these vortices, ultimately determining their strength, trajectory, and associated losses.

2.2.1 Blade Loading Effect

To analyze the effect of aerodynamic loading and blade induced flow deflection on secondary flows, Lampart [60] conducted numerical simulations on two different blade geometries:

- a) a relatively low flow turning and low load configuration (stagger angle -13° , inlet 63° , exit -63° , $Ma = 0.2$);
- b) a high flow turning and high load configuration (stagger angle -18° , inlet 75° , exit -72° , $Ma = 0.4$).

Overall, the author observed that flow turning and cascade loading increase enhances both the passage and tip leakage vortices, with the latter delaying or suppressing the passage vortex and determining both the location and magnitude of secondary flow losses. Specifically, as shown in the secondary flow vector maps and total pressure contours upstream and downstream of the trailing edge in Figure 2.14, in the low flow turning, low

load case (Figure 2.14.a), the passage vortex is weak and rapidly disintegrates due to interaction with the tip leakage vortex. In contrast, in the high flow turning, high load case (Figure 2.14.b), cross-flow in the endwall region is more pronounced, and the tip leakage vortex delays or suppresses the passage vortex, pushing it toward midspan and rapidly dissipating any new vortical structures near the pressure surface.

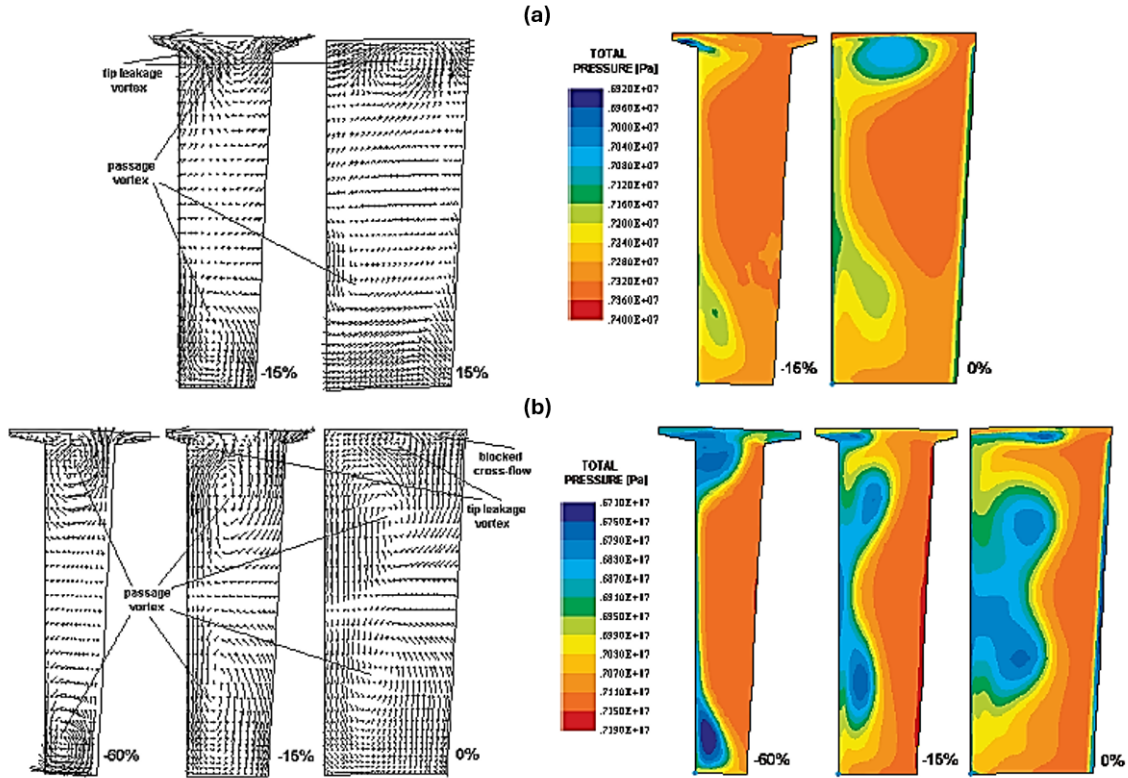


Figure 2.14: Secondary flow vectors and total pressure contours upstream and downstream of the trailing edge, in the (a) low flow turning, low load case and in the (b) high flow turning, high load case [60].

Moreover, to further elucidate the effect of blade loading distribution on secondary flow, several studies have investigated the mechanisms of secondary flow generation in front- and aft-loaded blades sharing the same overall aerodynamic load. Specifically, Weiss and Fottner [61] conducted an experimental study on a front-loaded T104 and an aft-loaded T106 cascade, both with the same Zweifel coefficient and no tip clearance. They observed that in the front-loaded cascade, strong flow deflection immediately downstream of the inlet produces intense transverse pressure gradients, resulting in a strong passage vortex and relatively high secondary losses. In contrast, in the aft-loaded cascade, the peak pressure gradient occurs only at the throat, where it interacts with a very thin, newly formed boundary layer, leading to lower secondary losses compared to the

front-loaded cascade. These findings were subsequently confirmed by Zoric *et al.* [62] and later by Qu *et al.* [63] through comprehensive experimental and numerical studies. In particular, as shown in the vortex iso-surfaces and total pressure loss coefficient maps in Figure 2.15, the latter authors observed that the maximum blade loading near the leading edge of the front-loaded blade induces a rapid increase in the cross-passage pressure gradient, strengthening the pressure side leg of the horseshoe vortex (V_{ph}) and amplifying secondary flow losses. They also noted that this trend persists even in the presence of separation bubbles (e.g., at lower Reynolds numbers): the secondary flow remains weaker for the aft-loaded blade because the larger open separation bubble on the suction surface restricts the flow passage and reduces the cross-passage pressure gradient, also resulting in a smaller outlet angle deviation.

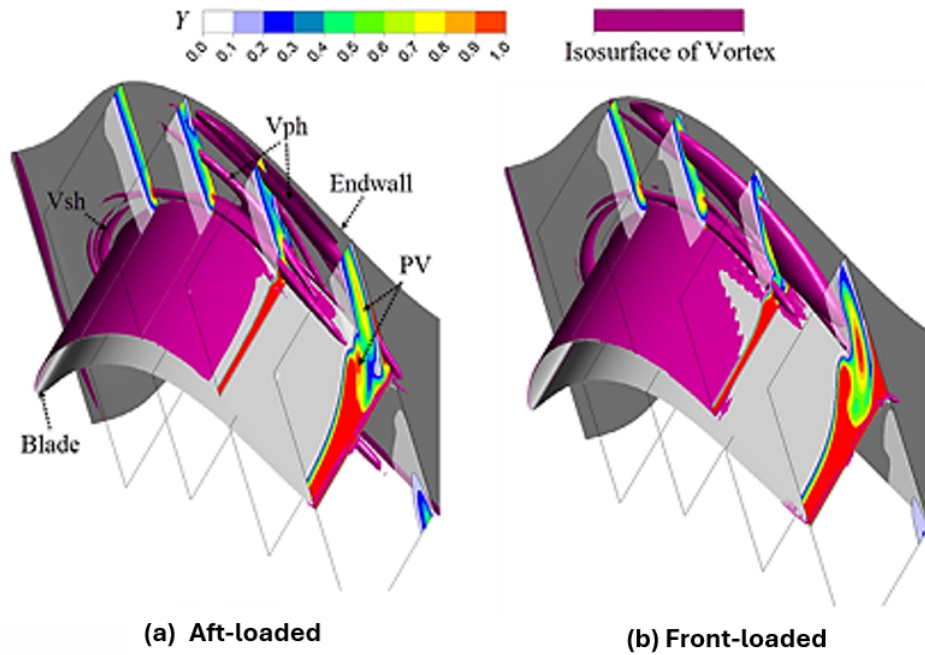


Figure 2.15: Vortex iso-surfaces determined using the Q-criterion and total pressure loss coefficient in the cascade passage of (a) aft- and (b) front-loaded blades [63].

Huang *et al.* [64] also performed numerical simulations to investigate the effect of blade loading distribution on the tip leakage vortex in unshrouded axial turbine blades, considering a forward-loaded and an aft-loaded tip section, both matched in overall work. The authors observed that the effect of blade loading distribution on the tip vortex is opposite to that on endwall secondary flows, as shown in the color plots of velocity and dissipation in Figure 2.16. In the forward-loaded blade, the reduced pressure rise alleviated vortex deceleration, suppressing breakdown and reducing clearance losses by ap-

proximately 15% compared to the aft-loaded blade, despite a 14% higher leakage mass flow. Furthermore, for a given cascade, different solidities were tested, and a lower solidity (hence higher blade loading) resulted in larger losses associated with a stronger tip leakage vortex, as expected.

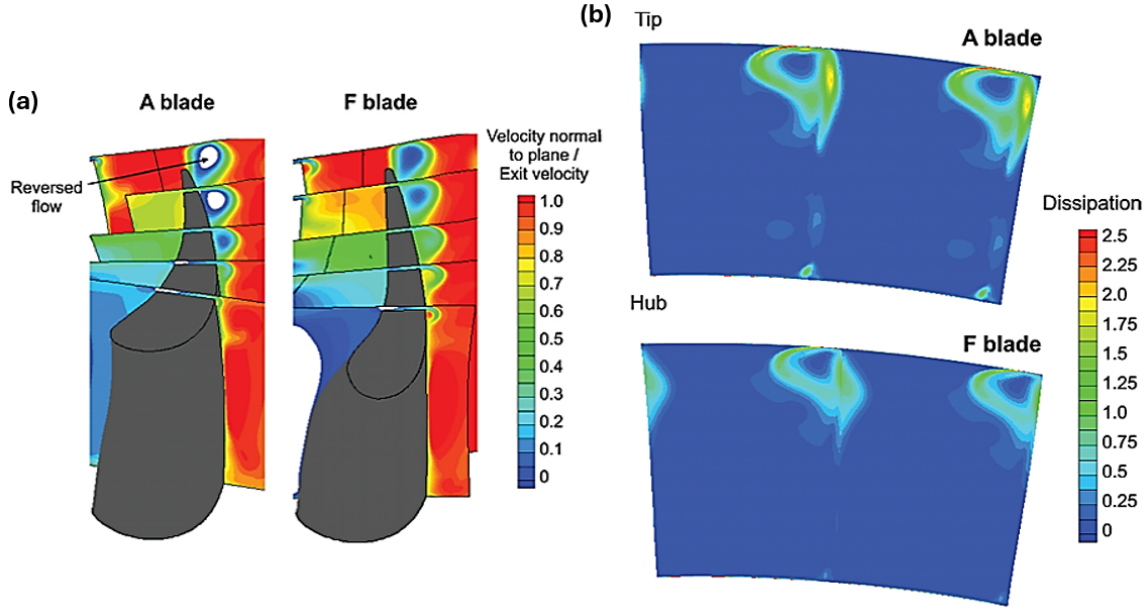


Figure 2.16: Colored contours of (a) plane normal velocity at several cross-flow planes and (b) normalized volumetric viscous dissipation at the location with maximum loss per unit leakage flow, for (F) forward- and (A) aft-loaded blade tip section [64].

2.2.2 Aspect Ratio Effect

The aspect ratio ($AR = h/C$, blade height-to-chord ratio) is a key parameter influencing secondary flows. As shown by Vinuesa *et al.* [65], although total losses remain nearly constant or decrease only slightly with variations in AR , profile losses dominate in high AR cascades, whereas in low AR configurations secondary flow induced losses may account for more than 40% of the total losses. This highlights that secondary losses cannot be neglected, even in low pressure turbines, as also reported by Wegener *et al.* [3] and Satta *et al.* [66]. Such behavior is attributed to the compact geometry of low AR blades, which enhances three-dimensional flow features, promoting the formation and persistence of large secondary vortices along the blade height, significantly increasing energy dissipation and dominating overall losses. In such cascades, it is often difficult to identify regions affected solely by profile losses due to the deep penetration of secondary flows. This is

evident from the experimental and numerical results of Teia [67], shown in the total pressure iso-contours in Figure 2.17, which indicate that secondary vortices remain largely unaffected until a critical AR of approximately 0.35, below which vortex merging begins to modify secondary losses. Govardhan *et al.* [68] further observed that the endwall loss cores do not physically move away as AR decreases; instead, they expand to occupy more of the blade passage.

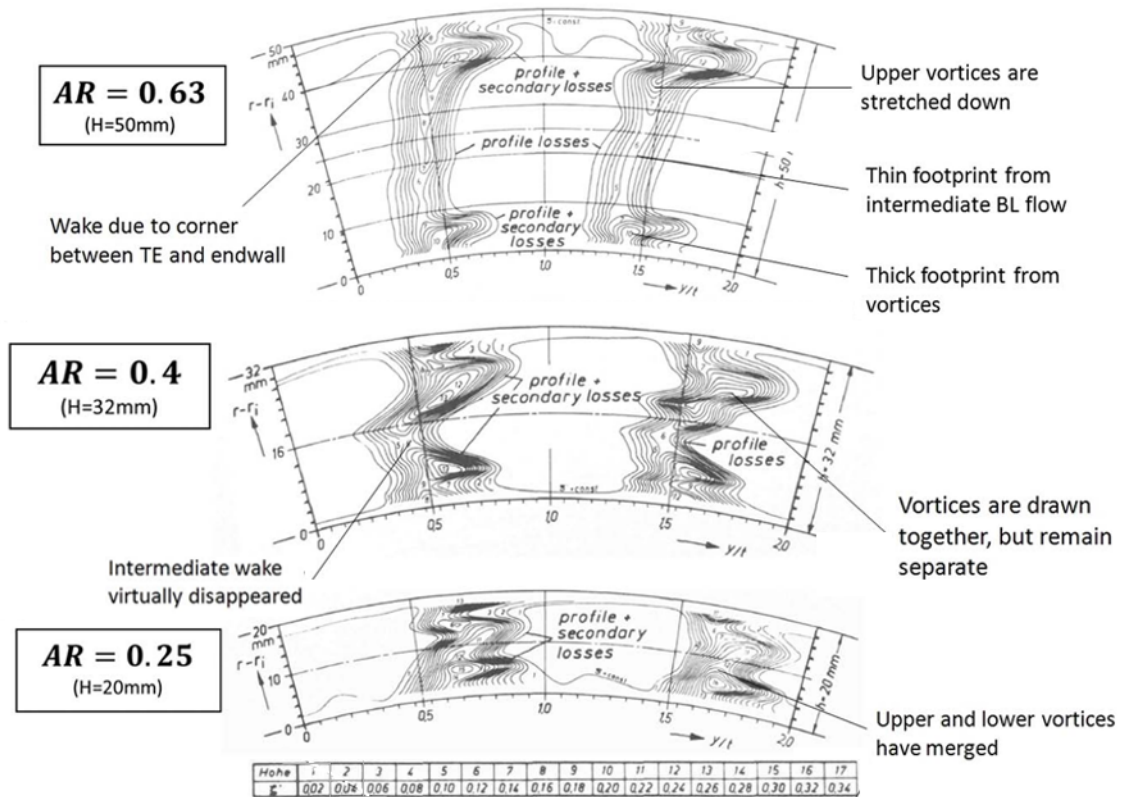


Figure 2.17: Total pressure iso-contours obtained from experiments on blade cascades with $AR = 0.63, 0.4$, and 0.25 [67].

2.2.3 Tip Clearance Effect

Yamamoto [69] analyzed the influence of tip gap height variation on aerodynamic losses, showing that the gap size controls the extent of the separation zone and directly impacts turbine performance [70]. As the tip clearance increases, a larger mass flow passes through the gap, which enhances the size and strength of the leakage vortex and the associated losses [71]. The tip leakage vortex therefore intensifies, shifts downward, and enlarges the disturbed region, eventually becoming dominant over the passage vortex,

whose size decreases with increasing clearance, as highlighted by Lampart [60] in Figure 2.18, which shows secondary flow vectors in a section located at 25% of the axial chord downstream of the blade trailing edge. Building on this, Lee *et al.* [72] investigated the effect of the tip gap height-to-chord ratio ($h/C = 1\text{--}4\%$) on the flow structure and heat/mass transfer over the plane tip surface of a high turning rotor blade. Their visualizations and measurements (summarized in Figure 2.19) highlighted that the tip gap flow is primarily governed by the interaction of tip gap vortices with a pressure side separation/reattachment process.

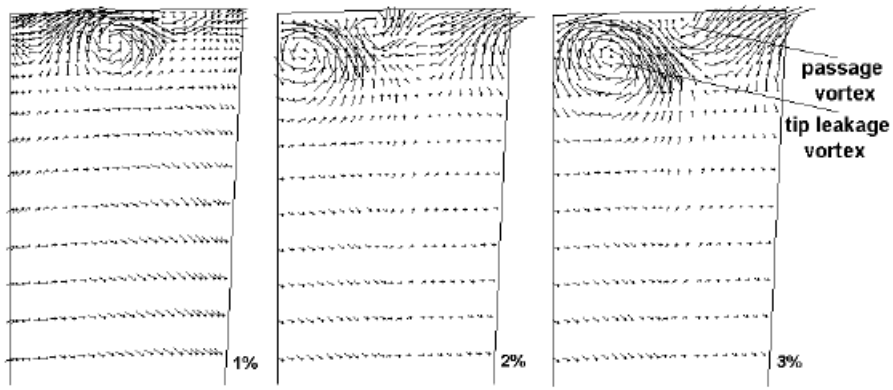


Figure 2.18: Secondary flow vectors in a section located 25% of the axial chord downstream of the rotor trailing edge for different tip gap height-to-chord ratio [60].

At low h/C (1%), the flow remains nearly two-dimensional, with fluid separation and immediate reattachment at the upstream tip edge and no significant vortical structures. Conversely, at high h/C (4%), the flow becomes fully three-dimensional, with the formation of strong suction and pressure side vortices, extended separation/reattachment, and mid-chord flow convergence. These effects markedly intensify local heat and mass transfer, with the pressure side vortex being more dominant and spatially extended than its suction side counterpart. Moreover, Rogo [73] noted that an increase in relative clearance on the rotor reduces tip performance less at very low blade heights than at higher aspect ratios.

2.2.4 Reynolds Number Effect

Among the aerodynamic parameters, the effect of Reynolds number variation on secondary flows is particularly noteworthy. Vazquez *et al.* [74] conducted an experimental study on a low pressure turbine annular cascade to assess its influence on both profile and secondary losses. With regard to secondary losses, they observed that the overall secondary flow patterns (the location of under-turning peaks and loss cores) remain almost

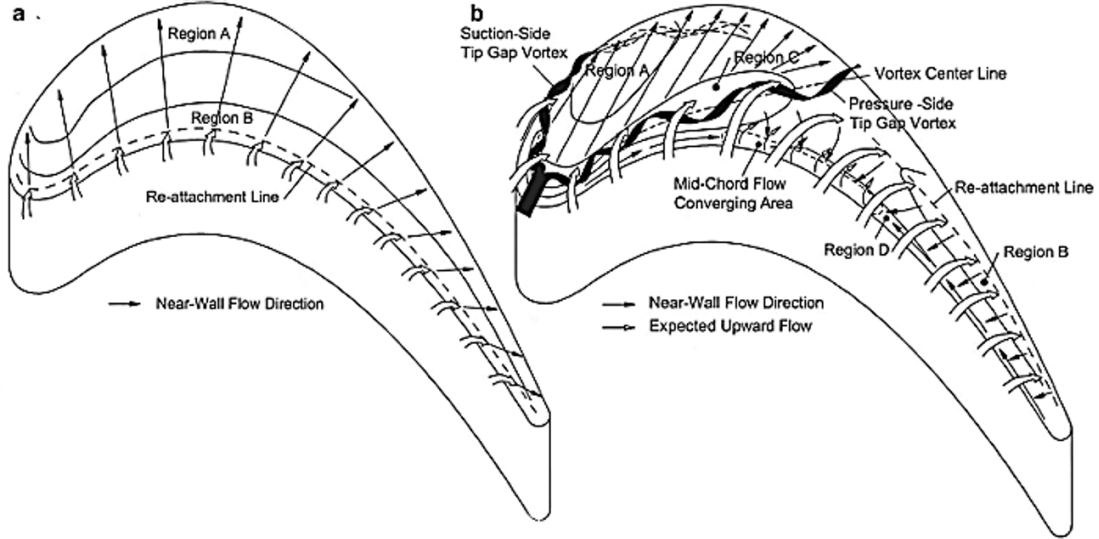


Figure 2.19: Tip gap flow models over a plain tip for (a) $h/C = 1\%$ and (b) $h/C = 4\%$ [72].

unchanged with varying Reynolds number. However, their intensity slightly increases and the structures result a bit more broadened at lower Re , as a consequence of a higher boundary layer. The variation of secondary losses with Reynolds number is therefore very smooth: nearly constant at low values and only slightly decreasing at high values. This trend is consistent with the findings of Satta *et al.* [66] for steady inflow conditions, while for unsteady inflow cases secondary losses appear to be nearly Reynolds number independent.

Additionally, the effect of Reynolds number (based on the tip gap height τ and the mean velocity in the tip gap region $\bar{v}_{tip}$ as $Re_{tip} = \bar{v}_{tip} \cdot \tau / \nu$) on tip leakage losses and heat transfer was analyzed numerically by Moore *et al.* [75]. At very low Reynolds numbers ($Re \leq 100$), the tip gap flow remains laminar, fully developed, and dominated by wall shear losses, with no separation. At $Re_{tip} \sim 1000$, separation occurs at the tip corner, forming a vena contracta whose contraction coefficient depends on corner curvature; limited turbulence causes reattachment around $x = 2h$, with nearly parabolic velocity profiles. At $Re_{tip} \sim 10'000$, the overall behavior is similar, but stronger turbulent mixing shifts reattachment farther downstream, typically between $x = 2h$ and $3h$. Complementary experimental studies by Li *et al.* [76] and Matsunuma [77] reported changes in the distributions of three-dimensional flow, but no effects in mass-averaged tip clearance losses for Re (based on the true chord length and the exit flow velocity from the turbine cascade) between 44'000 and 260'000, indicating a limited effect of Reynolds number on tip leakage flow. Specifically, decreasing Re slightly enhances leakage flow intensity and vortex strength without significantly altering its structure or size. Moreover, Matsunuma [77],

comparing experimental results from an annular cascade with and without tip clearance at two Reynolds numbers, showed that, despite higher secondary losses near the hub and a wider wake at low Re , the interaction between leakage and passage vortices is strongly Reynolds number dependent. As illustrated in Figure 2.20, at high Re , the leakage vortex (A_1) weakens the passage vortex (B_1), whereas at low Re , the passage vortex (C_3) is strengthened by the upward flow induced by the tip clearance (D_3), which modifies the entire flow field.

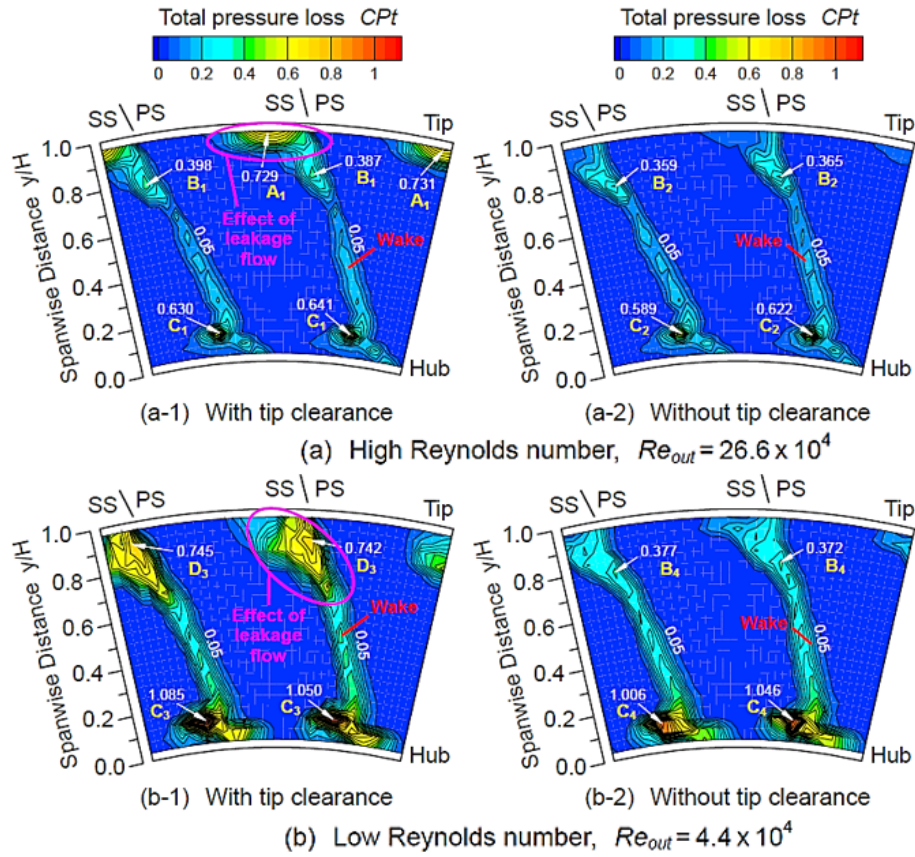


Figure 2.20: Reynolds number effect on the total pressure loss coefficient maps at the exit of cascades with and without tip clearance [77].

2.2.5 Other Geometrical and Flow Parameter Effects on Secondary Losses

Among the other geometric and fluid-dynamic parameters influencing secondary losses, the following can be found in the literature for turbine cascades:

- Leading edge geometry: the leading edge shape primarily modifies the three-dimensional flow field at the cascade exit rather than affecting overall losses, as observed by Moore and Ransmayr [78] and Benner *et al.* [79] that however found only a minor influence on the secondary flow field.
- Maximum profile thickness: Roelke and Haas [80] showed that thinning the blade profiles reduces losses, attributing most of the performance improvement between thick and thin rotor blades—both with the same surface finish—to reduced trailing edge losses.
- Trailing edge thickness: Prust [81] observed that losses increase with thicker trailing edges, and that rounded or streamlined trailing edges are preferable to squared designs.
- Incidence: Murty and Venkatrayulu [82] found that the passage vortex downstream of the cascade is strongest at the maximum positive incidence and weakens as incidence decreases, with low momentum fluid on the suction side rapidly diminishing. Perdichizzi and Dossena [83] observed that this behavior is due to high streamwise vorticity and intense endwall cross-flows at large incidence, which also significantly modify deviation angles, with the maximum over-turning angle varying from nearly zero to 15° . Instead, according to Vazquez *et al.* [74], the tip passage vortex strengthens at lower incidence due to radial migration of low momentum fluid. Indeed, Gao *et al.* [84] further showed that at negative incidences, incoming leakage flow aligns more downstream, increasing streamwise velocity along the vortex core, whereas higher incidence reduces the adverse pressure gradient near the blade tip, causing the tip leakage vortex breakdown location to shift upstream or downstream.
- Turbulence intensity: Matsunuma *et al.* [85] observed that net secondary losses decrease with increasing turbulence intensity for lower levels ($Tu \leq 0.4\%$) but they become constant at higher intensities ($Tu \sim 0.6\%$), consistently across all Reynolds numbers tested. Hence, both tip clearance losses and endwall secondary vortices remain nearly constant across varying freestream turbulence intensities up to 4% [77].
- Inlet boundary layer: Coull *et al.* [86] found that thickening the inlet boundary layer disperses secondary vorticity along the passage, producing larger secondary flow structures and higher secondary kinetic energy (SKE). Giovannini *et al.* [87] highlighted that secondary flow generation is primarily driven by differential convection of near wall vorticity rather than by the outer portion of the boundary layer. Schubert *et al.* [88] emphasized the dominant role of near wall vorticity, showing that reducing inlet boundary layer thickness also decreases peak of over-/under-turning

and secondary losses, shifting these regions toward the endwall. These findings contrast with Chan *et al.* [89], who reported negligible changes in secondary and tip losses with boundary layer thickening.

- Unsteady phenomena: Satta *et al.* [66] observed that under unsteady inflow, secondary flow effects in LPT cascades are generally reduced, accounting for approximately 20% of total losses compared to 40% under steady inflow. Qu *et al.* [63] further observed the effect of unsteadiness in both aft- and front-loaded blades, finding that upstream wakes delay passage vortex formation and suppress secondary flows: in aft-loaded blades, passage and counter-vortex strengths decrease by 21.6% and 13.5%, respectively, while in front-loaded blades, reductions of 17.5% and 28.6% are observed. Moreover, Schubert *et al.* [88] reported a notable reduction not only in maximum secondary losses but also in over-/under-turning, with a spanwise shift toward the endwall.
- Mach number: Secondary losses are highly sensitive to Mach number, showing variations up to 70% between Mach 0.5 and 0.9. In this range, Duden and Fottner [90] observed a general decrease in secondary loss intensity and under-turning with increasing Mach number, while Vazquez *et al.* [74] noted this trend only near the outer wall. Complementary findings by Perdichizzi [91] suggest that net secondary losses tend to increase for $M_{2is} > 0.5$, although their relative importance compared to profile losses diminishes as the Mach number rises. Furthermore, Perdichizzi [91] observed that the passage vortex and loss cores move closer to the endwall while corner vortices gain importance; however, secondary losses undergo a marked reduction in the transonic regime, likely due to turbulence effects and shock wave-boundary layer interactions in supersonic flows.

2.3 Secondary Loss Correlations

The effects of these parameters on secondary losses have been used to define different models in the past. Among these, the momentum-based correlation proposed by Ainley and Mathieson [92] is the most well-known. The authors separated the contributions of secondary and tip losses, describing them as dependent on a parameter Z :

$$Z = \frac{C_L^2}{(s/C)^2} \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} \quad (2.1)$$

which accounts for the effect of the lift coefficient based on mean velocity C_L , the blade pitch s and chord C , and the gas flow angle α measured relative to the axial direction, with α_1 and α_2 referring to the rotor row inlet and outlet, respectively, and $\alpha_m = \tan^{-1}[(\tan \alpha_1 + \tan \alpha_2)/2]$. The secondary Y_s and tip Y_{tip} losses were then expressed as:

$$Y_s = \lambda Z \quad ; \quad Y_{tip} = B \frac{\tau}{h} Z \quad (2.2)$$

In this context, Y represents the total pressure loss coefficient, defined as the stagnation pressure drop across the row normalized with respect to the downstream dynamic pressure. Specifically, the authors considered secondary and tip leakage losses as a drag force acting on the blade, suggesting a linear variation of secondary losses with λ , defined as a function of $\left(\frac{A_{n2} \cos \alpha_2}{A_{n0} \cos \beta_1}\right)^2 \left(1 + \frac{I.D.}{O.D.}\right)$, where A_{n2} and A_{n0} are the annulus areas at the rotor and stator outlet, respectively, and β_1 the blade angle at the rotor inlet, while $I.D.$ and $O.D.$ are the inner and outer turbine diameters. A linear variation of TLV losses was suggested with clearance size made non-dimensional by the blade height τ/h , with $B = 0.5$ for unshrouded blades and $B = 0.25$ for shrouded ones.

By multiplying Y_s and Y_{tip} by $\frac{Re^{-0.2}}{2 \cdot 10^5}$, the effect of Reynolds number variation could also be considered. Matsunuma [77] found a $Re^{-0.5}$ proportionality, while classical correlations assumed $Re^{-0.2}$. Nevertheless, results were closer to the findings of Hodson and Dominy [93], who proposed a $Re^{-0.15}$ exponential law for a more similar type of airfoil.

Dunham and Came [94] later improved the previous correlations, leading to:

$$Y_s = 0.0334 \frac{C \cos \alpha_2}{h \cos \beta_1} Z \quad ; \quad Y_{tip} = B \frac{C}{h} \left(\frac{\tau}{C}\right)^{0.78} Z \quad (2.3)$$

with $B = 0.47$ for plain tips and $B = 0.37$ for shrouded blades. Kacker and Okapuu [95] also confirmed this class of momentum-based models.

It was then with Michael H. [96] that models shifted to energy-based approaches, particularly with Sharma and Butler [52], who focused exclusively on secondary losses, excluding tip losses. They described secondary losses through a semi-empirical model as a portion of the profile loss Y_p , dependent on flow turning θ , convergence ratio $CR = \frac{\rho_2 v_2}{\rho_1 v_1}$ (ratio of density and velocity at the row outlet and inlet), blade pitch s , height h , exit angle α_2 , and trailing edge thickness TET :

$$Y_s = \left[1 + \frac{4\theta}{\sqrt{CR}} \frac{s \cdot (90 - \alpha_2) - TET}{h} \right] Y_p \quad (2.4)$$

Yaras and Sjolander [97] developed an improved energy-based tip-loss model to account for the kinetic energy associated with the velocity component normal to the blade chord in the gap flow, also distinguishing between tip leakage losses in the gap and outside of it, proposing:

$$Y_{tip} + Y_{gap} = 2K_E \frac{C}{s} \frac{\tau}{h} c_D \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} C_L^{1.5} + 0.007 K_G \frac{C}{s} \frac{1}{h/C} \frac{c_D C_L^{0.5}}{\cos \alpha_m} \quad (2.5)$$

where K_E and K_G are constants related to blade loading distributions ($K_E = 0.5$, $K_G = 1$ for mid-loaded blades; $K_E = 0.566$, $K_G = 0.943$ for aft-loaded blades), and c_D is the

discharge coefficient. The same authors [98] later refined the model to include the relative motion between rotor blades and casing, experimentally assessed using a moving-belt tip wall in a linear cascade.

Further correlations for tip loss prediction were established by Denton [37] (and later by Lampart [60]), who sought to better distinguish the mechanisms of loss generation, identifying entropy generation on endwall and blade surfaces, with the former accounting for about one-quarter of the latter.

In 2006, Benner *et al.* [99] extended the secondary loss correlations by incorporating geometric and fluid-dynamic contributions including airfoil loading, flow acceleration, endwall surface area, and inlet boundary layer vorticity, into two formulations depending on the cascade aspect ratio:

$$Y_s = \frac{0.038 + 0.41 \tanh(1.20\delta^*/h)}{\sqrt{\cos\gamma} CR \left(\frac{h}{C}\right)^{0.55} \left(\frac{C \cos\alpha_2}{C_x}\right)^{0.55}} \quad \text{for } h/C \leq 2 \quad (2.6)$$

$$Y_s = \frac{0.052 + 0.56 \tanh(1.20\delta^*/h)}{\sqrt{\cos\gamma} CR \frac{h}{C} \left(\frac{C \cos\alpha_2}{C_x}\right)^{0.55}} \quad \text{for } h/C > 2 \quad (2.7)$$

where γ is the stagger angle measured relative to the axial direction, and δ^* is the boundary layer displacement thickness.

These correlations were later revised by Farokhi [100] and by Beschoner *et al.* [101], who proposed a model with the same structure as Benner *et al.* [99], but modified coefficients fitted to their secondary loss data. They also introduced a simple correlation for tip losses, expressed as a second-order fitting function depending only on τ/h . Buske [102] and others further contributed to this topic by extending correlations to improve accuracy and generalizability.

The most recent contribution is from Liu *et al.* [103], who developed a correlation for secondary losses that explicitly accounts for the height of secondary flows. They introduced Z_{TE}/h , corresponding to the passage vortex penetration depth at the trailing edge, defined as:

$$\frac{Z_{TE}}{h} = f(h/C, \delta^*/h, \alpha_1, \alpha_2, \gamma, s/C, AVR) \quad (2.8)$$

where AVR is the axial velocity ratio. This formulation was proposed to replace the λ coefficient of Ainley and Mathieson [92], as already suggested by Sharma and Butler [52], Benner *et al.* [99] and Teia [67]. Additionally, the loading parameter Z was replaced by a lift coefficient based on the outlet velocity $C_{L,2}$. To account for the radial shift of streamlines and compressibility, this coefficient and the mean flow angle α_m are generalized as follows:

$$\frac{C_{L,2}}{s/C} = \left(1 + \frac{1}{AVR}\right) \left(\frac{\kappa}{AVR} \tan\alpha_1 - \tan\alpha_2\right) \frac{\cos^2\alpha_2}{\cos\alpha_m} \quad (2.9)$$

$$\alpha_m = \tan^{-1} \left(\frac{\frac{1}{AVR} \tan \alpha_1 + \tan \alpha_2}{1 + \frac{1}{AVR}} \right) \quad (2.10)$$

where κ is the meanline radius ratio. Hence, the basic secondary loss coefficient is defined as:

$$Y_s = \frac{1}{2} \frac{Z_{TE}}{h} \left(\frac{C_{L,2}}{s/C} \right)^2 \quad (2.11)$$

Chapter 3

Secondary Flow Control Devices

Secondary flows can have a strong impact on the overall aerodynamic performance of turbine cascades, particularly in configurations where specific vortex structures dominate, such as the passage vortex in low aspect ratio cascades or the trailing edge vortex in unshrouded blades. Therefore, the implementation of control and mitigation strategies is essential to reduce associated losses and the non-uniformity of the flow at the cascade exit.

Flow control systems aimed at reducing secondary flows can generally be categorized into two main types: active and passive systems [104]. Active control systems require an external energy input to operate, typically through the use of devices such as suction, blowing, or plasma actuators, which directly influence the flow field [6]. These systems may act on the incoming boundary layer—reducing its thickness and, consequently, the strength of the secondary vortices—or interact with the vortical structures after their formation. In contrast, passive control systems do not require any external energy supply. They rely on geometric modifications of the blade and channel or on the integration of devices designed to alter the local flow field through their shape alone. Such solutions act on the development of secondary flow structures through momentum redistribution and pressure gradient modification, affecting the boundary layers across the entire passage.

3.1 Active Systems

Bons *et al.* [6] reviewed several active flow control systems, highlighting that there are essentially four common approaches to control fluid flow: flow momentum injection, boundary layer suction, streamwise vorticity generation and excitation of natural instabilities. More recent active flow-control devices will also be described.

3.1.1 Flow Momentum Injection

Among active flow control techniques, streamwise momentum injection—or blowing—has been extensively investigated as a means to manipulate the development of secondary vortices. Blowing can convect vorticity away from regions where it would otherwise intensify aerodynamic losses. For instance, Aunapu *et al.* [105] observed that endwall jets positioned near the passage centerline can modify the trajectory of the pressure side leg of the horseshoe vortex, partially inhibiting its migration toward the suction surface, although without significantly weakening its intensity. This behavior is clearly depicted in Figure 3.1, which presents the endwall ink-dot flow visualizations for (a) the baseline configuration and (b) the case featuring centerline wall jets. Despite these potential benefits, McAuliffe and Sjolander [106] reported that this technique is among the most energy-demanding, and its effectiveness strongly depends on the inlet turbulence level. At low turbulence, total pressure losses can be reduced by up to 75%, whereas under representative engine conditions, increased turbulence instead leads to higher losses.

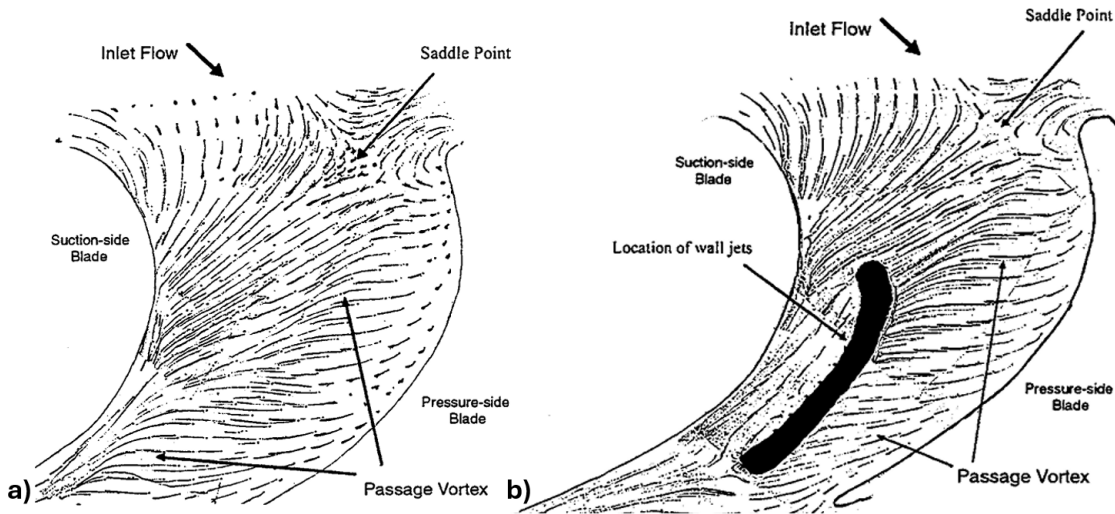


Figure 3.1: Ink-dot flow visualizations on the endwall for (a) the baseline configuration and (b) the case featuring centerline wall jets [105].

Numerical simulations by Romero and Gross [107] demonstrated that steady or harmonic blowing through endwall holes can alter the passage vortex (PV) trajectory, achieving a total pressure loss reduction of approximately 8%. Similarly, Fletcher *et al.* [108] investigated steady and pulsed blowing, exploring different duty cycles and frequencies to identify the most effective configuration for PV mitigation.

Blowing and coolant injection have been widely applied as active flow control strategies to mitigate tip leakage vortices while simultaneously providing blade tip cooling. In this context, cooled air can be injected through holes located on the casing or, more

commonly, directly from the blade tip surface (as shown in the schematic illustration in Figure 3.2), producing a blockage effect [33] that weakens vortex rotation and shields the blade tip from high thermal loads by limiting heat transfer between the mainstream flow and the blade, as discussed by Kim *et al.* [32] and Zhou and Hodson [109]. Mischo

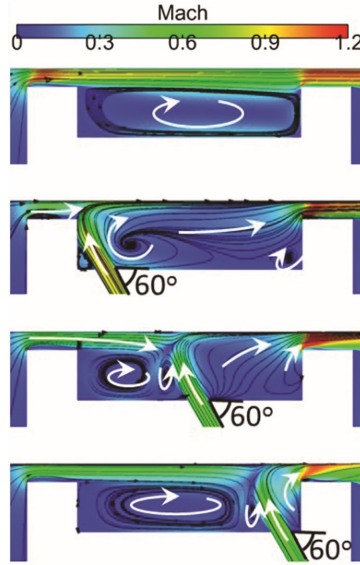


Figure 3.2: Mach contouring serving as schematic description of the impact of tip cooling and its position on transonic tip leakage flow [110].

et al. [31] and Behr *et al.* [111] injected cooling air from the stationary casing toward the rotor tip to simultaneously control the tip leakage and passage vortices. By varying parameters such as injection mass flow, hole axial position, and hole size, the authors observed a variation in vortex size and turbulence intensity, along with a displacement of both vortices in the channel blade passage. An appropriate combination of injection mass flow rate and axial position improved the stage isentropic efficiency by 0.55 percentage points. Instead, Volino [112] investigated tip blowing and hole positions, reporting that blowing normal to the tip is generally ineffective and, in some cases, even detrimental. Conversely, jets issuing from holes located near and inclined toward the pressure side can significantly reduce the total pressure drop by weakening the TLV. The most effective holes are positioned near the region of maximum loading, whereas those placed closer to the leading and trailing edges provide limited benefit. Moreover, Krishnababu *et al.* [35] found that aligning a row of holes beneath the separation line forming on the blade tip surface [113] improves both the uniformity of thermal loading along the tip and the aerodynamic performance of the cascade. Modest increase in jet velocity can further enhance the effect, yielding a total pressure drop reduction of about 20%. Conversely, excessive cooling mass flow (above 0.6% of the main passage flow) increases tip leakage losses,

as analyzed by Rao and Camci [114] and Choi *et al.* [115]. Rao and Camci [34] further observed that the number of coolant holes significantly influences the mitigation of the tip leakage vortex. Finally, the approach proved to be more effective with smaller tip gaps where the blockage effect is stronger, as highlighted also by Zhou and Hodson [109]. For comprehensive discussions on turbine blade tip heat transfer and both internal and external tip cooling technologies, the reviews by Bunker [116] and Sunden and Xie [117] can be taken into consideration. Overall, blowing serves as a dual-purpose solution for vortex mitigation and blade tip cooling, often combined with geometric tip modifications. For this reason, multi-variable and multi-objective optimization approaches, including artificial intelligence techniques such as genetic algorithms, have proven effective in tailoring tip cooling designs to specific cases [118].

3.1.2 Boundary Layer Suction

Suction can effectively remove vorticity together with the associated low momentum fluid. As observed by Bloxham *et al.* [119], this technique can lead to a 10–20% reduction in total pressure losses due to the weakening of the horseshoe vortex as visible in the flow visualizations of Figure 3.3, while Bons *et al.* [6] reported a reduction of about 7%. Moreover, Bloxham and Bons [120] compared two different suction strategies: a removal method, in which suction holes were placed directly along the passage vortex trajectory, and a redirection method, where suction was applied from holes near the pressure side of the adjacent blade to retain the PV in the mid passage and prevent corner stall. The removal approach proved to be more efficient, especially at high stage loading ratios, where the characteristic total pressure loss core of the PV was almost completely eliminated, resulting in up to a 28% reduction in area-averaged total pressure loss. Indeed, Xiao *et al.* [121] demonstrated that slot suction applied along the endwall boundary layer can significantly suppress secondary flows by extracting low momentum fluid, thereby reducing the secondary kinetic energy by up to 44.2%. The authors also observed that increasing the suction rate near the blade leading edge weakened the HSV by disrupting the interaction between the pressure side and suction side legs before the vortex developed into a passage vortex. Under unsteady inflow conditions, the combined effects of incoming wakes and endwall suction further suppressed secondary flows at higher suction mass flow ratios.

Endwall suction has also been adopted to control tip leakage flows. Zhang *et al.* [122] investigated an endwall suction scheme and found that it directly modified the TLV structure, shifting the vortex onset downstream, weakening its intensity, and reducing its spatial extent. This led to improved cascade performance in the tip region (the study was conducted on a compressor configuration). Later, the same authors [123] analyzed the use of a suction slot on the blade tip and introduced an oscillating suction scheme. Their results, shown in Figure 3.4 in terms of 3D streamlines associated with the tip leakage vortex (top) and total pressure loss coefficient (bottom), demonstrate that oscillating suction

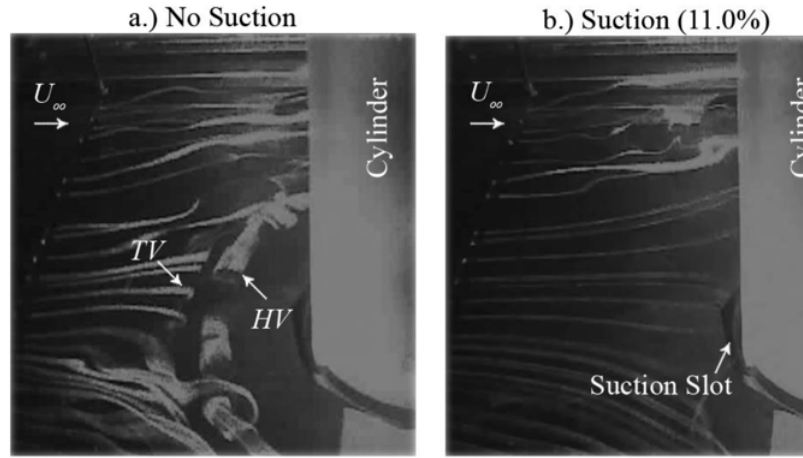


Figure 3.3: Flow Visualization depicting the impact of boundary layer removal on the horseshoe vortex [119].

results in lower time-averaged losses compared to steady suction. When the excitation frequency matched a characteristic frequency or harmonic of the flow field, the control effect was maximized, yielding a total pressure loss reduction of up to 11.14% compared to the uncontrolled configuration.

3.1.3 Streamwise Vorticity Generation

This active flow control can counteract undesirable secondary vortices by injecting vorticity of opposite sign, as demonstrated by Khan and Johnston [124], thereby reducing the strength of the passage vortex. Specifically, Benton *et al.* [125] showed that skewed vortex generator jets produce a single, coherent streamwise vortex, which can generate vorticity opposing that of the PV. Depending on the VGJ orientation relative to the blade surfaces, this method can push the PV away from the suction side. The combined effect can yield up to a 40% reduction in area-averaged endwall total pressure loss, although it requires additional suction at the endwall corner corresponding to about 4% of the main flow. Moreover, given the cost of steady VGJ blowing, the authors observed that reducing the duty cycle from 100% (steady) to lower values can be advantageous. Unsteady blowing may further enhance control effectiveness by generating streamwise starting vortices, hysteresis, or exciting natural flow instabilities. Interestingly, Sullerey and Pradeep [126] observed that VGJs in S-duct diffusers (whose geometry is reported in Figure 3.5.a) with uniform inflow can reduce total pressure loss and flow distortion by more than 30%, outperforming passive tapered-fin vortex generators alone. But under distorted inflow conditions, VGJs alone achieved only an 8% reduction; while, when combined with passive VGs for separation control, total pressure losses were reduced by approximately 25%.

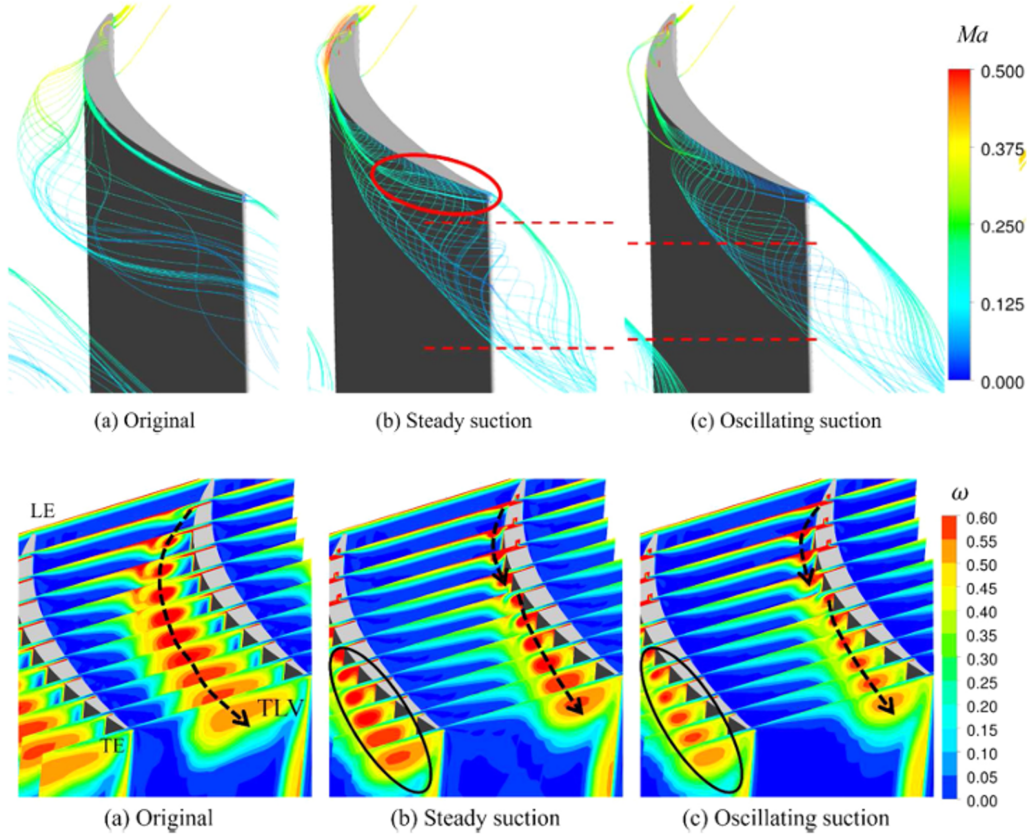


Figure 3.4: 3D streamlines associated with the tip leakage vortex (top) and total pressure loss coefficient (bottom) for the (a) original, (b) steady suction and (c) oscillating suction cases [123].

For this reason, the passive counterpart (the vortex generator visible in Figure 3.5) is also widely employed for boundary layer management and separation control, to mitigate secondary vortices at their origin. The low profile VGs, or “minimal near wall protuberances,” described by Lin [127] can overcome flow separation and the endwall submerged VGs of Paul *et al.* [128] can reduce boundary layer momentum thickness by up to 60%. But as described by Volino [7] and Sondergaard *et al.* [129], their effect alone on secondary flow mitigation is generally limited. Similarly, Santner *et al.* [130] showed that while low profile VGs reduced separation in 2D ducts, their influence in turbine passages was limited due to wakes and strong secondary flows. Ng *et al.* [131] also noted that although VGs can reduce localized high pressure losses, they cannot fully mitigate the strong three-dimensionality induced by secondary flow structures along the blade span. For unshrouded blades, Cao *et al.* [133] and Stephens *et al.* [24] investigated casing mounted passive VGs, reporting that appropriately sized and positioned VGs can reduce

region, which may diminish the positive influence of VGs on separation onset, potentially increasing separation and associated losses. Overall, the effectiveness of active vorticity generation systems is sensitive to inlet unsteadiness and channel curvature, and they do not always improve duct efficiency [8]. Simultaneous application of both passive and active systems can provide significant benefits, whereas the use of passive VGs alone does not directly control or mitigate secondary flows.

3.1.4 Excitation of Natural Instabilities

This active system is based on the suppression effect of instability excitation, as documented by Zaman and Hussain [135], which results from excitation induced modifications of the flow structure and occurs at the excitation frequency corresponding to the “maximally unstable” flow disturbance or to a resonance between the vortex external strain field and the long wave bending mode of the vortex line. This suppression effect is evident in Figure 3.7, which shows the $v'(x,y)$ component of the flow for an (a) excited and (b) unexcited cases. Consequently, it can rapidly diffuse a vortical flow, as reported by Bae *et al.* [136], who actively controlled a tip gap vortex in a compressor cascade and observed that the tip vortex was most susceptible to excitation at a reduced frequency near 0.9, enabling effective modulation and weakening of the vortex structure.

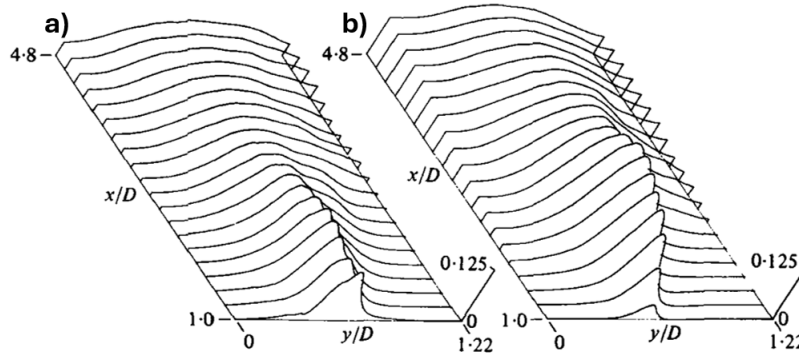


Figure 3.7: (a) Excited and (b) unexcited cases illustrating the suppression effect via instability excitation of the $v'(x,y)$ component of the flow [135].

3.1.5 Other Active Techniques

The choice of fluidic actuator and its implementation depends on specific design constraints and desired outcomes. Overall, injecting or extracting air in the low pressure turbine involves significant cycle penalties. Injection requires air bled from the compressor, bypassing the combustor and reducing overall efficiency since the compressed flow is not used for work. Suction is equally problematic, as the hot extracted flow represents

a complete energy loss unless reinjected downstream, while discarding it externally may act as a drag. Moreover, in the confined turbomachinery environment, high turbulence and unsteadiness make it difficult to selectively excite specific instabilities [6].

Although most documented results involve fluidic actuation, the underlying principles are applicable to other methods. Active flow control using a Dielectric Barrier Discharge (DBD) plasma actuator has been experimentally investigated by Matsunuma and Segawa [137] as a potential technique for passage vortex reduction in a linear turbine cascade. The actuator, driven by sinusoidal excitation, induced tangential jets that reduced passage vortex velocity and vorticity by approximately 55% and 74%, respectively; however, at higher freestream velocities, vorticity reduction dropped to 17%. Figure 3.8 shows (a) a schematic of the Dielectric Barrier Discharge plasma actuator operation and (b) its effect on the vorticity induced by the passage vortex. Subsequently, Stephens *et al.* [24] reported that three plasma actuators reduced total pressure losses associated with the tip leakage vortex by 7%. Matsunuma and Segawa [138] observed that, for small clearances, the actuator effectively reduced secondary losses related to the passage vortex, homogenizing the outflow velocity profile. Conversely, for larger gaps where the TLV dominates, the plasma actuator increased TLV related losses, necessitating two separate actuators with

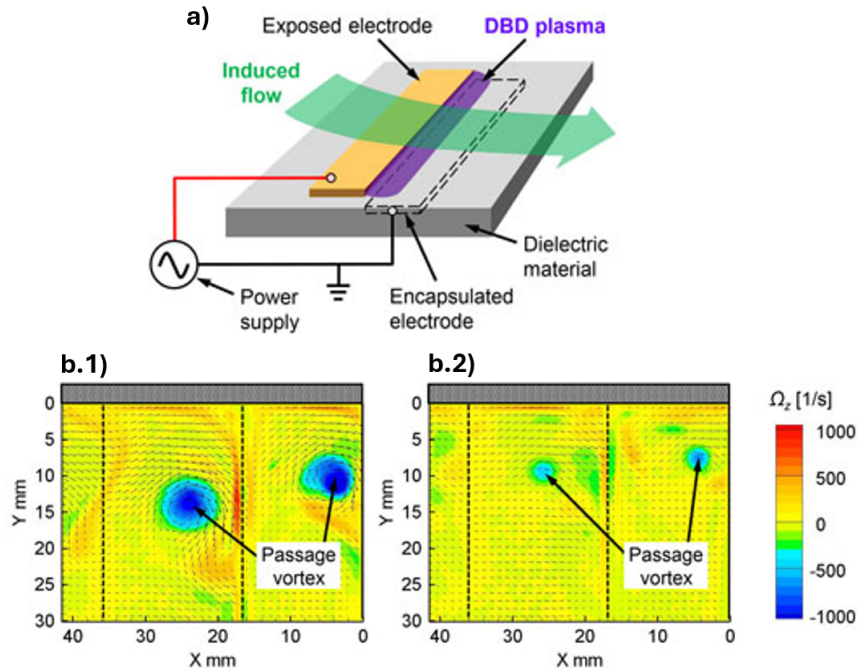


Figure 3.8: (a) Schematic of the Dielectric Barrier Discharge actuator and its operating principle; (b) effect on PV vorticity, with (b.2) showing the controlled case compared to the baseline without active flow control (b.1) [137].

different input voltages to independently mitigate PV and TLV losses. The same authors [139] and Y. *et al.* [25] also investigated plasma actuators acting directly on the boundary layer in the tip clearance and the formation of the leakage vortex, demonstrating promising results, albeit at a low experimental Technology Readiness Level (TRL). This study extended the work of Sharma and Butler [52], who, as shown in Figure 3.9, observed that, compared to conventional film cooling techniques, plasma actuators effectively redirect the cooling jets toward the blade tip surface, significantly enhancing cooling efficiency in highly loaded turbine squealer tips under extreme thermal conditions. On the other hand, the forced adherence of the flow along the cavity floor reduces the blockage effect of the coolant mass flow on the tip leakage flow, which explains why the plasma actuator performs well primarily at low gap heights.

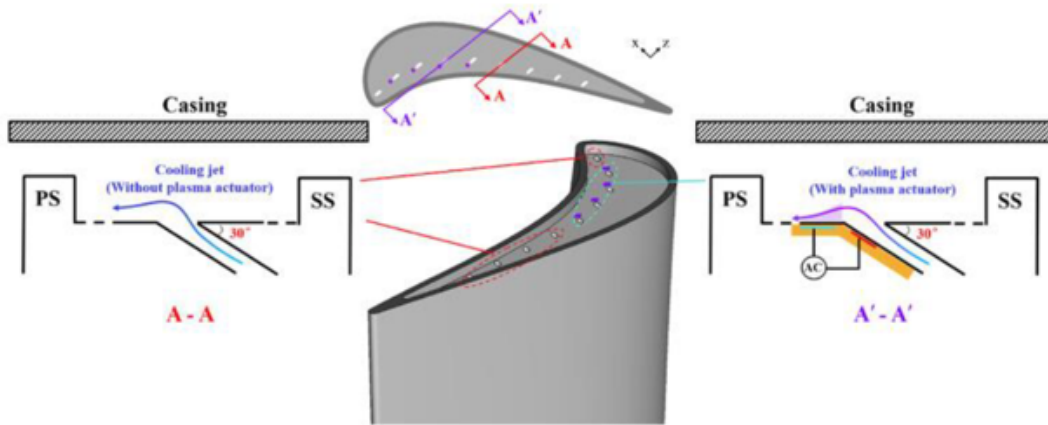


Figure 3.9: Conventional cooling system and cooling jet enhanced by a plasma actuator in a squealer turbine blade tip [52].

Other actuation methods include: fluidic oscillators, which generate oscillating jets over a wide fan angle without moving parts, making them attractive actuator concepts [140]; piezoactuators, such as elastomer piezoadaptive stator blades, where the leading edge oscillates in phase with incidence fluctuations to compensate for adverse effects [141]; combustion driven devices, capable of producing net pressure gain alongside temperature increases during the combustion process [142]; acoustic actuators, in which active noise generates secondary antiphase sound fields that interfere with primary radiated waves [143].

3.2 Passive Devices

Passive approaches for secondary flow control and mitigation involve both structural modifications of the channel and blades as well as the application of add-on devices.

3.2.1 Structural Modifications

Channel modifications are particularly widespread in intermediate turbine diffusers (ITDs). Over the years, a variety of optimized meridional shapes have been proposed and tested: Gräsel *et al.* [9] employed parametric design optimization to reduce boundary layer separation and attenuate secondary flow structures, while Wallin *et al.* [144] reported a total pressure loss reduction of 25%. More recently, Yang *et al.* [145] redesigned an arc-line-arc ITD configuration, achieving a 10.27% reduction in total pressure loss coefficient without compromising diffusion performance, which increased by 34.05%. A comprehensive overview of ITD designs visible in Figure 3.10 and including the integration of active and passive flow control systems, is provided by Göttlich [132]. Spataro *et al.* [146] also highlighted the benefits of integrating secondary flow mitigation systems in S-shaped channels with wide chord vanes.

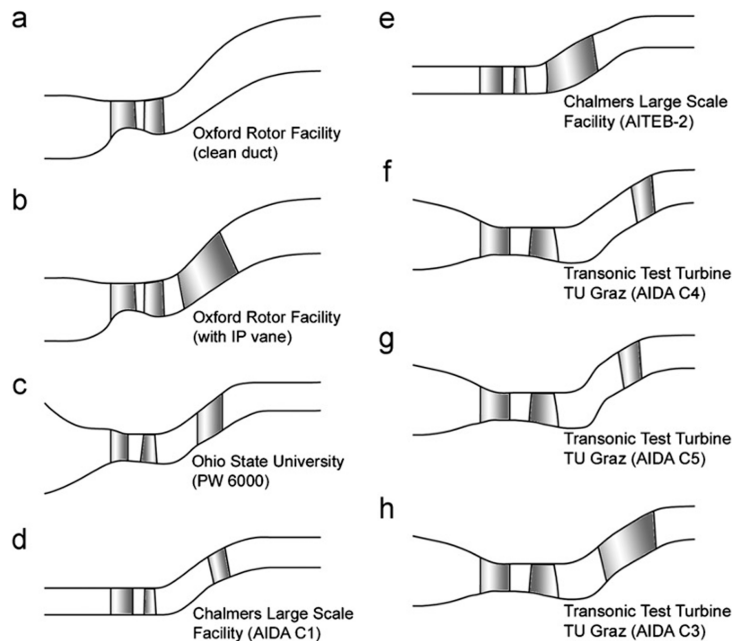


Figure 3.10: Intermediate Turbine Diffuser designs [132].

3.2.2 Endwall Contouring

Among the available methods, endwall contouring is the most widely adopted approach to weaken the passage vortex in intermediate turbine diffusers [147] modifying the endwall shape. An example of this approach is shown in Figure 3.11 (top), which presents the geometry of (a) flat and (b) contoured endwalls. Following the endwall design principles

first introduced by Deich *et al.* [10], this method involves shaping the casing to produce non-axisymmetric endwalls in both streamwise and pitchwise directions. By redistributing the endwall pressure gradients across the passage from the pressure to the suction surface, Ingram *et al.* [148] observed that endwall profiling has a remarkable impact on the flow structure development within the blade row, leading to a total pressure loss reduction, as experimentally assessed by Kopper *et al.* [149]. Then, Gregory-Smith *et al.* [150] validated a design method for endwall profiling, confirming its capability to mitigate secondary flow development. Using a linear design framework, Harvey *et al.* [151] developed non-axisymmetric endwalls to control the pressure gradient across the blade passage, reducing secondary kinetic energy by up to 40% and minimizing exit flow angle deviation, which resulted in an overall loss reduction of approximately 4%. Endwall contouring proved to be particularly beneficial in low aspect ratio nozzle guide vanes, where both primary and secondary losses were mitigated [152]. Also in this case the latter were reduced by 4%, and, interestingly, a quasi-linear distribution of the exit flow angle was achieved over most of the span—an advantage in real engines, as it improves the uniformity of the inlet flow angle to the downstream rotor, enhancing stage performance.

Further optimization strategies were introduced by Wallin and Eriksson [153], who employed a surrogate model based design method to generate endwalls with pronounced contraction. This configuration locally accelerated the flow, suppressing boundary layer separation and inhibiting secondary flow formation in critical regions of aggressive ducts [154]. Praisner *et al.* [155] extended these concepts to conventional and high lift low pressure turbine airfoils, demonstrating that endwall contouring primarily mitigates losses associated with the passage vortex. Snedden *et al.* [156] reported a 0.4% improvement in rotor efficiency and observed that contouring weakened the cross-passage pressure gradient, as evident from the streamlines of the two branches of the horseshoe vortex shown at the bottom of Figure 3.11 compared to the baseline case. Moreover, the study highlighted that endwall contouring can also affect tip clearance flows, underlining additional potential for performance gains through tailored endwall shape optimization. Consequently, adjoint-based optimization techniques were employed by Luo *et al.* [157] and Shahpar *et al.* [158], showing a significant reduction in secondary losses due to decreased pitchwise pressure gradients.

Moreover, a properly tailored endwall curvature near the blade tip can effectively limit the formation and migration of secondary tip vortices. Ameri *et al.* [159] were among the first to examine casing recesses—cavities machined into the casing above the blade—which modify the leakage flow path and mitigate its interaction with the main passage flow. Similarly, Wei *et al.* [160] and Bohn *et al.* [161] demonstrated that arc-shaped contouring of the casing and rotor tip can yield an overall efficiency improvement of approximately 0.14%. Expanding on these concepts, Ahmed and Masud [162] investigated three recess locations and three depths, achieving an efficiency gain of about 6.2%.

In an overview provided by Gao *et al.* [163], the authors experimentally investigated

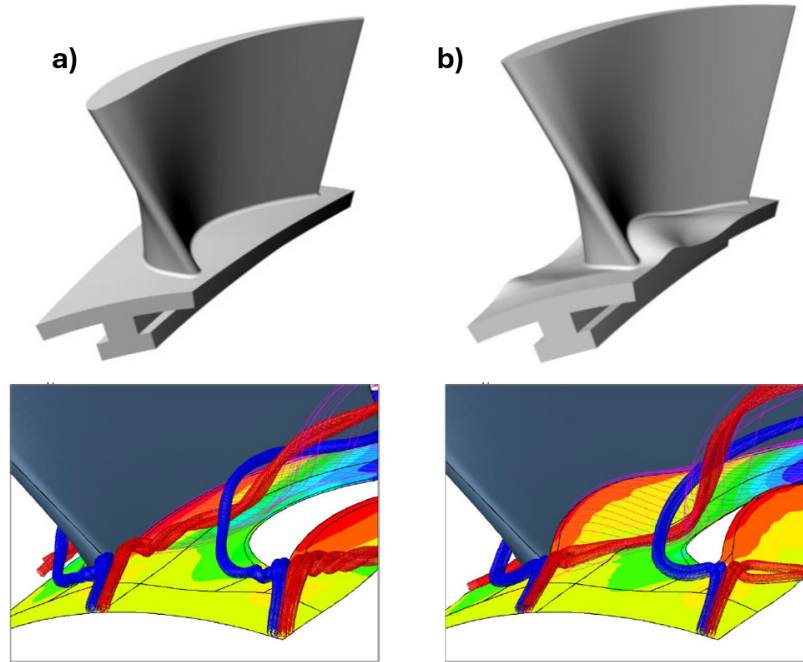


Figure 3.11: Visualizations of the (top) geometry and (bottom) pressure contours, with a close-up of the SS-HSV in blue and PS-HSV in red, for (a) flat and (b) contoured endwalls [156].

four casing geometries—standard, step (recessed), trench, and arc-shaped (see Figure 3.12)—in an unshrouded turbine rotor under both design and off-design conditions. The aim was to identify an optimal casing capable of minimizing direct tip leakage and exploiting the interaction between the tip leakage vortex and the passage vortex to reduce total losses. The arc casing performed best at design incidence, while the step casing showed superior off-design behavior, effectively controlling tip leakage and passage vortex interactions. Nevertheless, further optimization was deemed necessary to minimize tip clearance losses. In parallel, Hilfer *et al.* [164] applied the Durham endwall design system to assess the influence of alternative profiled endwalls on tip clearance flow, confirming their impact on vortex dynamics. More recently, Jiang *et al.* [165] proposed a novel casing contour based on a bimodal Gaussian function curve, numerically optimized in a linear turbine cascade for the suppression of tip leakage flow. The optimized configuration achieved a 17.75% reduction in leakage mass flow rate and a 10.87% drop in total pressure losses. Thermal aspects have also been addressed. Lynch *et al.* [166] measured up to a 20% reduction in local heat transfer on contoured endwalls compared to flat ones, attributed to the attenuation of secondary vortices, with negligible dependence on Reynolds number. Likewise, Panchal *et al.* [167] observed a 10% decrease in the overall average heat transfer coefficient, while Roy *et al.* [168] reported a more uniform heat

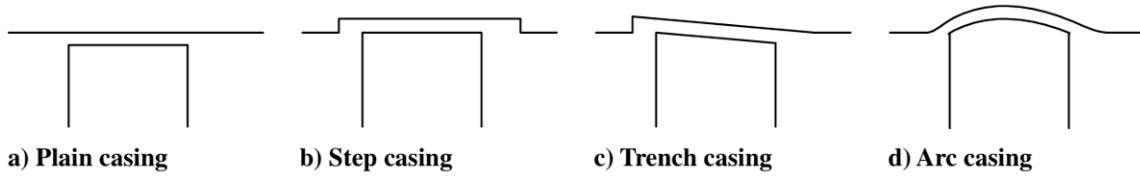


Figure 3.12: Four casing geometries illustrating casing contouring for the reduction of the tip leakage vortex [163].

transfer distribution across the entire endwall.

However, Schobeiri and Lu [169] observed that despite a 0.5% performance improvement and reduced secondary flow intensity, endwall contouring can suffer from geometric complexity. It often performs suboptimally under off-design conditions and may interfere with cooling design, thereby complicating manufacturing and maintenance. For these reasons, the adoption of dedicated devices represents a promising alternative, offering increased design flexibility without requiring substantial modifications to the meridional channel or endwalls. Such secondary flow control systems are among the most effective solutions for maintaining low aerodynamic losses and a uniform flow field as described by Lord *et al.* [11].

3.2.3 Endwall Fences

As early as 1972, Prümper [12] exploited the endwall as a mounting location for adaptable, robust, and easily installed add-on flow control devices for real aircraft engines, proposing the concept of endwall fences—small barriers placed on the endwall within the blade passage—to block cross-flow and inhibit the migration of the passage vortex toward the blade suction surface, as shown in Figure 3.13. Later, Kawai *et al.* [170] identified the optimal fence configuration as having a height equal to one-third of the inlet boundary layer thickness and positioned half a pitch from the blades. This arrangement effectively reduced the streamwise vortices, the thickness of the endwall loss region, secondary kinetic energy, and maximum flow under-turning, leading to a 22% reduction in secondary losses and a 25% decrease in total losses, including kinetic energy dissipation. Chung and Simon [171] further investigated endwall boundary layer fences by analyzing in detail the effects of their leading and trailing edges. Specifically, they observed that a sharp leading edge mitigates the horseshoe vortex formed ahead of the fence by reducing the local deceleration region, while a sharp trailing edge promotes smoother boundary layer release, thereby reducing secondary flow effects and enhancing the potential for efficient film cooling on the suction surface near the endwall. Similarly, Aunapu *et al.* [105] demonstrated that boundary layer fences effectively alter the vortex path, weaken its strength, and minimize the detrimental effects of the pressure side leg of the horseshoe vortex. Importantly, they observed that the fence is only crucial in the middle portion of the inter blade passage,

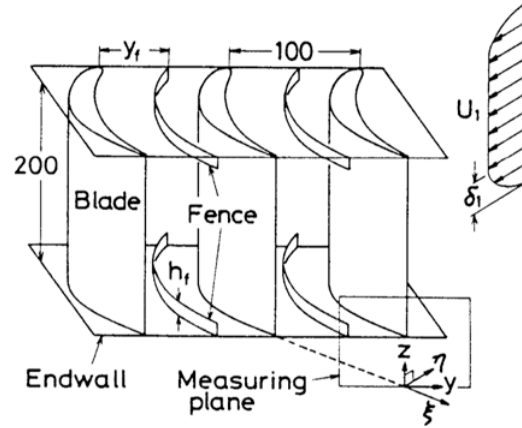


Figure 3.13: Example of endwall fence implementation in a turbine blade cascade [170].

meaning its length can be shortened without significant performance degradation.

Moon and Koh [172] provided additional insights into the formation of counter-rotating streamwise vortices over fences, analyzing in detail the mechanism by which endwall fences mitigate the development of the passage vortex, reduce secondary flow penetration, and enhance the pitchwise uniformity of vorticity. Duden *et al.* [173] also combined endwall contouring with fences in the endwall region, achieving further improvements in the radial extent and strength of secondary flow. Interestingly, Sullerey *et al.* [174] extended the application of endwall fences to non-vaned S-duct diffusers, reporting a noticeable improvement in efficiency, with a 6% reduction in exit flow angle distortion, thereby enhancing the uniformity of the inlet flow field to the downstream blade rows.

Subsequently, Govardhan *et al.* [175] confirmed that endwall fences weaken the cross-flow kinetic energy and inhibit the migration of the horseshoe vortex toward the suction side, thereby reducing low energy fluid accumulation on the endwall and decreasing the mixing-out of vortices downstream of the blade passage. This led to an increase in lift coefficient and a substantial reduction in drag, demonstrating an overall improvement in turbine cascade performance. With the advent of advanced optimization techniques, Kumar and Govardhan [176] proposed an innovative fence with height varying linearly from the leading to the trailing edge and positioned centrally within the flow passage. This configuration minimized the cross-flow secondary kinetic energy, by 27% relative to the baseline case, and significantly mitigated both over-turning and under-turning of the flow. Consequently, the magnitude and spanwise penetration of the passage vortex were substantially reduced, with reported performance improvements in low aspect ratio NGVs ranging from 1% to 2%. Finally, Mingardo *et al.* [177] and Yuan *et al.* [178] investigated non-uniform height endwall fences, demonstrating their effectiveness in preventing boundary layer cross migration and suppressing passage vortex development. They reported a 14.82% reduction in total pressure loss at the cascade outlet, and based on

feature importance analyses, designed a 53% shorter fence to reduce additional weight while maintaining the same aerodynamic performance. These findings can be observed from the total pressure loss contours and streamlines shown in Figure 3.14. Compared to the baseline configuration (a), the configuration with endwall fences (b) shows a noticeable reduction in the PV intensity and an improved flow alignment, which is further enhanced in the case with optimized endwall fences (c). However, in reviewing several endwall modification systems aimed at reducing losses and heat transfer, Simon and Piggush [179] highlighted that endwall fences can locally increase heat transfer, thereby posing additional challenges for cooling.

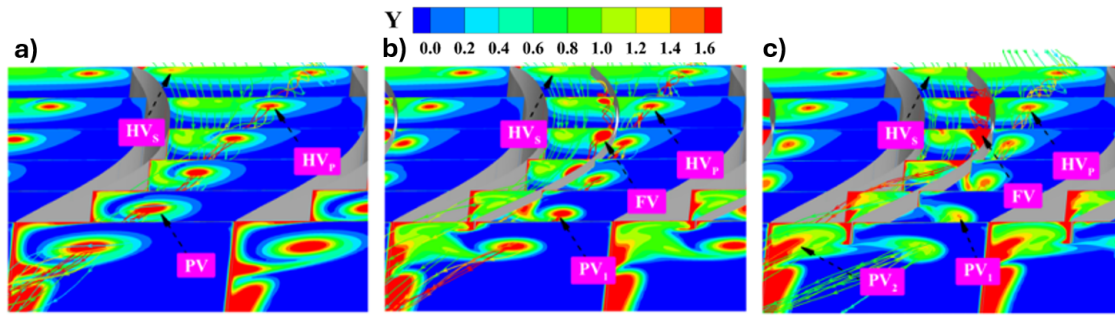


Figure 3.14: Total pressure loss contours and streamlines for (a) baseline, (b) endwall fence and (c) optimized endwall fence configurations [178].

3.2.4 Endwall Splitter

Inspired by endwall fences, Kreitmeier and Greim [13] demonstrated that endwall mounted splitters—higher aspect ratio vanes than the endwall fences, positioned in the middle of the inter-blade channel—offer an alternative passive solution to limit the propagation of secondary vortices from the blade pressure side to the adjacent suction side, also reducing outlet secondary kinetic energy by splitting the inlet circulation. Spataro *et al.* [146] highlighted their particular effectiveness when integrated as secondary flow mitigation systems in S-shaped passages with wide chord vanes. Clark *et al.* [180] reported that splitters reduced secondary flow penetration compared to the baseline configuration, as shown in Figure 3.15. This led to a 6.9% reduction in NGV row losses and a 77.4% reduction in NGV exit mixing losses, resulting in a 0.88% improvement in stage efficiency relative to the baseline. Multi-splitter turbine cascades (such as that reported in Figure 3.15.c) were also investigated by Zhou *et al.* [181], showing that performance increases with the number of splitter blades (even with limited turning), outperforming designs with struts or blunt blades. However, Solano *et al.* [182] noted that while flow follows the aero-vane profiles under design conditions, the structural vane can adversely affect the aerodynamics of neighboring aero-vanes in off-design operation. Moreover, Russo *et al.* [14] confirmed

that although multi-splitter configurations effectively minimize secondary flow interaction and reduce NGV exit losses, their broader application is limited by cooling constraints.

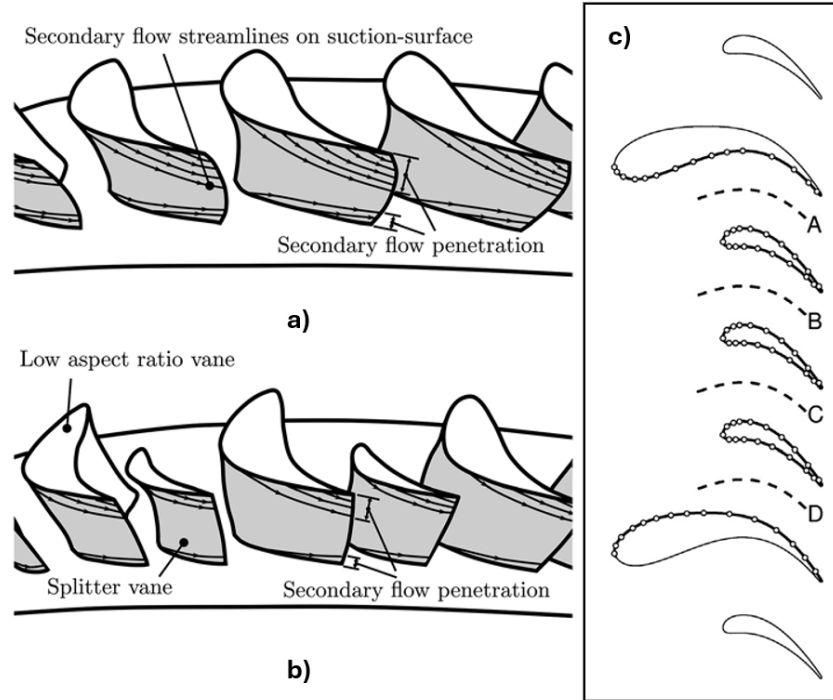


Figure 3.15: (a) Baseline, (b) splitter [180] and (c) multi-splitter cascade configurations [182].

Pioneeringly, Doerffer and Amecke [183] combined the concept of endwall fences with a multi-splitter approach, testing multiple endwall fences installed within the blade passage channel. The fences were oriented both in the transverse and streamwise directions. The transverse fences, while producing a passage vortex with low vorticity, generated a strong pressure side branch of the horseshoe vortex, resulting in higher local losses. In contrast, the streamwise fences shifted the loss peak corresponding to the suction side horseshoe vortex, significantly reducing the spanwise extent of the secondary flow. However, the fence height tested appeared excessive requiring further detailed investigation.

3.2.5 Endwall Ridging

A more advanced and innovative passive approach to secondary flow control, involving endwall ridging—grooved structures applied on the endwall and aligned with the streamwise flow—originated from this study [183] and was inspired by the functioning of riblets. Riblets are microstructures designed to manipulate the near wall boundary layer

flow, reducing skin friction by altering turbulent vortices and thereby minimizing energy dissipation [15]. They were originally developed for external aerodynamic applications, such as aircraft nacelles and wings, by Viswanath [184] and Stockman *et al.* [185] who reported drag reductions of up to 8%. Subsequently, Hage *et al.* [186] and Lv *et al.* [187] identified trapezoidal concave grooves as the most effective geometry for loss reduction, noting that although riblets typically require alignment with the main flow direction for maximum efficiency, trapezoidal grooves remain effective even under moderate misalignment. Building upon these principles, and considering the economic advantages of riblets and ridges in terms of manufacturing and maintenance costs highlighted by Lynch and Klinge [16], several subsequent studies investigated the feasibility of medium-scale engineered grooves or ridges. These structures are neither micro-scale, as in riblets, nor large-scale, as in endwall fences or splitters, and were applied to endwall surfaces to mitigate secondary vortices in internal aerodynamic applications [17], [18], [188]. An example of such implementation is shown in Figure 3.16 (top). Interestingly, Miao *et al.*

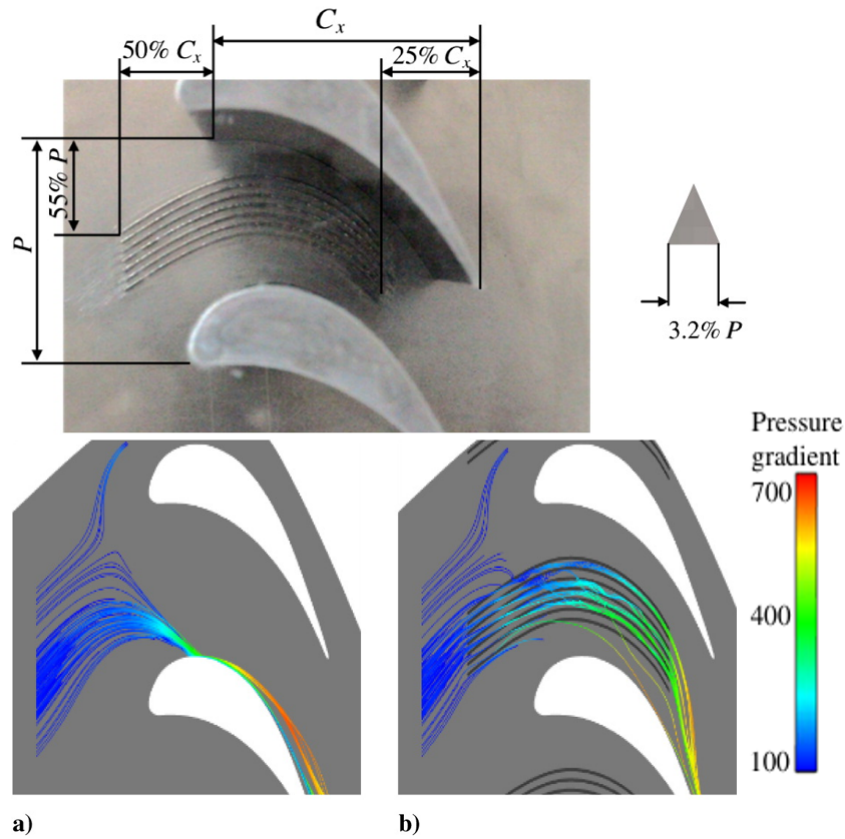


Figure 3.16: Top: ridge geometry in a turbine blade cascade; bottom: endwall fluid streamlines with pressure gradient contours inside the channel for (a) baseline and (b) ridged cases [20].

[20] were the first to explore the potential of ridges applied to the endwall of axial turbines (Khader and Sayma [19] analyzed their effects on endwall radial turbines), demonstrating their ability to weaken secondary flow structures, reducing the pressure gradient inside the blade channel, thereby blocking the migration of the passage vortex, as evidenced by the streamlines at the bottom of Figure 3.16. Moreover, the formation of small scale vortices between the crests contributes to reduce the passage vortex lift up, enabling performance improvements of up to 1%. Nonetheless, the authors emphasized the need for further design optimization to fully exploit the potential of endwall ridging.

At the same time, and in ongoing research efforts, several studies have focused on the effect of transverse grooved casing treatments on tip leakage flow and, consequently, on the aerodynamic performance of linear unshrouded blade turbine cascades. Although grooved casings are widely employed in compressors to improve stall margin [189], Kavurmacioglu *et al.* [190] investigated four different configurations of circumferential grooves also on the casing of a single stage axial turbine, as presented at the top of Figure 3.17. Their results in terms of total pressure losses are shown at the bottom of the same figure. A comparison between the baseline case (a) and the grooved casing case (b) indicates that the grooves reduce the size of the leakage vortex in both spanwise and pitchwise directions, although a slight increase in peak losses is noted. Moreover, the authors observed that both converged and diverged circumferential grooves decrease the leakage flow rate and the total pressure loss, unlike square grooves, while the triple diverged grooved configuration provides the best aerodynamic performance. These findings were later confirmed by Shi *et al.* [191] in a study on guide vanes of a variable geometry

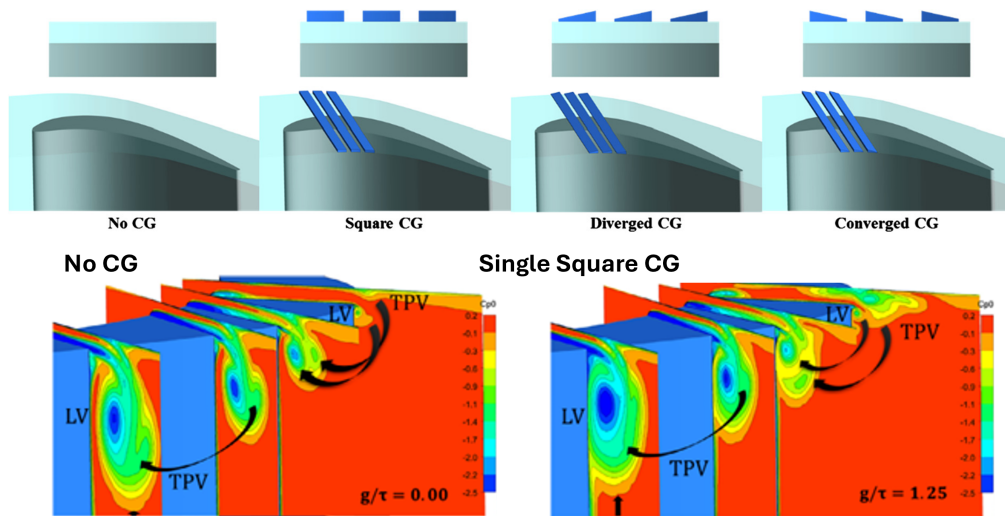


Figure 3.17: Top: circumferential grooves on the casing of a axial turbine; Bottom: total pressure loss contours for the (left) baseline and (right) grooved cases [190].

turbine, where the circumferentially grooved vane configuration significantly attenuated the intensity and size of the horseshoe and passage vortices, leading to a marked reduction in endwall flow losses associated with these structures. Furthermore, the authors in [192] found that increasing the guide vane opening angle did not exacerbate the aerodynamic performance degradation, highlighting the potential of casing groove based designs as a transformative approach to optimize turbine aerodynamic performance also in off-design conditions through effective management of endwall secondary flow losses.

3.2.6 Blade Fillet

Beyond passive solutions involving structural endwall modifications or devices mounted on it, several studies have focused on redesigning the blades themselves or integrating flow control devices directly on their surfaces. In this context, Wang *et al.* [53] first suggested that rounding the sharp intersections between the blade and the endwall—particularly near the leading edge—could mitigate the generation of secondary structures. The aerodynamic potential of such fillets was soon confirmed by Shih and Lin [193], whose fillet geometries are illustrated in Figure 3.18, and by Zess and Thole [194], whose results are presented in Figure 3.19. Specifically, Han and Goldstein [195] observed that fillets

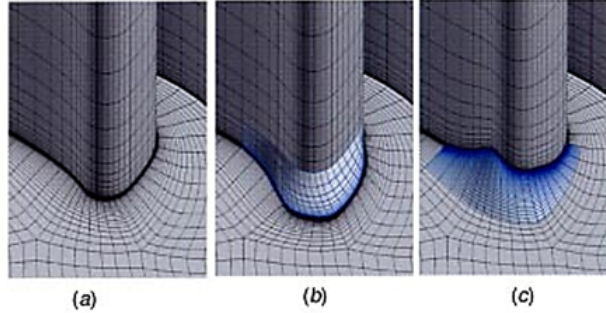


Figure 3.18: Blade (a) without and (b-c) with different fillet geometries [193].

significantly weaken the horseshoe vortex and delay the formation of the passage vortex, especially under high turbulence intensity. Mahmood and Acharya [196] compared linear and parabolic leading edge fillets, reporting a reduction in the size and strength of the suction side leg of the horseshoe vortex, along with decreases in pressure loss coefficients and pitch angles. Further downstream, the total pressure loss and vorticity distributions revealed that fillets lift the passage vortex higher above the endwall and shift it closer to the suction side, while reducing its strength near the trailing edge. Overall, the linear profile fillet achieved the best overall performance. Wei *et al.* [197] clarified the physical mechanisms behind the effectiveness of leading edge fillets, identifying three main aspects: (i) the fillet fills and suppresses the corner separation vortex region, (ii) it reduces

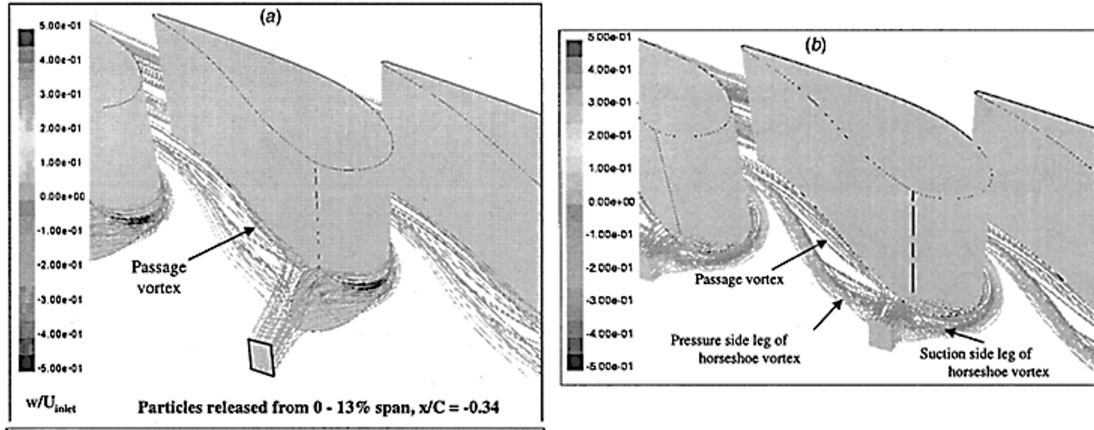


Figure 3.19: Streamlines over a (a) filleted and (b) un-filleted vane [194].

the local cross-pressure gradient, thereby decreasing loss coefficients and flow angle deviation at the exit plane, and (iii) it weakens both leading edge horseshoe vortex and downstream passage and shed vortices, collectively reducing endwall secondary losses. More recently, Hawes *et al.* [198] analyzed the influence of fillet curvature, showing that a uniform large radius fillet (16% of the axial chord) actually increased endwall losses by about 10%. Conversely, an optimized variable radius fillet—identified via genetic algorithm optimization—minimized losses by combining a large leading edge radius with a slowly decreasing pressure side radius, a rapidly decreasing suction side radius, and a medium radius near the trailing edge. Similarly, Ananthakrishnan and Govardhan [199] found that variable leading and trailing edge fillet geometries with streamwise radial variation outperform constant radius designs for both stator and rotor blades. These configurations effectively reduced the height and strength of the secondary flow region, suppressed trailing edge separation bubbles, minimizing boundary layer growth in this region. Increasing the leading edge radius further decreased the intensity and size of the horseshoe vortex, hence reducing secondary flow losses. Very recent numerical simulations by Li *et al.* [200] extended this concept by analyzing leading edge fillets with different bending directions, lengths and thicknesses. The study demonstrated that both forward- and reverse-bending leading edge fillets improve turbine efficiency and reduce losses by weakening the passage vortex, suppressing its lift-up, and splitting the horseshoe vortex into less energetic structures. Moreover, the authors observed that the total pressure loss coefficient is more sensitive to variations in fillet length than in thickness. Although fillets are commonly implemented at the blade endwall junction for structural purposes—such as alleviating thermal and mechanical stress concentrations—they can also act as localized endwall contouring, potentially providing both aerodynamic and thermal benefits. For instance, Burigana *et al.* [201] reported a stage efficiency gain of 0.3% with a more uni-

form heat transfer distribution. However, improper design may lead to increased losses, as evidenced by Han and Goldstein [195] through the enhancement of corner vortices near the leading edge resulting in locally intensified mass transfer, by Mank *et al.* [202] with a 10% increase in losses relative to the baseline, and by Turgut and Camci [203] through the deterioration of overall stage performance caused by specific fillet geometries.

3.2.7 Blade LE-bulb

Another method to reduce secondary flows involves modifying the blade profile near the endwall by introducing a leading edge endwall bulb, as described by Sauer *et al.* [55]. Here, the authors experimentally investigated the effectiveness of the leading-edge end-wall bulb (see the left hand side of Figure 3.20) applied to a highly loaded low pressure turbine cascade. In this configuration, the suction side branch of the horseshoe vortex is intensified by the bulb in the endwall region. Since it rotates counter to the passage vortex, it displaces and deforms this structure, thereby moving it away from the suction side boundary layer. This interaction results in a substantial reduction of endwall losses—up to approximately 50%, as evident on the right of Figure 3.20 for certain LE bulb geometries. Shih and Lin [193] further analyzed the leading-edge bulb, also in combination with various sharp-pointed and rounded leading edge fillets, showing the potential of such

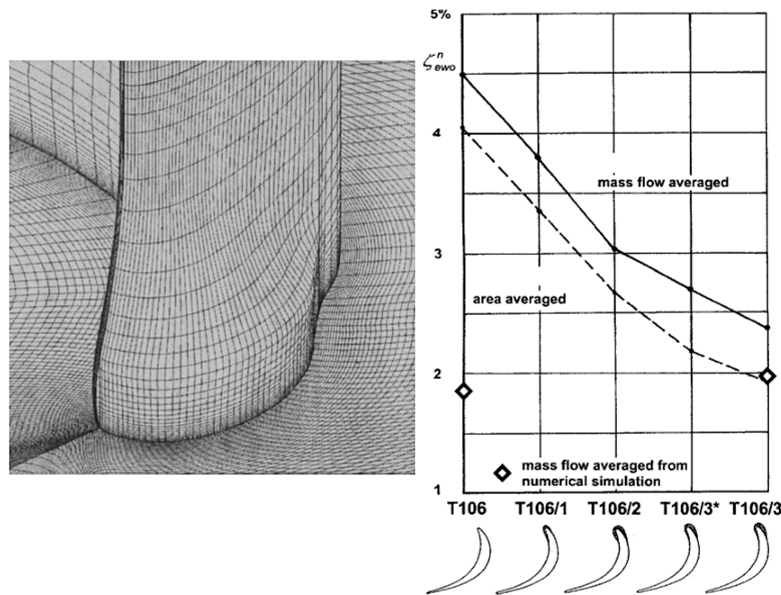


Figure 3.20: Leading-edge bulb: (a) geometry and (b) effect in terms of loss reduction [55].

geometries to simultaneously reduce aerodynamic losses and surface heat transfer. However, Becz *et al.* [204] reported that, while the fillet configuration reduced the overall loss by about 7%, the bulb alone did not exhibit a measurable loss reduction. For the fillet case, the overall turning was slightly reduced, whereas the bulb slightly increased the turning, indicating its potential to enhance airfoil loading without an associated penalty in loss. Interestingly, the regions of greatest loss reduction differed between the bulb and fillet configurations, suggesting distinct underlying flow control mechanisms. The authors highlight the possibility of combining the most effective features of both designs to define an optimal hybrid shape for minimizing secondary flow losses.

3.2.8 Blade Leaning and Re-Staggering

Another technique investigated for secondary flow control, though with limited success and cited here for completeness, is blade leaning, which consists of stacking the blade at an angle differing from 90° relative to the endwall, as represented in Figure 3.21. In addition to the pressure gradient from the pressure to the suction surface, spanwise static pressure differences along the blade surfaces contribute to feeding the passage vortex and generating cross-flow. Leaning the blade redistributes the load along the surfaces, creating a radial component that can counteract the cross-flow. Harrison [205] studied this approach, comparing a leaned blade with a prismatic blade of identical airfoil. The authors observed a weakened passage vortex near the tip endwall, where the boundary layer was thinner and separated regions were minimal, whereas the hub endwall experienced a thickened boundary layer and stronger vortex. Despite local improvements, the overall losses were higher in the leaned blade case. Later, D'Ippolito *et al.* [206] performed an optimization of the lean angle and concluded that even with an optimal lean of 10° , the improvements were negligible. More recently, Neipp and Riedelbauch [207] noted that positive compound lean in optimized rotor blades can slightly increase efficiency by modifying the shape and intensity of the horseshoe vortices at the leading edge, and that combining lean with positive compound sweep shifts blade load upstream and towards midspan, improving static pressure distribution and cavitation behavior. Nevertheless, the practical benefit of blade leaning for secondary flow loss reduction remains limited.

Interestingly, Gregory-Smith *et al.* [208] investigated a combination of reverse compound lean, leading edge extension, and endwall profiling, highlighting the synergistic effect of these structural modifications. By inducing a spanwise pressure gradient, blade leaning accelerates axial velocity toward the endwall, moving the affected fluid closer to it. While this increases secondary flow intensity near the endwall, the overall losses are reduced thanks to endwall contouring, which mitigates the cross-passage pressure gradient despite increasing aft loading and local flow over-turning near the endwall. At the same time, the streamwise shift toward the endwall also lowers midspan velocities, reducing midspan losses; hence, downstream, the yaw angle decreases while tangential velocity

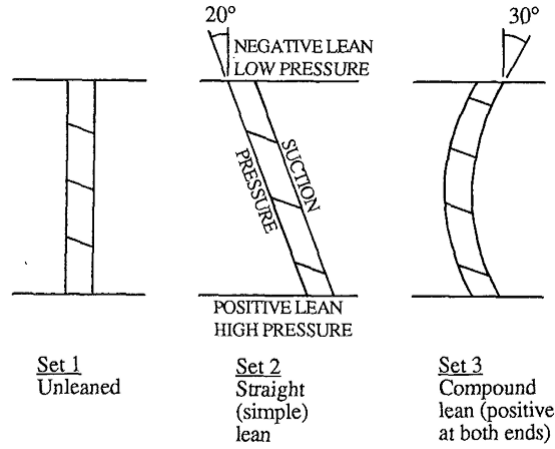


Figure 3.21: Blade leaning techniques [205].

remains constant. However, the authors note that, although an overall loss reduction was achieved for the present cascade, results may vary for other configurations, emphasizing the need for careful optimization.

Luo *et al.* [209] also investigated two additional approaches to reduce secondary flows: re-staggering the blade along the span and a combined stagger and profile modification. By optimally re-staggering the blade, the overall total pressure loss is slightly reduced by decreasing near wall over-turning, while midspan over-turning is increased to maintain the average exit flow angle. This provides a modest but positive improvement in blade row performance, especially for low aspect ratio blades, where the reduction in secondary flow dominates, yielding a net efficiency gain despite a slightly higher profile loss at midspan. In contrast, the combined stagger and profile modification approach adjusts both stagger angle and blade shape. The stagger angle changes account for most of the reduction in secondary flow loss but reduce overall turning. Blade shape modification counteracts this by restoring flow turning and reducing hub loading, further inhibiting secondary flow generation. Hence, the combined effect significantly reduces secondary flow losses with only a minor increase in profile loss.

3.2.9 Blade Snaking

Finally, Giovannini *et al.* [210] proposed a novel turbine blade redesign concept, the “snaked” blade, which is achieved through shape morphing of a conventional blade to reduce secondary flow losses and exhibits a snake-like shape with modifications concentrated near the endwall regions, as illustrated in Figure 3.22. The authors observed that this configuration smooths out the exit swirl distribution and compensates for over- and under-turning associated with secondary flows, resulting in noticeable performance

improvements—even under varying inlet boundary layer thicknesses and corresponding secondary flow intensities.

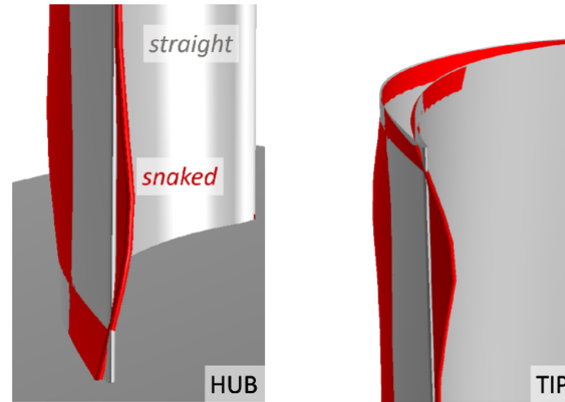


Figure 3.22: Snaked blade compared with the straight configuration [210].

3.2.10 Blade Grooves

Regarding passive devices applicable to blade surfaces, grooves and blade fences are among the most widely used. Grooved surfaces, similar to endwall ridging, consist of ridges aligned with the flow direction and are inspired by riblets, being particularly attractive due to their low production and maintenance costs [16] and their feasibility for large scale application [17]. An example of these is shown in Figure 3.23, where different ridge shapes are presented at the top, and the ridge orientation on the blade surfaces is illustrated in the bottom plot. Among the earliest applications in internal aerodynamics, Lake *et al.* [211] tested a blade cascade with modified blade section surfaces using recessed dimples, recessed V-grooves, and a trip wire to assess their impact on reducing separation losses and consequently secondary flows. Later, Lietmeyer *et al.* [18] examined their application on compressor blades, focusing solely on a modest reduction of profile losses by 1.4% without evaluating secondary flow effects, while Leitl *et al.* [188] numerically evaluated trapezoidal ridges applied to the pressure and suction sides of turbine vane frames, showing that this optimized shape can achieve overall efficiency gains of approximately 0.16%. In any case, grooves on blade surfaces primarily act on the boundary layer, influencing secondary flows only indirectly.

3.2.11 Blade Fences

Instead, blade fences are shelf-like structures mounted on the suction side and/or pressure side to limit secondary vortex penetration along the blade height (see Figure 3.24). They

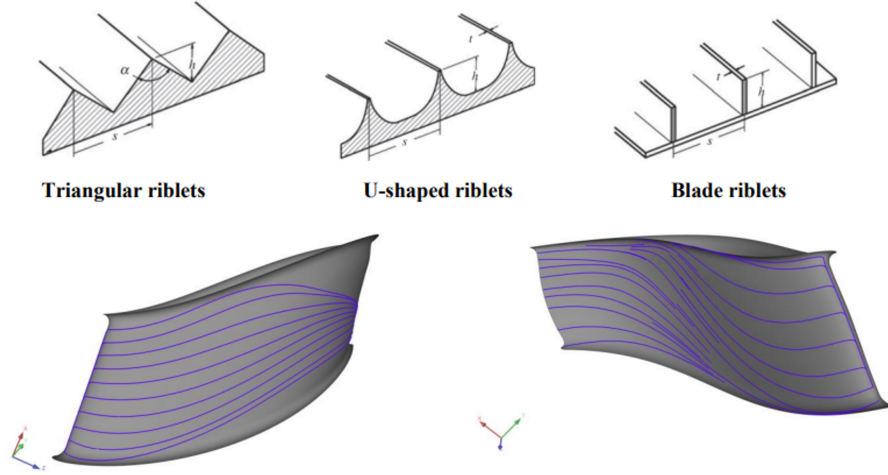


Figure 3.23: Top: different ridge shapes; Bottom: ridge orientation on the blade surfaces [188].

were first studied by Kawai and Adachi [212], who observed that optimal suction side (SS) fences are located at a distance from the endwall equal to the inlet boundary layer thickness, effectively reducing secondary flow and associated losses. At shorter distances, fences primarily minimize vortex size and secondary kinetic energy, while at distances between $2/3$ and $4/3$ of the boundary layer thickness, fences can trap passage and trailing vortices. Specifically, Prümper [12] and Kawai [21] showed that SS installations intercept the passage vortex on the suction side of the adjacent blade, whereas Rubechini *et al.* [22]

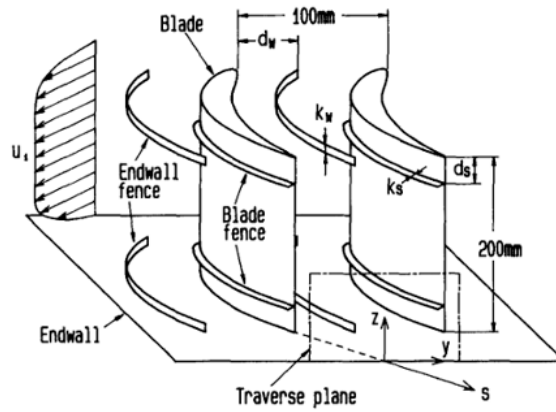


Figure 3.24: Blade fences applied on the blade suction side and endwall fences in the inter-blade passage [21].

highlighted the importance of pressure side (PS) placements, which attenuate the onset of passage and horseshoe vortices, particularly for high lift profiles, as evident in the endwall streamlines shown in Figure 3.25 for a specific fence geometry. The studies by Rubechini *et al.* [22] and Giovannini *et al.* [23] fully elucidated the working mechanism of blade fences, demonstrating that their primary benefit lies in enhancing flow uniformity at the cascade exit and improving downstream row interaction and overall stage efficiency. This effect is achieved by the formation of a counter-rotating vortex above the shelf-like structure, which interacts with the secondary flow. For this reason, Rubechini *et al.* [22] also proposed multi-span fence arrangements, which confine secondary flows near the endwall (see the streamlines visualization for a four-fence geometry configuration in Figure 3.26)

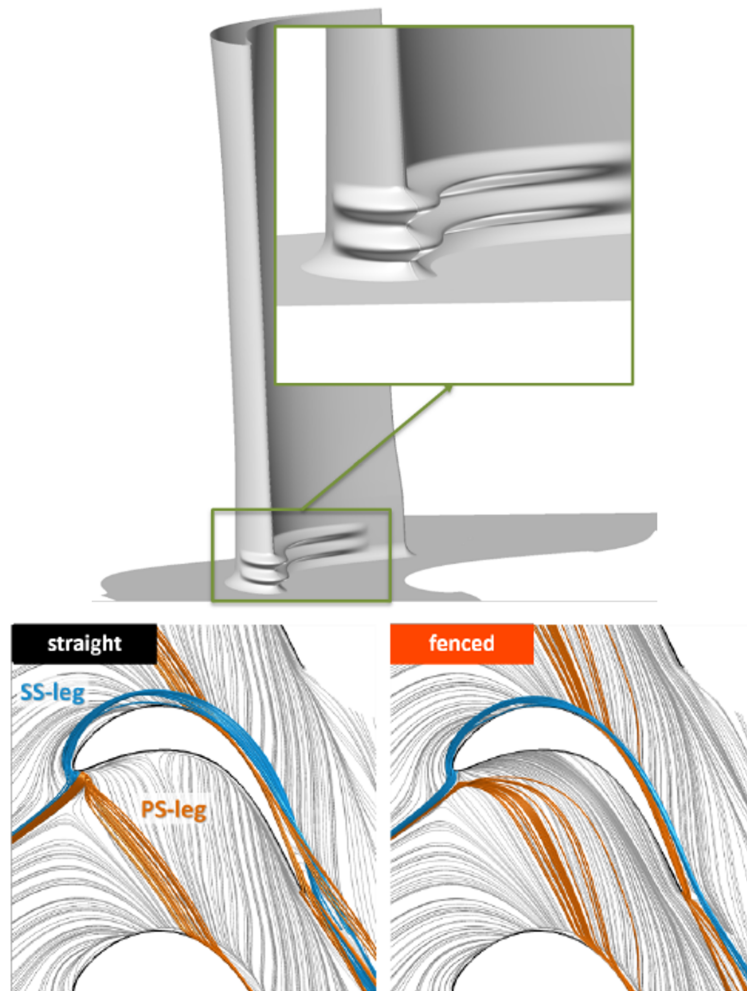


Figure 3.25: Blade fences geometry and its impact on the PS and SS legs of HSV [23].

by generating smaller vortices between fences positioned at different spanwise locations, hence producing a more uniform exit flow and optimized stage performance. Despite their simplicity, blade fences are easy to manufacture and install, making them a practical and effective solution to mitigate secondary flow penetration. These qualities also make them well-suited for combination with other devices; for instance, Kawai [21] investigated the simultaneous use of endwall and blade fences in the same cascade, observing greater loss reductions than with either optimal blade or endwall fences alone, with secondary losses reduced by up to 42%.

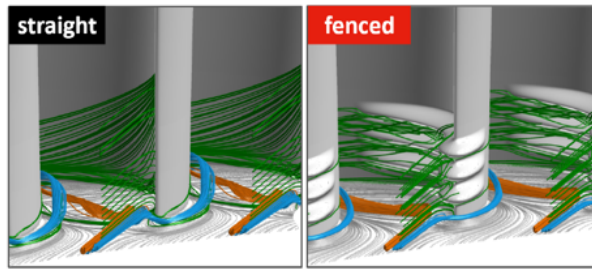


Figure 3.26: Reduction of secondary flow penetration achieved through a multiple-fence arrangement consisting of four fences [22].

3.2.12 Squealer Tip

To counteract the tip leakage vortex developing at the tip of unshrouded blades, the most common approach involves modifying the blade tip geometry or, equivalently, applying a specially shaped tip insert. Bindon and Morphis [26] initially investigated alternative tip shapes, replacing the standard flat tip with a radiused pressure edge (rather than a sharp-edged one) to channel the radial leakage flow, reduce the gap separation bubble, and improve overall blade performance. However, the most widely adopted solutions soon became those involving squealers and winglets [213]. Squealers are raised rims along the blade tip that reduce the leakage mass flow rate [214], while winglets are tangential extensions of the blade edge able to mitigate the leakage vortex strength [215].

Heyes *et al.* [216] first compared suction and pressure side squealers, showing that suction side configurations can decrease tip leakage losses by approximately 3.5%. This finding was later confirmed by Camci *et al.* [217], who demonstrated that partial suction side squealers are more effective than pressure side ones, even in their inclined squealer tip configuration [218]. Furthermore, Ameri [219] combined experimental and numerical analyses to assess the aerodynamic and thermal performance of radiused tips equipped with mean-camberline squealers. More recently, Jung *et al.* [220] investigated both this configuration and a mean-camberline rib embedded within a squealer tip, finding that neither geometry led to a significant reduction in losses. Subsequent studies, however,

demonstrated that orienting the ribs transversely rather than along the camber line inside a full squealer can yield markedly improved effectiveness (as discussed in the following) [221]. Finally, Nho *et al.* [27] tested eleven different blade tip geometries—including conventional plane, squealer, grooved, chamfered, and dimpled tips—and found that double (or full) squealer tips most effectively reduce clearance mass flow, improving cascade efficiency by up to 11% (as later confirmed by Ma and Wang [222]). Their eleven geometries are shown on the left of Figure 3.27, while the corresponding flow effects are presented on the right in terms of total pressure loss coefficient. Moreover, Key and Arts [223] assessed the robustness of the squealer configuration under varying operating conditions, observing its almost independent effectiveness of the Reynolds number. To improve the squealer geometry, Lee and Chae [224] experimentally investigated the effect of rim height on the aerodynamic efficiency, demonstrating that rim height-to-chord ratios above 5% provide optimal aerodynamic benefits—exhibiting a minimum loss behavior, as later confirmed by Jeong and Lee [225]. Furthermore, Lee *et al.* [226] found that a higher

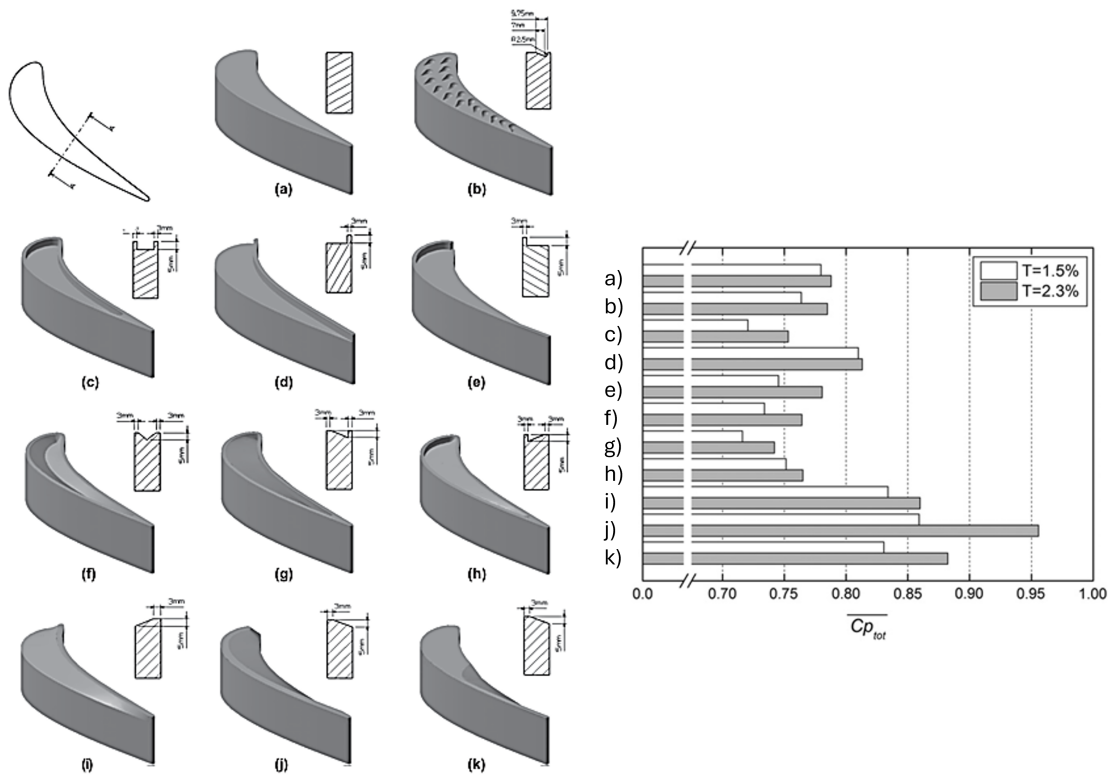


Figure 3.27: On the right: eleven different tip geometries; on the left: the associated benefits in terms of total pressure loss coefficient [27].

pressure side rim configuration, exhibited markedly lower aerodynamic losses than both lower suction side rim and plane tip configurations. Given the importance of squealer height, Duan and He [227] consequently analyzed in detail how rim height generates a blockage effect that reduces tip leakage mass flow and increases aerodynamic efficiency. From an aerothermal standpoint, also Senel *et al.* [228] emphasized that the combination of squealer width and height is crucial to achieve improved performance by simultaneously reducing aerodynamic losses and heat transfer to the tip platform. The enhanced vortical activity within the cavity introduces a blockage effect that reduces the tip leakage mass flow rate. Although leakage generally decreases with increasing squealer height and decreasing width, the relationship remains highly non-linear due to complex interactions in the tip gap region. Moreover, in terms of heat transfer, decreasing squealer width and increasing height significantly reduce the overall Nusselt number on the tip platform, cavity floor, and squealer sidewalls. Instead, Lee and Kim [229], [230], and Krishnababu *et al.* [231] focused on the influence of tip gap height on the aerodynamic performance of cavity squealer tips, while Mischo *et al.* [28] numerically demonstrated that recessed cavity tips experience a reduction in thermal load. Nevertheless, although Newton *et al.* [232] observed that suction side and cavity squealers reduce heat transfer within the tip gap, the extremely high thermal loads experienced in the blade tip region make the application of coolant mass flow essential to ensure component durability—particularly under worn conditions, where squealer tips may lose effectiveness and increase the kinetic loss coefficient by 10–15%, as highlighted by Vass and Arts [233].

For this reason, cooling strategies have long been a focus of several investigations [234]. Early studies by Acharya *et al.* [235] showed that, while on flat tips, film cooling injection lowers the local pressure ratio and modifies the leakage vortex structure, enhancing cooling effectiveness, the internal cavity flow of squealer tips deflects coolant jets, reducing their efficiency. Conversely, Hofer and Arts [236] observed that full squealer tips outperform suction side and flat tip configurations enhancing tip gap resistance, slowing the clearance flow and reducing both mass flow and heat transfer, while suction side squealers without a pressure side rim remain largely insensitive to cooling. Hence, driven by the need for more effective designs, Andreoli *et al.* [237] optimized over 350 tip geometries, identifying Pareto optimal profiles that improve efficiency by up to 0.53% and reduce heat flux by 65%, primarily through more favorable mixing processes rather than direct leakage reduction. In contrast, Duan and He [227] applying a multi-objective genetic algorithm to optimize squealer height and tip cooling concurrently, showed that the injected coolant actively interacts with the tip flow: it can block over tip leakage or form a protective air layer, reducing leakage losses—a mechanism contrary to the conventional assumption that added coolant always increases losses. Moreover, Sakaoglu and Kahveci [36] numerically evaluated twelve cooling configurations, finding that squealer tips with four-seven holes aligned along the mean camber line provide the best compromise between aerodynamic performance and thermal protection. All tested tip and cooling ge-

ometries are reported with their details in Figure 3.28, where the aerodynamic effects of the best tip configuration, both with and without cooling, are also illustrated in terms of velocity contours, compared to uncooled and cooled flat tips. Recently, Bang *et al.* [238] investigated suction and pressure side slots for coolant injection in rotor blade squealer tips, further exploring the interplay between cooling and aerodynamic behavior.

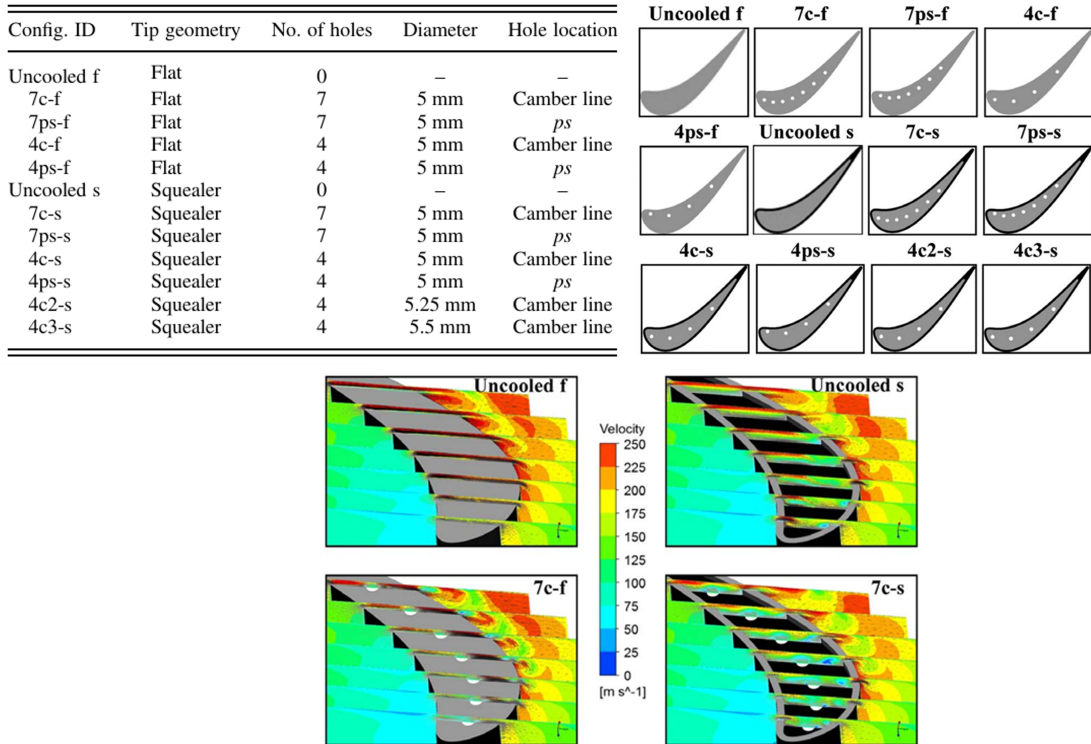


Figure 3.28: Tip and cooling geometries with their specifications, and velocity contours of the optimal tip shape, both without and with cooling, compared to uncooled and cooled flat tips [36].

For completeness, several squealer tip variants developed over time—also thanks to the advent of advanced optimization techniques—are briefly reviewed here. As discussed by Caloni *et al.* [239], such methods can refine the original squealer design, enabling tailored geometries according to specific performance requirements. Lomakin *et al.* [240] investigated full squealers and partial configurations with cutbacks near the trailing edge on either the pressure or suction side. Later, Jiang *et al.* [241] demonstrated that pressure side cutbacks significantly increase aerodynamic losses compared to full-cavity tips, whereas suction side cutbacks exhibit nearly identical overall losses. A further evolution was proposed by Kim *et al.* [242], who introduced a shelf-type squealer tip, obtained by shifting the pressure side rim inward into the cavity and orienting it either perpendicu-

larly or at an inclination. The area-averaged heat transfer coefficients decreased by 17% and 13% for the perpendicular and inclined shelf tips, respectively, compared to the conventional squealer tip. Meanwhile, total pressure loss coefficients increased by 2% and decreased by 3%, respectively—identifying the inclined shelf squealer tip as the most effective configuration for simultaneously reducing both thermal load and total pressure loss. More recently, He *et al.* [243] proposed novel suction side and pressure side modifications, including bulges, partial inclined rims, and cutbacks as visible in Figure 3.29, achieving a 33% reduction in leakage loss and a 0.77% stage efficiency gain relative to a flat tip. Another line of research focused on optimizing the cavity geometry to mitigate

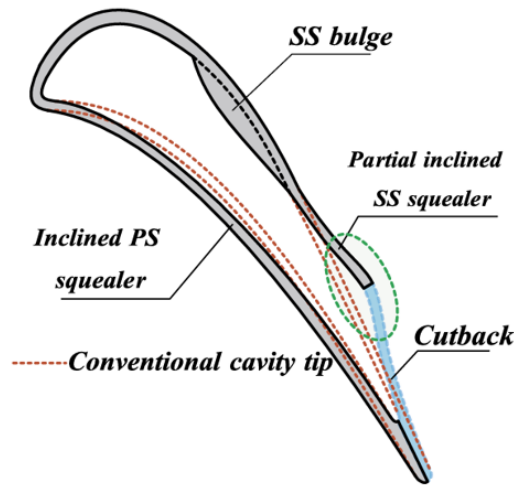


Figure 3.29: Novel suction side and pressure side modifications for blade tip optimization [243].

tip losses by promoting vortical structures that form a labyrinth-like flow pattern within the cavity. These enhance flow resistance, limit leakage, and reduce mixing losses between leakage and main flow, as shown by Zou *et al.* [244]. Similar concepts include the introduction of localized pressure losses via honeycomb surfaces [245] or transverse and streamwise ridges [246]. Following this approach, De Maesschalck *et al.* [247] used topology optimization to create a fully carved blade tip design consisting of multiple fixed height rims. By dividing the tip into over 200 discrete zones—each defined as either rim or cavity—the study demonstrated the potential of such advanced design methodologies to yield entirely new tip configurations with enhanced aero-thermal performance. Park *et al.* [248] experimentally analyzed the effects of rib number and inclination on multi-cavity squealer tips, with the different tested geometries shown in Figure 3.30. Interestingly, the author observed that adding ribs inside the squealer cavity reduced total pressure loss in the cascade, while improvements in heat transfer were limited. Instead, rib inclination was found to have a negligible influence. Consistently, Pátý *et al.* [249] quantified that

multi-cavity tips outperformed single-cavity and flat tips, improving rotor efficiency by 0.15% and 0.3%, respectively. The multi-cavity tip of Jiang *et al.* [250], which resulted in a 4% reduction in total pressure loss, was subsequently optimized from a thermal standpoint. Rib-slot geometries within the multi-cavity configuration achieved up to an 11.3% reduction in the area-averaged heat transfer coefficient. The film-cooling performance of this rib-slot tip was later experimentally validated by Zhang *et al.* [251]. Finally, Bi *et al.* [221] analyzed the combined influence of multiple cavities and coolant injection, showing that a three-rib squealer tip with coolant injection reduced the area-averaged heat transfer coefficient by 19.55% compared to a conventional squealer tip—demonstrating the synergic effect of rib structuring and active cooling.

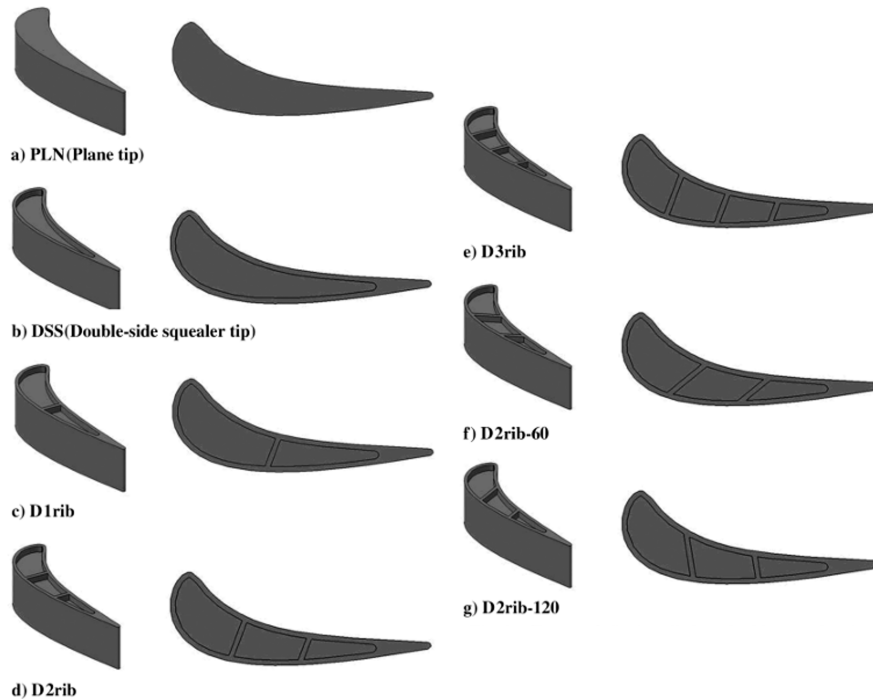


Figure 3.30: Different tip geometries featuring several multi-cavity configurations, varying in rib number and inclination [248].

3.2.13 Winglet Tip

Later, winglet-based configurations (see Figure 3.31) were investigated by Cheon and Lee [252] as a solution able to reduce the driving pressure difference between the pressure and suction sides of the blade, hence mitigating the tip vortex and the associated losses.

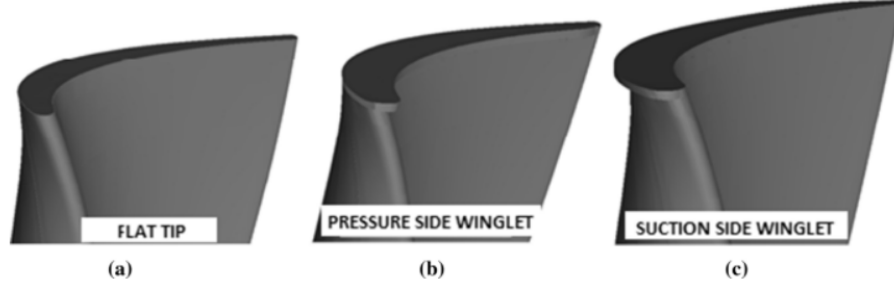


Figure 3.31: (a) plain tip, (b) PS winglet, and (c) SS winglet tip concepts [253].

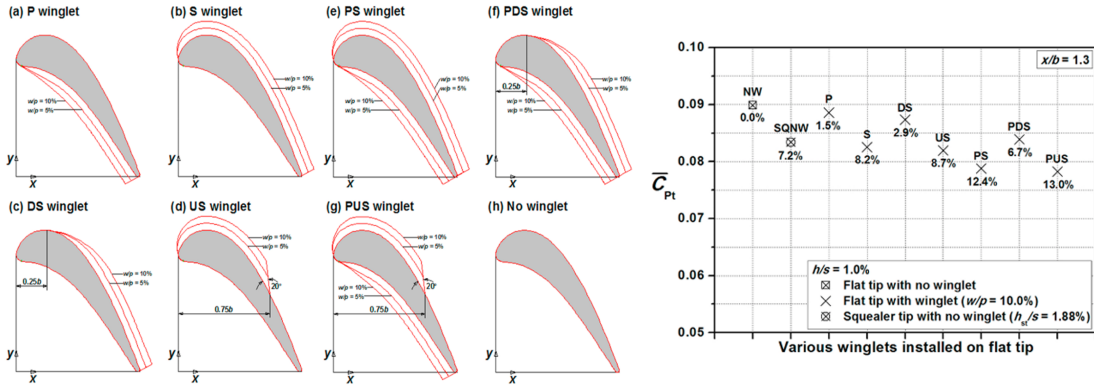


Figure 3.32: Eight different winglet flat tip geometries and their associated losses [29].

Nonetheless, Zhou *et al.* [254] observed that pressure side winglets have little influence on this pressure difference, whereas suction side winglets reduce it only near the leading edge while increasing it from mid chord to the trailing edge, however both still reducing losses compared to the plain tip. These findings resulted consistent with the work of Lee *et al.* [255], who showed that while PS winglets reduce losses near the leakage vortex region, the addition of a leading edge section further decreases losses near the casing. Moreover, Seo and Lee [256] analyzed suction side winglets of constant and varying width, showing that the varying width design reduced aerodynamic losses by 7.4% relative to flat tips and Cheon and Lee [257] found that extended (full-coverage) winglets enhance performance, recommending thin, slanted geometries with minimal outer edge thickness. Only later, Jeong *et al.* [29] confirmed the best design as that covering the pressure side and the upstream suction side, analyzing eight different winglet flat tip geometries shown on the left of Figure 3.32 (the corresponding losses are presented on the right). Finally, Jo *et al.* [258] quantified the effectiveness of this geometry by comparing pressure side winglets with squealer tips, finding that although losses rise beyond 90% of the span, the counter-rotating passage vortex reduces losses in the 70–80% span region, yielding an

overall 28.3% loss reduction relative to the squealer tip. From an aerothermal perspective, Zhou *et al.* [259] investigated the impact of coolant mass flow rate, showing that coolant injection partially blocks tip leakage, reducing downstream flow deviation but slightly increasing leakage loss. Despite a higher overall heat load, the cooled winglet exhibits a significantly lower heat flux per unit area than an uncooled flat tip. Then, O'Dowd *et al.* [260] also confirmed the aerodynamic benefits of coolant injection, showing through detailed kinetic energy loss maps and pitchwise averaged loss coefficients that it effectively shrinks the over tip leakage vortex and its associated loss core. In contrast with the purely aerodynamic effects, Coull *et al.* [261] demonstrated that adding a suction side overhang interestingly weakens the impingement of the leakage vortex on the blade suction surface, lowering both heat transfer and aerodynamic losses by unloading the tip section, reducing leakage flow rate, and aligning it more streamwise—effects most pronounced near the leading edge where the flow is subsonic. More recently, Chen and Li [262] compared steady and sweeping-jet actuators, reporting that a winglet tip combined with the sweeping-jet system achieved the best aerodynamic performance, yielding a 20.1% reduction in total pressure loss—exceeding the 16.2% reduction observed with the steady-jet configuration.

3.2.14 Squealer-Winglet Tip

Then, Ledezma *et al.* [263] and Schabowski *et al.* [264] investigated hybrid tip configurations combining squealer and winglet concepts to achieve both aerodynamic and thermal improvements. A schematic view is shown in Figure 3.33. Earlier, Saha *et al.* [265] re-

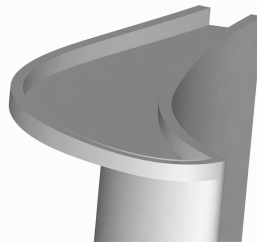


Figure 3.33: Squealer-winglet tip concept [264].

ported that a pressure side winglet locally reduces the heat transfer coefficient by up to 30%, and by about 7% on average compared to a plain tip. However, in the presence of a squealer, the PS winglet effect becomes marginal, yielding only a 5% pressure loss reduction and negligible thermal benefit. Conversely, a suction side squealer combined with a constant-width PS winglet achieved 5.5% lower heat transfer and 26% lower pressure losses than its baseline. Accordingly, Yan *et al.* [266], observing that removing the PS rim from a conventional squealer decreases cascade losses, proposed a full-coverage

PS winglet with a suction side rim, achieving a 12.2% reduction in total pressure loss and 10% in tip heat transfer. Optimizing the leading edge angle and coverage further enhanced these reductions to 13.8% and 15.8%, respectively. Subsequent studies, such as Zhou *et al.* [267], demonstrated that a suction side winglet combined with a full squealer tip provides superior aerodynamic performance compared to its PS counterpart. Similarly, Jiang *et al.* [30] found that in combination with a full-squealer, a suction side winglet reduced losses by 8.5%, while dual-side winglet achieved a 2.5% reduction; conversely, a PS winglet increased losses. Nonetheless, all configurations consistently decreased tip heat transfer by up to 19%. Later, Seo and Lee [268] tested flat, squealer, and winglet configurations (including combinations thereof), confirming the conclusions of the two previous studies, whereas incorporating the suction side winglet with a full-squealer geometry proved essential for achieving the best aerodynamic performance. Some of the tested tip geometries, including the optimized ones, are illustrated in Figure 3.34, along with their corresponding performance. Zhong and Zhou [269] further investigated the role of tip gap size, finding that the squealer–winglet configuration is particularly effective at smaller gaps, reducing tip leakage mass flow and reorienting the leakage jet while mitigating pitchwise pressure gradients and suppressing endwall separation losses.

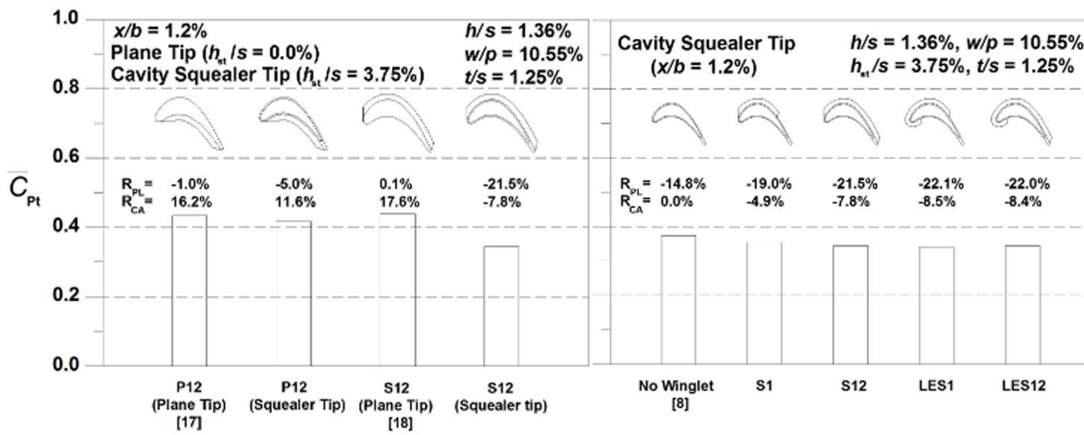


Figure 3.34: Different flat, squealer, and winglet geometries, including combinations of these, and their corresponding performance [268].

From a thermal standpoint, Papa *et al.* [270] showed that the squealer-winglet tip exhibits an extended low heat transfer region near the leading edge and a more uniform, clearance insensitive mass-transfer distribution, in contrast to the conventional squealer tip, where a large recirculating vortex enhances local heat exchange—particularly near the suction side rim—and intensifies with increasing clearance. Cooling studies by Yan *et al.* [271] demonstrated that squealer in combination with pressure side winglet tips provide superior film coverage on the cavity floor, especially near the PS rim, with further

improvements at larger clearances, outperforming conventional squealers. Subsequently, Zhou *et al.* [272] showed that simultaneous blade tip and casing coolant injection further reduces heat load and improves film-cooling effectiveness.

For completeness, additional recent winglet-based variants are also reported. Chen *et al.* [273] investigated three configurations of squealer and squealer-winglet tips, differentiated by the pressure side shaping: a straight rim, an inclined step, and a pressure side winglet. Garg *et al.* [274] introduced hybrid designs in which the squealer rim was either aligned with the blade surface or positioned at the winglet extremities; among these, the latter exhibited the best aerothermal performance, achieving a 15% reduction in total pressure loss and a 22% reduction in average Nusselt number compared to a flat tip, both with and without relative casing motion. Wang and Xuan [275] proposed a synergic suppression method combining blade shape modification with spontaneous injection (see Figure 3.35). By investigating squealer, squealer-winglet, and squealer-winglet-spontaneous

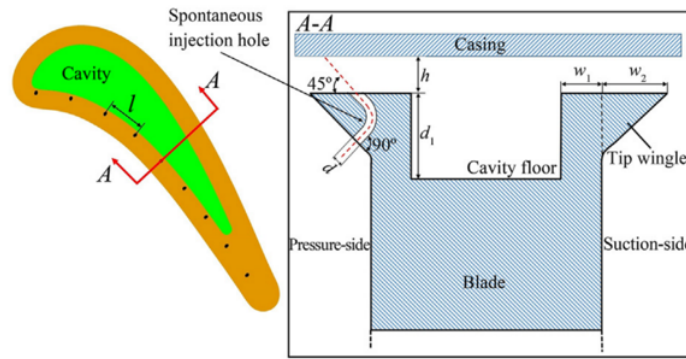


Figure 3.35: Synergetic suppression method combining blade squealer-winglet tip with spontaneous injection [275].

injection hole composite configurations, the authors observed that while the squealer-winglet tip provides a stronger suppression effect on the TLV compared to the simple squealer tip, the addition of spontaneous injection holes further reduces the leakage flow rate and mixing losses thanks to the blocking effect of the injected flow. Finally, Zhang *et al.* [276] proposed an extreme variant, the winglet shroud visible in Figure 3.36, in which a linkage segment connects adjacent blades to form a continuous shrouded passage. The optimized shroud reduced total pressure loss by 16% and tip leakage mass flow by 13%, although further gains may be achieved by accounting for coupled aerothermal effects near the tip.

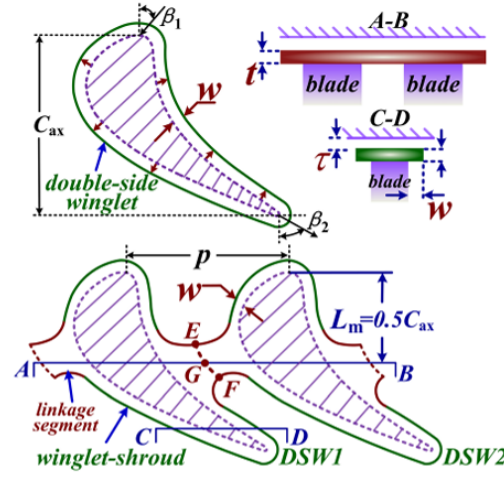


Figure 3.36: Winglet shroud geometry [276].

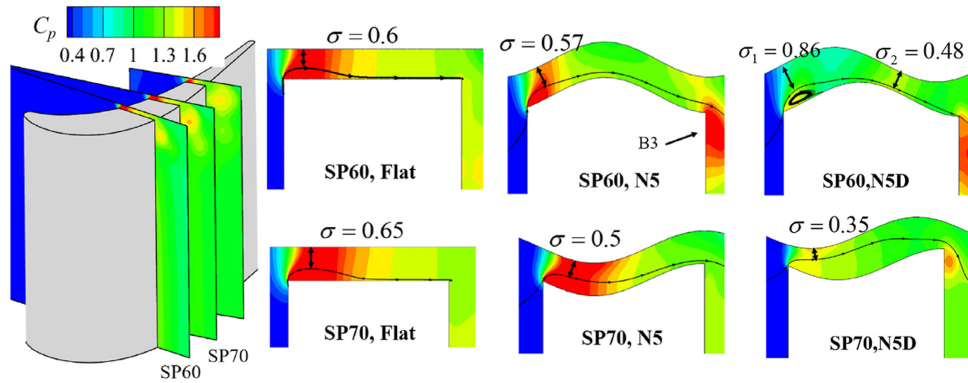


Figure 3.37: Wave-shaped tip geometry [278].

3.2.15 Other Tip Solutions

In addition to squealer and winglet concepts, attempts to actively profile the blade tip surface, though less extensively studied, are worth noting. Kusterer *et al.* [277] first discussed blade tip and endwall contouring, introducing novel shapes with winglets or hooked extensions. Then, Pátý *et al.* [249] tested a fully three-dimensional contoured tip, which proved most effective in reducing blade tip heat transfer, though secondary losses were better mitigated by the squealer–winglet configuration. Finally, Zhou and Zheng [278] proposed wave-shaped tip geometries capable of reducing secondary losses by approximately 3%, such as those illustrated in Figure 3.37.

Many of these approaches for tip shaping and vortex mitigation have been recently

reviewed and experimentally validated by Li *et al.* [279], who systematically tested various tip geometries and coolant ejection strategies in a turbine cascade rig. Overall, the literature highlights the considerable potential of tip shape modeling, offering multiple degrees of freedom for innovative designs and suggesting substantial scope for further optimization.

3.2.16 Summary of Secondary Flow Control Devices

In the following section, Tables 3.1, 3.2, 3.3, and 3.4 provide a comprehensive summary of the secondary flow control and mitigation strategies previously discussed. These tables categorize the devices based on their operational nature and application zones, ranging from active flow devices to passive geometric modifications of the endwalls and blade surfaces.

Table 3.1: Classification and performance impact of active flow control strategies.

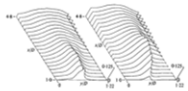
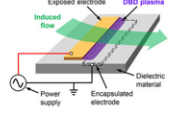
Active Devices			
DEVICE	MECHANISM	APPLICATION	PERFORMANCE
Flow Momentum Injection 	Injection or blowing to convey vorticity away from endwalls, modifying vortex trajectory	On endwall and blade tip surfaces to mitigate all secondary flow structures	$\omega_s \downarrow$ - Tip thermal load uniformity $\uparrow$
Boundary Layer Suction 	Remove vorticity and low-momentum fluid	On endwall and tip surfaces to minimize all secondary vortices	$\omega_s \downarrow$ - SKE $\downarrow$
Streamwise Vorticity Generation 	Injection of vorticity with an opposite sign to counteract detrimental secondary vortices. Passive counterpart: BL management and separation control	On endwall surfaces for reducing all secondary flows Passive: to be used with active ones	$\omega_s \downarrow$ - Flow distortion $\downarrow$
Excitation of Natural Instabilities 	Suppression and modulation of vortex structures through flow instabilities excitation	On blade tip surfaces for TLV control	Vorticity field modifications
Plasma Actuator 	Generation of tangential body forces to re-energize BL and redirect flow near solid surfaces	On endwall and tip surfaces for mitigating all secondary flows	- Vortex velocity and vorticity $\downarrow$ $\omega_s \downarrow$ - Outflow velocity profile homogenization $\uparrow$

Table 3.2: Classification and performance impact of passive flow control strategies applied on endwall surfaces.

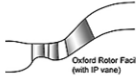
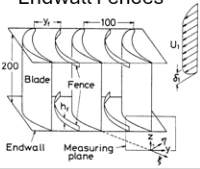
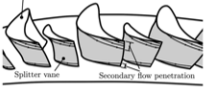
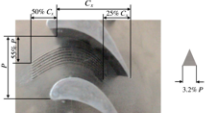
Passive Devices on endwalls			
DEVICE	MECHANISM	APPLICATION	PERFORMANCE
Structural Modifications 	Optimized meridional shapes to reduce BL separation and attenuate secondary vortex development	For PV mitigation	$\omega_s \downarrow$ - No diffusion performance penalties
Endwall Contouring 	3D endwall shaping to redistribute pressure gradients from PS to SS	For reducing all secondary flows	$\omega_s \downarrow$ - SKE $\downarrow$ - Exit flow angle $\downarrow$ - Leakage mass flow $\downarrow$ - Local heat transfer $\downarrow$
Endwall Fences 	Barriers to block cross-flow and inhibit transverse migration of vortices in the channel	For PV control	$\omega_s \downarrow$ - SKE $\downarrow$ - Flow distortion $\downarrow$
Endwall Splitter 	Partial blade to limit the propagation of secondary vortices	For PV reduction	$\omega_s \downarrow$ - Stage efficiency $\uparrow$
Endwall Ridging 	Grooved endwall surfaces to counteract the transverse pressure gradient	For the mitigation of all secondary flows	- Vortex lift up $\downarrow$ $\omega_s \downarrow$

Table 3.3: Classification and performance impact of passive flow control strategies applied on blade surfaces.

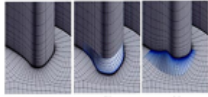
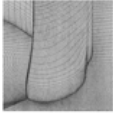
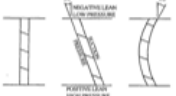
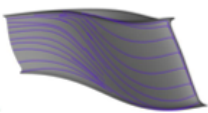
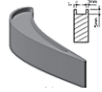
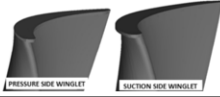
Passive Devices on blades			
DEVICE	MECHANISM	APPLICATION	PERFORMANCE
Blade Fillet 	Fillet radius between blade LE and endwall to streamline the flow and mitigate corner vortex formation	For HSV mitigation and delay of PV formation	• Potentially $\omega_s \downarrow$ • Blade turning $\downarrow$
Blade LE-bulb 	Intensify SS-leg of HSV to counteract PV	For PV suppression	• $\omega_s \downarrow$ • Blade turning $\uparrow$
Blade Leaning and Re-Staggering 	Redistribute the load along the blade surface to create a radial component to counteract the cross-flow	For HSV and PV reduction across the span	• $\omega_s \downarrow$ • Potentially $\omega_p \uparrow$ • Exit yaw angle deviation $\downarrow$
Blade Snaking 	Blade shaping to compensate for secondary flow over- and under-turning	For PV mitigation	• $\omega_s \downarrow$ • Exit flow uniformity $\uparrow$
Blade grooves 	Ridges on blade surfaces to energize BL and indirectly influence secondary flows	For HSV and PV control	• $\omega_p \downarrow$ • $\omega_s \downarrow$
Blade Fences 	Shelf-like structure to limit the penetration of secondary vortex on the blade height	For HSV and PV control	• $\omega_s \downarrow$ • Exit flow uniformity $\uparrow$

Table 3.4: Classification and performance impact of passive flow control strategies applied on tip blade surfaces.

Passive Devices on blade tips			
DEVICE	MECHANISM	APPLICATION	PERFORMANCE
Squealer Tip 	Raised rims along the blade tip to induce a blockage effect in the tip gap to reduce leakage mass flow rate	For TLV mitigation	• $\omega_{tip} \downarrow$ • Leakage mass flow rate $\downarrow$ • Heat transfer $\downarrow$
Winglet Tip  PRESSURE SIDE WINGLET SUCTION SIDE WINGLET	Tangential extensions of the blade edge able to mitigate the leakage vortex strength	For TLV reduction and control	• $\omega_{tip} \downarrow$ • Heat transfer $\downarrow$
Squealer-Winglet Tip 	Hybrid geometry combining the blockage effect of the squealer rims with the vortex-weakening properties of the winglet	For TLV mitigation	• $\omega_{tip} \downarrow$ • Leakage mass flow rate $\downarrow$ • Optimized heat transfer distribution

Chapter 4

Experimental Campaigns: Setup and Cascade Configurations

Within the aim of this doctoral research, two different turbine cascades were investigated to study various active and passive strategies for the control and mitigation of secondary flows. Both cascades are derived from next-generation lightweight, compact, and high specific thrust aero-engine configurations.

The first cascade is an NGV designed to replace the intermediate turbine diffuser and featuring low aspect ratio vanes that guide the flow directly from the smaller HPT radius to the larger LPT radius. Moreover, to achieve the necessary pressure recovery while maintaining a reduced axial length of the turbomachinery shaft, the NGVs are installed within short, aggressive diffuser channels, as identified by Sovran [2], resulting in highly diffusive blade passages, particularly near the tip region. Under these conditions, pronounced secondary flows develop, with a particularly strong passage vortex, whereas the horseshoe vortex is relatively weak due to the lightly loaded front portion of the vanes. To confine and mitigate these secondary flows, two passive approaches were applied in this cascade configuration: fences and endwall ridging. Two variants of the latter were investigated: one derived from the literature and also tested in combination with the fences, and a second variant obtained through an optimization process conducted on the cascade itself.

The second cascade is a highly loaded LPT blade row designed to meet the lightweight and compact requirements of modern aero-engines. It features fully unshrouded blades, where the tip clearance required to prevent rubbing between the blade row and the casing induces a strong tip leakage vortex. In this cascade, the TLV was mitigated using squealer and squealer-winglet tip geometries. In addition to tip surface modifications, various coolant flows were employed.

Both cascades were faithfully reproduced in large scale linear models to be tested in the wind tunnels of the Aerodynamics and Turbomachinery Laboratory at the University of Genova. Specifically, the NGV cascade was tested in the wind tunnel located at

the Genova site, while the unshrouded LPT cascade was tested in the Savona laboratory facility.

4.1 Test Rigs

The two low-speed wind tunnels of the University of Genova used for the present campaign are shown in Figures 4.1 and 4.2. Specifically, the wind tunnel used to test the NGV cascade at the Genova site is shown in Figure 4.1, while the facility adopted for the unshrouded LPT cascade tests at Savona laboratory is shown in Figure 4.2. Both wind tunnels are open-loop, blow-down facilities for low-speed testing of turbine cascades. These facilities are designed to operate under Reynolds number similarity with respect to actual engine conditions, thereby reproducing the same viscous effects while taking advantage of low-speed testing. Lower velocities and larger spatial scales allow higher quality measurements, improving both temporal and spatial resolution. The Reynolds number in these campaigns is defined based on the axial chord of the blades, C_x , and the isentropic exit velocity from the cascade, $v_{2,is}$, as

$$Re_x = \frac{C_x v_{2,is}}{\nu}, \quad (4.1)$$

where the kinematic viscosity is set to $\nu = 1.5 \cdot 10^{-5} m^2/s$. Since it is not possible to simultaneously match Mach number similarity, compressibility effects—although limited, particularly in LPTs—are mitigated by redesigning the three-dimensional low-pressure turbine blades to scale them from high-speed engine conditions to large-scale, low-speed test environments [74], [280]. According to the scaling guidelines proposed by Giovannini *et al.* [281], the core of this methodology lies in matching the three-dimensional blade loading distributions across the entire span. This approach ensures comparable boundary layer development and loss behavior, despite the inherently incompressible nature of low-speed experimental facilities. Furthermore, Vázquez and Torre [282] demonstrated that this procedure remains valid even for high Mach number operations, up to the onset of strong shocks ($Ma_2 \approx 0.9$). To achieve this scaling, an advanced multi-step optimization framework based on 3D RANS computations is required. This framework leverages artificial neural networks to navigate a complex design space exceeding several degrees of freedom, including 3D stacking, camber line parameters, and thickness distributions. In the present work, while the meridional channel divergence of the NGV cascade is rigorously accounted for (particularly in the region between the midspan and the casing) the unshrouded LPT cascade has been simplified: the redesign focused exclusively on the target tip section, which was then extruded linearly along the span to generate a prismatic blade cascade geometry.

Both facilities share the same basic architecture, schematically illustrated in Figure 4.3 (see [283] for further details). The wind tunnels are powered by a centrifugal fan driven by



Figure 4.1: Genova wind tunnel.



Figure 4.2: Savona wind tunnel.

a three phase asynchronous motor controlled via an inverter, allowing precise regulation of the mass flow throughout the tunnel and, consequently, of the Reynolds number in the test section. An intake bell is installed at the fan inlet, carefully shaped to prevent flow separation, and equipped with a filter to protect the instrumentation and ensure reliable measurements by preventing the ingress of environmental impurities. Downstream of the

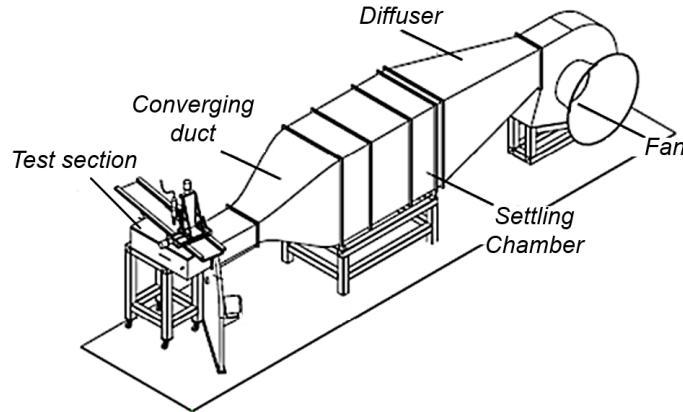


Figure 4.3: Wind tunnel architecture.

fan, which introduces turbulence due to the interaction of air with the fan blades, a diffuser duct is installed to reduce flow velocity and minimize total pressure losses, while avoiding local flow separation. Its design follows the guidelines of Sovran [2]. To further prevent separation and improve flow uniformity at the diffuser exit, a fine-mesh screen is installed. The flow then enters the constant-section settling chamber, designed to laminarize and homogenize the flow along the tunnel axis and to ensure isotropic turbulence. Three fine-mesh metallic screens are used to break large scale turbulent structures into smaller vortices, producing a more uniform velocity profile. Since the screens can deflect the main flow, they are combined with an axial honeycomb, a metallic structure with hexagonal cells of 10 mm side length, extending approximately 50 mm along the tunnel axis, which channels the flow and reduces transverse velocity components, thereby aligning the flow with the main direction. The low flow velocity in this maximum section zone minimizes losses. At the exit, the flow is accelerated through a convergent section, promoting a uniform velocity profile with thin endwall boundary layers. This is followed by a short constant section equipped with a turbulence generating grid, which introduces a controlled and uniform turbulence level at the test section inlet. The grid position and bar dimension (d) are carefully selected to regulate vortex scale and intensity, ensuring a turbulence level consistent with the target experimental conditions and a sufficiently homogeneous and isotropic flow downstream. For both test rigs, a square mesh grid oriented perpendicular to the flow is installed to achieve a turbulence intensity (Tu), defined as the root mean square of the velocity fluctuations normalized by the mean flow velocity, typical of fully developed turbulent conditions at the test section inlet. Without the grid, the tunnel exit turbulence would be approximately $Tu = 0.1 - 0.2\%$. The turbulence values achieved with the installed grids, along with other construction specifications of the wind tunnels, are summarized in Table 4.1. The flow downstream of the turbulence grid then enters the blade cascade. In both wind tunnels, the cascade consists of a linear cascade of seven

prismatic blades representative of the real profiles, machined from aluminum using 5-axis milling and coated with a white filler paint to closely reproduce the surface roughness of the actual blades. Since the flow is periodic in the real turbine operation, the linear cascades with a finite number of blades are equipped with adjustable bleeds and tailboards to enforce periodicity. The bleeds are adjustable openings located upstream of the cascade that allow control of the flow incidence, while the tailboards are hinged panels at the lateral blade exit that regulate the flow leaving the cascade. Before measurements, bleeds and tailboards are set to achieve optimal superposition of the loading distributions on the three central blades, which are instrumented with static pressure taps on the blade surfaces for this purpose. Both facilities are also equipped with systems to adjust the test section incidence and generate upstream wakes to reproduce unsteady effects caused by an upstream rotor. In the present experimental campaigns, however, all tests were conducted at fixed incidence and stationary conditions to ensure maximum reproducibility of the boundary conditions (BCs) across the various cascade configurations tested (i.e., before and after the application of fences, ridges, or tip modifications). Different Reynolds numbers were tested to characterize the influence of the devices on secondary flows under varying operating conditions.

4.1.1 Genova Laboratory Test Section

The test section of the wind tunnel installed in Genova laboratory is designed to house the large scale linear cascade replicating the NGV integrated within the intermediate turbine diffuser of a lightweight, compact engine. As shown in Figure 4.4, these vanes are long chord blades that simultaneously fulfill aerodynamic and structural functions, since they act also as struts within the S-shaped duct.

They feature a low aspect ratio, $AR = \frac{\bar{h}}{C_x} = 1.0$ (based on the average blade height), and a low solidity, $\sigma = \frac{C_x}{s} = 0.85$. The vanes also exhibit low turning angle ($\theta = 55^\circ$), and Zweifel number ($Zw = \frac{2s}{C_x} \cdot \cos(\alpha_2)^2 \cdot (\tan(\alpha_2) - \tan(\alpha_1)) = 0.7$). The NGV cascade

Table 4.1: Specifications of the Genova and Savona wind tunnels.

Component	Genova	Savona
Fan max flow	8 m ³ /s at 4800 Pa	6 m ³ /s at 2500 Pa
Engine power	200 kW	20 kW
Diffuser area ratio	$\frac{1}{6}$	$\frac{1}{3}$
Settling chamber length	2 m	5 m
Converging duct area ratio	6	4
Turbulence Tu	3.5%	5.5%
Grid bar width d	5.8 mm	5.8 mm

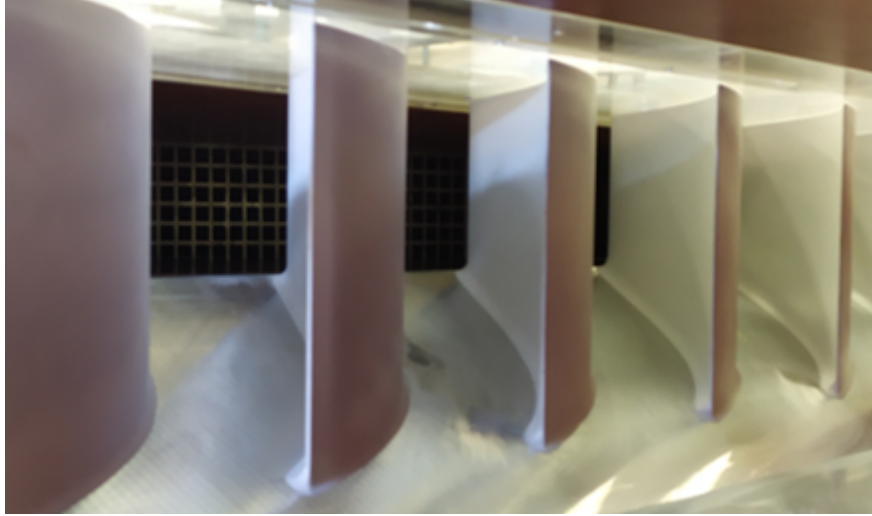


Figure 4.4: NGV cascade installed in the Genova test section.

is located within a diffusive meridional channel, characterized by an area ratio $\frac{A_2}{A_1} - 1 = 1$ and a characteristic duct length normalized by the inlet duct height equal to $L/h_1 = 1.8$. The duct features flat endwalls up to $5\% \cdot C_x$ upstream of the blade leading edge, followed by a combination of flat and divergent endwalls extending to $30\% \cdot C_x$ downstream of the trailing edge, ensuring aggressive diffusion from the HPT outlet to the LPT inlet (see Figure 4.5.b). This configuration results in a symmetric inlet channel and an asymmetric outlet, with the blade tips positioned at the bottom of the test section, aligned with the divergent endwall for technical reasons. Overall, the meridional channel shape provides a linear connection between the upstream and downstream regions of the NGV, faithfully reproducing the real geometry of an engine intermediate turbine diffuser.

The lower part of the test section is made of aluminum and consists of a horizontal plate and an inclined ramp at approximately 25° relative to the plate (see Figure 4.4). The ramp is equipped with a series of pre-drilled holes to mount the blades using screws, allowing precise and repeatable positioning and ensuring accurate alignment of the blades within the cascade. Downstream of the blade trailing edges, on the horizontal plate, a housing accommodates an aluminum plate equipped with static pressure taps, used to measure the static pressure downstream of the cascade, p_2 . This measurement is adopted both prior to the experimental campaign to correctly set the position of the tailboards and during the tests to accurately evaluate the Reynolds number for setting the facility to specific operating conditions. The positions of the bleeds and tailboards in this test section are highlighted in Figure 4.5, where the schematic (a) top and (b) meridional views of the section are presented. These adjustable partitions are essential to achieve periodicity, which is verified through the aerodynamic loading measurement, c_p , via multiple static pressure taps located at midspan and near the divergent endwall at 80% of the blade height

on both suction and pressure surfaces of the three central blades.

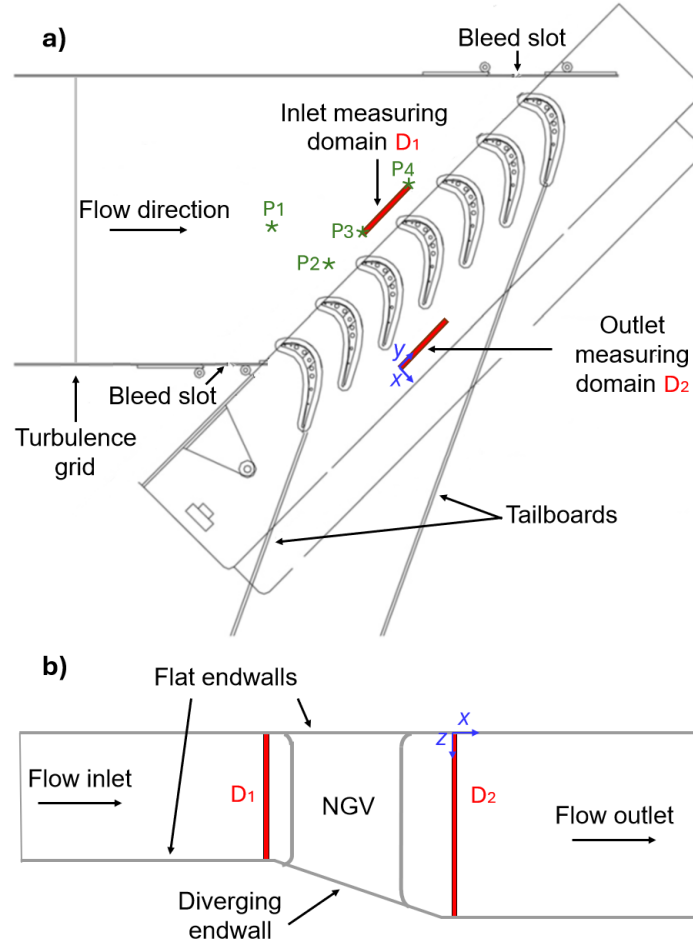


Figure 4.5: Schematic (a) top and (b) meridional views of the Genova test section.

The upper plate is made of plexiglass to provide optical access to the test section and measurement instrumentation during the experiments. The plexiglass features two tangential traverses, designed to allow the insertion and movement of pneumatic probes used during the measurement campaigns (see Figure 4.5). Specifically, the inlet domain D_1 is located at a distance of $0.13 \cdot C_x$ upstream of the blade leading edge, while the D_2 domain is positioned $0.58 \cdot C_x$ downstream of the blade trailing edge. The upstream position was chosen to be as close as possible to the blade leading edges for an accurate characterization of the inflow, while remaining sufficiently far to avoid potential flow effects. Conversely, the downstream location was selected to minimize probe intrusiveness near the trailing edge and to allow for the adequate development of secondary flows and mixing losses as the wakes merge with the freestream.

A hot-wire anemometer was installed in the upstream traverse for the detailed characterization of the 2D, time-resolved velocity and turbulence properties of the flow field at the cascade inlet. Moreover, positioning the probe at midspan at different locations (indicated by *) along the streamwise direction x_t downstream of the turbulence grid, allowed the characterization of the turbulence decay Tu , providing accurate boundary conditions for the 3D numerical simulations. Specifically, P_1 , P_2 , P_3 , and P_4 are located downstream of the turbulence grid at $x_t/C_x = 1.89, 2.61, 2.9$, and 3.19 , respectively.

Additionally, in both traverses, a Kiel probe and a five-hole probe were used to measure the total pressure p_t and velocity v fields entering and exiting the NGV aggressive duct. This enabled a detailed characterization of the upstream boundary layer and the secondary flow structures at the cascade exit plane. Due to its high accuracy, the Kiel probe was used to quantitatively evaluate the aerodynamic performance of the cascade via the total pressure loss coefficient C_{pt} and the integral loss ω , while the five-hole probe was used to provide qualitative information on the flow field upstream and, in particular, downstream of the cascade, allowing analysis of secondary flow structures developing within the cascade.

Measurements in both domains were performed on a 30×34 grid, with the probes traversing along the pitchwise (y) and spanwise (z) directions using a numerically controlled two-dimensional traversing system, which ensures precise movement both tangentially and along the blade height, with a minimum step of $8\mu m$. Moreover, the probes are equipped with a probe holder fitted with a goniometer, enabling rotation about their axis for proper alignment with the investigated flow direction.

The instrumentation used for both test sections will be described in detail in the following section.

4.1.1.1 NGV Configurations

The experimental campaign began with the baseline cascade representative of the NGV of interest (see Figure 4.4), without any passive devices applied to counteract secondary flows. This configuration hence serves as a reference to evaluate the effectiveness of the selected passive devices in mitigating secondary flows. These devices were designed as add-on elements, manufactured using high-resolution 3D printing technology, and mounted directly on the existing cascade to preserve the original setup and boundary conditions of the experiment.

The first set of modifications included a dual-fence layout, optimized for the case under investigation, installed on the blade surface. Next, a classical endwall ridging, derived from the literature, was positioned on the diverging endwall. The third configuration consisted of a combined setup, featuring both fences and the conventional ridge layout. Finally, the effectiveness of an optimized ridges configuration was evaluated.

4.1.1.2 Fences

The design of the fences was obtained through a comprehensive Computational Fluid Dynamics (CFD) based parametric design campaign carried out by Morfo Design to evaluate the ability of fences to mitigate the complex secondary flow structures characteristic of the baseline NGV, with particular focus on regions where the flow field is more challenging due to the high level of endwall diffusion. Multiple fenced configurations were analyzed using RANS calculations with STAR-CCM+ as the flow solver, simulating a stage environment to establish an extensive database of CFD solutions. See Appendix 2 for further details on the numerical setup. The parameterization strategy for advanced geometry management followed the methodology outlined by Giovannini *et al.* [23], and the capability of the design tools to generate a wide variety of fence types on both suction and pressure sides, as well as around the leading edge, is shown in Figure 4.6.



Figure 4.6: Example of blade fences in a 3D annular NGV.

In the present design campaign, the parameters defining the fence geometry that were subject to optimization included both their shape in the planform plane (start and end positions, and maximum thickness) and their distribution in the spanwise plane (number of fences and start position along the span). The design space was sampled by generating geometries with varying parameter values and evaluating their performance through RANS calculations. Subsequently, Artificial Neural Networks (ANNs) were trained on this CFD database to construct a response surface. The response surface was then used to systematically explore the design space, and identify the optimal configuration. Stage efficiency was selected as the objective function, while ensuring preservation of the stage operating point (mass flow rate). This approach allows a direct assessment of the beneficial carry-over effect on the downstream rotor, enabling a comprehensive evaluation of overall performance improvements. Thus, rather than focusing solely on the performance of the NGV where the fences are applied—which does not necessarily require improvement—this method captures the broader impact on the entire stage. As mentioned earlier, the final solution resulting from this optimization process differs from those pre-

viously reported in the literature, reflecting the specific features of this application, where the secondary flow structures deviate significantly from those observed in conventional LPT blades. Specifically, the current solution features a double fence originating on the pressure side, wrapping around the leading edge, and extending along the suction side, following the divergence of the meridional channel and gradually tapering toward the trailing edge. Hence, the present study complements existing literature by demonstrating how a double blade fence, positioned on the suction side, can reduce the penetration of a large passage vortex. The fence characteristics are reported in Table 4.2 [284]. Once the

Table 4.2: Fence characteristics.

Description	Value
<i>Position on planform plane (from LE)</i>	
Axial starting position on the PS	$\sim 0.3 C_x$
Axial position of max thickness	$\sim 0.65 C_x$
Axial ending position on the SS	$\sim 1.0 C_x$
<i>Position on spanwise plane (from hub)</i>	
Spanwise location	75%-95% h
Lower fence max thickness	$\sim 90\% h$
Upper fence max thickness	$\sim 80\% h$

fence design was finalized and 3D-printed, the device was experimentally validated by installing it on the blade surfaces of the baseline NGV, thereby enabling the investigation of the modified NGV cascade, shown in Figure 4.7.

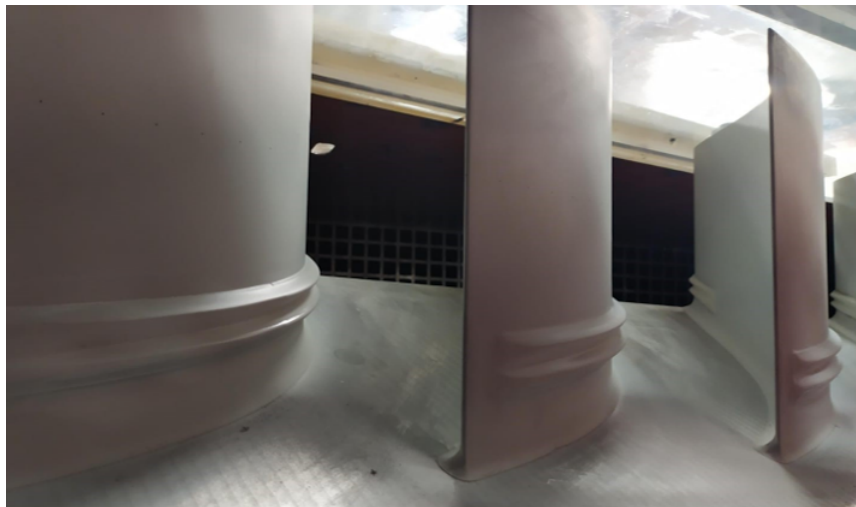


Figure 4.7: NGV cascade with fences.

Figure 4.8 shows a close-up view of the fences installed on the cascade.

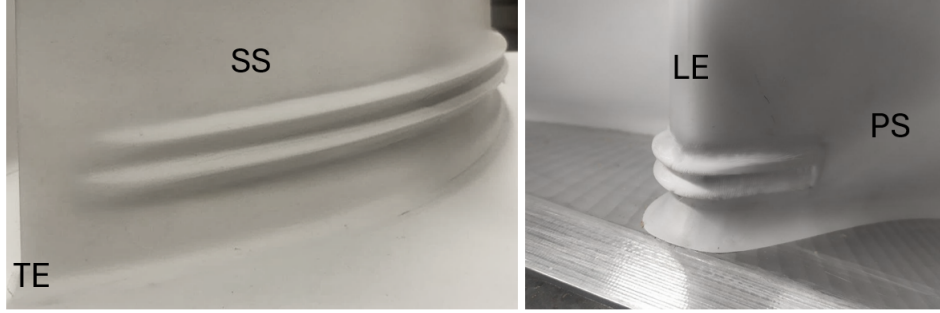


Figure 4.8: Close-up views of the fences on the NGVs.

4.1.1.3 Conventional Endwall Ridging

Subsequently, classical ridges, derived from the literature, were used as a proof of concept to evaluate the general effectiveness of the EWR concept. No parametric optimization was performed, and the literature based layout consisted of six trapezoidal grooves aligned with the main streamline direction. Their specific characteristics are summarized in Table 4.3 [285].

Table 4.3: Conventional EWR features.

Description	Value
Axial starting position from LE	$\sim 0.15 C_x$
Axial ending position from LE	$\sim 1.04 C_x$
Number of ridges	6
Height of ridges	$1.6\% h$
Width of ridges	$10\% s$

Similarly, once designed in CAD and exported for 3D printing, the ridges were applied to the divergent endwall of the baseline cascade, as shown in Figure 4.9.

A close-up view of the ridges, including their tangential section, is shown in Figure 4.10.

4.1.1.4 Fences and Conventional Endwall Ridging

Then, a combined setup, featuring both fences and conventional ridge layout while maintaining the geometric characteristics reported in Tabs. 4.2 and 4.3, was tested to explore potential synergies between the two passive flow control strategies to mitigate secondary flows. The NGV in this third configuration is shown in Figure 4.11.

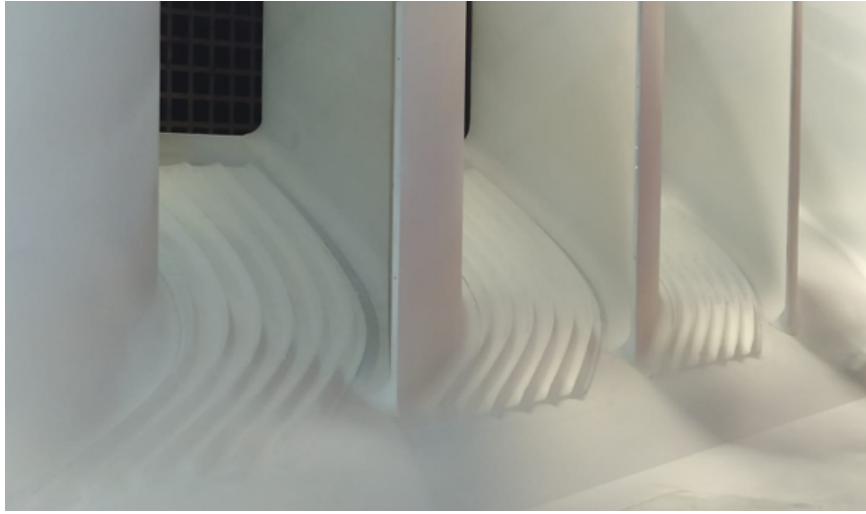


Figure 4.9: NGV cascade with endwall ridging.



Figure 4.10: (a) Close-up view and (b) tangential section of the ridges on the NGV cascade endwall.

4.1.1.5 Optimized Endwall Ridging

In this case, the optimized endwall ridging design was obtained through a CFD-based optimization procedure carried out by Morfo Design. Starting from the conventional ridged configuration, the EWR optimization was performed under realistic conditions within an annular stage environment. The process employed a parametrization framework combined with CFD simulations, consisting of successive phases: representation of parametric geometries through an automated CAD generation tool, definition of the design space identifying key parameters and their variation ranges, and generation of an extensive CFD database of candidate EWR geometries leveraging High Performance Computing (HPC) to ensure both flexibility and computational efficiency. In particular, as shown in Figure

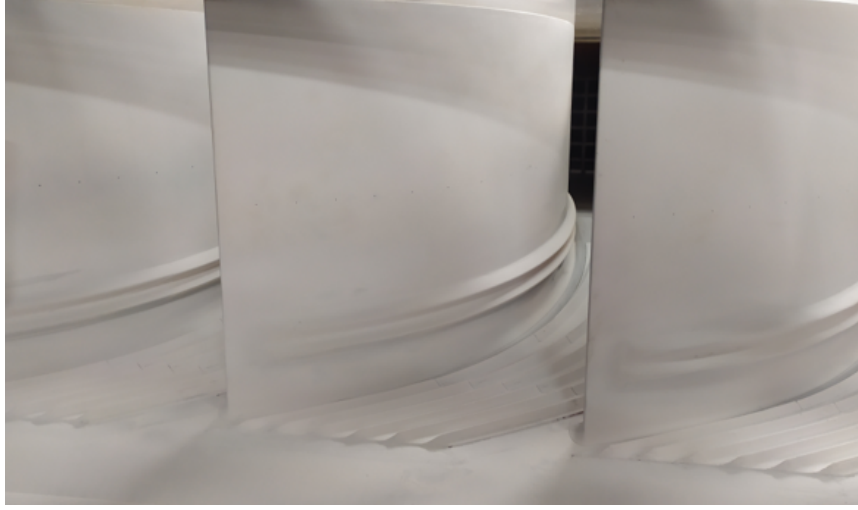


Figure 4.11: NGV cascade with both blade fences and conventional endwall ridging.

4.12, the EWR shape was constrained to follow the streamlines of the undisturbed flow, while the number of peaks, their height and width, as well as their start and end positions, were selected as optimization parameters and systematically varied.

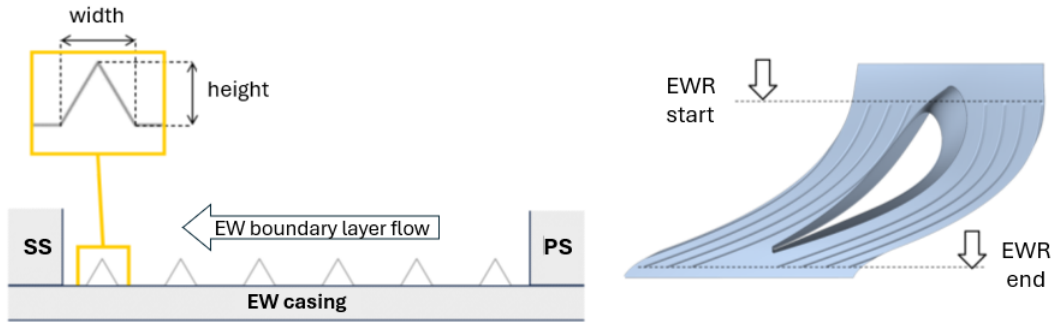


Figure 4.12: EWR geometry parameterization.

Then, the optimization was carried out in two steps, as illustrated in Figure 4.13. The first step consisted of a coarse sampling of the design space to explore a wide range of configurations and generate data for an initial response surface. Artificial Neural Networks trained on this dataset were then employed to provide a comprehensive understanding of the design landscape, enabling the redefinition of constraints and objective functions and the selection of second-step candidate geometries within the most promising region in terms of improved stage efficiency and reduced flow angle non-uniformity downstream of the stator. A total of 500 geometries were subsequently evaluated through stage RANS CFD simulations, requiring approximately 200 core hours. The results were then used to

retrain the ANNs and identify the optimal solutions in terms of stage efficiency, which effectively balances the competing objectives of maximizing flow uniformity at the NGV exit plane while minimizing associated loss penalties, all while preserving the stage operating point (mass flow rate). The computational results clearly depict the Pareto front between stage efficiency and the variance of the flow angle at the NGV discharge, capturing the impact of secondary flows.

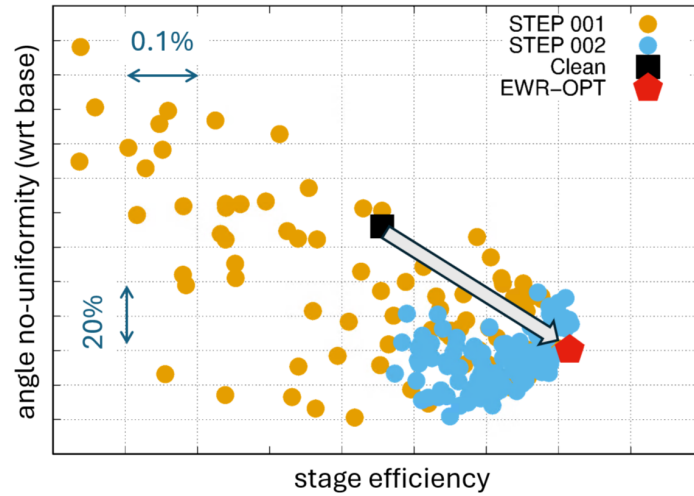


Figure 4.13: EWR optimization.

The analysis revealed that ridge height is by far the most critical parameter affecting performance, playing a dominant role in determining the overall aerodynamic efficiency of the stage and highlighting its importance in the design process. The optimal ridge height was found to be approximately 1.5% of the NGV axial chord, ensuring effective interaction with the secondary flows, enhancing stage efficiency while minimizing adverse flow phenomena. In contrast, the EWR start and end positions showed no significant impact on performance, provided that, as in the present optimization campaign, they cover the rear portion of the NGV passage. This arrangement allows the ridges to effectively interact with the passage vortex where its influence is most pronounced, while minimizing sensitivity to small variations in placement. The results further indicate that the optimal number of ridges falls within the range of 6 to 8, balancing sufficient flow control with avoidance of excessive blockage. A mean value of 7 ridges was therefore selected. All optimized EWR features are summarized in Table 4.4 [285].

Then, the optimized trapezoidal grooves were experimentally validated. They were fabricated using 3D printing technology and installed on the diverging endwall of the existing cascade, as shown in Figure 4.14.

In addition, Figure 4.15 presents a (a) close-up view of the device installed in the wind tunnel and (b) the tangential section of the optimized ridge geometries (scaled to

Table 4.4: Optimized EWR features.

Description	Value
Axial starting position from LE	$\sim 0 C_x$
Axial ending position from LE	$\sim 1.12 C_x$
Number of ridges	7
Height of ridges	$0.8\% h$
Fillet radii	$0.4\% h$
Width of ridges	$12\% s$

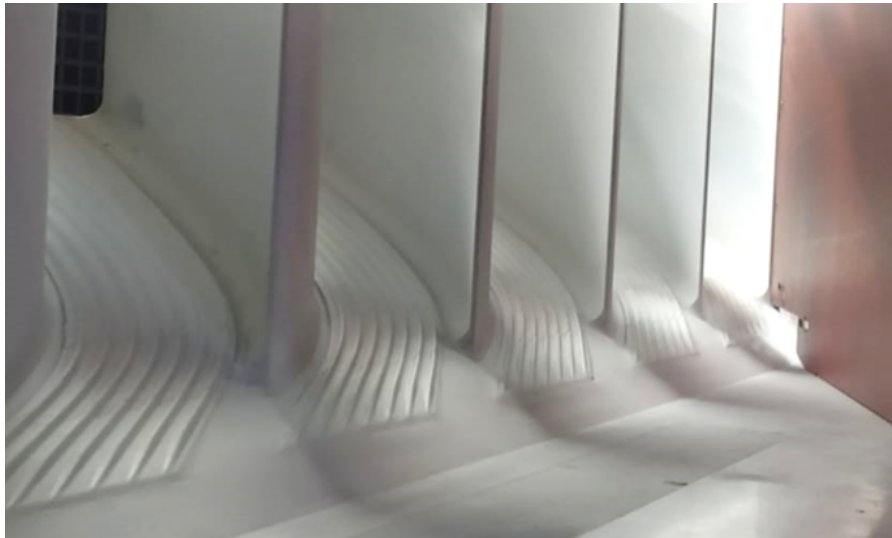


Figure 4.14: Optimized ridged NGV cascade.

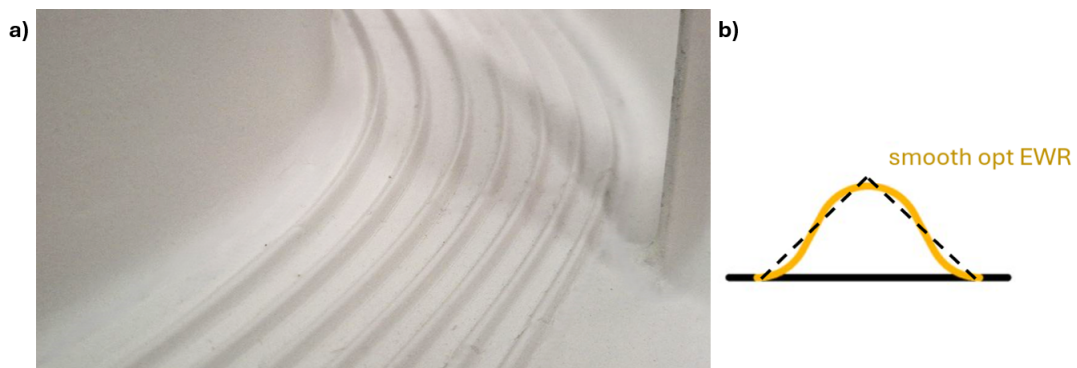


Figure 4.15: (a) Close-up view and (b) tangential section of the optimized ridges.

the conventional EWR in Figure 4.10.b for comparison). The final ridge design features smoother profiles, slightly reduced height, and fillet radii that make it representative of

a realistically manufacturable component. Moreover, the optimized configuration incorporates seven ridges instead of six, extending slightly beyond the trailing edge compared to the demonstrator. As previously discussed, these refinements yield only marginal improvements over the conventional design.

4.1.1.6 NGV Test Matrix

All NGV cascade configurations were tested at fixed incidence under steady inflow conditions. To assess the effect of the passive devices under different operating conditions, two Reynolds numbers based on the axial chord C_x , $Re_x = 400'000$ and $Re_x = 700'000$, representative of cruise and take-off/landing conditions, respectively, were investigated. The experimental test matrix for the NGV cascade configurations investigated in the Genova wind tunnel is summarized in Table 4.5.

Table 4.5: Experimental test matrix of the NGV cascade.

	Base	Fences	Conv. ridges	Fences + conv. ridges	Opt. ridges
$Re_x = 400k$	✓	✓	✓	✓	✓
$Re_x = 700k$	✓	✓	✓	✓	✓

Complementary to these experimental investigations, 3D numerical simulations, combining Reynolds-Averaged Navier–Stokes (RANS) and Large Eddy Simulation (LES) approaches, were also carried out. These simulations further supported the analysis by clarifying the mechanisms through which fences and ridges mitigate passage vortex effects within the vane channel.

4.1.2 Savona Laboratory Test Section

The test section of the wind tunnel installed in Savona is instead designed to accommodate the seven-blade linear cascade representative of the highly loaded unshrouded LPT cascade, reported in Figure 4.16.

The cascade is characterized by a high aspect ratio ($AR > 3$), a solidity of nearly $\sigma = 1.0$, a flow turning of approximately $\theta = 95^\circ$, and an incompressible Zweifel number

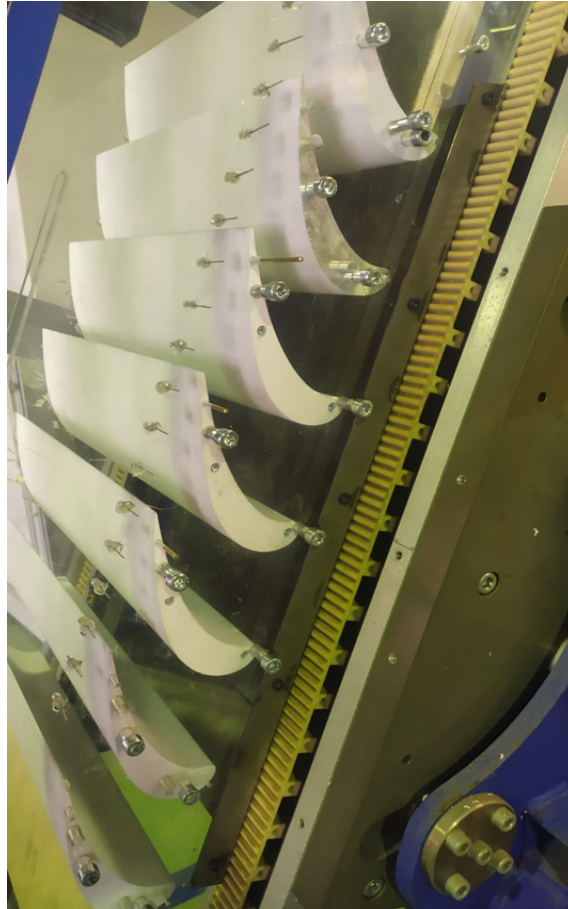


Figure 4.16: Unshrouded LPT cascade installed in the Savona test section.

of about $Z_w = 0.8$. Contrary to the NGV cascade, in this wind tunnel the cascade is mounted on a vertical plane, so that the blades deflect the flow upward, as shown in the lateral view of the test section in Figure 4.17.a. It is worth noting that this orientation is not expected to affect the flow, as gravity effects are negligible.

From the exit view sketch of the test section (Figure 4.17.b), it can be observed that in this configuration the blades are mounted between two flat and parallel plexiglass endwalls. The plexiglass near the blade tips is fixed and rigidly connected to the test section entrance to ensure a controlled inlet boundary layer for the cascade across different tests, since the tip region is the primary area of interest for these experimental investigations. Downstream of the blades, static pressure taps are integrated into this plexiglass to measure the downstream static pressure, similarly to the NGV cascade. The blades are mounted via screws onto the hub plexiglass to guarantee precise positioning within the cascade. This plexiglass, rigidly connected to the blades, can be moved closer to or farther from the fixed case plexiglass to regulate the tip gap height (τ) for all blades. To

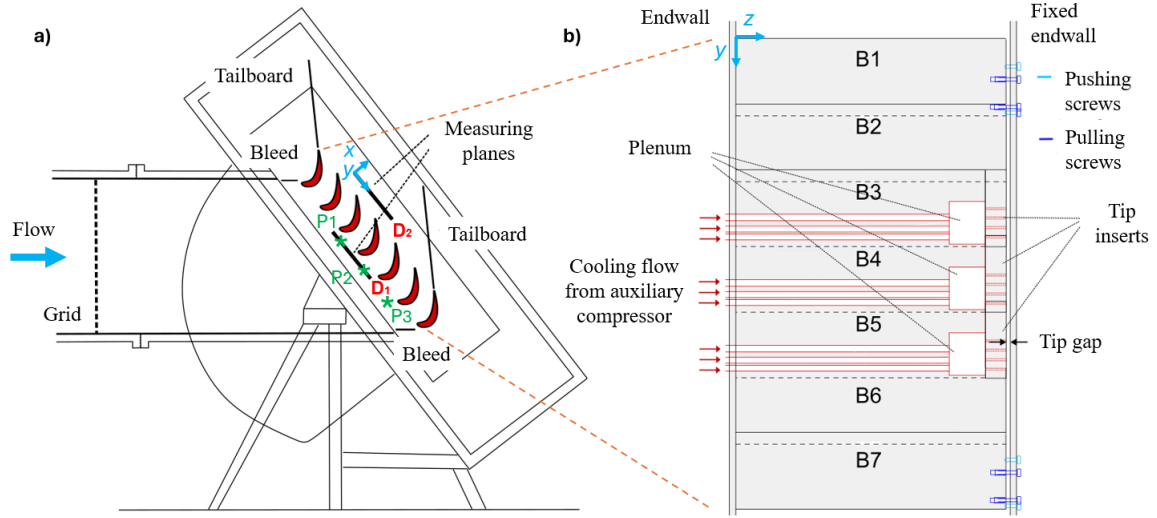


Figure 4.17: (a) Lateral and (b) exit view sketches of the test section arrangement used for the investigation of tip leakage losses.

this end, a system of pushing and pulling screws on the two lateral blades allows fine adjustment of the clearance. The tip gap has been measured with gages, with an accuracy of ± 0.05 mm. The five central blades are free of screws on the case, allowing the tip leakage flow to develop without obstructions, while on the three central blades it is possible to test alternative tip geometries. This approach has notable merits, as in the literature several studies simulate tip clearance only on the central blades [252], [286], and testing alternative tip shapes is often limited to a single blade [109]. Specifically, the three blades are manufactured in two parts: an aluminum core that hosts a plenum for testing cooled tips, and a 3D printed interchangeable tip, allowing the investigation of different tip geometries, a detailed photo of which is shown in Figure 4.18.

The coolant flow is supplied into the plenum via an auxiliary compressor. The plenum and coolant supply channels are highlighted in red in the sketch of Figure 4.17.b. A Venturi-meter at the compressor inlet is used to measure the coolant mass flow rate, $\dot{m}_c$. Specifically, the Venturi-meter is connected to a manometer (accuracy ± 1 Pa) that measures the pressure difference between the ambient inlet air and the throat of the Venturi-meter. Since this pressure difference is uniquely related to the mass flow rate through the Venturi-meter throat, assuming negligible ambient air velocity and knowing the throat dimensions, the coolant flow can be set based on the measured pressure difference, with a measurement accuracy of approximately $\pm 2\%$ of the coolant mass flow rate. Downstream of the Venturi, a duct containing steel wool to uniformize the flow exiting the compressor, along with several flexible tubes, directs the flow to the plenums of the three central blades. Here too, the steel wool ensures a uniform flow before exiting through the dedicated coolant holes on the blade tip surface. Moreover, Figure 4.17.a highlights the

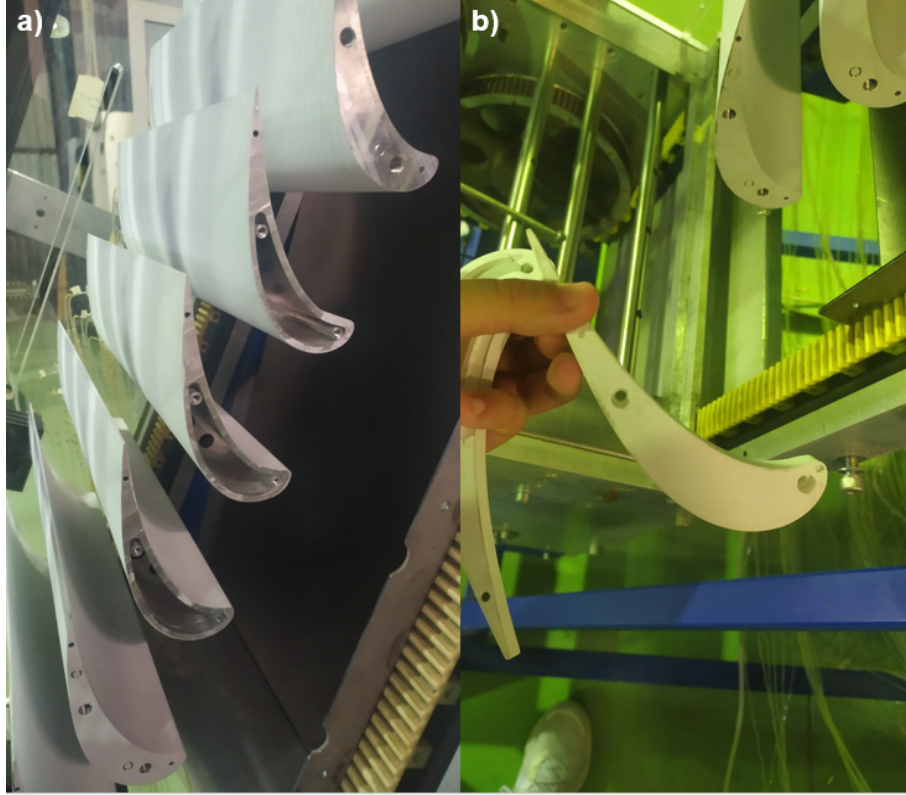


Figure 4.18: (a) Plenum and (b) flat tip insertion.

presence of adjustable bleeds and tailboards to set cascade periodicity, verified through blade loading measurements via several pressure taps at the midspan of the three central blades. From the sketch, the measurement traverses in the hub endwall, D_1 and D_2 , are also visible. These are positioned at $10\% \cdot C_x$ upstream of the blade leading edge and $30\% \cdot C_x$ downstream of the trailing edge, respectively, allowing the probes to sample the flow field entering and exiting the cascade in the pitchwise and spanwise directions. Similarly to the NGV case, a hot-wire anemometer was used to characterize in detail the time-resolved velocity and turbulence of the incoming flow in the D_1 plane. Moreover, measurements along different streamwise positions at midspan (indicated with *) allowed characterization of the turbulence decay, essential for setting proper boundary conditions in numerical simulations. Specifically, turbulence measurements were taken at P_1 ($x_t/C_x=4.5$), P_2 ($x_t/C_x=4.84$), and P_3 ($x_t/C_x=5.19$). Subsequently, both a Kiel probe and a five-hole probe were used in the two measurement traverses. The Kiel probe, due to its high accuracy, was used to detail the total pressure profile of the incoming flow, which is also necessary for defining proper inlet boundary conditions, and to compute the total pressure loss coefficient. The five-hole probe was used to characterize the flow velocity field, particularly at the cascade exit in the D_2 plane, for the analysis of secondary flow

structures. Each probe measures one pitch with 31 points and moves from the endwall to midspan with 21 points, using a bidirectional, numerically controlled traverse driven by a stepper motor with an integrated lead screw (800 steps per revolution, corresponding to a 2 mm screw pitch), allowing a minimum probe displacement of 0.0025 mm. The probes are also mounted on a holder with a goniometer, enabling rotation and precise alignment with the flow direction.

4.1.2.1 Unshrouded LPT Configurations

The experimental campaign began with the unshrouded LPT cascade equipped with a flat tip, serving as the reference condition to assess the effectiveness of alternative tip geometries and the potential influence of coolant injection. Accordingly, the tip of the three central blades was subsequently redesigned to host squealer and squealer–winglet configurations. Both geometries were equipped with cooling holes, allowing an assessment of the tip cooling mass flow rate effect on the tip leakage vortex. The tip inserts were realized as add-on elements, manufactured using high-resolution 3D printing technology and mounted directly onto the existing cascade, hence preserving the original setup and boundary conditions of the experimental tests. For improved comparison among the tested configurations, schematic top and cross-sectional views of the (a) flat, (b) squealer, and (c) squealer–winglet tips are shown in Figure 4.19.

4.1.2.2 Squealer Tip

The first tip modification involved the adoption of an optimized squealer-tip geometry designed by Morfo Design. In this configuration, the optimization process targeted the reduction of tip leakage losses by parameterizing both the rim width and height, alongside the pressure gradient distribution on the squealer surface at the LE and TE. The squealer was also equipped with cooling holes, designed with reference to configurations available in the literature. Specifically, the squealer tip was conceived as a continuous cavity tip with an equal rim width on both pressure and suction sides, and a rim height-to-chord ratio exceeding 5%. It features four coolant ejection holes, equally spaced along the camber line, providing tip cooling. The holes have a diameter equal to the rim width and are oriented orthogonally to the cavity floor. These specifications are illustrated in Figure 4.19.b [287]. Once designed in CAD and manufactured using 3D printing, this tip geometry was experimentally validated. Figure 4.20 shows the unshrouded LPT cascade equipped with the squealer tip geometry on the tips of the three central blades.

4.1.2.3 Squealer-Winglet Tip

Then, the unshrouded LPT cascade was tested with a squealer–winglet tip configuration. Also in this case, the geometry was obtained through a CFD-based parametric design

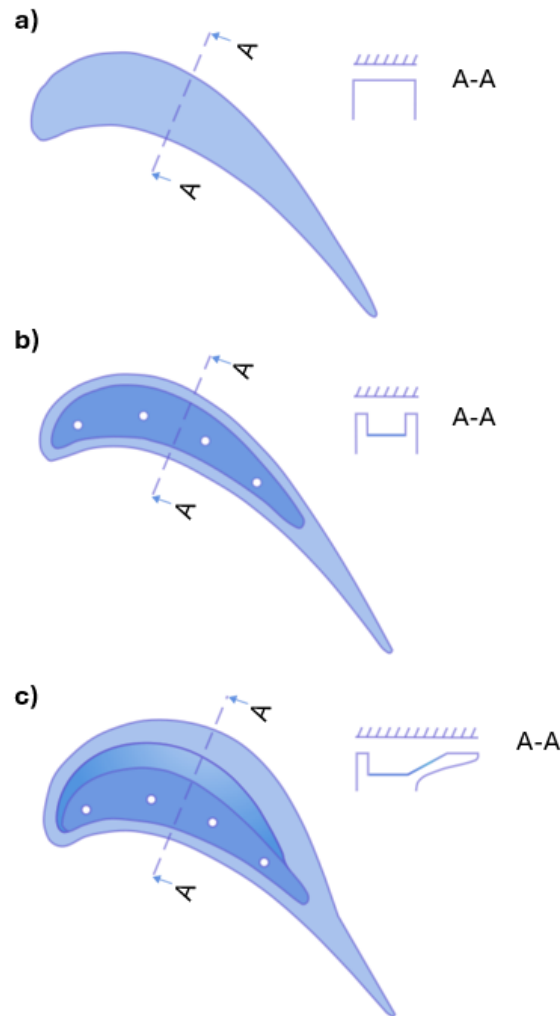


Figure 4.19: Schematic top and cross-sectional views of the (a) flat, (b) squealer, and (c) squealer–winglet tips.

campaign carried out by Morfo Design using STAR-CCM+ as the flow solver. The design preserved the pressure side rim geometry and cooling holes of the baseline squealer tip, while the winglet shape was parametrized. In addition to the pressure gradients on the tip surface at both LE and TE, the parameters varied during the optimization included the start position and streamwise extent of the winglet along the blade SS, as well as the winglet thickness at both tip and base. These latter variations therefore accounted for the inclination angle of the inner suction side rim and for the chordwise distribution of the winglet tangential width. The optimal geometry resulted in a suction side rim featuring a variable inclination along the chord: perpendicular to the cavity floor in its leading and trailing portions, where it connects to the pressure side rim, and gradually increasing up



Figure 4.20: Unshrouded LPT cascade equipped with the squealer tip geometry on the blade tips.

to 45° in correspondence with the blade maximum thickness region. The suction side winglet, characterized by a smooth connection with the squealer rims, exhibits a variable tangential width, reaching its maximum—approximately equal to the rim height—near the blade maximum thickness location (see Figure 4.19.c). As for the previous configuration, this tip was manufactured as an add-on element and mounted on the existing unshrouded LPT cascade to experimentally assess its effectiveness in controlling and mitigating the tip leakage vortex. Figure 4.21 shows the unshrouded LPT cascade equipped with the squealer–winglet tip geometry on the three central blades.

4.1.2.4 Unshrouded LPT Test Matrix

All the tested LPT configurations were performed at fixed incidence and under steady incoming flow conditions, in order to preserve the boundary conditions among the different tip modifications. In addition to the characterization of the gap-free blade performance, the flat, squealer, and squealer–winglet tips were tested for three different tip gap sizes ($\tau/C_x = 1\%$, 3% , and 5%) at a Reynolds number of $Re_x = 200'000$. Besides the condition



Figure 4.21: Unshrouded LPT cascade equipped with the squealer-winglet tip geometry on the blade tips.

without coolant flow ($\dot{m}_c/\dot{m}_0 = 0\%$, where $\dot{m}_0$ denotes the mass flow rate inside a single blade channel), both squealer and squealer–winglet configurations were additionally tested with two coolant flow ratios ($\dot{m}_c/\dot{m}_0 = 0.45\%$ and 0.9%) for each tip gap, in order to characterize the interaction between the coolant, the leakage, and the main passage flows. For selected operating conditions, a low Reynolds number case at $Re_x = 100'000$ was also investigated to assess the effect of Reynolds number variation on tip leakage losses. The experimental test matrix conducted on the unshrouded LPT cascade is summarized in Table 4.6.

Complementary, a large numerical campaign based on 3D RANS and LES simulations was performed to provide a deep insight into the mechanisms of leakage loss generation, analysing the effectiveness of the cavity–squealer and squealer–winglet tip configurations in reducing the leakage mass flow and the consequent mixing losses, as well as the influence of different coolant flow ratios by investigating their interaction with the leakage flow inside the cavity and, downstream, with the main flow. See Appendix 2 for further details on the numerical setup.

Table 4.6: Experimental test matrix of the unshrouded LPT cascade.

		Flat				Squealer				Sq.-Winglet			
τ/C_x [%]		0	1	3	5	0	1	3	5	0	1	3	5
$Re_x = 200k$	$\dot{m}_c/\dot{m}_0 = 0$	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
	$\dot{m}_c/\dot{m}_0 = 0.45\%$						✓	✓	✓		✓	✓	✓
	$\dot{m}_c/\dot{m}_0 = 0.9\%$						✓	✓	✓		✓	✓	✓
$Re_x = 100k$	$\dot{m}_c/\dot{m}_0 = 0$	✓				✓				✓			
	$\dot{m}_c/\dot{m}_0 = 0.9\%$						✓	✓	✓		✓	✓	✓

4.2 Measurement Techniques and Acquisition Chains

4.2.1 Hot-Wire Anemometer with Constant Temperature Anemometer Module

The hot-wire anemometer (HWA) is an instrument capable of measuring the time-resolved velocity and hence the turbulent properties of a flow field by means of a sensitive element consisting of a thin tungsten wire exposed to the flow. This wire typically has a length of about 1 mm and a diameter of 5 μm . The tungsten wire is traversed by an electric current, which is dissipated through Joule heating effect at a rate that increases with the flow velocity surrounding the element. The wire is inserted as an active resistance in one arm of a Wheatstone bridge. Its electrical resistance depends on its thermal characteristics and temperature, which in turn are governed by the heat dissipated by forced convection. Hence, when the flow velocity changes, the convective heat transfer rate varies, altering the wire temperature and consequently its electrical resistance. This unbalances the Wheatstone bridge, prompting the power supply to adjust the circuit voltage and current intensity in order to compensate for the convective power losses. Through this mechanism, the wire voltage E is related to the flow velocity v according to King's law, expressed as $E^2 = E_0^2 + Bv^n$, where E_0 is the supply voltage at zero flow velocity, and B and n are empirical calibration coefficients. Due to this operating principle, the Hot-Wire Anemometer features a high-frequency response, making it an essential tool for accurate measurements of turbulent flow quantities.

In the present work, a Dantec single-sensor miniature boundary layer Hot-Wire Anemometer probe (type 55P14) was employed in combination with a Constant Temperature Anemome-

ter (CTA) module, which includes the electronic components such as the Wheatstone bridge and operational amplifiers [288]. This setup provides a voltage signal proportional to the measured flow quantity. Figure 4.22 shows (a) the technical image of the probe and (b) the CTA module used in the experiments.



Figure 4.22: (a) Technical image of the probe and (b) CTA module.

Specifically, an overheat ratio of 0.8 has been set. The anemometer output voltages have been sampled using a DAQ-NI acquisition system, collecting 400'000 samples at a sampling frequency of 40 kHz for each measuring point. An anti-aliasing low-pass filter with a cutoff frequency of 20 kHz has been applied. The uncertainty in the velocity measurements has been estimated to be below 1%, considering a 95% confidence level (see Appendix 1).

The HWA was calibrated by exposing the sensor wire to an air jet exiting a dedicated calibration nozzle at various velocities. The reference velocity was uniquely determined by measuring the static pressure within the nozzle plenum (since the flow velocity inside the plenum is negligible and the expansion through the nozzle is assumed to be isentropic, this measurement represents the total pressure at the nozzle exit) and considering the static pressure at the exit to be equal to the ambient pressure. Measurements were performed using a differential pressure manometer with an accuracy of ± 3 Pa. Simultaneously, the corresponding output voltages from the HWA instrument were recorded, including the voltage at zero velocity to serve as a reference condition. Then, the empirical coefficients B and n have been obtained through a least squares fitting technique, allowing the instantaneous velocity to be determined during the actual measurements once the output voltage is known. The calibration curve is reported in Figure 4.23.

Then, the HWA has been employed in the inlet measurement domain to characterize the boundary layer and the turbulent properties of the incoming flow to the cascade. It has also been used at midspan at several locations upstream of the cascade and downstream of the grid along the flow direction to characterize the turbulence decay, thereby providing accurate boundary conditions to properly define the inflow in 3D numerical simulations. Referring to the sketches of the test section shown in Figures 4.5 and 4.17.a, the HWA has been operated in both pitchwise and spanwise directions within the upstream measurement planes D_1 , and at midspan in the measurement points indicated by *.

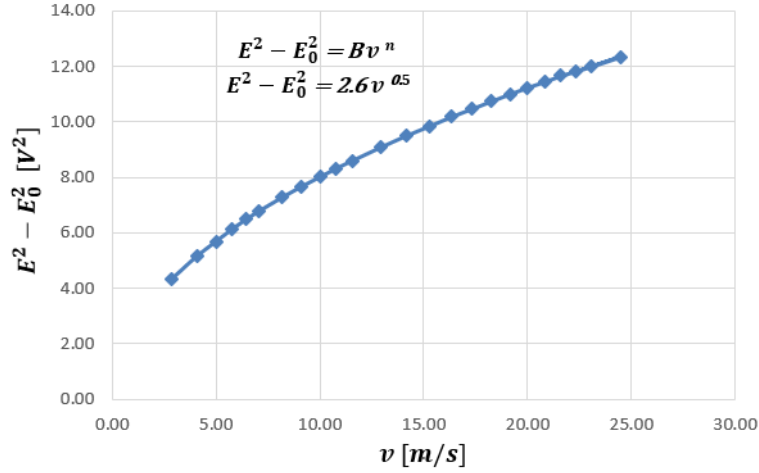


Figure 4.23: HWA calibration curve.

4.2.2 Static Pressure Taps with Digital Sensor Array

The static pressure taps are small holes made on the blade and endwall surfaces, with their axes perpendicular to the local surfaces, specifically designed for the measurement of the static pressure in these points. For this reason, they are sharp-edged holes with an internal diameter of 0.6 mm, ensuring that the pressure inside the channel corresponds to the local wall static pressure. This dimension prevents the dynamic pressure component from recirculating within the hole and stagnating, while also minimizing the risk of clogging due to impurities in the air.

Both cascades under investigation feature three instrumented central blades. The NGV cascade includes 26 static pressure taps (13 on the pressure side and 13 on the suction side) located at midspan, and 27 taps (13 on the pressure side and 14 on the suction side) positioned near the divergent endwall, at 80% of the blade height. The unshrouded LPT cascade, on the other hand, is equipped with 12 pressure taps on the pressure side and 13 on the suction side, all positioned at midspan. These pressure taps are used to determine the aerodynamic load distribution at the respective spanwise locations. Moreover, considering that the NGV and LPT cascades are scaled by 5:1 and 3:1, respectively, the relatively high number of taps ensures high spatial resolution in the measurement of the aerodynamic loading.

Additional 53 pressure taps are installed on the NGV endwall surfaces, and 16 further taps are located downstream of the LPT cascade, allowing the measurement of the downstream static pressure. This latter measurement is essential for the facility operating setup, as it is used to regulate and fix the position of the movable tailboards downstream of the cascade. The tailboards are adjusted to obtain an approximately zero and spatially uniform static pressure distribution at the cascade exit, hence ensuring atmospheric outlet

conditions and the periodicity of the flow across adjacent passages.

The pressure taps are connected to the pressure transducers through small pneumatic lines consisting of tubes with a diameter of 0.8 mm. To create the internal pneumatic channels within the instrumented blades—linking the blade surface pressure taps to the outside without altering the blade geometry—steel capillaries were embedded in grooves of appropriate shape and size machined into the blade surface. The external profile was then restored using filler material (see Figure 4.24.a–b). Similarly, steel tubes were integrated within the endwall to connect its pressure taps to the outer wall of the wind tunnel (see Figure 4.24.c). Finally, flexible silicone tubes were used to connect the ends of the steel tubes to the pressure transducers.

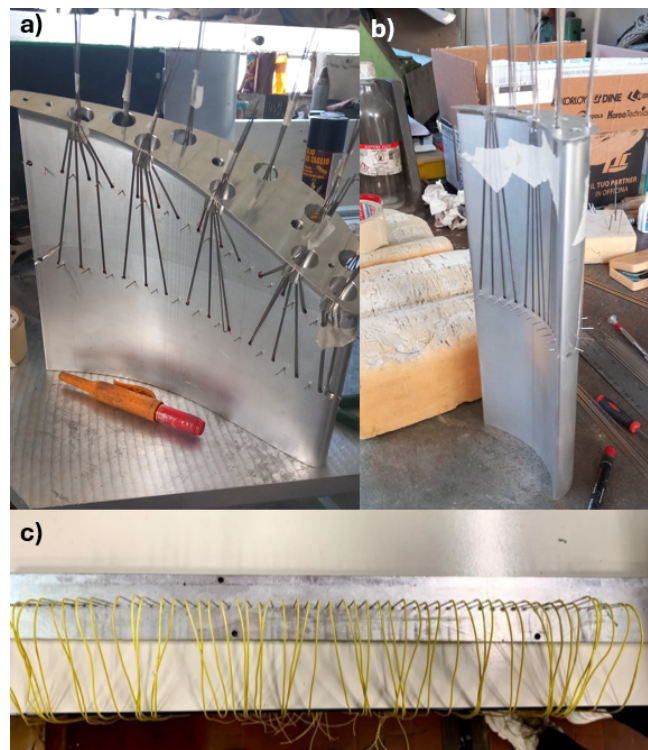


Figure 4.24: Blade pressure taps and internal pneumatic channels in the instrumented (a) NGV and (b) unshrouded LPT blades; (c) endwall pressure taps with steel and silicone tubes.

Specifically, for the blade loading measurements and the static pressure measurement downstream of the cascades, three Scanivalve Digital Sensor Array (DSA) units [289], each equipped with 16 differential transducers with a range of 2500 Pa and an accuracy of $\pm 0.08\%$ of the full scale, were used. These units are shown in Figure 4.25. The uncertainty associated with the static pressure coefficient was estimated to be below 1.5% (see Appendix 1). The communication with the DSA units was managed via Ethernet through



Figure 4.25: Scanivalve DSA transducer units.

the LabVIEW software. The procedure involved an initial calibration of the DSAs (zero offset acquisition), followed by their configuration — setting 250 samples per pressure tap and a sampling frequency of 500 Hz — and finally the acquisition of the static pressure values directly in Pascal.

4.2.3 Kiel and five-hole pressure probes with differential transducers

The Kiel probe is an instrument designed to measure the total pressure of the flow. It consists of a stem and a head, which in this case has a diameter of 1.5 mm to minimize intrusiveness relative to the cascade dimensions, and houses the sensing element. The head features a truncated-cone shape at the inlet of the sensing element, which aligns the incoming flow with the probe axis, allowing the capture of both static and dynamic components of the flow. Hence, this geometry makes the Kiel probe insensitive to a wide range of yaw and pitch angles, allowing accurate total pressure measurements even under moderately misaligned flow conditions. To fully exploit this insensitivity range, the probe is mounted on a frame equipped with a goniometer, enabling precise orientation and alignment with the expected local freestream direction. Owing to its high accuracy and robustness, the Kiel probe is employed for total pressure mapping at both inlet and outlet of the cascades, as well as to set the facility to a specific operating condition. A schematic view of the Kiel probe is shown in Figure 4.26.

The five-hole probe allows the simultaneous measurement of total pressure, static pressure, and flow direction in terms of both yaw (α) and pitch (β) angles. From these quantities, the magnitude and direction of the velocity vector can be determined, hence providing the complete velocity field. This makes the probe particularly suitable to visualize secondary flow structures and to determine the flow exit angle downstream of the blades. Similarly to the Kiel probe, it consists of two main components: a long stem and a miniaturized head, which acts as the sensing element. In the present configuration, the head has a diameter of 2.5 mm, selected to minimize flow distortion given the large scale

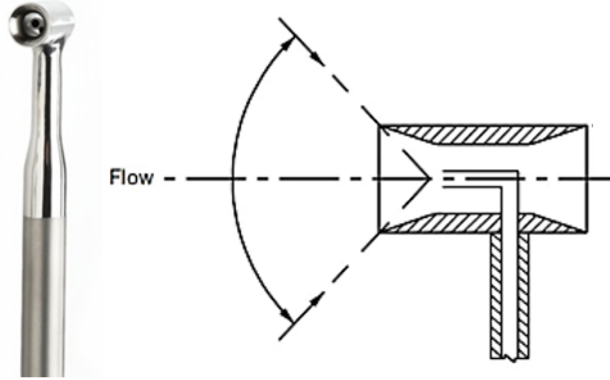


Figure 4.26: Kiel probe.

dimensions of the test section. The probe operates on the principle that the pressure distribution over the head surface uniquely depends on the local direction of the incoming velocity vector, in addition to the static pressure. For this reason, the truncated-cone nose of the probe features five pressure holes. The central one (hole 1) is most sensitive to total pressure variations; holes 2 and 3, located symmetrically on the lateral sides, are sensitive to changes in yaw angle; while holes 4 and 5, positioned above and below, are sensitive to pitch angle variations. The probe is typically operated in its maximum sensitivity position, with the head axis aligned to the expected mean flow direction. As in the case of the Kiel probe, a goniometer mounted on the support frame enables fine orientation and alignment of the probe with the freestream. A schematic of the five-hole probe is shown in Figure 4.27.

The calibration phase of the five-hole probe plays a crucial role in accurately determining the total and static pressures, as well as the flow angles, from the pressure readings acquired by the five ports. The calibration of the probe used in this work was performed on a dedicated calibration rig equipped with a numerically controlled positioning system, which allows the probe to rotate about its two axes in discrete steps. This setup enables a full characterization of the probe response with respect to both yaw and pitch angles, while keeping the probe head fixed in space. During calibration, the probe head was positioned at the exit of a calibrated nozzle, where the static pressure in the settling chamber was maintained constant throughout the procedure. Since the flow velocity inside the settling chamber is negligible and the flow through the nozzle is isentropic, this static pressure represents the total pressure at the nozzle exit seen by the probe. To generalize the calibration for different flow regimes, a set of dimensionless pressure coefficients has been defined: CP_{32} , based on the pressure difference between holes 3 and 2, associated with the yaw angle (α); CP_{54} , based on the pressure difference between holes 5 and 4, associated with the pitch angle (β); CP_{din} and CP_t , related respectively to the dynamic and total pressures. During calibration, these coefficients are known, and maps of CP_{32} and CP_{54} can

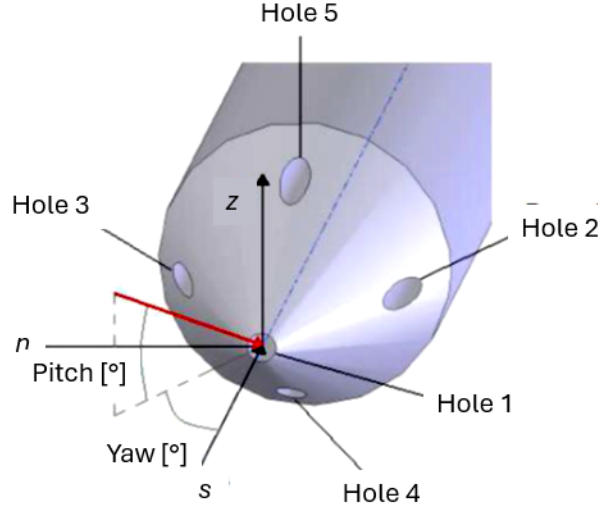


Figure 4.27: Five-hole probe.

be constructed by varying α and β within a range of $\pm 30^\circ$. In the measurement phase, the inverse procedure is applied: the coefficients CP_{32} and CP_{54} are derived from the measured pressures, and an iterative procedure is used to determine the corresponding yaw and pitch angles. These angles are then used to access the calibration curves of CP_{din} and CP_t , from which the dynamic pressure (i.e., flow velocity) and total pressure are obtained, and consequently the static pressure as well. Since the five-hole probe utilized in this study features a hemispherical head geometry, it is inherently less sensitive to Reynolds number variations, ensuring a more stable response and consistent pressure coefficients across a broader range of operating conditions. In any case, the Reynolds number range tested during the experimental campaign remained within the range characterized during the probe calibration phase, thereby ensuring the reliability and accuracy of the measured flow field downstream of the cascade.

Both probes have been operated through a LabVIEW interface, which automatically controlled their positioning to sample the flow over a predefined measurement grid located upstream D_1 and downstream D_2 of the cascades (see the measurement planes sketched in Figures 4.5 and 4.17.a). The Kiel probe has been employed to measure the total pressure, while the five-hole probe has been used to characterize the velocity field of the flow entering and exiting both the NGV aggressive duct and the unshrouded LPT cascade. These measurements enabled a detailed evaluation of the aerodynamic performance of the cascades. At the inlet domain, measurements allowed the reconstruction of the two-dimensional flow field and the detailed characterization of the upstream boundary layer, which is crucial to accurately define inlet boundary conditions for both low- and high-fidelity simulations [290]. At the outlet domain, the data provided insight into the

secondary flow structures developing downstream of the cascades. For both probes, the pressure signals were acquired by collecting 5000 samples at a 1 kHz sampling frequency, using several SETRA differential pressure transducers [291] with a measurement range of ± 620 Pa and an accuracy of $\pm 0.075\%$ of the full scale. The uncertainty in the total pressure loss coefficient, estimated via the propagation of uncertainty, was approximately 2%, while the uncertainty in the yaw angle was 0.2° (see Appendix 1). A picture of the SETRA transducers is shown in Figure 4.28. Specifically, this type of transducer communicates

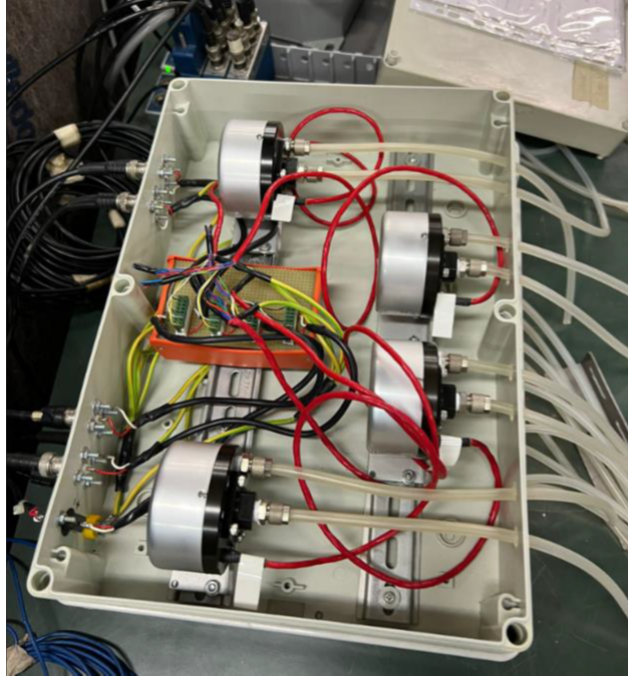


Figure 4.28: SETRA differential pressure transducers.

with the computer through a DAQ-NI acquisition system, providing an output voltage $X[V]$ proportional to the measured pressure $X[Pa]$. Knowing the individual calibration coefficient C_{trasd} of each transducer, and recording its zero offsets X_0 at the beginning and at the end of each test (to account for ambient pressure and temperature variations), the output voltage from the SETRA transducers can be accurately converted into pressure values according to the following relation:

$$X[Pa] = (X[V] - X_0[V]) \cdot C_{trasd} \quad (4.2)$$

4.3 Data Analysis

The following sections present the data analysis techniques applied to the measurements acquired from the various probes. The correspondence between each measurement technique and the related output is summarized below:

- **Hot-Wire Anemometer** → Time-resolved velocity measurements → Flow turbulence characterization
- **Pressure Taps** → Static pressure measurements → Blade loading distributions
- **Kiel Probe** → Total pressure measurements → Aerodynamic loss evaluation
- **Five-Hole Probe** → Total and static pressure, flow angle measurements → Velocity field reconstruction

The comprehensive synthesis of dedicated probes and acquisition systems, as well as the detailed analysis of the error propagation for the uncertainty assessment of the main variables and the repeatability of the experimental measurements, are reported in Appendix 1.

4.3.1 Flow Turbulence Characterization

Time-resolved flow velocity signals acquired at each measurement point by means of the HWA have been processed to fully characterize the turbulence properties. For each time series, the turbulence intensity Tu has been evaluated as the ratio between the root mean square value (v_{rms}) and the time-mean value ($\bar{v}$) of the streamwise velocity component:

$$Tu = \frac{v_{rms}}{\bar{v}} \quad (4.3)$$

The turbulence intensity values measured at midspan at different axial locations downstream of the turbulence generating grid also enabled the evaluation of the turbulence decay, as a function of the streamwise distance from the grid x_t , made non-dimensional by the grid bar width d (reported in Tab. 4.4). This decay trend can be compared with the correlation proposed by Roach [292]:

$$Tu = K \frac{x_t}{d}^{(-5/7)} \quad (4.4)$$

where K is a geometric parameter primarily dependent on the turbulence grid, linking the turbulence intensity with the grid geometry and its downstream position.

Thanks to the time-resolved inlet flow velocity data, additional quantities of interest — especially useful for accurately defining the boundary conditions in numerical simulations

— can be derived. For the evaluation of the flow integral scales, the Power Spectrum (PS) of the velocity signal has been computed according to the Wiener–Khinchin theorem [293] as:

$$PS = FT(v(t))FT^*(v(t)) \quad (4.5)$$

where $FT(\cdot)$ denotes the Fourier Transform, $v(t)$ is the streamwise velocity time trace, while $(\cdot)^*$ indicates the complex conjugate. The auto-correlation function is then obtained through the inverse Fourier Transform of PS , normalized by its value at $t = 0$:

$$R(t) = \frac{FT^{-1}(PS)}{FT^{-1}(PS)|_{t=0}} \quad (4.6)$$

The integral time scale (T_0) is calculated by integrating $R(t)$ from $t = 0$ to the first zero-crossing time, where the auto-correlation function becomes null:

$$T_0 = \int_{t=0}^{t=t_{R(t)=0}} R(t)dt \quad (4.7)$$

The integral length scale (L_0) is then derived using Taylor’s frozen turbulence hypothesis as:

$$L_0 = T_0 \cdot \bar{v} \quad (4.8)$$

Once the integral length scale has been computed, the turbulence dissipation rate ε can be evaluated following Batchelor [294] as:

$$\varepsilon = \frac{\bar{k}^{3/2}}{L_0} \quad (4.9)$$

where $\bar{k}$ indicates the time-mean turbulent kinetic energy. This provides the characterization of the smallest dissipative eddies in the flow. The rate of dissipation per unit turbulent kinetic energy (ω) has been derived as:

$$\omega = \frac{\varepsilon}{\bar{k}} \quad (4.10)$$

4.3.2 Blade Loading Distributions

Through the static pressure taps installed on the blade surfaces at midspan (and, in the case of the NGV cascade, also at 80% of the blade height near the divergent endwall) and downstream of the blade row, it is possible to:

- Correctly orient the tailboards to ensure that the static pressure at the cascade exit, p_2 , is approximately zero and nearly constant across the different blade channels.

- Set the facility to a specific steady operating condition. By using a Kiel probe positioned upstream at midspan and within the inter-blade channel (away from the potential effects induced by the blade leading edges), the upstream total pressure $p_{t,1}$ can be measured. Under the assumption of an isentropic flow, this value can be considered equal to the total pressure at the cascade exit $p_{t,2,is}$. Knowing p_2 , the isentropic dynamic pressure at the cascade outlet can then be calculated as:

$$p_{t,2,is} - p_2 = \frac{1}{2} \rho v_{2,is}^2 \quad (4.11)$$

which is directly related to the Reynolds number (Eq. 4.1) at which the test is to be performed.

- Verify the periodicity condition of both cascades through the measurement of the aerodynamic loading. The pressure coefficient, c_p , is calculated as:

$$c_p = \frac{\overline{p_{t,1}} - p(x)}{\frac{1}{2} \rho v_{2,is}^2} \quad (4.12)$$

where the area-averaged total pressure at the cascade inlet $\overline{p_{t,1}}$ was measured at the inlet domain D_1 of Figures 4.5 and 4.17.a using a Kiel probe. The local dynamic pressure along the blade surface in Eq. 4.12 is made non dimensional by the isentropic dynamic pressure at the cascade outlet (Eq. 4.11)

4.3.3 Aerodynamic Loss Evaluation

To directly define the efficiency η , it is necessary to first define the enthalpy loss coefficient:

$$\zeta = \frac{h_2 - h_{2,is}}{h_{t,2} - h_2} \quad (4.13)$$

However, during experimental tests, a net balance of the total pressure flux, which is unambiguously related to η under specific assumptions regarding the inlet boundary conditions and flow properties, can be more readily established (see the work of Denton [37]).

To assess the aerodynamic performance of the cascade, the total pressure maps obtained by the Kiel probe at the inlet $p_{t,1}$ and the outlet $p_{t,2}$ were compared point-by-point. From this comparison, the two-dimensional distribution of the total pressure loss coefficient $C_{pt}(y, z)$ was evaluated as:

$$C_{pt}(y, z) = \frac{p_{t,1}(y, z) - p_{t,2}(y, z)}{\frac{1}{2} \rho v_{2,is}^2} \quad (4.14)$$

Total pressure difference in Eq. 4.14 is made non dimensional by the area-averaged downstream isentropic dynamic pressure. The pitchwise average of the total pressure loss coefficient has also been evaluated and expressed in terms of the spanwise distribution, $\overline{C_{pt}}(z)$, obtained integrating along the pitchwise direction:

$$\overline{C_{pt}}(z) = \frac{1}{s} \int_0^s C_{pt}(y, z) dy \quad (4.15)$$

To evaluate the cascade total pressure losses ω_t , the map of $C_{pt}(y, z)$ has been integrated over both pitchwise and spanwise directions:

$$\omega_t = \frac{1}{s \cdot h} \int_0^h \int_0^s C_{pt}(y, z) dy dz \quad (4.16)$$

It is recalled that s and h are the cascade pitch and height, respectively.

This represents an area-averaged loss evaluation, whereas to account for the local mass flow rate, the mass-averaged loss coefficient $\omega_t^{(MA)}$ has been defined by mass-averaging inlet and outlet total pressure distributions indicated by the superscript $\cdot^{(MA)}$.

$$\omega_t^{(MA)} = \frac{\overline{\overline{p_{t,1}}}^{(MA)} - \overline{\overline{p_{t,2}}}^{(MA)}}{\frac{1}{2} \rho v_{2,is}^2} \quad (4.17)$$

This represents a net inlet–outlet balance, whereas for the unshrouded LPT cascade, to account for the stagnation pressure of the coolant, a second total loss coefficient $\omega_{t,c}^{(MA)}$ has been adopted. Specifically, the energy contributions from the total pressure of the flow entering the cascade $p_{t,1}$ and from the total pressure of the coolant $p_{t,c}$ have been averaged according to their relative mass flow rates. The latter has been assumed to be equal to the total pressure within the plenum $p_{t,pl}$, as reported, for example, in [109] and [286]:

$$\omega_{t,c}^{(MA)} = \frac{\left(\frac{\dot{m}_o \overline{\overline{p_{t,1}}}^{(MA)} + \dot{m}_c p_{t,pl}}{\dot{m}_o + \dot{m}_c} - \overline{\overline{p_{t,2}}}^{(MA)} \right)}{\frac{1}{2} \rho v_{2,is}^2} \quad (4.18)$$

For clarity, it should be emphasized that for these last two equations, each quantity $p_t^{(MA)}$ has been obtained as:

$$\overline{\overline{p_t}}^{(MA)} = \frac{\int_0^h \int_0^s p_t(y, z) \cdot v_x(y, z) dy dz}{\int_0^h \int_0^s v_x(y, z) dy dz} \quad (4.19)$$

where $v_x(y, z)$ is the axial velocity of the flow obtained from the five-hole probe, as will be outlined in Eq.4.25.

The total losses in the cascade result from both profile ω_p and secondary ω_s losses. Profile losses are evaluated as

$$\omega_p = \frac{\overline{P_{t,1,mid}} - \overline{P_{t,2,mid}}}{\frac{1}{2}\rho v_{2,is}^2} \quad (4.20)$$

where the subscript $\cdot_{mid}$ identifies the spanwise location representative of the two-dimensional flow. For the LPT cascade this corresponds to the geometrical midspan ($z/h = 0.5$). For the NGV cascade, due to outlet channel asymmetry, the 2D flow condition is instead found at $z/h = 0.35$, which is therefore used to compute ω_p without interference from secondary flows. The losses associated with secondary flows are then determined as:

$$\omega_s = \omega_t - \omega_p \quad (4.21)$$

The contribution to loss production associated with the tip leakage flow, ω_{tip} , in the unshrouded LPT cascade can be isolated within the secondary loss component by comparing the gap-free results with those obtained in the presence of a tip gap. In accordance with Eq. 4.19, both ω_s and ω_{tip} can be rewritten as mass-averaged quantities.

Finally, to compensate for any temporal drift of both wind tunnels in the cascades, the total pressure in the settling chamber has also been monitored and recorded as a reference, enabling the definition of a temporal correction coefficient. This coefficient has been determined by referencing the settling chamber pressure signals to the first measurement point and subsequently applying the correction point by point to the total pressure distributions and the corresponding coefficients obtained from the cascades.

4.3.4 Velocity Field Reconstruction

The five-hole pressure probe enables the analysis of secondary flow structures in terms of velocity vector maps. Defining (s, n, z) as the reference frame in the probe coordinate system and $v(y, z)$ as the velocity of the flow incident on the probe measured in each pitchwise and spanwise position in the upstream and downstream cascade domains, the individual velocity components can be obtained by projecting v according to the measured angles $\alpha(y, z)$ and $\beta(y, z)$ as:

$$v_s(y, z) = v(y, z) \cdot \cos\beta(y, z) \cdot \cos\alpha(y, z); \quad (4.22)$$

$$v_n(y, z) = v(y, z) \cdot \cos\beta(y, z) \cdot \sin\alpha(y, z); \quad (4.23)$$

$$v_z(y, z) = v(y, z) \cdot \sin\beta(y, z); \quad (4.24)$$

Then, these velocity components can be projected into the cascade reference system (x, y, z) :

$$v_x(y, z) = v(y, z) \cdot \cos \beta(y, z) \cdot \cos(\alpha(y, z) + \alpha_s); \quad (4.25)$$

$$v_y(y, z) = v(y, z) \cdot \cos \beta(y, z) \cdot \sin(\alpha(y, z) + \alpha_s); \quad (4.26)$$

where α_s represents the angle between the probe head axis and the cascade reference system, which is approximately equal to the blade throat angle and is the same for every point in the measurement map.

It should also be noted that, for the NGV cascade, which features a divergent endwall, the z -component of velocity has been corrected to account only for secondary flows by subtracting the portion that would be associated with an inviscid flow evolving through the divergent channel.

$$v_z(y, z) = v_x(y, z) \cdot \tan(\beta(y, z) - \beta_{inv}) \quad (4.27)$$

where β_{inv} is obtained by assuming a linear distribution of β from -25° at the divergent endwall to 0° at the flat endwall.

The Secondary Kinetic Energy SKE has then been obtained as:

$$SKE(y, z) = \frac{v_z^2(y, z) + v_n^2(y, z)}{v_s^2(y, z)} \quad (4.28)$$

and the non-dimensional streamwise $v_{n,ad}(y, z)$ and spanwise $v_{z,ad}(y, z)$ velocity components have been obtained by dividing by $v_{2,is}$.

Analogously to the evaluation of $\overline{C_{pt}}(z)$, the spanwise distributions of the velocity components in the probe reference system, $\overline{v_x}(z)$, $\overline{v_y}(z)$ and $\overline{v_z}(z)$, can be obtained, from which the spanwise distributions of the flow angles at the cascade exit can in turn be calculated as:

$$\overline{\alpha_2}(z) = \arctan\left(\frac{\overline{v_y}(z)}{\overline{v_x}(z)}\right) \quad (4.29)$$

$$\overline{\beta_2}(z) = \arctan\left(\frac{\overline{v_z}(z)}{\overline{v_x}(z)}\right) \quad (4.30)$$

These distributions are essential for understanding how the flow exiting the cascade is over- or under-turned due to the secondary flows along the blade span.

In addition, a flow yaw angle distortion coefficient, averaged over the exit domain, can be calculated to assess the non-uniformity of the flow downstream of the cascade as:

$$c_{\alpha,2} = \frac{|\overline{\alpha_2}(z) - \overline{\alpha_{2,mid}}|}{\overline{\alpha_{2,mid}}} \quad (4.31)$$

Chapter 5

Results

In this chapter, the results concerning the passive and active flow control and secondary flow mitigation strategies investigated on the two aeronautical turbine cascades under study are presented. The chapter first reports the experimental findings related to the effectiveness of fences, ridges, and their combined application in attenuating the passage vortex developing within the NGV. Subsequently, the influence of squealer and squealer–winglet tip geometries, together with different cooling mass flow rate conditions, on the tip leakage vortex of an unshrouded LPT blade is discussed for several values of the tip clearance height. These two main result sections are further divided into several subsections, addressing:

- A detailed characterization of the boundary conditions, which is essential to define the flow state at the test section inlet and to enable a consistent replication of the experimental setup within numerical simulations, since secondary flows are strongly influenced by the inlet boundary conditions [87].
- The characterization of the flow field developing in the baseline cascades, i.e., without any flow control or mitigation device. This baseline cascade serves as a reference condition to assess the effectiveness of the adopted devices in suppressing secondary flow structures.
- The analysis of the controlled cascades is then presented. The results obtained on the aggressive NGV and on the unshrouded LPT equipped with the different devices are discussed, and their effectiveness is evaluated in comparison with the baseline cases.
- A dedicated section on the experimental–numerical comparison, where detailed flow visualizations based on streamlines are used to elucidate the mechanisms through which the proposed devices reduce secondary losses, thereby improving the performance of the cascades and/or the entire turbine stages.

5.1 Aggressive NGV Results

5.1.1 Detailed Experimental Characterization of the Inlet Boundary Conditions

For a correct characterization of the inlet boundary conditions, the turbulence levels acquired with the HWA probe at midspan in the four measurement points marked with * in Figure 4.5.a (P_1 , P_2 , P_3 , and P_4) are first reported. The resulting turbulence decay is shown in Figure 5.1 as a function of the distance from the grid in the streamwise direction, made non-dimensional by the bar width, x_t/d . As shown in the figure, the experimental points (in blue) are in close agreement with the turbulence decay correlation proposed by Roach (red dashed line), reported in Eq. 4.4.

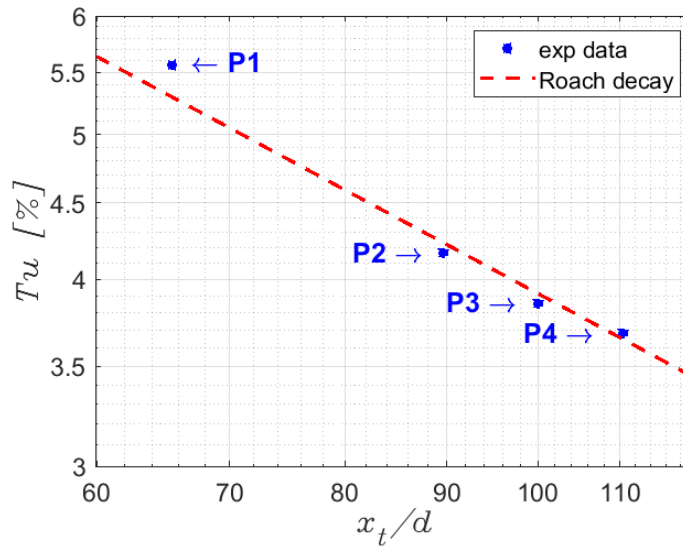


Figure 5.1: Turbulence decay along the flow direction.

Using the HWA measurements collected on plane D_1 , both in the pitchwise and spanwise directions, the turbulent characteristics of the cascade inlet flow can be evaluated. Figure 5.2.a shows the contour plot of the turbulence intensity, made non-dimensional with its midspan value Tu/Tu_{mid} , as a function of the tangential coordinate normalized by the blade pitch y/s and the spanwise coordinate normalized by the blade height z/h . Only half of the span is shown because the profiles are symmetric, owing to the geometric symmetry of the channel at the cascade inlet. As it can be seen, the turbulence intensity increases from midspan when moving toward the endwall boundary layer, also indicating potential effects induced by the leading edge of the central blade (located at $y/s = 0.5$). Figure 5.2.b reports the spanwise distribution of turbulence obtained by

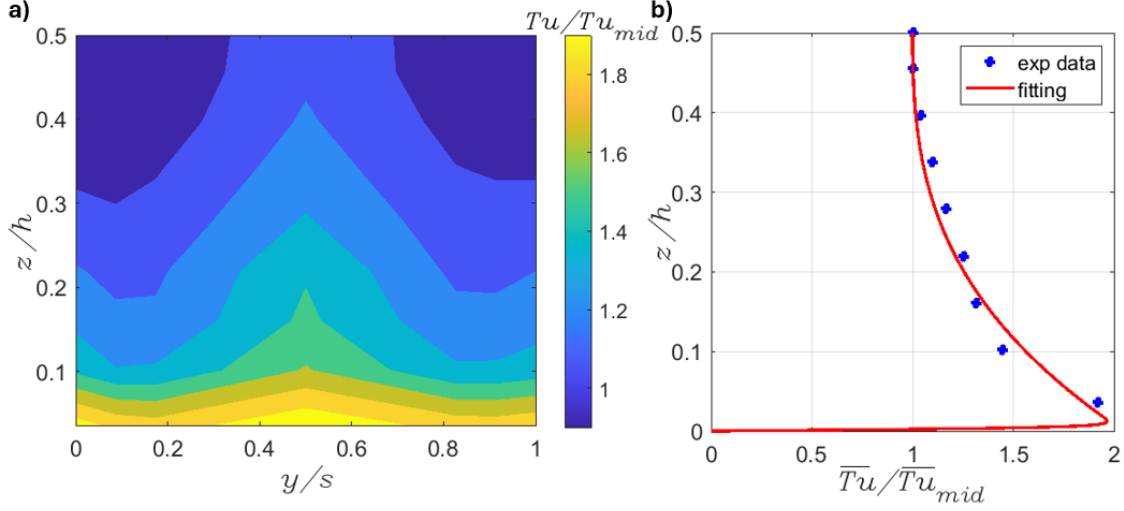


Figure 5.2: (a) Contour plot of Tu/Tu_{mid} along the pitchwise and spanwise directions, and (b) its pitchwise-averaged distribution along the spanwise direction.

pitchwise-averaging the previous Tu map. A fitting curve is also shown in red, which will be discussed in the following.

Then, as previously described, additional turbulence properties can be extracted from the same time-resolved velocity measurements acquired with the HWA on plane D_1 , from which the turbulence intensity Tu has already been obtained. Specifically, from the auto-correlation function $R(t)$ computed at each spatial point of the measurement domain—examples of which are reported in Figure 5.3 for two positions near the endwall

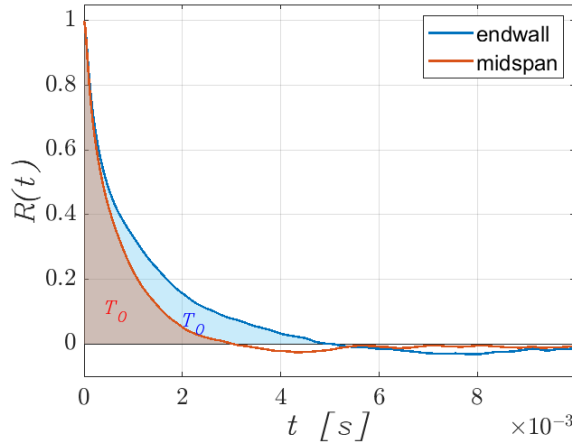


Figure 5.3: Examples of normalized auto-correlation functions $R(t)$ computed near the endwall (longer decay) and at midspan (shorter decay). The shaded areas indicate the corresponding integral time scales T_0 .

and at midspan—it is possible to derive the integral time scale T_0 (corresponding to the shaded areas), the integral length scale L_0 , the dissipation rate ε , and the rate of dissipation per unit turbulent kinetic energy ω .

The spanwise distributions of these inlet quantities, made non-dimensional by their midspan values, are plotted as blue points in Figure 5.4 as a function of the distance from the endwall, normalized by the blade height. The integral time scale increases toward the endwall (as also shown in Figure 5.3), implying a corresponding increase of the length scale within the boundary layer; its value at midspan, $L_{0,mid}$, is roughly 0.012 m. Moreover, since the velocity fluctuations increase even further toward the wall, the dissipation rate also rises, according to Eq. 4.9. Interestingly, the specific dissipation exhibits only minor variations moving toward the wall. To derive suitable equations for the turbulence-property data, each turbulence property has been fitted with the solid red curves shown

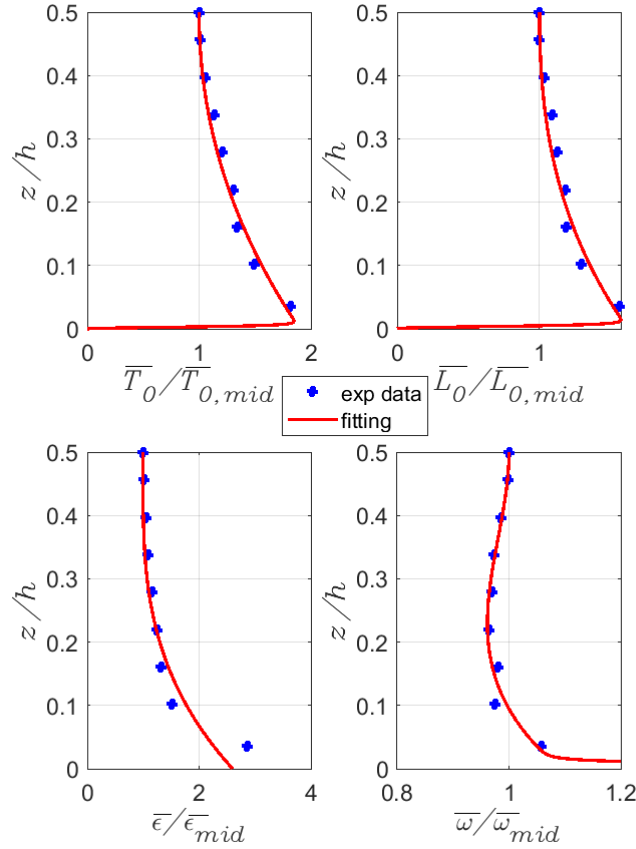


Figure 5.4: Spanwise distributions of the integral time scale T_0 , the integral length scale L_0 , the dissipation rate ε , and the rate of dissipation per unit turbulent kinetic energy ω .

in Figures 5.2.b and 5.4. The fitting procedure employed an equation of the form $Q(z)$, as reported in Eq. 5.1, consisting of a third-order polynomial multiplied by a hyperbolic tangent profile, which enforces the appropriate limiting values at the wall (zero for Tu , T_0 , and L_0 ; a constant for ε ; and tending to infinity for ω), in accordance with Wilcox et al. [295].

$$Q(z) = \left(A \left(\frac{z}{\delta} \right)^3 + B \left(\frac{z}{\delta} \right)^2 + C \left(\frac{z}{\delta} \right) + D \right) \tanh^F \left(E \frac{z}{\delta} \right) \quad (5.1)$$

where A , D , E , and F are coefficients determined by imposing $Q(h/2) = 1$ and $Q'(h/2) = 0$, and subsequently fitting the experimental data. The fitting curve equations are defined in the author's work [290] with the aim of providing detailed spanwise distributions of the inlet boundary conditions, which can be easily rescaled by readers depending on the channel height, the boundary layer thickness (accounting for Reynolds number variations), and the midspan values of the turbulence quantities, in order to properly set up numerical simulations. These equations will be used to correctly configure the simulations carried out by Morfo Design on both the NGV and the unshrouded LPT cascades.

To conclude the characterization of the inlet boundary conditions, Figure 5.5.a shows the contour plot of the normalized upstream total pressure distribution, $p_{t,1}/p_{t,1,mid}$, measured with the Kiel probe on plane D_1 , along the spanwise (z/h) and pitchwise (y/s) directions. Contrary to the Tu map shown in Figure 5.2, the pressure distribution appears uniform along the pitchwise direction, exhibiting variations only along the blade height due to the momentum deficit associated with the endwall boundary layer. Its pitchwise-averaged distribution, shown in Figure 5.5.b, provides a quantitative estimation of the

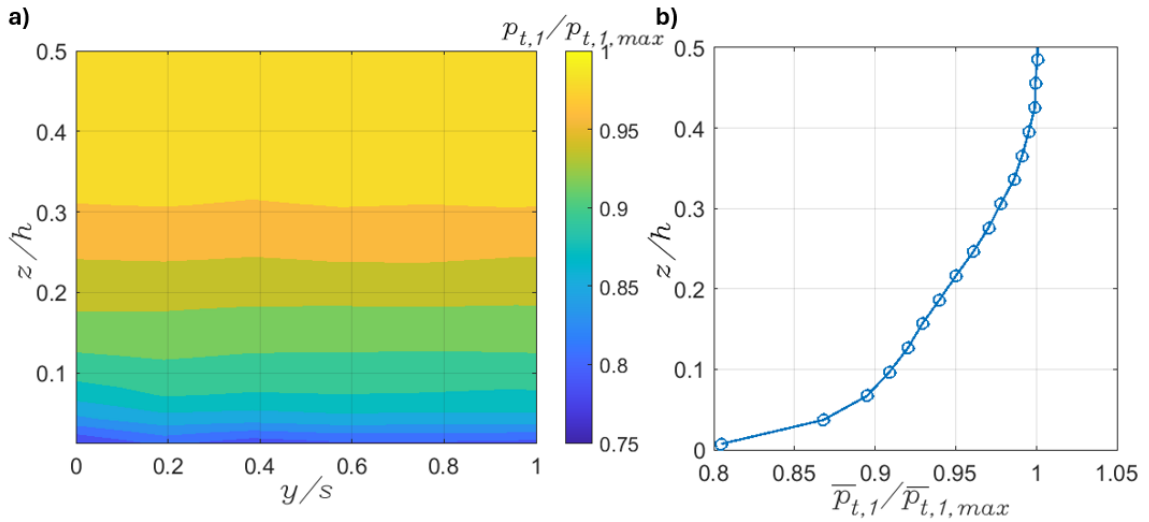


Figure 5.5: (a) Contour plot of $p_{t,1}/p_{t,1,mid}$ along the pitchwise and spanwise directions and (b) its pitchwise averaged distribution along the spanwise direction.

boundary layer thickness (here $\delta/h = 0.45$), which can be used in Eq. 5.1 to define the distributions of turbulence-related quantities to properly set up the boundary conditions in numerical simulations. It is worth noting that such a significant boundary layer thickness (high in proportion to the channel height though less substantial in absolute terms) provides an ideal environment for the present study. This condition intentionally enhances the prominence of secondary flow structures, thereby creating a more challenging and appropriate test case to evaluate the effectiveness of the proposed loss-reduction devices.

5.1.2 Baseline NGV

The experimental campaign began with the analysis of the baseline cascade, without any secondary flow control device applied, which served as a reference case to assess the effectiveness of the flow control and mitigation devices once implemented. This baseline case also allows isolating and analyzing the effect of the Reynolds number. Starting from the upstream region of the cascade, Figure 5.6 shows the spanwise total pressure distributions of the flow entering the cascade, measured with the Kiel probe in both pitchwise and spanwise directions within the measurement domain D_1 of Figure 4.5, for the two Reynolds numbers investigated. Both distributions are normalized by the pitchwise averaged total pressure at midspan. In contrast to the thick incoming boundary layer observed previously in Figure 5.5.b at $Re_x = 400'000$, a slightly thinner boundary layer ($\delta/h = 0.4$) is observed at $Re_x = 700'000$, where a fuller velocity profile is expected due to the re-

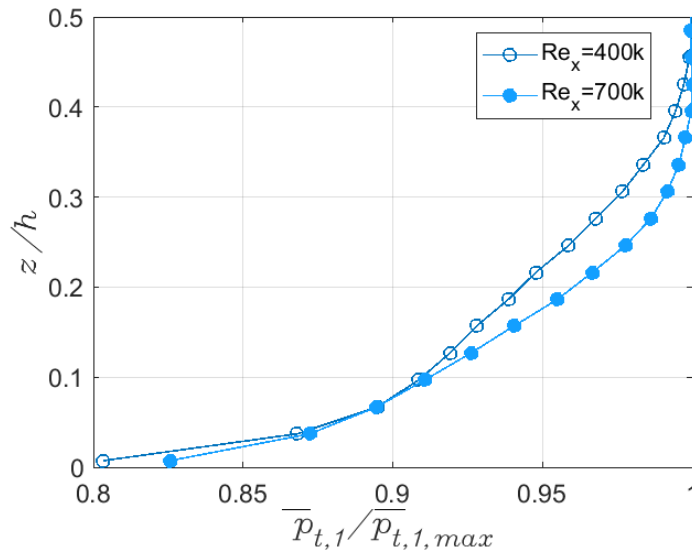


Figure 5.6: Spanwise distributions of the total pressure at the inlet of the baseline cascade for the two operating conditions.

duced influence of viscous effects. Figure 5.7 shows the blade loading distributions of the baseline blade, measured at both (a) midspan and (b) 80% of the blade height with the static pressure taps, for the two operating conditions. Both c_p distributions are made non-dimensional by the maximum value measured for the baseline blade under cruise conditions, i.e., $c_{p,max}$, for comparability. The nearly identical overlap of the c_p distributions at both Reynolds numbers indicates that no significant variation of the viscous phenomena occurs, likely due to the high turbulence levels induced by the turbulence grid. At the

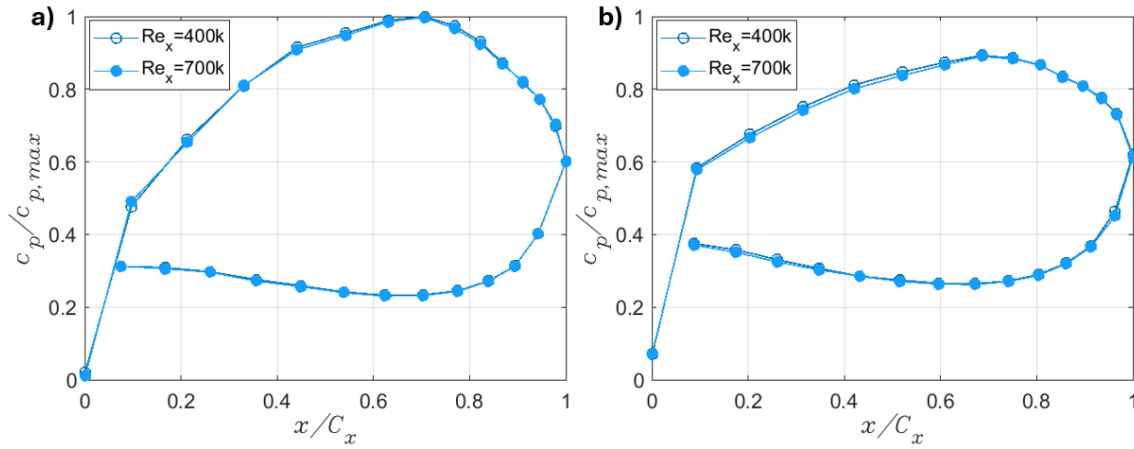


Figure 5.7: Blade loading distributions at (a) midspan and (b) 80% of the blade height for the baseline cascade under both Reynolds number conditions.

cascade outlet (D_2), which is of particular interest, detailed maps of the total pressure loss coefficient, C_{pt} , obtained with the Kiel probe, together with velocity vectors measured by the five-hole probe, can be analyzed to investigate the secondary flows at the cascade exit and to evaluate the influence of the Reynolds number on the vortex structures. The C_{pt} distributions, made non-dimensional by the maximum value measured in the baseline cascade at $Re_x = 400'000$, i.e., $C_{pt,max}$, are shown in Figure 5.8 for the (a) cruise and the (b) take-off conditions, respectively. In the downstream plane, the entire blade span is shown due to the asymmetry at the channel outlet. A loss region is visible downstream of the trailing edge (around $y/s = 0.5$) and extends over a large portion of the blade height, associated with the wake region. Two distinct loss cores are also observed near the flat and divergent endwalls, centered at $z/h = 0.18$ and $z/h = 0.65$, respectively, and located on the suction side of the blade. The velocity vector maps indicate that these two vortices are passage vortices, with the tip PV near the divergent endwall rotating clockwise, while the hub passage vortex rotates counterclockwise. More interestingly, the passage vortex near the blade tip, adjacent to the divergent endwall, is larger and stronger than the one near the flat endwall, due to the asymmetry of the meridional duct at the cascade outlet. Moreover, the combination of duct divergence and low blade aspect ratio results in sec-

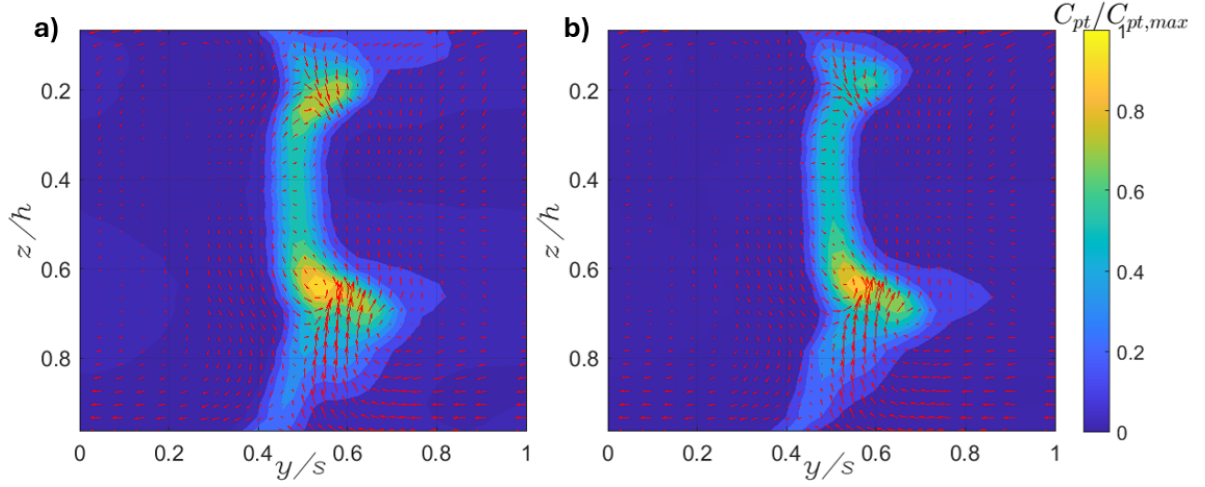


Figure 5.8: Total pressure coefficient maps downstream of the baseline cascade at (a) $Re_x=400'000$ and (b) $Re_x=700'000$.

ondary flow structures so extended that only a limited core region remains unaffected by the vortices and is primarily influenced by the blade wake. This is further highlighted by the strong three-dimensionality of the flow observed downstream of the cascade. With increasing Reynolds number, although the positions of the hub and tip passage vortices appear largely unchanged, a reduction in total pressure losses is observed, reflected in both width and intensity of the wake and vortex structures, which also leads to a slight decrease in the three-dimensionality of the flow. To quantitatively assess the effect of secondary flows on losses and the influence of Reynolds number, the total pressure loss coefficient maps have been pitchwise-averaged. Figure 5.9 shows the spanwise distributions of the averaged C_{pt} for the two Reynolds numbers tested, normalized by the maximum value obtained in the baseline configuration at $Re_x = 400'000$ i.e., $\overline{C_{pt,max}}$. In the baseline cascade, the two $\overline{C_{pt}}$ peaks correspond to the cores of the two passage vortices previously discussed, with the one near the blade tip (in the bottom part of the plots) exhibiting significantly higher intensity than the hub PV. Moreover, these peaks are sufficiently broad that only at $z/h = 0.35$ a region of minimum loss can be identified, where the limited influence of secondary flows, as visible from the maps, allows the computation of the profile loss coefficient, ω_p . Regarding the influence of Reynolds number, losses associated with secondary flows (particularly those related to the vortex structure developing near the divergent endwall) are significantly reduced at higher Reynolds numbers, as evidenced by the peak reduction at $z/h = 0.65$.

To analyze the flow distortion induced by secondary flows, Figure 5.10 shows the normalized distributions of the difference between the local pitchwise-averaged yaw angle ($\overline{\alpha_2}$) and the midspan pitchwise-averaged yaw angle ($\overline{\alpha_{2,mid}}$) for both Reynolds numbers.

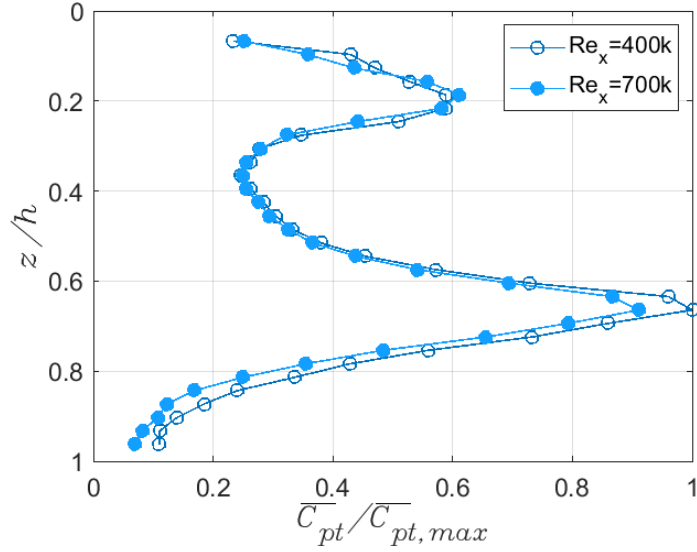


Figure 5.9: Distributions of the pitchwise averaged total pressure coefficient downstream of the baseline cascade for both Reynolds numbers.

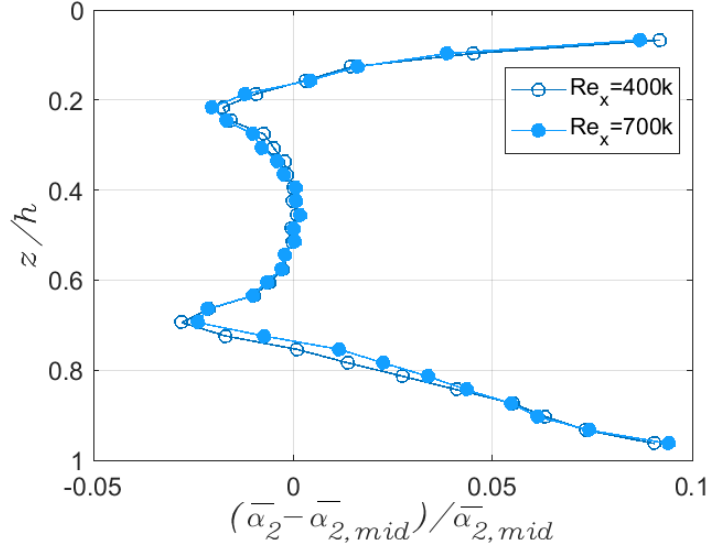


Figure 5.10: Distributions of the flow yaw angle deviation downstream of the baseline cascade for both Reynolds numbers.

Distinct regions of flow over-turning (near the endwalls) and under-turning (in correspondence to the vortex loss cores) can be identified, driven by the action of secondary vortices. As expected, the largest deviation in flow angle is associated with the dominant passage

vortex originating near the divergent endwall. Moreover, in this case, the influence of Reynolds number appears limited, resulting only in a very slight reduction of the underturning effect at the highest Re_x .

5.1.3 NGV with secondary flow control devices applied

Once the analysis of the baseline cascade was completed, the various devices were sequentially applied to test the other cascade configurations. To ensure a proper comparison among the five tested cases and to assess the effect of the different devices on secondary flow control and mitigation, it is first essential to verify that the boundary conditions remain unchanged before and after the installation of the devices on the vane surface and the divergent endwall.

The first step is to examine the endwall boundary layer entering the cascade, a fundamental assessment to evaluate the impact of secondary flow control devices on cascade performance, since the inlet boundary layer strongly influences the development of secondary flows [87]. Accordingly, Figure 5.11 shows the non-dimensional, pitchwise-averaged distributions of the upstream total pressure measured under cruise conditions for all configurations. Again, due to the symmetry of the vane and channel at the cascade inlet, only half of the span is shown. The nearly identical distributions confirm that the inlet boundary conditions remained unchanged after the installation of fences, conventional ridges, and optimized ridges, thereby ensuring a high level of comparability.

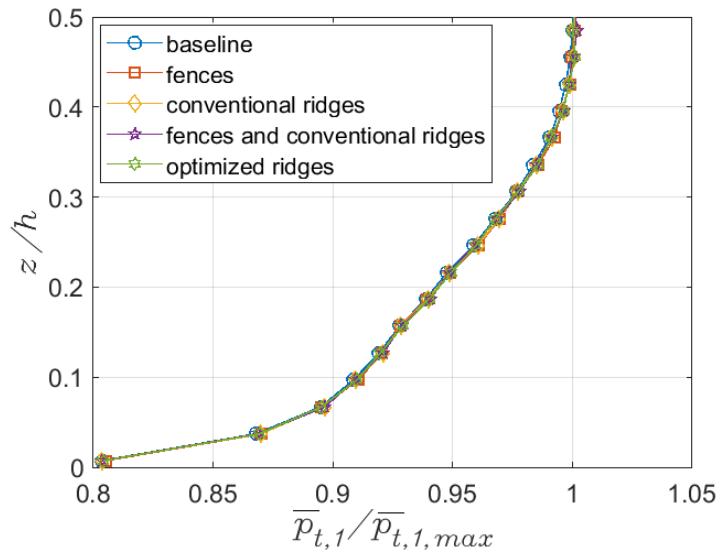


Figure 5.11: Spanwise distributions of the upstream total pressure for the different cascade configurations.

The blade loadings for the different cascade configurations, measured at the two spanwise positions for $Re_x = 400'000$, were also analyzed and are shown in Figure 5.12. In

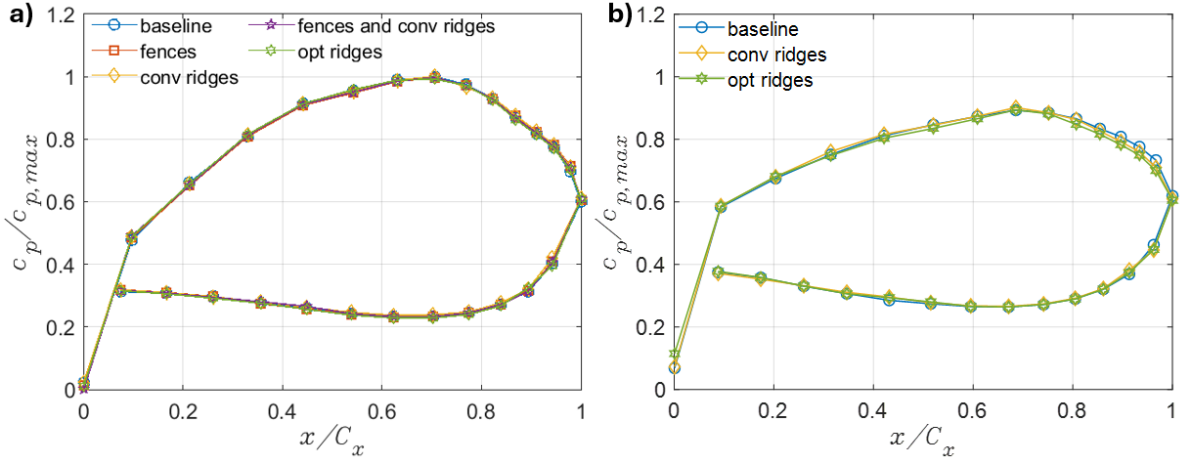


Figure 5.12: Comparison of blade loadings for the different cascade configurations at (a) midspan and (b) 80% of the blade height.

Figure 5.12.a, the aerodynamic loadings at midspan are almost perfectly superimposed, indicating no significant variation in the midspan flow among the five configurations. This result is expected, given the minimal influence of secondary flows at this blade height, as will be discussed in the following section. At the same time, this overlap further confirms the consistency of the experimental setup across all tests. In Figure 5.12.b, at 80% of the blade height, only the c_p distributions for the baseline, conventional ridge, and optimized ridge configurations are shown, since static pressures could not be measured at this blade height with fences installed. Here, although the three curves are largely overlapped across most of the blade surface, a small deviation appears on the rear part of the suction side, suggesting that ridges applied to the endwall may slightly affect the blade aerodynamic loading in the region near the tip. However, the impact of the devices on the secondary flow structures developing near the divergent endwall becomes much more evident when examining the total pressure loss coefficient maps, shown in Figure 5.13 for all cascade configurations and Reynolds numbers tested. Since the vortex developing near the flat endwall does not exhibit significant variations among the tested configurations, the following figures focus on the region from the blade height where profile losses are measured ($z/h = 0.35$) up to the blade tip. This allows the analysis to concentrate on the dominant secondary vortex developing near the divergent endwall. At this stage, only the cruise operating condition ($Re_x = 400'000$) is considered (left maps). The C_{pt} contour plot in Figure 5.13.b, measured downstream of the fenced cascade, shows that the large passage vortex at the blade tip is still present and has approximately the same intensity as in the baseline case (Figure 5.13.a). However, its core is confined closer to the diver-

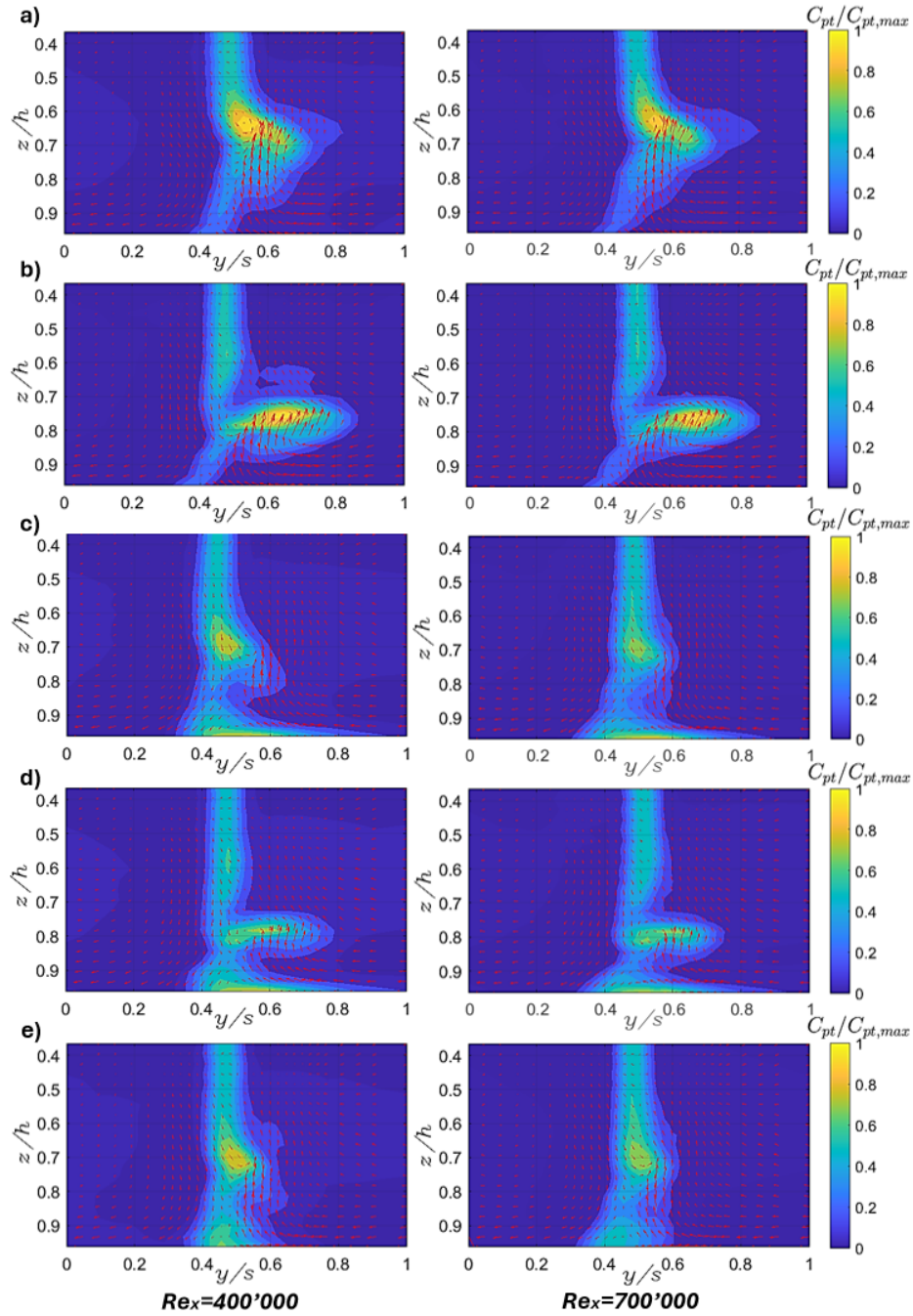


Figure 5.13: Maps of the total pressure loss coefficient downstream of the (a) baseline, (b) fenced, (c) conventional ridged, (d) fenced and conventional ridged and (e) optimized ridged cascades, for (left) $Re_x=400'000$ and (right) $Re_x=700'000$.

gent endwall, around $z/h = 0.75$, and appears more extended in the pitchwise direction. This behavior can be attributed to the shelf-like geometry of the fences, which pushes the vortex toward the wall, thereby reducing its spanwise penetration and allowing a larger portion of the blade height to remain unaffected by secondary flows. This reduction in secondary flow penetration results in a more uniform outflow from the fenced cascade compared to the baseline configuration. Such an improvement may induce a carry-over effect, potentially enhancing the efficiency of the downstream rotor and, consequently, the overall stage performance. The isolated effect of conventional ridges can be evaluated by analyzing the contour plot in Figure 5.13.c. In this case, the vortex core shifts to $z/h = 0.7$, indicating greater penetration along the blade height compared to the fenced case, but still reduced as compared to the baseline cascade. However, the large tip passage vortex is noticeably reduced in both spanwise and pitchwise directions. This change can be attributed to the grooved ridge design, which inhibits the development and propagation of the passage vortex from the pressure side of one blade to the suction side of the adjacent blade. Consequently, a larger portion of the blade channel remains unaffected by secondary flows. However, while endwall ridging enhances flow uniformity by limiting the development of secondary structures, it also induces additional losses near the endwall, most likely due to the increased wetted surface area and to the formation of small scale turbulent structures between the ridges. When conventional ridges are added to the fenced configuration (Figure 5.13.d), the reduced spanwise penetration and the elongated pitchwise shape of the vortex induced by the fences remain observable. However, a significant reduction in both size and intensity of the tip passage vortex becomes evident, likely due to the combined effect of the two control devices, with the endwall ridging still generating an additional high-loss region near the divergent endwall. Finally, in Figure 5.13.e, the optimized grooved design yields significantly lower losses in the vicinity of the endwall where the ridges are applied, compared to the conventional ridged configuration. Moreover, this setup achieves an additional—albeit modest—reduction in the size and three-dimensionality of the secondary vortical structures at the cascade outlet. Although the position of the tip passage vortex remains largely unchanged with respect to the conventional ridged configuration, the improved outflow uniformity results in a broader quasi-two-dimensional flow region. This is expected to further enhance stage efficiency in real engine applications, leveraging the benefits of optimized flow uniformity for improved downstream rotor performance. Figure 5.14 shows the spanwise $\overline{C_{pt}}$ distributions obtained by pitchwise-averaging the 2D maps measured for the different cascade configurations. The comparison highlights that, for the fenced cascade, the peak associated with the passage vortex near the divergent endwall is higher than in the baseline case. This is due to the larger extension of the loss core along the pitchwise direction when fences are applied, which results in higher pitchwise-averaged values of the total pressure loss coefficient, even though no significant differences are observed in the maximum values of the C_{pt} maps between the two configurations. Nevertheless, the core of the

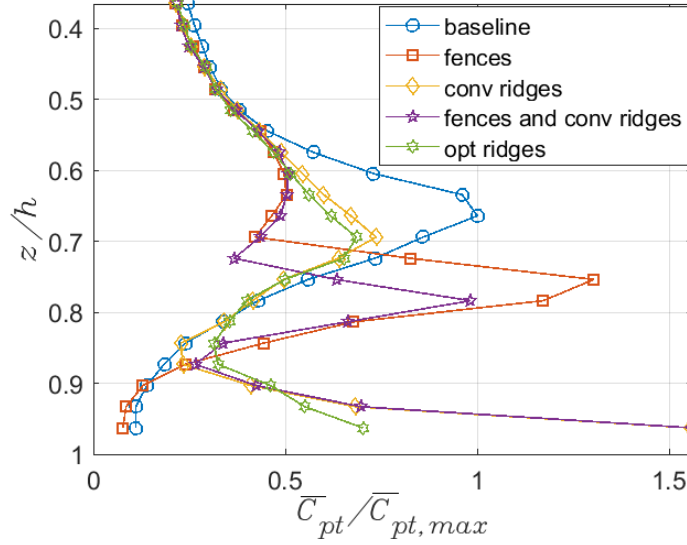


Figure 5.14: Comparison of total pressure coefficient distributions for the different cascade configurations.

secondary vortex in the fenced cascade is effectively more confined toward the divergent endwall, confirming the positive effect of fences in limiting the penetration of secondary structures along the blade height. Additionally, the $\overline{C_{pt}}$ distribution reveals another loss peak at $z/h = 0.62$, likely associated with the formation of vortical structures above the fence surfaces in combination with the passage vortex developing within the blade channel. In the configuration with only conventional ridges installed, the loss core related to the passage vortex at the divergent endwall is slightly shifted toward the blade tip compared to the baseline case, though not relative to the fenced configuration. In any case, the loss peak is significantly lower than in both the baseline and the fenced cases, resulting in a secondary loss region that is notably reduced and confined near the divergent endwall, as already evidenced by the C_{pt} maps. Furthermore, the additional loss peak observed in the 2D plot near the divergent endwall is clearly visible in the pitchwise-averaged distribution. When conventional ridges are added to the fenced configuration, the peak associated with the passage vortex is significantly reduced and slightly shifted further toward the divergent endwall compared to the fenced case. The secondary loss peak associated with the fences still persists, and the additional peak at the endwall due to the ridged geometry also appears. In the case of optimized ridges, the loss peak associated with the secondary vortex near the divergent endwall is slightly reduced compared to the conventional ridging configuration, and its penetration along the blade height is somewhat more contained. Additionally, the loss peak at the endwall induced by small scale vortical structures forming between the grooves is mitigated, thanks to the optimized height of the ridges.

As previously noted, the beneficial effects of these device configurations are more clearly observed in the reduction of flow three-dimensionality. This results in improved flow uniformity at the cascade exit, which may induce a carry-over effect on the downstream rotor row, enhancing its performance and ultimately contributing to increased overall stage efficiency. To illustrate this effect, Figure 5.15 shows the normalized distributions of the flow yaw angle distortion relative to the midspan-averaged flow angle. Here, it can

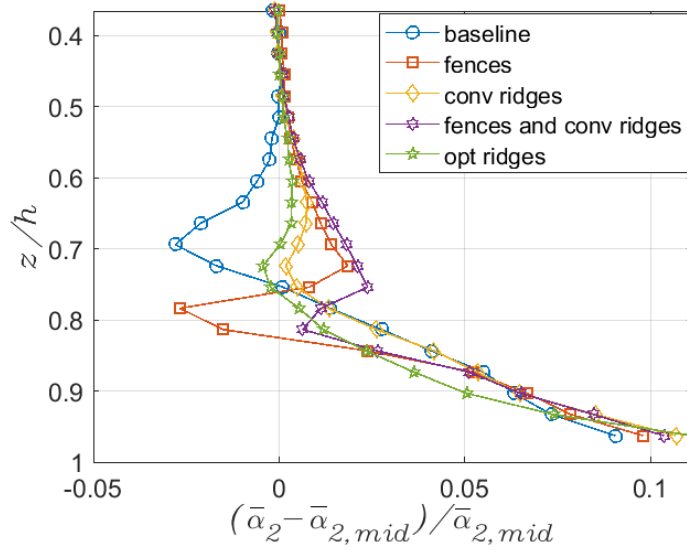


Figure 5.15: Comparison of exit yaw angle distributions for the different cascade configurations.

be observed that the maximum flow angle distortion is confined to regions closer to the divergent endwall when fences are installed on the blade, compared to the baseline configuration. As a result, a larger portion of the blade height exhibits only small deviations from $\overline{\alpha_{2,mid}}$. Specifically, in the fenced case, the local over-turning induced by the fences compensates the natural under-turning. When ridges are installed, the flow angle distortion is significantly mitigated, even with non-optimized ridges, and even more effectively with optimized ridges on the divergent endwall. This is achieved by reducing the strong under-turning caused by the tip passage vortex through the use of the grooved surface. When both conventional ridges and fences are installed in the cascade, a slight increase in the over-turning induced by fences leads to a further reduction in the under-turning with respect to the fenced case. However, this reduction is smaller than that observed when only ridges (conventional or optimized) are applied to the cascade. Hence, all the passive devices investigated contribute to a more uniform flow at the cascade exit, which is expected to optimize the performance of the downstream rotor row resulting in an overall improvement in turbine stage efficiency.

The effect of Reynolds number on the C_{pt} contour maps is clearly observable in the right plots of Figure 5.13, where the maps are made non-dimensional by the maximum C_{pt} value of the baseline cascade at the cruise condition to ensure a high level of comparability. From these maps, only a slight reduction in both intensity and size of the wake and secondary structures can be observed as the Reynolds number increases, in agreement with the baseline cascade behavior. For a more quantitative comparison, the distributions of $\bar{C}_{pt}$ and $\bar{\alpha}_2$ for the different cascade configurations are reported in Figure 5.16. Here, as

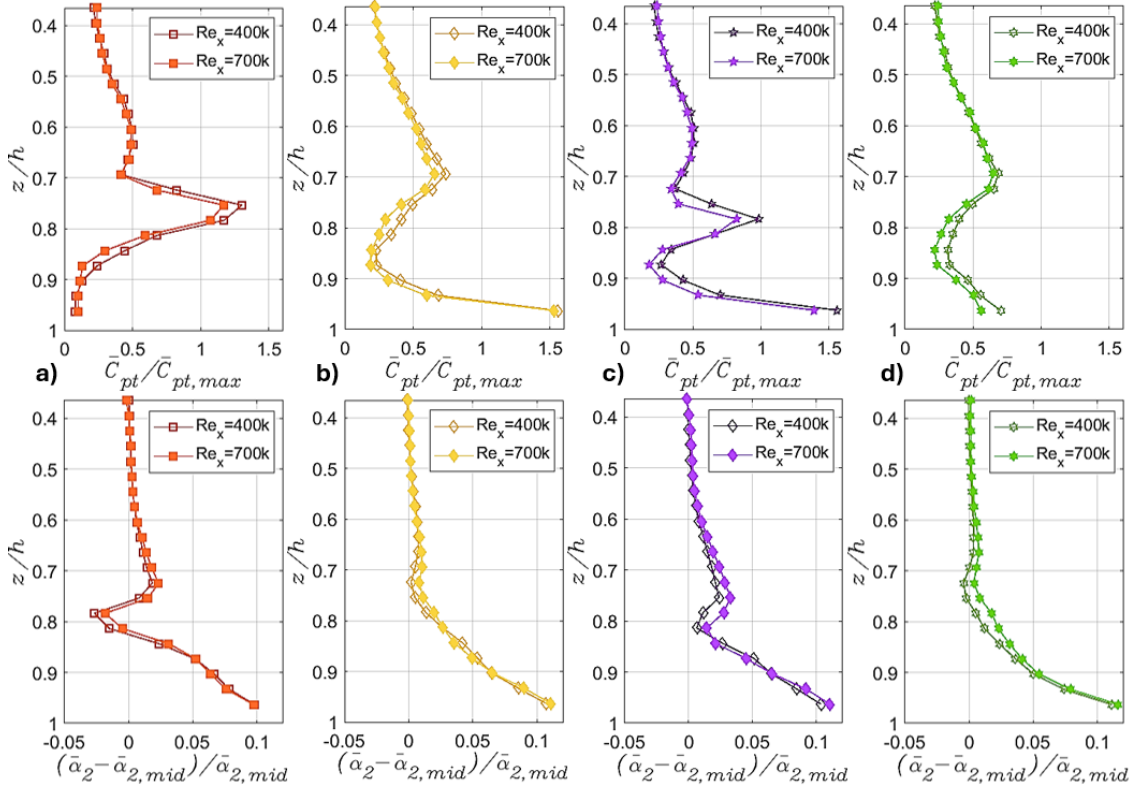


Figure 5.16: Distributions of the (top) total pressure loss coefficient and (bottom) exit flow yaw angle for the (a) fenced, (b) conventional ridged, (c) fenced and conventional ridged and (d) optimized ridged cascades at the two investigated conditions.

expected, similar to the baseline cascade, the total pressure loss and flow angle deviation slightly increase as the Reynolds number decreases. However, the distributions exhibit very similar shapes at the two operating conditions, confirming that the trends observed at $Re_x = 400'000$ also hold at $Re_x = 700'000$. In conclusion, to allow a quantitative comparison of the effectiveness of the different devices tested under both operating conditions, the profile and secondary losses are summarized in the bar chart of Figure 5.17, while the averaged flow angular distortion coefficient is reported in the bar chart of Figure 5.18. The area-averaged loss coefficients are presented normalized by the total losses of the base-

line cascade at cruise conditions, for confidentiality reasons. As shown in the bar chart, all tested configurations at the same Reynolds number exhibit similar profile loss values (blue) and significant secondary losses (red), roughly comparable to the profile losses. Regarding the effect of the devices, fences applied on the suction side increase ω_s by approximately 5% relative to the baseline configuration. The conventional endwall ridging solution also raises secondary losses (by about 2%), due to the increased wetted surface area introduced by the grooved geometry. When conventional ridges and fences are installed together, a slight further increase in ω_s is observed, attributable to the additional loss region near the endwall. However, when considering the optimized ridges, a reduction of approximately 2% in secondary losses relative to the baseline cascade is observed. This underscores the importance of optimizing the endwall ridging geometry to mitigate losses associated with secondary flow structures. Moreover, an increase in Reynolds number results in reduced profile and secondary losses for all configurations, regardless of the device applied. However, since the 2% reduction falls within the measurement uncertainty, the main potential of the devices lies in the increased flow uniformity at the NGV exit, as previously noted. The flow yaw angle distortion coefficient reported in Figure 5.18 is normalized by its value measured for the baseline cascade at $Re_x = 400'000$. From the bar chart, it is evident that the distortion coefficient decreases when passive devices are introduced. Fences achieve a reduction of approximately 12% compared to the baseline configuration, while conventional ridges alone yield a reduction of 8% at $Re_x = 400'000$ and 10% at $Re_x = 700'000$. When fences and ridges are both applied, the angle deviation is reduced by about 6% at $Re_x = 400'000$, and slightly less at $Re_x = 700'000$ relative to the baseline cascade. The largest mitigation of flow angle distortion is obtained with optimized ridges, achieving a reduction of approximately 32% at $Re_x = 400'000$ and 26% at $Re_x = 700'000$. Thus, also from the point of view of flow angle uniformity, optimized endwall ridging is confirmed to be the best performing solution.

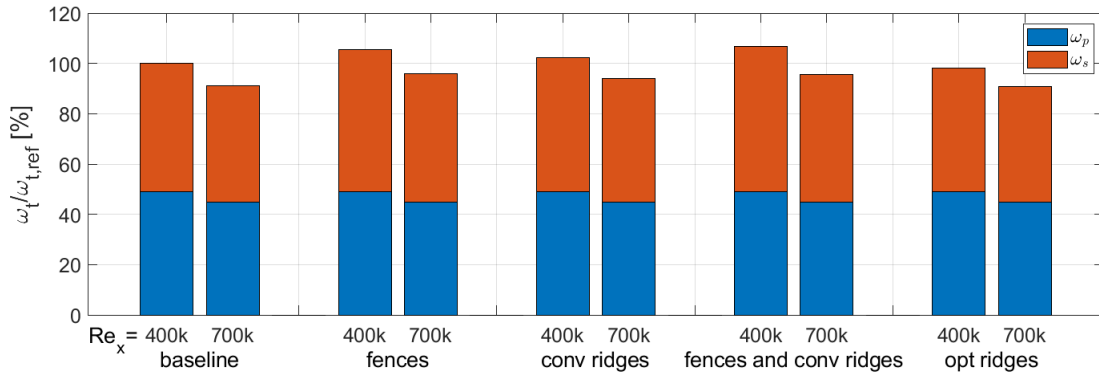


Figure 5.17: Bar chart of the normalized profile and secondary losses measured for the different cascade configurations at the two operating conditions.

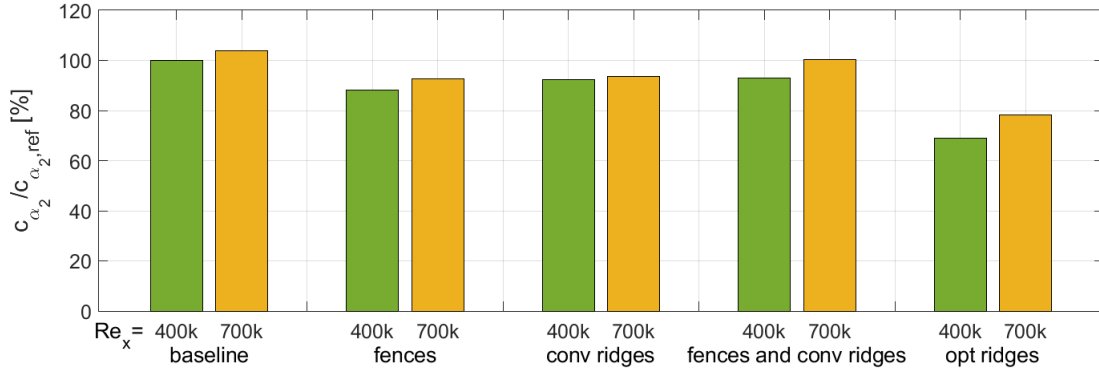


Figure 5.18: Bar chart of the normalized flow yaw angle coefficients measured for the different cascade configurations at the two operating conditions.

5.1.4 Further analysis with RANS and LES

To delve into how fences and ridges act to reduce the secondary structures within the NGV cascade, reference can be made to the numerical simulations conducted on the same cascade configurations by Morfo Design, using the experimentally measured boundary conditions presented previously [290]. See Appendix 2 for further details on the numerical setup [285]. After a comparison between numerical results (from both RANS and LES approaches) and experimental data to validate the use of low cost simulations in the design phase of the device geometries, streamline visualizations from high-fidelity simulations have been employed to gain deeper insight into the mechanisms by which these devices modify the flow within the channel. Specifically, for the NGV cascades, only the baseline, fenced, and conventional ridged cascade configurations have been analyzed numerically, serving as demonstrators of how these two different passive devices can mitigate secondary flow. Moreover, only the flow conditions at $Re_x = 400'000$ have been investigated.

The comparison between CFD results (low- and high-fidelity simulations) and experiments is reported in Figures 5.19 and 5.20 for the baseline, fenced, and conventional ridged cascade configurations, in terms of spanwise distributions of (a) total pressure loss coefficient and (b) cascade exit flow yaw angle, with the full blade span shown. For the baseline cascade, LES predictions of the spanwise $\overline{C_{pt}}$ distribution show very good agreement with experimental data, with only minor discrepancies near the casing and in the prediction of secondary flow penetration. Overall losses predicted by LES differ from experimental measurements by less than 3%. RANS captures the overall loss trend along the span, but predicts a deeper penetration of secondary flows, resulting in a shifted loss peak and a general overprediction of losses, particularly outside the casing region. This

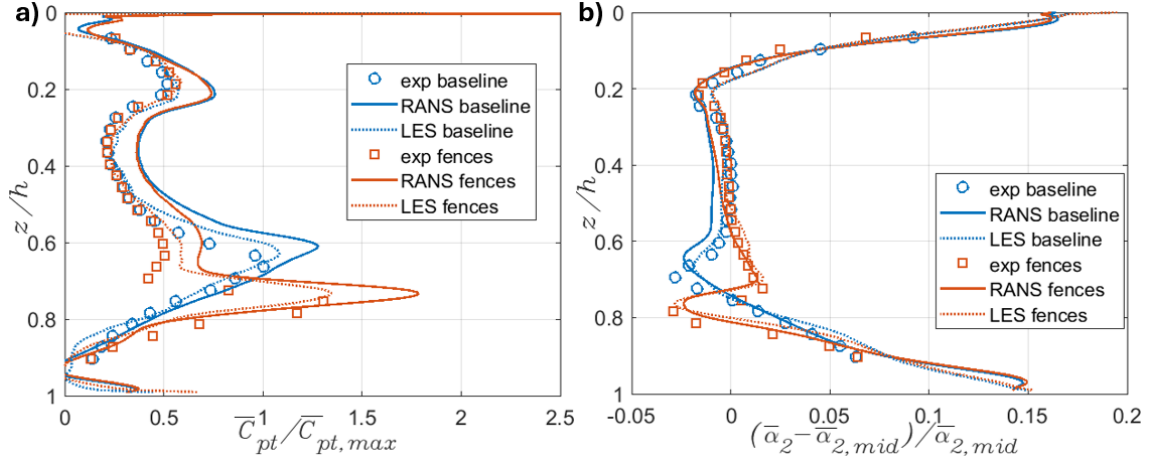


Figure 5.19: Experimental and numerical (RANS and LES) spanwise distributions of the normalized (a) total pressure coefficient $\bar{C}_{pt}$ and (b) yaw angle at the cascade outlet $\bar{\alpha}_2$ for the baseline and the fenced cascade configurations [284].

behavior is mainly attributed to an anticipated transition prediction, leading to an overestimation of shear stress and midspan losses, with overall losses exceeding experimental values by more than 10%. Similar trends are observed for the fenced (Figure 5.19.a) and conventional ridged cascades (Figure 5.20.a). LES accurately reproduces the experimentally observed effects of both devices, including loss peak reduction and redistribution along the span. RANS simulations capture the main impact of fences and ridges on sec-

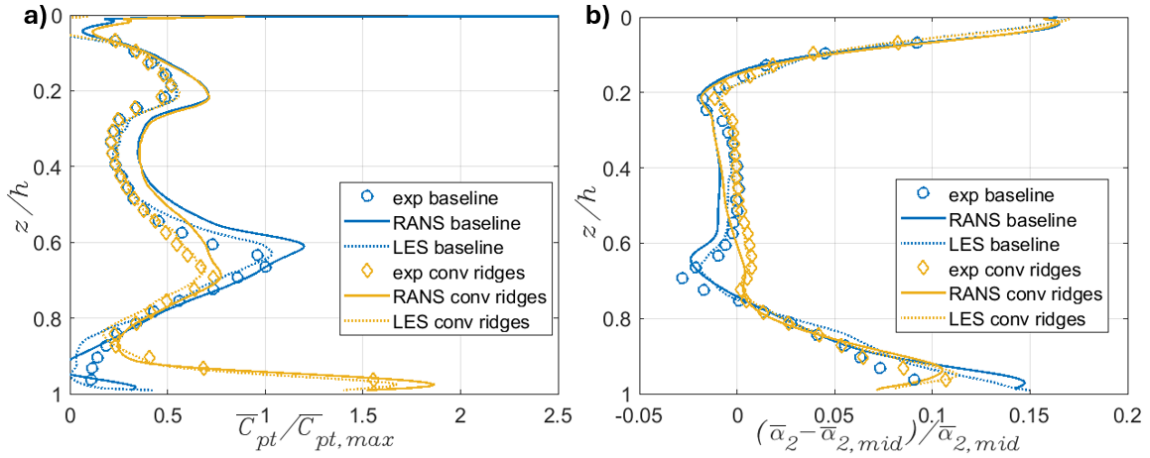


Figure 5.20: Experimental and numerical (RANS and LES) spanwise distributions of the normalized (a) total pressure coefficient $\bar{C}_{pt}$ and (b) yaw angle at the cascade outlet α_2 for the baseline and the conventional ridged cascade configurations [285].

ondary losses, although losses are generally overpredicted. For the conventional ridged configuration, RANS predicts the loss peak position more accurately than in the baseline case, showing improved agreement with both LES and experimental data. Regarding the pitchwise-averaged yaw angle distributions (Figures 5.19.b and 5.20.b), LES and RANS yield comparable results and show good agreement with experimental measurements. Both approaches slightly underpredict under-turning and overpredict over-turning for the baseline cascade, indicating deeper secondary flow penetration. The confining effect of fences on the passage vortex and the ability of endwall ridges to promote a more uniform exit flow are well captured by both low- and high-fidelity simulations. Minor discrepancies in RANS predictions are mainly associated with a slightly deeper secondary flow penetration, reflected in the shifted position of the minimum yaw angle.

Given that both RANS and LES are able to correctly predict the effect of fences and ridges on losses and flow angle deviation at the cascade exit, in agreement with experimental observations and reinforcing the choice of employing low cost approaches during the design phase of the device geometries, the high-fidelity simulations carried out by Morfo Design can be used to gain a deeper understanding of the flow phenomena within each cascade configuration. Starting with the baseline cascade, the time-averaged streamlines shown in Figure 5.21 provide insight into the development of the flow field inside the NGV channels. The classic effect of the passage vortex is clearly visible near the casing, where streamlines become almost tangential, and on the suction side of the blade, where a distinct deviation is observed (see Figure 5.21.a). This migration is driven by the pressure difference between the casing and the midspan of the blade, where lower pressure values are found. Focusing on the blade leading edge (Figure 5.21.b), the horseshoe vortex is found to be practically negligible, as previously mentioned. Indeed, the specific configuration of the meridional channel—characterized by high diffusion near the casing surface—combined with the presence of large fillets between the endwall and the blade, as well as the elevated values of the Eckerle and Awad [296] parameter, contributes to limiting the strength and development of the classical horseshoe vortex typically observed in other applications.

Hence, looking at the rear part of the cascade (Figure 5.21.c), only the effect of the passage vortex and the associated loss increase is visible. This distinct configuration of secondary flows and the relative significance of the primary vortices (passage vs. horseshoe) plays a crucial role in defining the optimal fence design. In cases where a significant horseshoe vortex is present, the optimal geometry is primarily designed to inhibit its initiation and the growth of the pressure side leg interacting with the passage vortex [22]. However, in the present case, where the passage vortex intensifies towards the aft section of the passage, fences applied on the SS have proven more effective in reducing the downwash on the blade surface and in containing flow distortions for the successive row. This is illustrated in Figure 5.22, where averaged LES results show how the installation of fences (bottom) modifies the development of secondary flows compared to the baseline

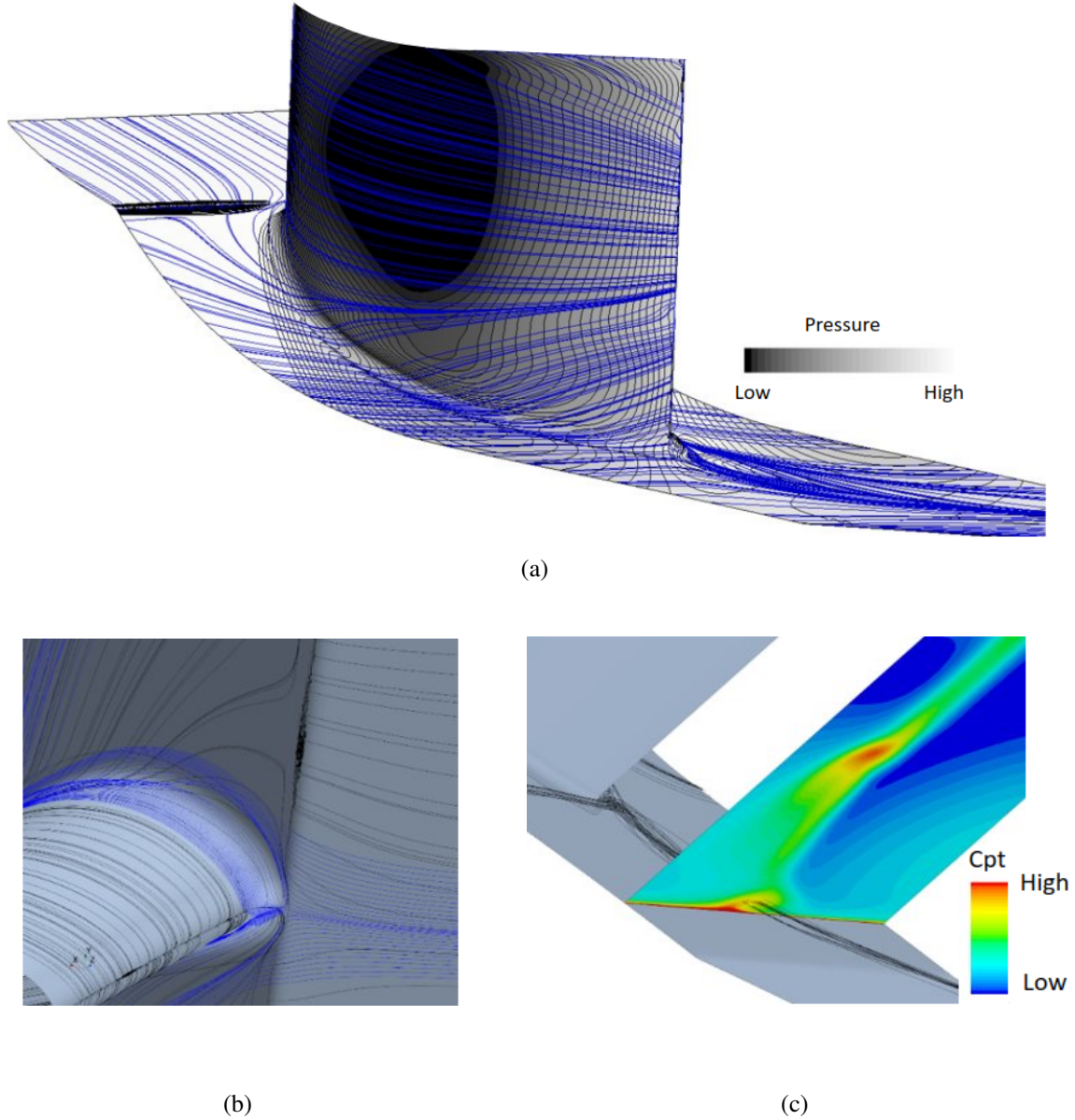


Figure 5.21: a) Time-averaged streamlines and contour of pressure on blade suction side and endwall. b) Streamlines close to blade LE and on endwall and blade surfaces. (c) C_{pt} contour on plane D_2 [284].

case (top).

For both configurations, the figures show total pressure losses on plane D_2 . The casing and blade surfaces display the static pressure contours, while the mid-vane plane section

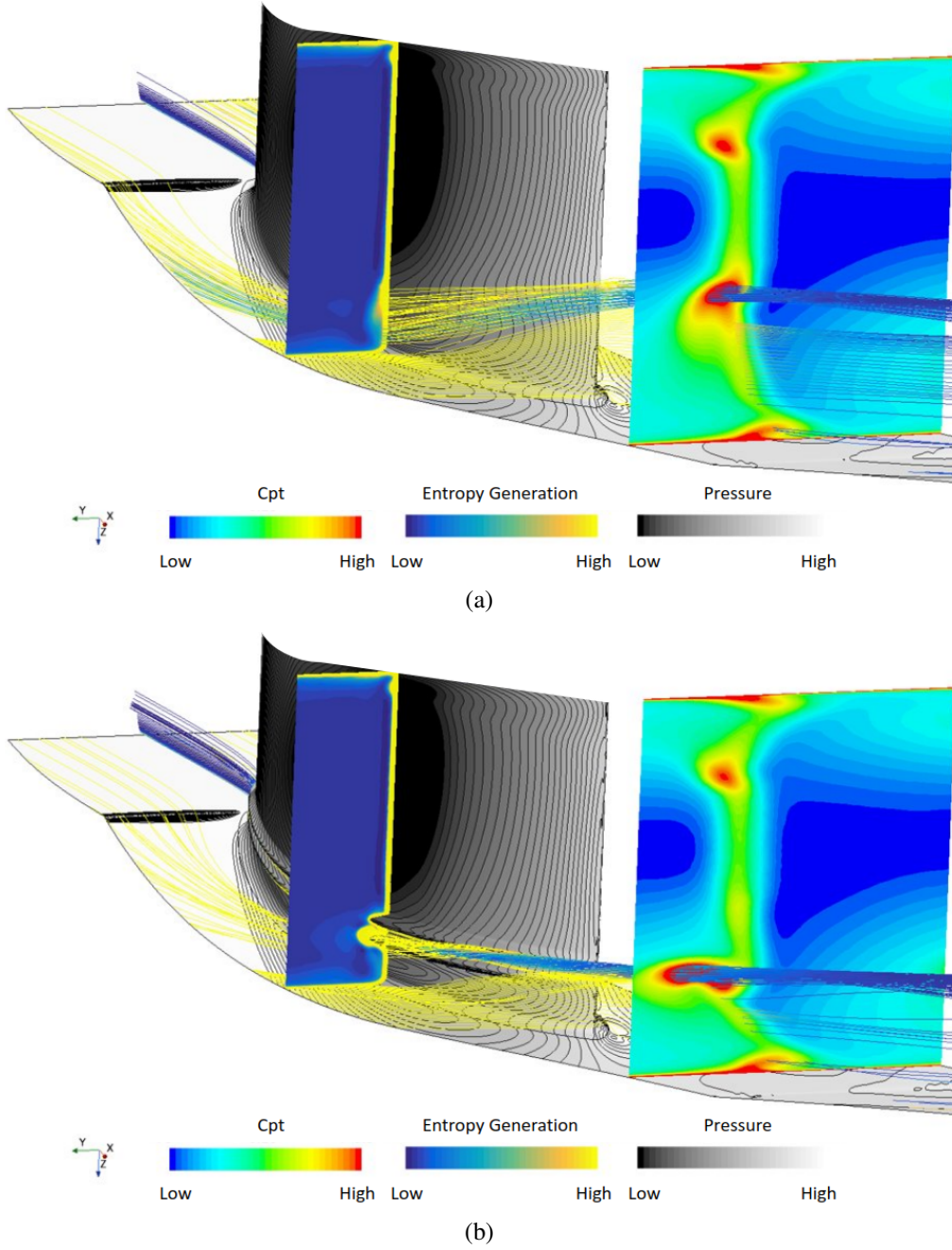


Figure 5.22: Streamlines visualizations for the (a) baseline and (b) fenced blade with the contour of pressure on endwall and blade suction side, the contour of C_{pt} on plane D_2 and the contour of local entropy generation on plane at $x/C_x = 0.6$. Streamlines are time-averaged and colored with local entropy generation [284].

highlights local entropy generation. Additionally, the streamlines associated with the secondary flow are colored according to the local entropy generation, computed following the procedure reported by Russo *et al.* [297], providing further insight into the flow dynamics and the losses induced by the secondary structures. In Figure 5.22.a, the migration of streamlines from the endwall toward the center of the channel, guided by the pressure field, is clearly evident. This secondary flow behavior is closely associated with regions of elevated losses. In contrast, Figure 5.22.b shows that the installation of fences on the suction side of the blade effectively restricts streamline migration, resulting in a more uniform downstream flow. However, in the regions where secondary flows interact with the fences, increased entropy generation occurs, leading to a stronger peak in total pressure losses for the fenced blade compared to the baseline, as highlighted in Figure 5.19. The effect of the fences on the time-averaged axial vorticity field, Ω_x , is illustrated in Figure 5.23. In the baseline case, the migration of streamlines from the casing toward the channel center is associated with an increase in axial vorticity. A similar trend is observed in the fenced case; however, as the streamlines are guided by the fence, they acquire opposite vorticity, generating a stronger region of negative vorticity in plane D_2 . On the fence side facing the casing, a region of positive vorticity develops, ultimately forming the distinct positive vorticity region observed in plane D_2 , which is absent in the baseline blade configuration.

Regarding the conventional ridged configurations, Figure 5.24 presents the time mean LES results, which allow understanding the mechanisms of the effectiveness of the EWR. Figure 5.24.a illustrates the development of the flow field around the blade for the baseline case (repeated here for completeness and to provide a direct comparison). The pressure losses at different axial positions indicate how these losses are generated: near the leading edge, they are confined to the boundary layer regions, while further downstream, two high loss regions emerge, eventually forming the typical pattern observed downstream of the blade in Figure 5.13. These loss peaks are caused by the migration of streamlines from the endwalls toward the channel center, as also depicted in the streamline visualization of Figure 5.21. Figure 5.24.c shows the significant impact of the EWR on the flow field. The migration of streamlines from the casing toward the midspan is substantially reduced, along with the associated loss peak. However, this improvement is accompanied by the formation of a high loss region near the casing. The mechanism through which the EWR effectively suppresses the migration of secondary flows from the casing toward the channel center is clearly illustrated by comparing Figures 5.24.b and 5.24.d. In particular, Figure 5.24.d shows how the streamlines interact with the first portion of the EWR. In the baseline configuration, the streamlines migrate from the casing toward the blade suction side, whereas in the EWR configuration, this migration is largely prevented by the ridges. The streamlines initially leap over them and subsequently flow through the gap formed between two adjacent ridges. The streamlines that still migrate toward the midspan are primarily those located between the suction side of the blade and the first ridge. The C_{pt} maps along the axial direction clearly reveal the development of a high loss region near

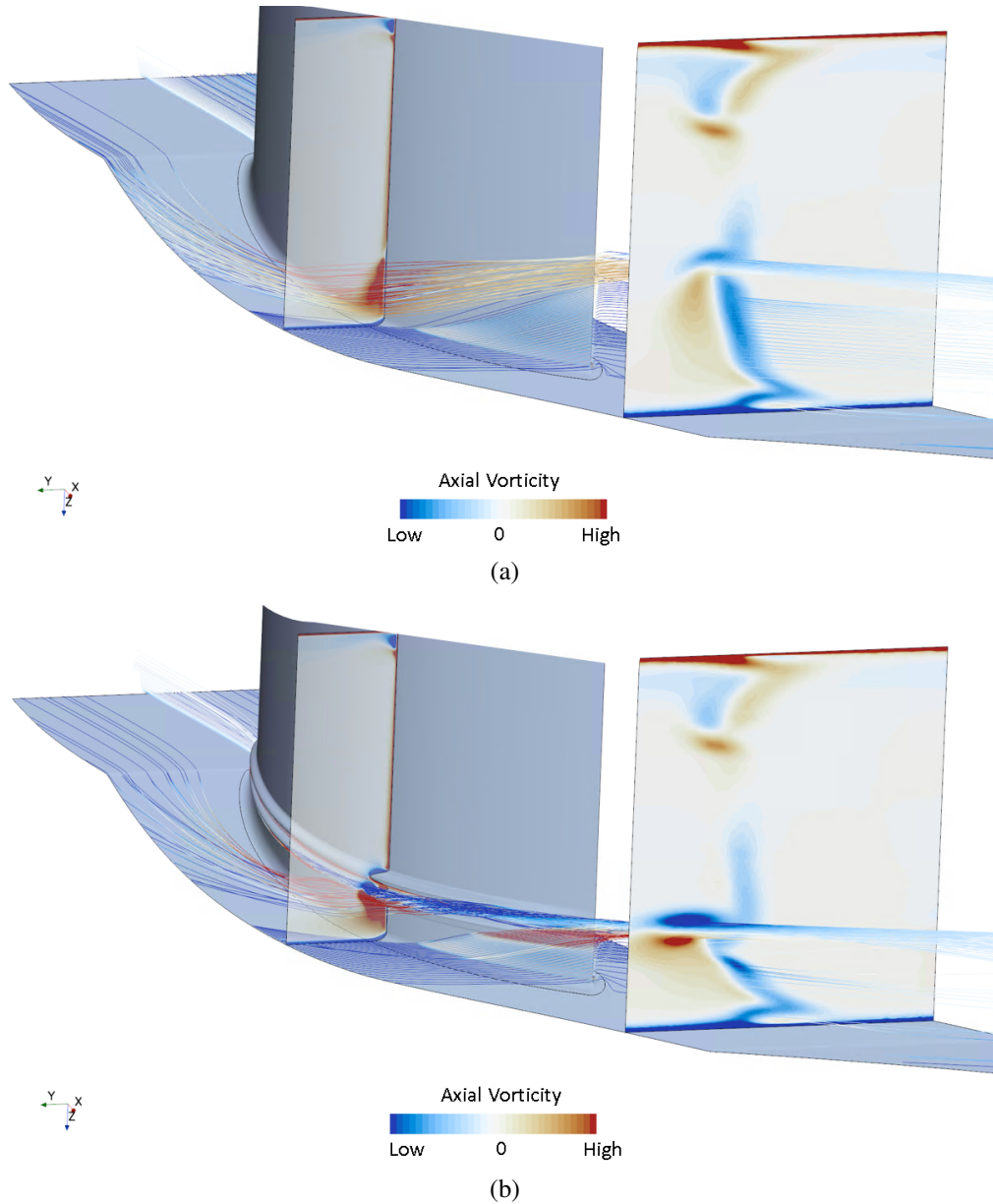


Figure 5.23: Contour of time-averaged axial vorticity on plane D_2 and on plane at $x/C_x = 0.6$ with streamlines colored with axial vorticity. (a) Baseline blade. (b) Fenced blade [285].

the endwall, also highlighted by the experimental maps, which show an increase in losses in these areas. As expected, this large loss region is primarily attributed to the interaction between the ridges and the wall boundary layers, further influenced by the inevitable increase in wetted surface area. This analysis hence provides a phenomenological expla-

nation of the mechanisms responsible for the reduction of downstream flow distortions, occurring at the expense of a slight increase in endwall losses.

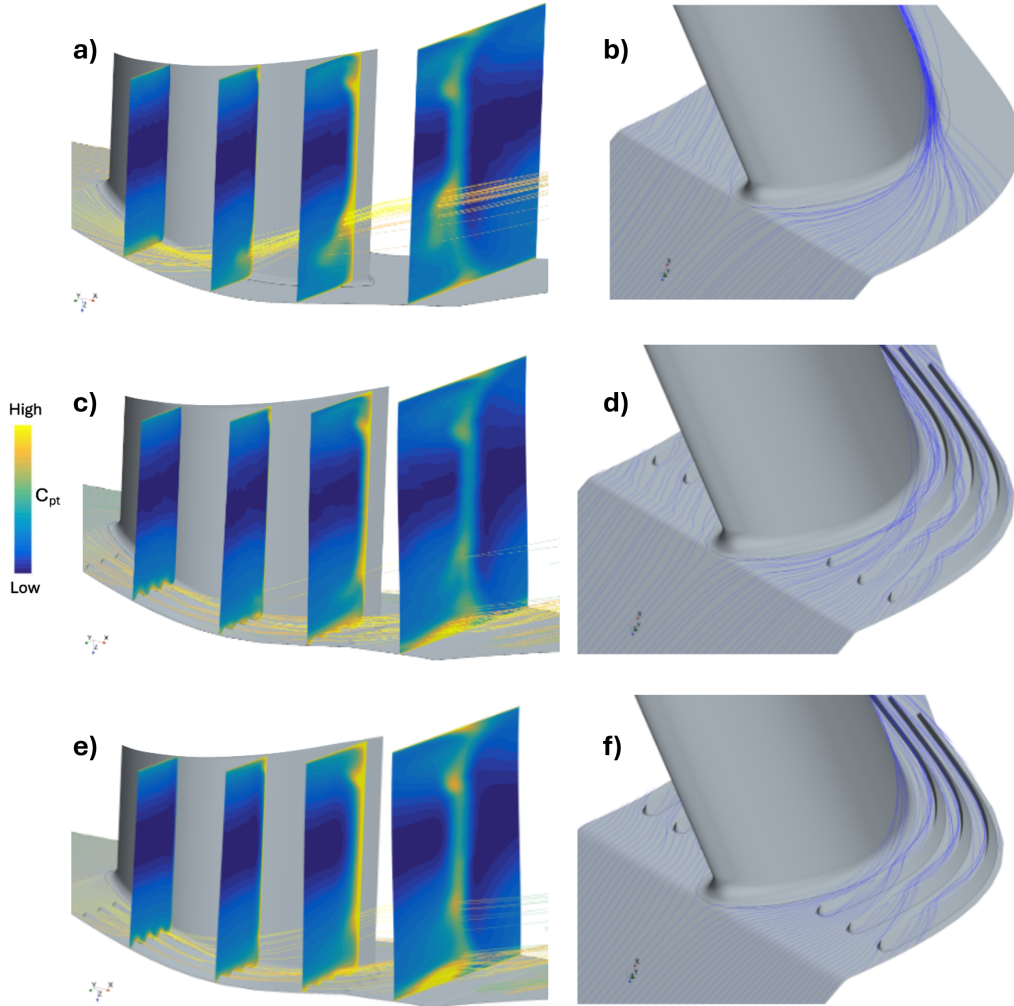


Figure 5.24: Left figures shows the pressure losses at different axial position together with streamlines colored by the pressure losses. Right figures show the details of the same streamlines in the first part of the blade. Top row corresponds to the LES of the baseline configuration, middle row to the LES of the EWR configuration, bottom row to the RANS of the EWR configuration. The time mean quantities are shown for the LES simulations [285].

Comparing the flow fields of the EWR configuration obtained from RANS simulations with those from LES (Figures 5.24.c–f) reveals a consistent scenario. In agreement with the observations for the loss distribution, the C_{pt} contours show high values near the casing as well as in regions affected by secondary flows. Furthermore, the interaction of stream-

lines with the ridges exhibits similar behavior across both computational approaches. A notable difference is that in the LES simulations, the streamlines tend to form a vortex immediately downstream of the ridges, whereas this effect is less pronounced in the RANS results.

To conclude, the comparison between numerical and experimental results validated the computational tools employed. Specifically, the assessment of low- and high-fidelity simulations demonstrates that low-cost approaches are capable of capturing the impact of these passive systems on secondary flow development, proving to be reliable tools for the design of such devices. For a closer agreement with experimental data, particularly regarding pressure losses, LES provides superior accuracy. Importantly, both simulations and experiments confirm the expected performance of the devices, showing that fences and ridges effectively reduce the flow distortions induced by secondary structures at the cascade exit, thereby creating favorable conditions for enhanced performance of downstream blade rows. Furthermore, this analysis provides valuable insights into the mechanisms responsible for mitigating secondary flow distortions, as well as the slight increase in losses observed in the fenced configuration. These findings are fully consistent with the results obtained in the stage environment, where the introduction of fences leads to an overall increase in stage efficiency of approximately 0.2%, despite a slight rise in NGV losses, while the use of endwall ridging results in an overall stage efficiency improvement of about 0.25%, without increasing NGV losses. Indeed, numerical simulations conducted in a representative annular stage environment show that both devices effectively mitigate flow non-uniformities at the NGV outlet (in line with the literature [22]), leading to a less distorted inflow to the rotor and ultimately improving its efficiency. Although the detailed results of these calculations cannot be disclosed due to confidentiality constraints, their consistency with the experimental observations reinforces confidence in the beneficial effect of fences and ridges on overall engine performance.

To this aim, further investigations have been conducted in parallel with the present study to evaluate the ease of manufacture and installation of these devices in a real engine. Extensive manufacturing and thermo-mechanical tests confirmed the robustness of the devices, addressing local heat transfer concerns through optimized geometries. Fillets were integrated to prevent stress concentrations, and high MRL testing demonstrated that both fences and ridges are straightforward to manufacture and install. Ongoing efforts aim to advance these strategies to higher TRLs for practical engine applications.

All these analyses are reported in several of the author's works: see [284] for the fence analysis and [285] for the study on ridges. A complete overview of all cascade configurations is provided in [298].

5.2 Unshrouded LPT Results

5.2.1 Characterization of the Inlet Boundary Conditions

Analogously to the NGV case, the experimental campaign began with a detailed characterization of the inlet boundary conditions. In Figure 5.25, the turbulence values measured via HWA at midspan of points P_1 , P_2 , and P_3 (indicated with * in Figure 4.17.a) are reported in blue as a function of x_t/d . For reference, the turbulence decay correlation by Roach (Eq. 4.4) is plotted as a red dashed line, which passes visibly through the experimental points. The inlet total pressure map, obtained by traversing the Kiel probe in

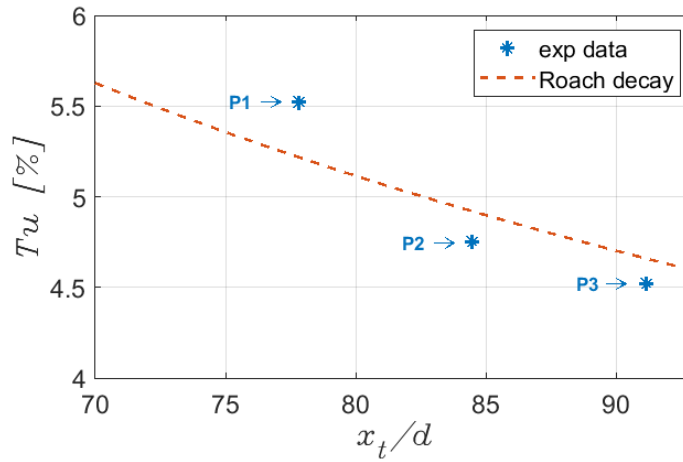


Figure 5.25: Turbulence decay along the flow direction.

D_1 along the pitchwise and spanwise directions, is shown in Figure 5.26.a along y/s and z/h , and normalized by its maximum value. For this cascade, since the inlet and outlet channels are symmetric, only half of the span is shown for all maps of the flow entering and exiting the cascade, as well as for the corresponding spanwise distributions. The map clearly highlights a significantly thinner boundary layer compared to that developing upstream of the NGV cascade. From its pitchwise-averaged distribution along the blade height, shown in Figure 5.26.b, the boundary layer thickness can be quantitatively estimated as $\delta/h = 0.12$.

It should be noted that these conditions are necessary to properly set up the numerical simulations conducted on the same cascade by Morfo Design, in order to gain a deeper insight into the phenomena developing in this LPT configuration (see Appendix 2). All other inlet boundary conditions, such as the spanwise distributions of the integral time and length scales, dissipation rate, and rate of dissipation per unit turbulent kinetic energy, were obtained using Eq. 5.1 (following the method proposed in the author's work [290]), by inputting the meridional channel height, the midspan values of the turbulence

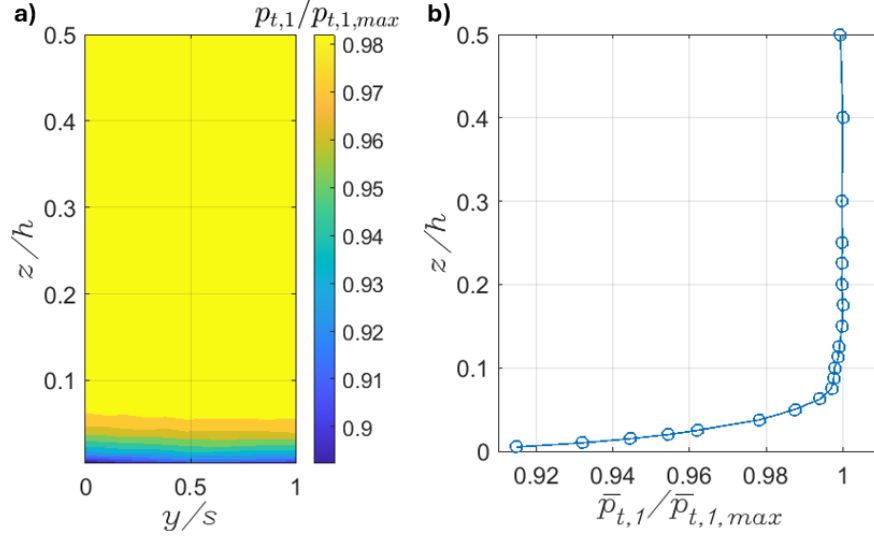


Figure 5.26: (a) Contour plot of $p_{t,1}/p_{t,1,max}$ along the pitchwise and spanwise directions and (b) its pitchwise averaged distribution along the spanwise direction.

quantities measured with the HWA at P_1 , and the boundary layer thickness estimated from the upstream total pressure distribution just presented.

5.2.2 Flat Tip

The experimental campaign began with the analysis of the unshrouded LPT cascade equipped with a flat tip, used as a reference case to assess the effectiveness of the squealer and squealer-winglet tip configurations in mitigating the tip leakage vortex and its associated losses. Since each tip configuration is tested for variations in tip gap, cooling flow rate, and Reynolds number, the discussion of the reference condition starts with the characterization of the no-gap case, obtained by bringing the blade tip into contact with the plexiglass casing. For this condition, the effect of Reynolds number is also isolated and analyzed. The analysis then proceeds with the evaluation of the impact of varying the tip gap height.

5.2.2.1 No Gap Characterization

In Figure 5.27.a, the spanwise distributions of the inlet total pressure are shown for both Reynolds numbers tested, made non-dimensional by the maximum inlet total pressure at the nominal condition ($Re_x = 200'000$). Figure 5.27.b reports the blade loadings measured at midspan for both Reynolds numbers, made non-dimensional by the maximum pressure coefficient at the nominal Reynolds number, $c_{p,max}$. Interestingly, as visible from the almost complete overlap of the distributions, the Reynolds number appears to have a

negligible effect on both the $p_{t,1}$ and c_p distributions. This suggests that no significant variations in viscous related phenomena occur on the endwall BL upstream of the cascade or on the blade surface BL, likely due to the high turbulence level induced by the turbulence grid, as also observed in the NGV case.

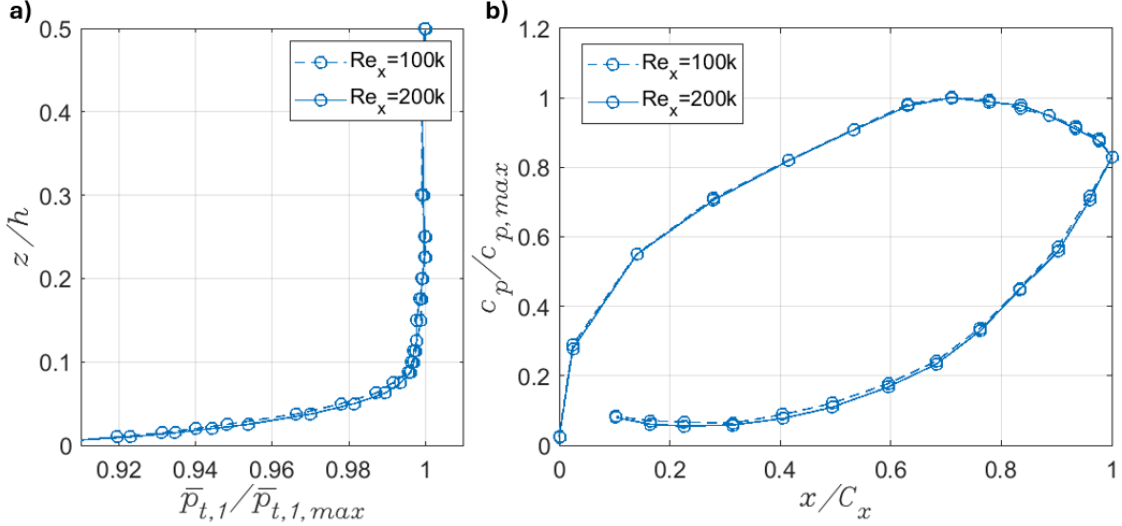


Figure 5.27: (a) Spanwise distributions of the total pressure at the inlet of the baseline cascade and (b) blade loading distributions at midspan for both Reynolds number conditions.

Moving on to the maps of the total pressure loss coefficient, C_{pt} , obtained with the Kiel probe, and the velocity vectors measured by the five-hole probe at the cascade outlet plane D_2 , it is possible to investigate the secondary flows at the cascade exit and to evaluate the influence of the Reynolds number on them. The C_{pt} distributions for (a) $Re_x = 100'000$ and (b) $Re_x = 200'000$ are shown in Figure 5.28. These distributions, as well as all the subsequent C_{pt} maps presented, are made non-dimensional by the maximum value, $C_{pt,max}$, measured for the flat tip cascade at $Re_x = 200'000$ with $\tau/C_x = 1\%$, to ensure a higher level of comparability across all the conditions included in the experimental test matrix. As it can be observed from the different shades of blue, the lower Reynolds number results in a wider wake with higher losses compared to the nominal condition, while the core of the secondary losses, located on the blade suction side at approximately $z/h = 0.92$, appears unaffected. This loss core is clearly associated with a strong three-dimensionality of the outflow from the cascade, suggesting the presence of a vortical secondary structure rotating clockwise, which can be identified as a passage vortex, given that the present condition corresponds to $\tau/C_x = 0$.

To provide a more quantitative estimation of the impact of secondary flows on the flat tip cascade at the no-gap condition and the influence of Reynolds number on them,

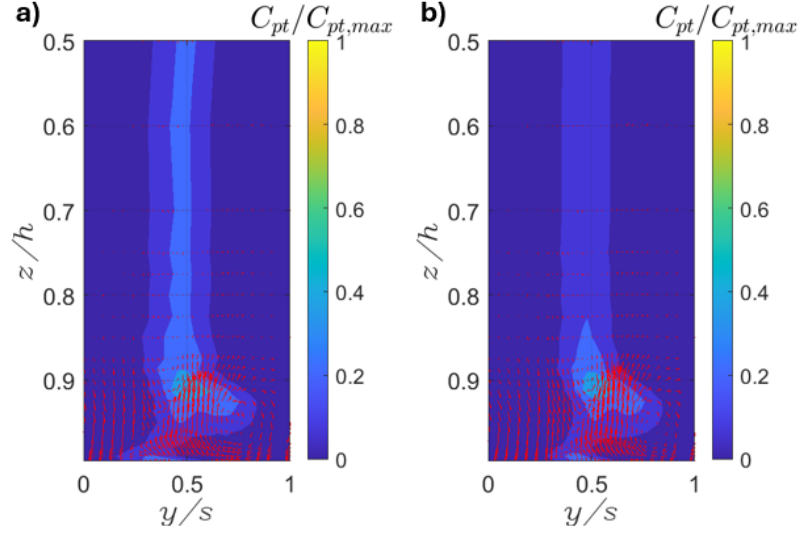


Figure 5.28: Total pressure coefficient maps downstream of the baseline cascade at (a) $Re_x=100'000$ and (b) $Re_x=200'000$.

Figure 5.29.a reports the normalized spanwise distributions of the total pressure loss coefficient, $\overline{C_{pt}}$, obtained by pitchwise-averaging the previously shown maps. For comparability between the two Reynolds numbers tested, the $\overline{C_{pt}}$ distributions are normalized by the maximum total pressure loss coefficient, $\overline{C_{pt,max}}$, obtained for the nominal condition ($Re_x=200'000$). A distinct peak in the total pressure loss coefficient is visible at $z/h = 0.9$, corresponding to the core of the passage vortex previously discussed. However, its penetration is limited due to the high aspect ratio of the cascade. For $z/h > 0.8$, a nearly constant value of $\overline{C_{pt}}$ is observed, associated with the development of the blade wake, i.e., the profile losses. Regarding the Reynolds number effect, a lower Re_x results in an almost rigid upward shift of the $\overline{C_{pt}}$ distribution compared to the nominal condition, leading to a noticeable increase in profile losses. In contrast, the secondary losses (which can be evaluated by the difference between total and profile losses) appear unchanged, both in terms of magnitude and in the location and penetration of the loss peak along the blade height. In Figure 5.29.b, the normalized distributions of the difference between the local pitchwise-averaged yaw angle ($\overline{\alpha_2}$) and its midspan value ($\overline{\alpha_{2,mid}}$) obtained for the nominal condition are reported. A strong over-turning near the endwall and under-turning in the vicinity of the $\overline{C_{pt}}$ peak are observed, corresponding to the presence of the passage vortex. Interestingly, in this case, the Reynolds number has an almost negligible effect on the flow yaw angle distribution, as suggested by the nearly complete overlap of the distributions.

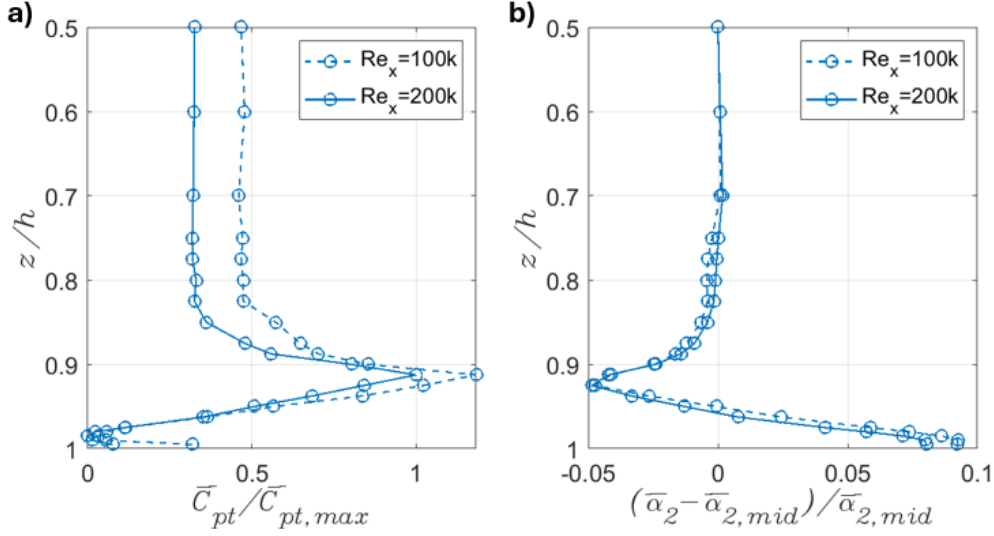


Figure 5.29: Distributions of the (a) pitchwise averaged total pressure coefficient and (b) flow yaw angle deviation downstream of the baseline cascade for both Reynolds numbers.

5.2.2.2 Gap Variation Characterization

Using the system of pull and push screws, it is then possible to analyze the effect of opening the tip gap on the creation of a leakage flow over the flat tip of the blades. This analysis is presented only for the nominal Reynolds number ($Re_x = 200'000$). However, in order to correctly compare the results among the different cases, it is first necessary to ensure that the boundary conditions remained unchanged before and after opening the gap, thereby confirming that the plexiglass casing remained rigidly attached to the rest of the wind tunnel, as described in the experimental setup. For this reason, Figure 5.30.a shows the spanwise distributions of the inlet total pressure measured with the Kiel probe in plane D_1 for each tip gap height investigated, normalized by the maximum value $\bar{p}_{t,1,max}$ of the no-gap case for comparability. Similarly, Figure 5.30.b reports the aerodynamic loadings measured at midspan of the LPT cascade for the different tip gaps, normalized by the maximum pressure coefficient $c_{p,max}$ obtained in the no-gap condition.

From the almost perfect superimposition of the inlet total pressure and midspan pressure coefficient distributions, it can be confirmed that the boundary conditions of the cascade remained unchanged with the opening of the tip gap, allowing an accurate comparison of the results obtained. Figure 5.31 shows the 2D C_{pt} maps along the tangential and spanwise directions for the different tip gap heights analyzed. The maps are made non-dimensional by the maximum C_{pt} obtained for the flat tip cascade configuration at $\tau/C_x = 1\%$ under the nominal condition. From left to right, the effect of increasing the tip gap is evident. Moving from the $\tau/C_x = 0$ condition, already presented in Figure 5.28.b, to higher τ/C_x values, it can be observed that while the region influenced solely by the

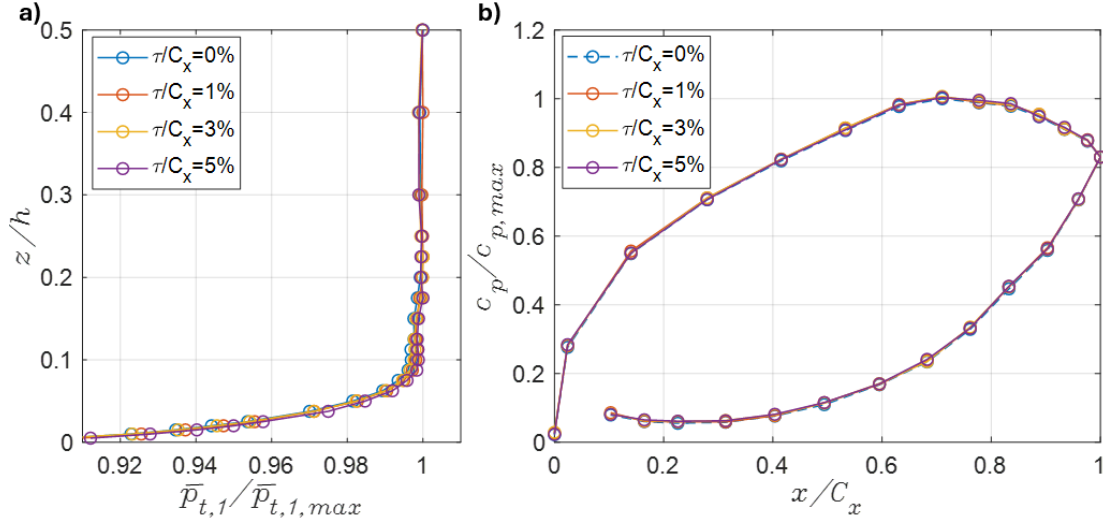


Figure 5.30: (a) Spanwise distributions of the total pressure at the inlet of the baseline cascade and (b) blade loading distributions at midspan for all tip gap heights investigated.

wake remains unchanged, the secondary flows developing at the blade tip are significantly altered. Specifically, at $\tau/C_x = 1\%$, two distinct loss cores are visible: one with clockwise rotation, corresponding to the passage vortex from the no-gap case, slightly shifted toward the midspan at $z/h = 0.9$ and of increased intensity; and a second, more intense loss region located near the case at approximately $z/h = 0.99$, characterized by counter-clockwise rotation relative to the PV, which can be associated with the tip leakage vortex formed due to the presence of the tip gap. Moreover, the intensity of the secondary velocity vectors that characterized the passage vortex in the no-gap case is significantly reduced, giving way to a pronounced three-dimensionality associated with the tip leakage vortex. As the tip gap increases to $\tau/C_x = 3 - 5\%$, the leakage flow inevitably intensifies, resulting in a TLV of progressively higher intensity and larger size, with a substantial increase in the three-dimensionality of the exit flow from the cascade. Interestingly, at $\tau/C_x = 5\%$ the TLV becomes so dominant that the PV is completely dissipated. Consequently, the overall penetration of the secondary flow system is reduced, becoming largely confined to the TLV development near the case, compared to the cases where both the PV and TLV are present.

For a more quantitative assessment of the effect of tip gap opening on secondary flows, the spanwise distributions of $\overline{C_{pt}}$ and $(\overline{\alpha_2} - \overline{\alpha_{2,mid}})/\overline{\alpha_{2,mid}}$ are reported in Figure 5.32. From the total pressure loss coefficient distributions for each tip gap height analyzed, made non-dimensional by the maximum value of the no-gap condition $\overline{C_{pt,max}}$ (Figure 5.32.a), it can be observed that while the no-gap case features a single peak at $z/h = 0.92$ corresponding to the passage vortex, at $\tau/C_x = 1\%$ two distinct loss peaks appear. The first peak, located at the same spanwise position, is still associated with the PV. How-

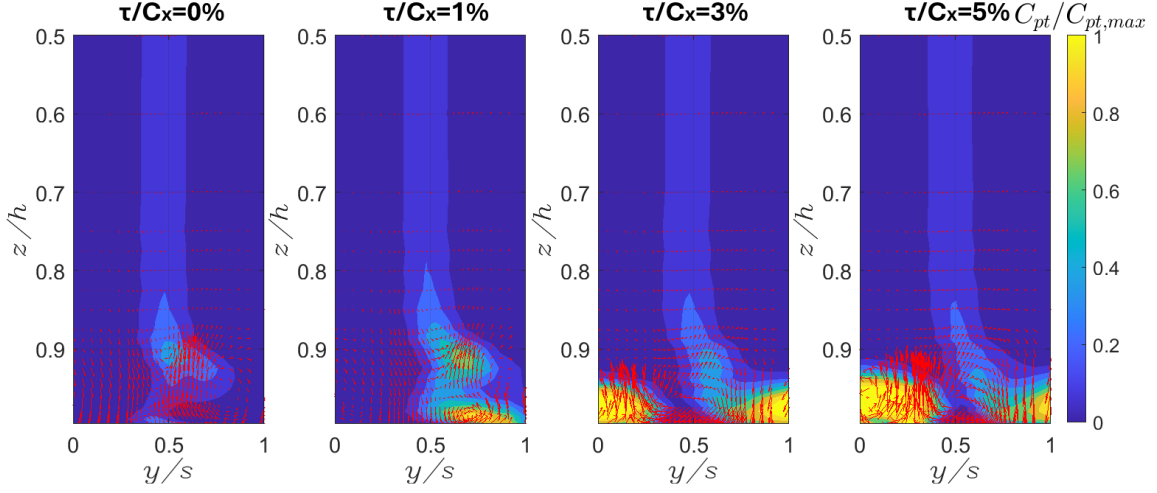


Figure 5.31: Total pressure coefficient maps downstream of the baseline cascade for increasing tip gap heights from left to right.

ever, its spanwise extent is increased, and its intensity is nearly doubled compared to the PV developed in the no-gap condition, induced by the opposing rotation of the tip leakage vortex, whose significantly higher loss peak is visible at $z/h = 0.99$. As the tip gap increases, the loss peak associated with the TLV grows due to the higher leakage flow from the pressure side to the suction side in the tip region, consistently with the trends observed in the C_{pt} maps. Its spanwise penetration also slightly increases because of the larger clearance, shifting toward $z/h = 0.98$ at $\tau/C_x = 3\%$ and toward $z/h = 0.97$ at $\tau/C_x = 5\%$. Conversely, the intensity of the loss peak associated with the PV decreases. Indeed, when the tip gap is opened, the endwall boundary layer that feeds the PV is less affected by the pressure gradient imposed by the blade, and the strong TLV contributes to the dissipation of the PV. Additionally, it should be noted that, as already observed from the C_{pt} maps, the losses associated with the wake region remain essentially unchanged across the different tip gap conditions. The normalized yaw angle distributions at the cascade exit, shown in Figure 5.32.b, reveal that the over-turning region near the case, visible in the no-gap condition, progressively shifts toward the midspan. In the meanwhile, an extended under-turning region induced by the presence of the TLV develops near the endwall. Conversely, the under-turning peak associated with the PV in the no-gap condition at $z/h = 0.92$ decreases in intensity and moves away from the endwall, especially for $\tau/C_x = 3 - 5\%$, where the PV is completely dissipated.

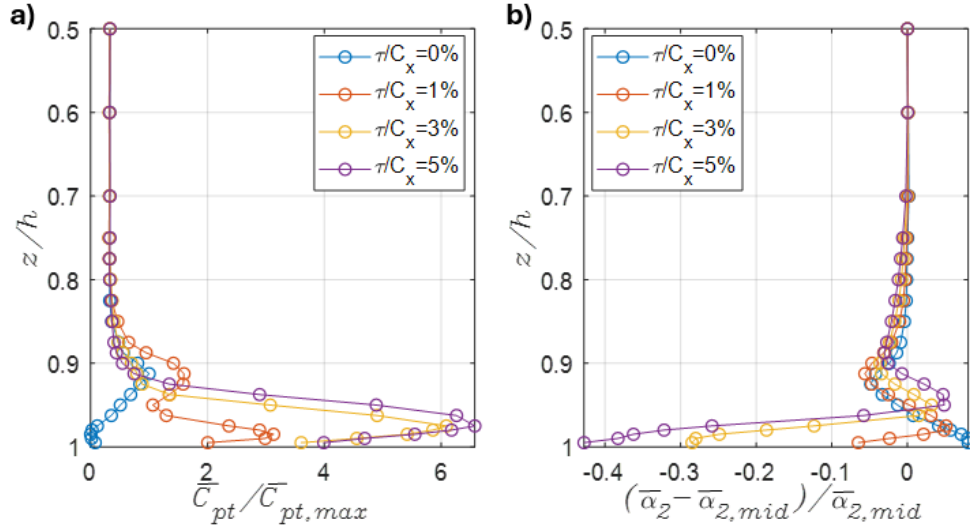


Figure 5.32: Distributions of the (a) pitchwise averaged total pressure coefficient and (b) flow yaw angle deviation downstream of the baseline cascade for all tip gap heights investigated.

5.2.3 LPT with secondary flow control devices

After characterizing the flat tip cascade configuration, the blade tips were modified by installing the squealer and, subsequently, the squealer-winglet configurations. For each tip configuration, the effect of gap variation on the TLV was analyzed, followed by an investigation of the influence of cooling and Reynolds number on the tip losses. The latter was studied both in the absence of a gap and in the presence of a tip clearance combined with a coolant mass flow rate. As previously done, to ensure high accuracy and comparability between the results, the experimental setup was verified to remain unchanged before and after replacing the tip inserts and in the presence or absence of cooling flow. For this purpose, the boundary conditions were analyzed and reported in terms of normalized inlet total pressure and midspan blade loadings in Figure 5.33.a and Figure 5.33.b, respectively.

The almost perfect superimposition of the distributions confirms that the experimental setup remained unchanged, as intended with the use of add-on tip inserts and the plexi-glass case rigidly attached to the wind tunnel. This ensures enhanced accuracy and comparability of the results obtained across the different cascade configurations.

5.2.3.1 Squealer Tip

In Figure 5.34, all the C_{pt} maps measured for the LPT equipped with the squealer tip are shown. The effect of the tip gap height is presented along the columns, from left to right corresponding to $\tau/C_x = 0$, $\tau/C_x = 1\%$, $\tau/C_x = 3\%$, and $\tau/C_x = 5\%$, while the

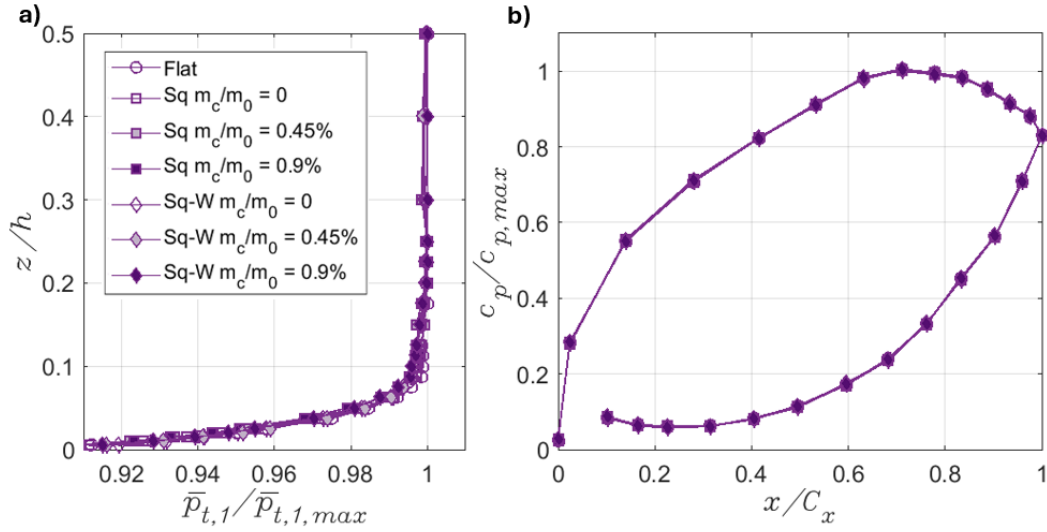


Figure 5.33: (a) Spanwise distributions of the inlet total pressure and (b) blade loading distributions at midspan for all tip configurations and coolant flow rates investigated (at $\tau/C_x = 5\%$).

effect of the coolant mass flow rate is shown along the first three rows, from top to bottom corresponding to $\dot{m}_c/\dot{m}_0 = 0$, $\dot{m}_c/\dot{m}_0 = 0.45\%$, and $\dot{m}_c/\dot{m}_0 = 0.9\%$. These conditions refer to the nominal Reynolds number ($Re_x = 200'000$). The fourth and last row illustrates the effect of Reynolds number, showing the C_{pt} maps at $Re_x = 100'000$ for each investigated tip gap height, and, where the tip gap region is present, including the highest coolant mass flow rate ($\dot{m}_c/\dot{m}_0 = 0.9\%$). For comparability, all maps are made non-dimensional by the maximum total pressure loss coefficient $C_{pt,max}$ of the flat tip case at $\tau/C_x = 1\%$ and $Re_x = 200'000$.

Starting with the squealer tip results in the absence of coolant, it can be observed that increasing the tip gap height leads to a behavior similar to that already observed for the flat tip. As the tip gap increases, the TLV strengthens, while the PV, when moving from the no-gap condition to $\tau/C_x = 1\%$, initially increases—likely due to the counter-rotation induced by the TLV and then disappears at larger tip gap heights. Comparing these initial C_{pt} maps of the squealer tip with those of the flat tip, it is evident that at no gap the two distributions are almost identical, since the outer structure of the squealer matches that of the flat tip. At $\tau/C_x = 1\%$, the effect of the internal cavity structure on the TLV is still minor due to the small clearance, and the maps for the flat and squealer cases appear very similar. However, at $\tau/C_x = 3 - 5\%$, a slight reduction in both extent and intensity of the TLV at the blade tip becomes noticeable, thanks to the internal cavity of the squealer, which likely recirculates the tip leakage flow, preventing it from fully spilling over the SS of the blade. Moreover, at the maximum tip gap height, the TLV appears to be partially dissipated (at $y/s = 0.35$, near the case), probably due to a strong separation induced

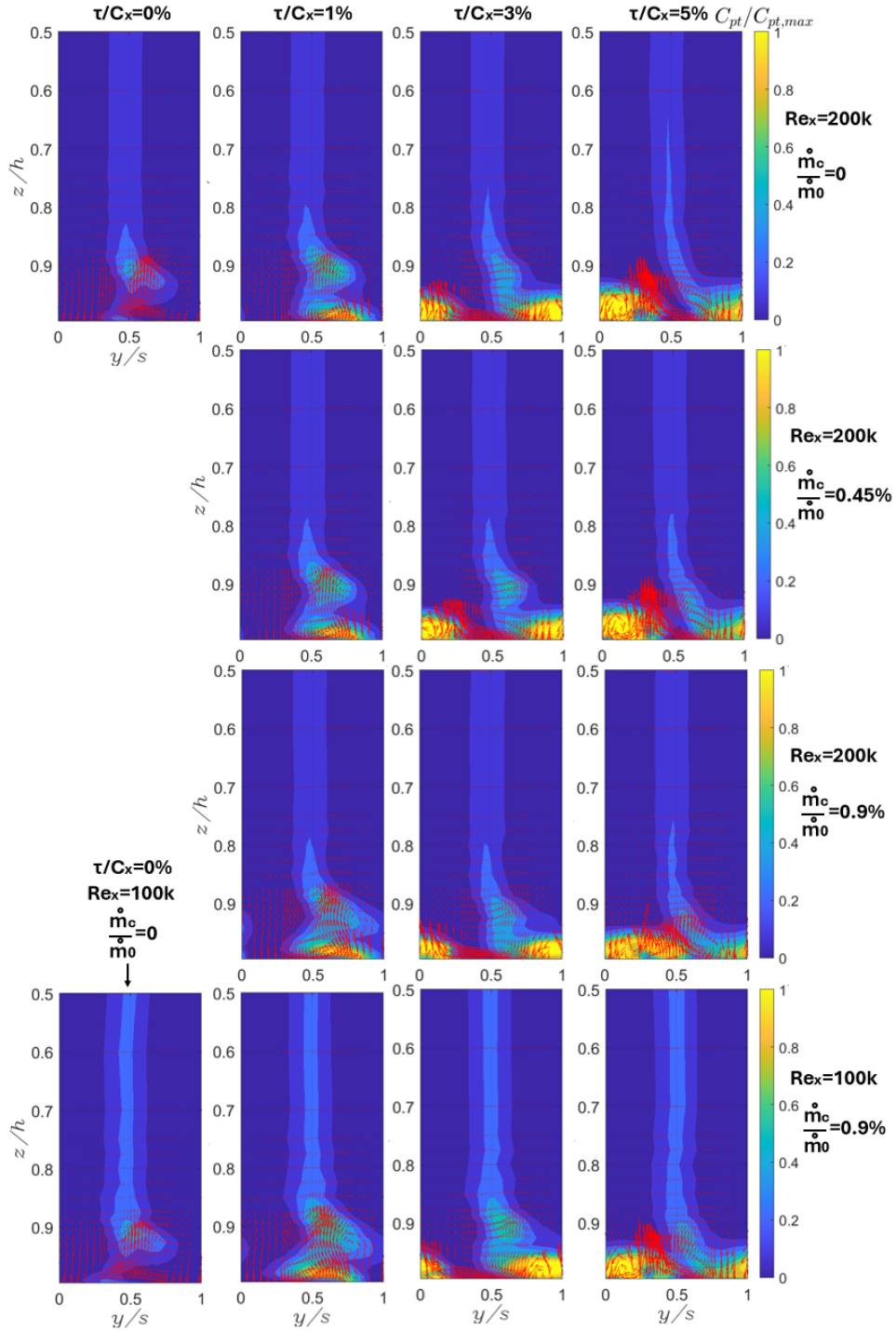


Figure 5.34: Normalized C_{pt} maps for all investigated cases with squealer tip.

by the squealer cavity structure near the PS rim, which attenuates the rotation of the TLV. Nevertheless, the TLV still induces a strong three-dimensionality of the flow at the cascade exit, particularly for the case with $\tau/C_x = 5\%$.

Looking at the case with a low coolant mass flow injected from the tip, no significant differences are observed compared to the no-coolant condition, except for a slight intensification of the PV, likely induced by the blockage effect of the coolant in the tip clearance (especially at small tip gaps), which makes the endwall boundary layer more sensitive to the full pressure difference imposed by the blade. Additionally, this coolant flow, interacting with the TLV, appears to further enhance the three-dimensionality of the exit flow. At the highest coolant flow rate, a marked change in the effect of the tip gap height is observed compared to the no-coolant condition. At $\tau/C_x = 1\%$, the addition of coolant appears detrimental, as it increases both size and intensity of the PV, and to a lesser extent also the TLV, likely due to an excessive coolant flow beyond what is required to achieve a blocking effect at this small tip gap. Conversely, at larger tip gap heights, the TLV is significantly reduced in both extent and intensity, and the flow exhibits reduced three-dimensionality. At the maximum tip gap height, the coolant flow even appears to split the TLV core into two adjacent vortical structures near the PS rim. Finally, considering the effect of the Reynolds number, it can be observed that, similarly to the flat tip no-gap condition, the PV remains largely insensitive to changes in Re_x , whereas the wake region shows a noticeable increase. In contrast, when combined with a high coolant mass flow ratio, the PV grows in both size and intensity at lower Reynolds numbers. Meanwhile, the TLV remains largely unchanged, albeit slightly shifted toward the blade SS. Interestingly, at the maximum tip gap height, the TLV appears noticeably reduced in both size and intensity near the PS rim. Nevertheless, the strong intensity of the secondary velocity vectors indicates a significant deviation of the exiting flow.

For a more quantitative analysis of the various effects on the squealer tip, the previously shown C_{pt} maps can be examined by studying the impact of tip gap height (displayed along the columns) for each tested coolant mass flow rate (presented along the rows), or by analyzing the effect of varying the coolant flow rate (shown along the rows) for each tip gap height in terms of spanwise distributions of the total pressure loss coefficient and exit flow yaw angle distortion. In Figure 5.35, the spanwise distributions of $\overline{C_{pt}}/\overline{C_{pt,max}}$ (on the left) and $(\overline{\alpha_2} - \overline{\alpha_{2,mid}})/\overline{\alpha_{2,mid}}$ (on the right) are presented for different tip gap heights and coolant flow rates ($\dot{m}_c/\dot{m}_0 = 0$, $\dot{m}_c/\dot{m}_0 = 0.45\%$, $\dot{m}_c/\dot{m}_0 = 0.9\%$); results at the lowest Reynolds number are reported in Figure 5.36.

For the no-coolant case, it can be observed that at $\tau/C_x = 1\%$, the squealer tip produces a relatively contained TLV, but its rotation leads to a pronounced increase in the PV, both in intensity and size, resulting in a slight increase in PV penetration toward the midspan. At $\tau/C_x = 3\%$, the PV is reduced; its characteristics are similar to the no-gap case, likely due in part to the reduced pressure gradient imposed by the blade on the flow with the presence of a tip gap, and in part because it is dissipated by the increasingly intense TLV. At the

maximum tip gap height, the PV entirely disappears, while the TLV, due to the larger

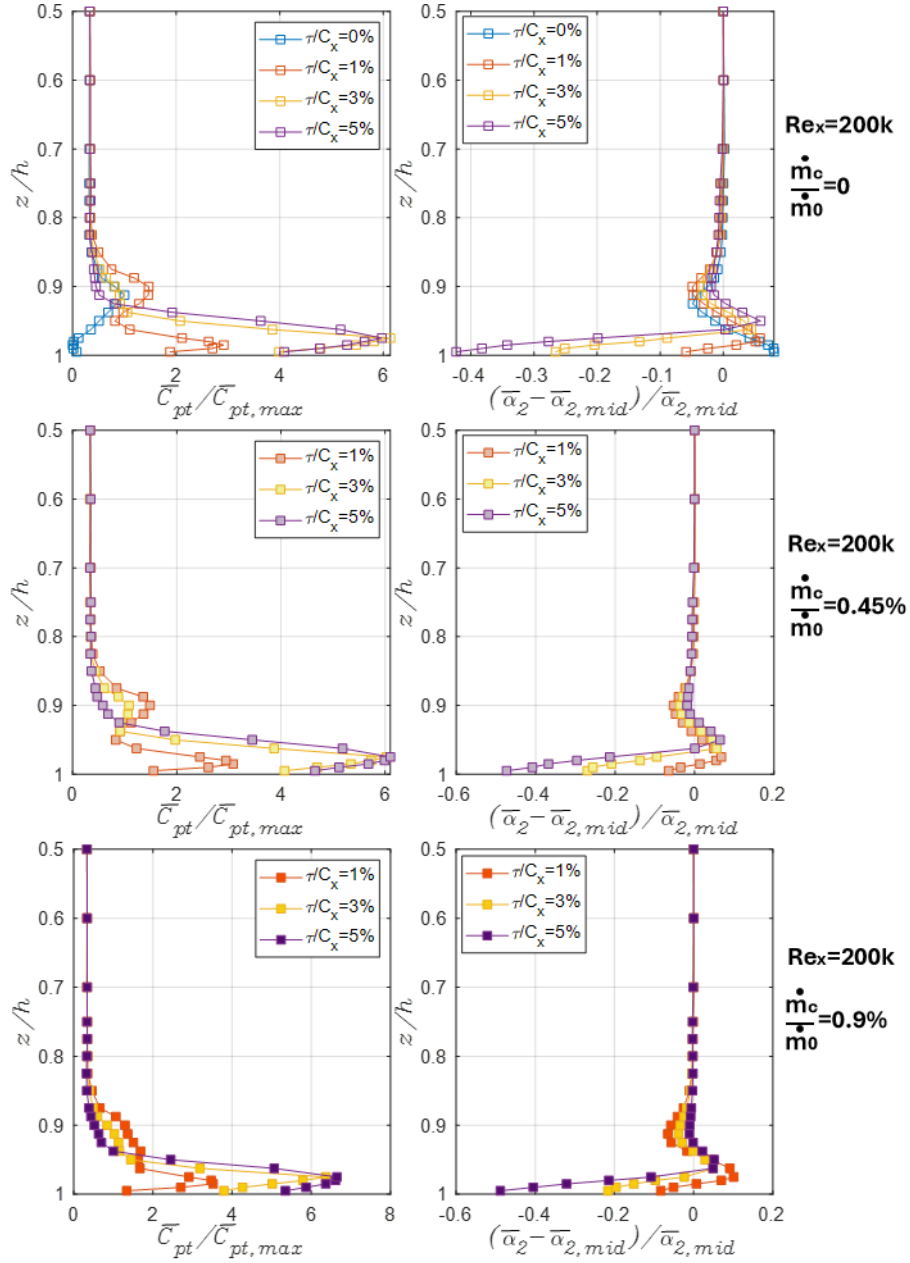


Figure 5.35: Tip gap height variation effect on the squealer tip in terms of spanwise distributions of (left) normalized total pressure loss coefficient $\bar{C}_{pt}/\bar{C}_{pt,max}$ and (right) exit flow yaw angle deviation $(\bar{\alpha}_2 - \bar{\alpha}_{2,mid})/\bar{\alpha}_{2,mid}$ for each coolant flow rate tested at the highest Reynolds number.

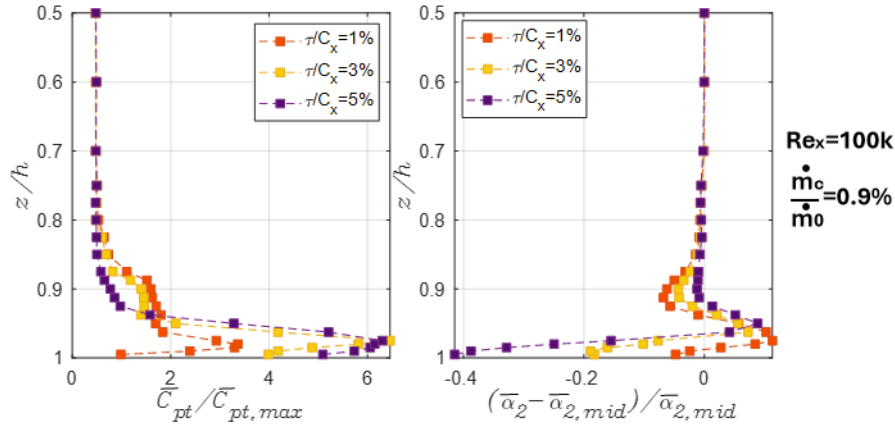


Figure 5.36: Tip gap height variation effect on the squealer tip in terms of spanwise distributions of (left) normalized total pressure loss coefficient $\bar{C}_{pt} / \bar{C}_{pt, max}$ and (right) exit flow yaw angle deviation $(\bar{\alpha}_2 - \bar{\alpha}_{2, mid}) / \bar{\alpha}_{2, mid}$ at the highest coolant flow rate and the lowest Reynolds number tested.

clearance, occupies a slightly larger portion of the blade height. Examining the exit flow yaw angle distortion, it can be seen that as the tip gap height increases, the TLV generates an increasingly large region of under-turning, whereas the reduced over-turning remains almost constant with increasing τ/C_x . At the same time, the under-turning induced by the PV decreases with increasing tip gap, with its peak shifting toward midspan.

The effect of tip gap height on $\bar{C}_{pt} / \bar{C}_{pt, max}$ and $(\bar{\alpha}_2 - \bar{\alpha}_{2, mid}) / \bar{\alpha}_{2, mid}$ under the lowest coolant flow ratio remains similar to the no-coolant case. In contrast, for the highest coolant flow rate, the PV region around $z/h = 0.92$ becomes more extended along the blade height compared to the low- and no-coolant cases, likely because the casing boundary layer is more influenced by the pressure difference between the pressure and suction sides. Moreover, at $\tau/C_x = 1\%$, a slight increase in over-turning is observed relative to the two previous cases. Interestingly, as already noted from the C_{pt} maps, the effect of tip gap height at the maximum coolant flow rate for the lowest Reynolds number exhibits the same trend as the maximum coolant flow case at the nominal Reynolds number. It can also be observed that the TLV $\bar{C}_{pt}$ peaks are almost comparable to those at the highest Reynolds number.

Observing the distributions of $\bar{C}_{pt} / \bar{C}_{pt, max}$ (top) and $(\bar{\alpha}_2 - \bar{\alpha}_{2, mid}) / \bar{\alpha}_{2, mid}$ (bottom), which highlight the effect of varying the coolant flow for $\tau/C_x = 1\%, 3\%, 5\%$ in Figure 5.37, it can be seen that for nearly all tip gap heights the no-coolant and low-coolant conditions are almost coincident. The only condition that stands out is the high-coolant case, which, however, differs less from the other coolant conditions at $\tau/C_x = 3\%$ and $\tau/C_x = 5\%$. The most significant differences are observed at $\tau/C_x = 1\%$, where, consistently with the previously shown C_{pt} maps, a more intense peak associated with the

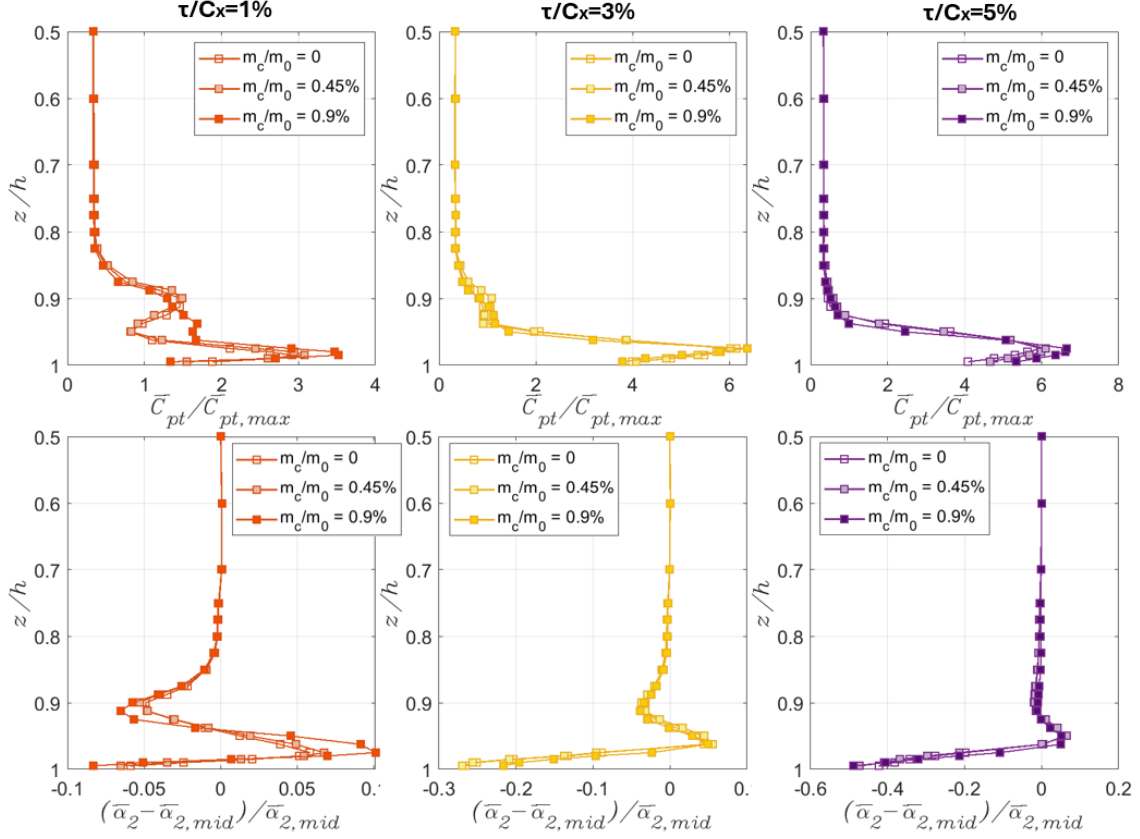


Figure 5.37: Coolant mass flow rate variation effect on the squealer tip in terms of spanwise distributions of (top) normalized total pressure loss coefficient $\bar{C}_{pt}/\bar{C}_{pt,max}$ and (bottom) exit flow yaw angle deviation $(\bar{\alpha}_2 - \bar{\alpha}_{2,mid})/\bar{\alpha}_{2,mid}$ for each tip gap height tested.

TLV appears at $z/h = 0.97$, along with a more extensive structure along the blade height attributable to the PV. The increased intensity of both secondary structures consequently leads to a TLV induced region of much stronger over-turning at $z/h = 0.96$, while the PV is responsible for enhanced under-turning at $z/h = 0.91$.

5.2.3.2 Squealer-Winglet Tip

Similarly to the squealer tip, Figure 5.38 shows all the $C_{pt}/C_{pt,max}$ maps measured for the different cases analyzed with the LPT equipped with the squealer-winglet tip, varying the tip gap height (columns) and the coolant mass flow rate (rows). Compared to all the conditions shown in the first three rows, obtained at the nominal Reynolds number ($Re_x = 200'000$), the fourth and final row presents the C_{pt} maps at $Re_x = 100'000$ for each of the investigated tip gap heights and, where the tip gap region is present, also in the presence of the highest coolant mass flow rate, $m_c/m_0 = 0.9\%$.

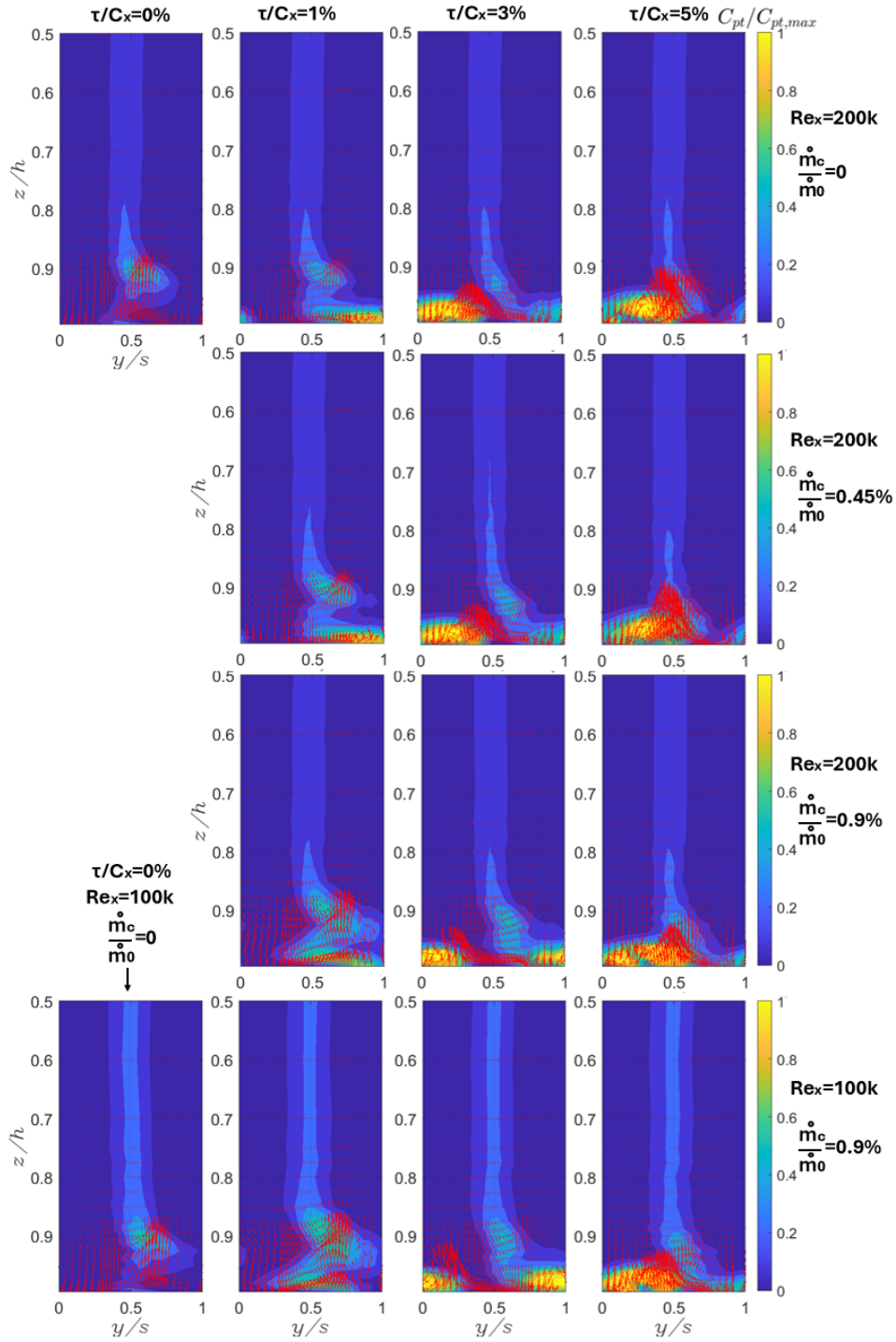


Figure 5.38: Normalized C_{pt} maps for all investigated cases with squealer-winglet tip.

Starting by examining the no-coolant conditions, it can be observed that, with increasing tip gap height, unlike the flat and squealer tip cases, the PV only decreases in both intensity and size. The TLV, on the other hand, behaves similarly to the other two tip geometries, increasing with the tip gap. Interestingly, comparing these first C_{pt} maps with the flat and squealer tip cases, the PV in the no-gap condition appears slightly larger. With the gap opened, at $\tau/C_x = 1\%$, the PV is reduced and the TLV is very confined near the casing and slightly more extended in the tangential direction, with its core positioned around $y/s = 0.9$ —hence much farther from the blade SS compared to the flat and squealer tips, where the TLV core was located around $y/s \approx 0.6$. Interestingly, this displacement of the vortex core becomes even more pronounced at $\tau/C_x = 3 - 5\%$, where a TLV of lower intensity compared to the flat and squealer cases, although characterized by higher three-dimensionality of the exiting flow, is positioned near the blade PS.

Similarly to the squealer tip case, no significant differences are observed between the no-coolant and low-coolant conditions. At high coolant flow, however, an increase in the PV is observed for all tip gap heights, likely induced by a coolant blockage effect. Due to its clockwise rotation, this PV significantly counteracts the TLV at the blade tip, also reducing the flow deviation caused by the TLV. Moreover, as in the squealer case, at the maximum tip gap the TLV loss core again splits into two smaller adjacent cores. Finally, examining the maps at the lower Reynolds number, it can be noted (similarly to the squealer tip) that only the wake region changes in the no-gap condition, while the PV exhibits slight variation with increasing tip gap at the high coolant mass flow rate. In contrast, the TLV remains almost unchanged compared to the high-coolant conditions at the nominal Reynolds number, except at the maximum tip gap, where a single TLV core appears to recombine, albeit with slightly reduced intensity. However, the secondary velocity vectors appear longer, indicating more pronounced flow deviation than in the $Re_x = 200'000$ case.

Similarly, to analyze the effect of varying the tip gap height for each coolant mass flow rate, Figure 5.39 shows the distributions of $\overline{C_{pt}}/\overline{C_{pt,max}}$ (on the left) and $(\overline{\alpha_2} - \overline{\alpha_{2,mid}})/\overline{\alpha_{2,mid}}$ (on the right) for each τ/C_x tested and each $\dot{m}_c/\dot{m}_0$ investigated; results at the lowest Reynolds number are reported in Figure 5.40.

In the case of the squealer-winglet tip, it can be observed that in the absence of coolant, the $\overline{C_{pt}}$ peak associated with the PV at $z/h = 0.91$ remains unchanged between the no-gap condition and $\tau/C_x = 1\%$. Interestingly, it then partially disappears at larger tip gaps, partly because the blade pressure gradient acting on the flow is reduced, and partly because it is dissipated by the TLV, which—similarly to what was observed for the flat and squealer tips—increases in intensity as the tip gap height increases. Specifically, for $\tau/C_x = 3 - 5\%$, the TLV exhibits essentially the same $\overline{C_{pt}}$ peak, although at the maximum tip gap it becomes slightly more extended along the span due to the larger clearance. As a result, the deviation of the exit flow yaw angle is characterized by an under-turning associated with the PV at $z/h = 0.91$, which remains nearly identical between the no-gap and $\tau/C_x = 1\%$

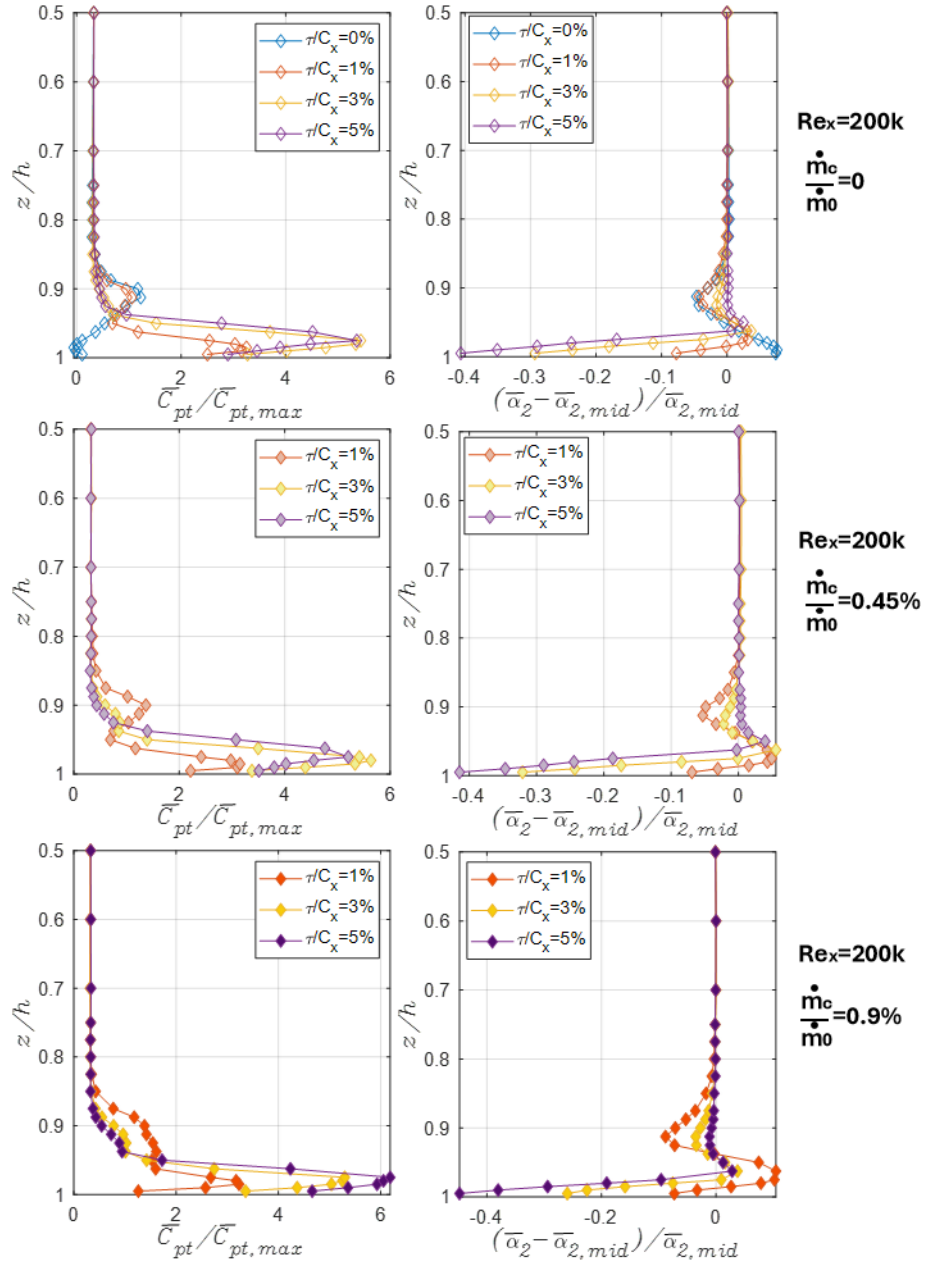


Figure 5.39: Tip gap height variation effect on the squealer-winglet tip in terms of spanwise distributions of (left) normalized total pressure loss coefficient $\bar{C}_{pt}/\bar{C}_{pt,max}$ and (right) exit flow yaw angle deviation $(\bar{\alpha}_2 - \bar{\alpha}_{2,mid})/\bar{\alpha}_{2,mid}$ for each coolant flow rate tested at the highest Reynolds number.

conditions, and then progressively decreases with increasing tip gap height, eventually

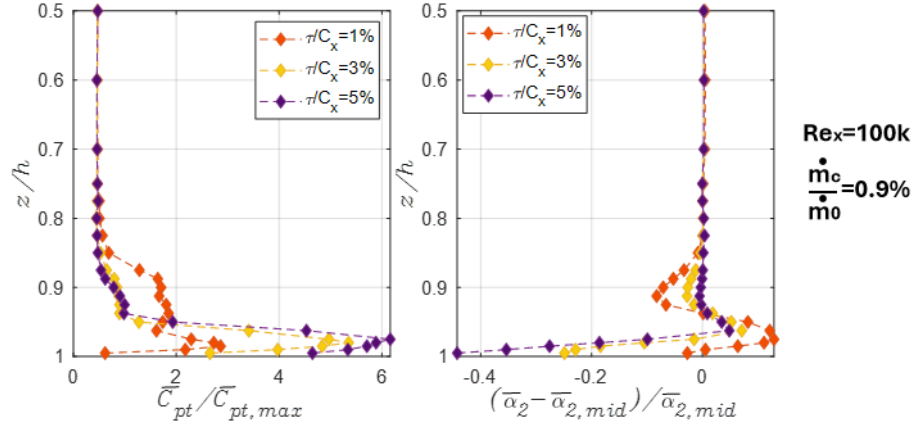


Figure 5.40: Tip gap height variation effect on the squealer-winglet tip in terms of spanwise distributions of (left) normalized total pressure loss coefficient $\bar{C}_{pt}/\bar{C}_{pt,max}$ and (right) exit flow yaw angle deviation $(\bar{\alpha}_2 - \bar{\alpha}_{2,mid})/\bar{\alpha}_{2,mid}$ at the highest coolant flow rate and the lowest Reynolds number tested.

disappearing at the maximum gap. Conversely, the over-turning caused by the PV near the casing in the no-gap condition diminishes as the tip gap increases and is replaced by an under-turning induced by the TLV, which becomes more pronounced as the tip gap height grows. As observed for the squealer tip, the no-coolant and low-coolant conditions remain essentially unchanged, with only minor differences at $\tau/C_x = 3\%$: a smaller PV peak at $z/h = 0.92$, much weaker than the one found at $\tau/C_x = 1\%$, and, at the maximum tip gap, a slightly less pronounced TLV related $\bar{C}_{pt}$ peak at $z/h = 0.97$. These trends result in a slightly stronger PV induced under-turning at $\tau/C_x = 3\%$ around $z/h = 0.92$, and a less pronounced TLV induced under-turning at $\tau/C_x = 5\%$ near the casing. Moving to the high-coolant cases, a PV driven loss region appears around $z/h = 0.92$, significantly more extended in the spanwise direction (likely due to a stronger blockage effect). This region is more intense at $\tau/C_x = 1\%$ but remains clearly visible also at $\tau/C_x = 3 - 5\%$. A strong variation in the TLV loss peak is also observed, with its magnitude increasing markedly with tip gap height (due to the increased leakage mass flow). Overall, the normalized exit flow yaw angle distributions are not particularly different from those of the low-coolant conditions, except for a more extended PV induced under-turning region and a notably stronger over-turning at $\tau/C_x = 1\%$, caused by the simultaneous presence of strong PV and TLV. As already noted for the squealer tip, at the lower Reynolds number the $\bar{C}_{pt}$ and $\bar{\alpha}_2$ distributions closely resemble those obtained at nominal Reynolds number for the same coolant flow rate. Substantial differences arise only at $\tau/C_x = 1\%$, where the PV increases in intensity and size and the TLV decreases in intensity, in agreement with the trends identified from the C_{pt} maps.

The effect of the coolant mass flow rate for each tip gap height is instead presented in

terms of $\overline{C_{pt}}/\overline{C_{pt,max}}$ (top) and $(\overline{\alpha_2} - \overline{\alpha_{2,mid}})/\overline{\alpha_{2,mid}}$ (bottom) in Figure 5.41.

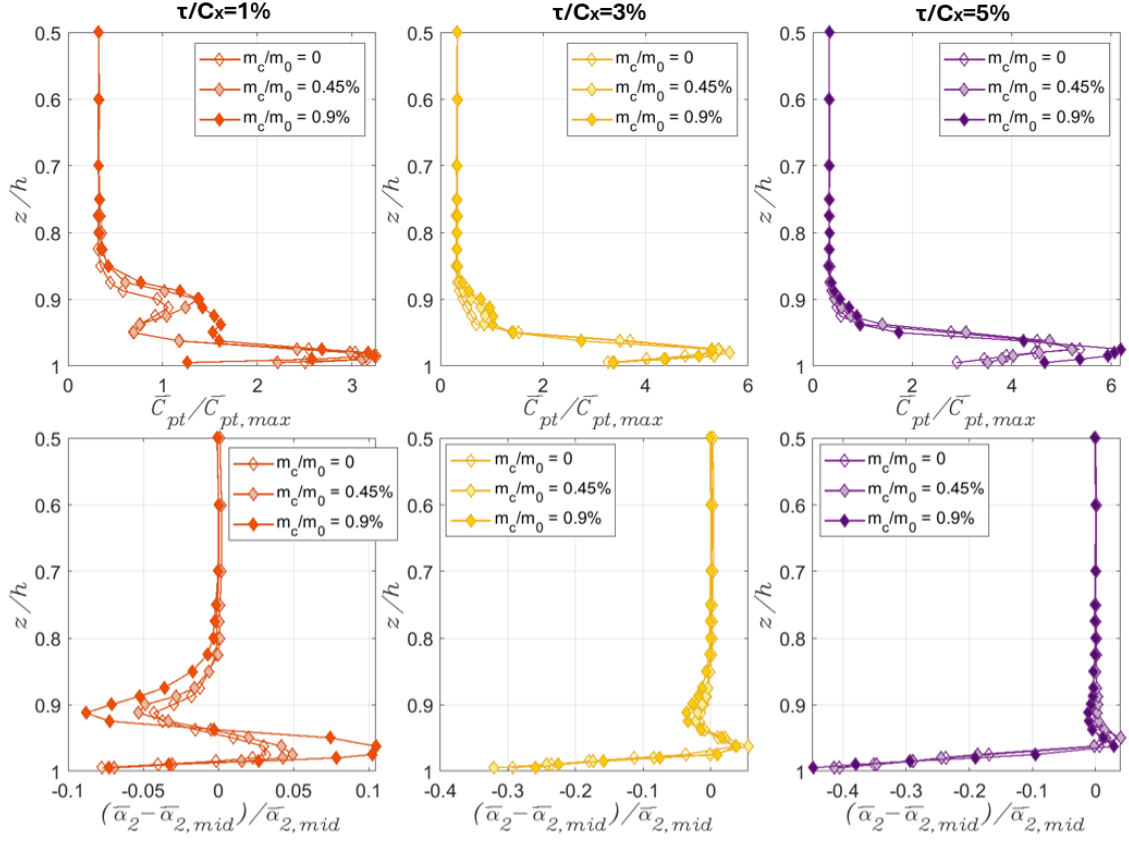


Figure 5.41: Coolant mass flow rate variation effect on the squealer-winglet tip in terms of spanwise distributions of (top) normalized total pressure loss coefficient $\overline{C_{pt}}/\overline{C_{pt,max}}$ and (bottom) exit flow yaw angle deviation $(\overline{\alpha_2} - \overline{\alpha_{2,mid}})/\overline{\alpha_{2,mid}}$ for each tip gap height tested.

For the squealer-winglet configuration, the influence of the coolant mass flow rate appears more articulated. At $\tau/C_x = 1\%$, increasing the coolant flow leads to a higher $\overline{C_{pt}}$ peak associated with the PV for $\dot{m}_c/\dot{m}_0 = 0.45\%$, whereas at $\dot{m}_c/\dot{m}_0 = 0.9\%$ the most prominent effect is the spanwise enlargement of the PV loss region. These trends are consistently reflected in the exit flow yaw deviation, where a higher coolant flow enhances the under-turning at $z/h = 0.92$ and increases the over-turning at $z/h = 0.96$. The impact of the coolant on the PV progressively weakens as the tip gap height increases. At $\tau/C_x = 3\%$ only a slight increase in the PV peak is observed, while at $\tau/C_x = 5\%$ its influence becomes negligible. Conversely, the coolant mass flow has a more pronounced effect on the TLV at larger tip gaps; indeed, at $\tau/C_x = 5\%$, higher coolant flow rates noticeably intensify the TLV $\overline{C_{pt}}$ peak at $z/h = 0.96$. Instead, variations in coolant flow

rate at $\tau/C_x = 3 - 5\%$ do not produce significant changes in the exit flow yaw angle deviation.

5.2.3.3 Comparison of Different Tip Leakage Flow Control Devices

After characterizing the effect of the tip gap height variation for each of the investigated tip geometries, as well as the influence of the coolant mass flow variation for both the squealer and squealer-winglet tip configurations, it is essential to compare the overall effectiveness of the different control devices.

Figures 5.42 and 5.43 illustrate the effect of different tip geometries at tip gap heights of $\tau/C_x = 0-1\%$ and $\tau/C_x = 3-5\%$, respectively, comparing passive devices (squealer and squealer-winglet without cooling) with the flat tip in terms of $\overline{C_{pt}}/\overline{C_{pt,max}}$ and $(\overline{\alpha_2} - \overline{\alpha_{2,mid}})/\overline{\alpha_{2,mid}}$. This comparison provides a first insight into how the various tip configurations influence the secondary flow structures. Starting from the no-gap condition, the $\overline{C_{pt}}$ distributions—characterized by a single peak associated with the passage vortex (PV)—are almost perfectly superimposed for the flat and squealer tips, as expected from their identical external tip geometry. In contrast, the external structure of the squealer-winglet configuration enhances the intensity of the PV, leading to an increase in the corresponding $\overline{C_{pt}}$ peak. However, in terms of exit flow angle deviation, no significant differences are observed among the three tip configurations. Interestingly, when the gap is opened, the PV $\overline{C_{pt}}$ peak at $z/h = 0.92$ is reduced with respect to the flat tip case when employing a squealer geometry, and is further decreased when using the hybrid squealer–winglet configuration. Regarding the TLV $\overline{C_{pt}}$ peak at $z/h = 0.98$, its magnitude remains essentially the same for all investigated tip configurations, consistently with the small clearance, but appears slightly lower for the squealer tip. In contrast, the effect of the tip geometry becomes more evident in the exit flow deviation. The region of under-turning induced by the PV decreases progressively from the flat tip to the squealer and then to the squealer–winglet configuration. Conversely, the over-turning region associated with the TLV becomes slightly more pronounced for the flat tip case and, in particular, for the squealer–winglet configuration. As the tip gap height increases, both PV intensity (at $\tau/C_x = 3\%$) and TLV strength (at $\tau/C_x = 3-5\%$) become progressively reduced when moving from the flat tip to the squealer, and further when transitioning from the squealer to the hybrid squealer–winglet configuration. This attenuation is likely attributed to the recirculating flow induced by the squealer recess and even more effectively by the squealer–winglet design. Accordingly, the exit flow deviation follows the same trend. The squealer tip, compared with the flat configuration, mitigates both the under-turning and over-turning regions. This effect is even more pronounced for the squealer–winglet tip, which, as clearly visible from the distributions, acts strongly towards uniformizing the exit flow angle across the blade span.

In Figure 5.44, the effect of the active flow control strategies (squealer and squealer–winglet tips with coolant injection) is shown in comparison with the flat tip configuration for each

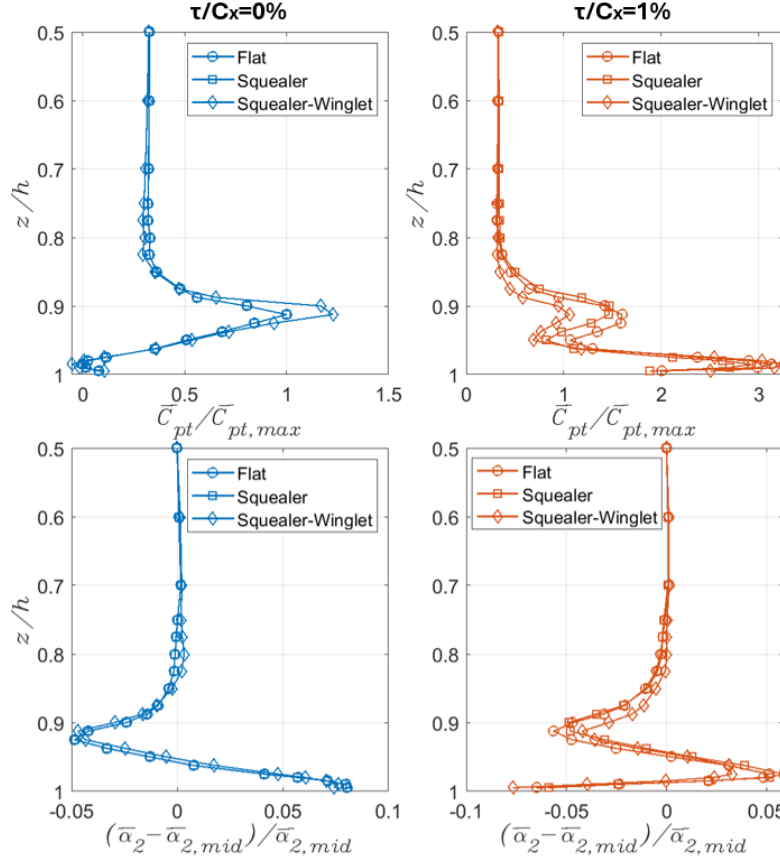


Figure 5.42: Tip geometry variation effect in terms of spanwise distributions of (top) normalized total pressure loss coefficient $\bar{C}_{pt}/\bar{C}_{pt,max}$ and (bottom) exit flow yaw angle deviation $(\bar{\alpha}_2 - \bar{\alpha}_{2,mid})/\bar{\alpha}_{2,mid}$ for $\tau/C_x = 0 - 1\%$.

investigated tip gap height. At $\tau/C_x = 1\%$, it is clearly visible that both squealer and squealer-winglet tips with a low coolant mass flow act to reduce the PV compared to the flat tip case, while they have little effect on the TLV. In contrast, both tip geometries with high coolant mass flow increase the PV loss region and, in the case of the squealer tip, also the $\bar{C}_{pt}$ peak associated with the TLV. As previously mentioned, this latter behavior is likely due to the excessive coolant flow rate in the small gap, which induces a strong blockage effect and interacts with the TLV, thereby amplifying it. Looking at the exit flow yaw angle, low coolant mass flow reduces the under-turning and does not significantly affect the over-turning region compared to the flat tip, whereas high coolant mass flow significantly increases both under- and over-turning. At $\tau/C_x = 3\%$, only a slight reduction in the penetration of the TLV loss region is observed in the presence of coolant, while the PV remains unchanged across the different conditions. This effect is even more pronounced at $\tau/C_x = 5\%$, where it can also be noted that the squealer tip with

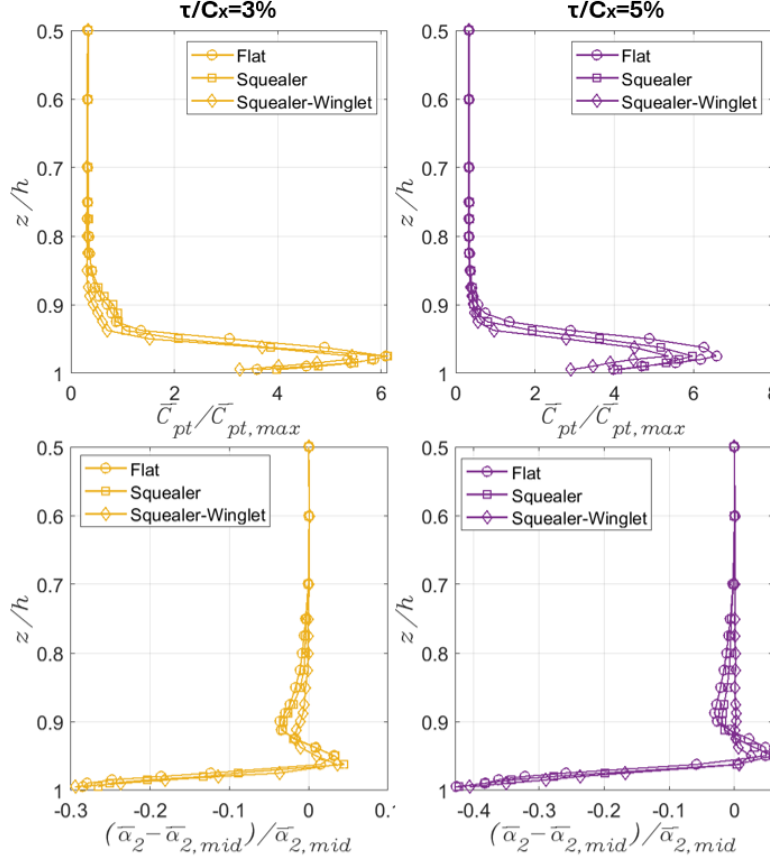


Figure 5.43: Tip geometry variation effect in terms of spanwise distributions of (top) normalized total pressure loss coefficient $\bar{C}_{pt}/\bar{C}_{pt,max}$ and (bottom) exit flow yaw angle deviation $(\bar{\alpha}_2 - \bar{\alpha}_{2,mid})/\bar{\alpha}_{2,mid}$ for $\tau/C_x = 3 - 5\%$.

low coolant mass flow is able to reduce the TLV loss peak compared to the flat tip. Regarding the yaw angle of the exit flow, no significant variations are observed between the use of a flat tip and the deployment of low or high coolant mass flow through the squealer or squealer–winglet tips.

Finally, Figures 5.45 and 5.46 show the effect of Reynolds number at tip gap heights of $\tau/C_x = 0-1\%$ and $\tau/C_x = 3-5\%$, respectively, for the three tip geometries. For the no-gap condition, the distributions of $\bar{C}_{pt}/\bar{C}_{pt,max}$ (top) and $(\bar{\alpha}_2 - \bar{\alpha}_{2,mid})/\bar{\alpha}_{2,mid}$ (bottom) at $Re_x = 100'000$ and $Re_x = 200'000$ are reported for the flat, squealer, and squealer–winglet tips. It can be seen that a lower Reynolds number induces a rigid rightward shift of the $\bar{C}_{pt}$ distribution obtained at the nominal Reynolds, suggesting an increase in profile losses (as expected) while the portion of losses exclusively due to secondary flows remains unchanged. Only the squealer–winglet tip configuration appears somewhat more sensitive to changes in this operating parameter, as it exhibits a more pronounced $\bar{C}_{pt}$ peak associated

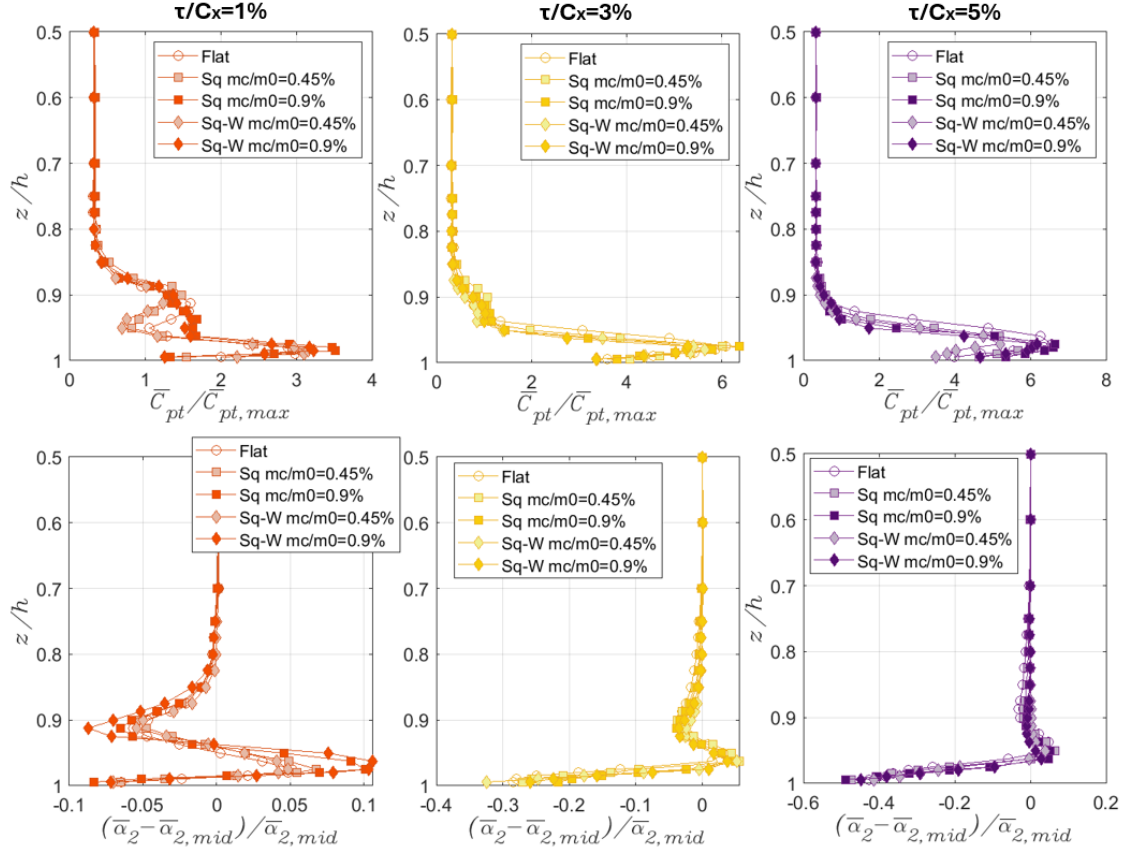


Figure 5.44: Coolant mass flow rate variation effect on the different tip configurations in terms of spanwise distributions of (top) normalized total pressure loss coefficient $\overline{C_{pt}}/\overline{C_{pt,max}}$ and (bottom) exit flow yaw angle deviation $(\overline{\alpha_2} - \overline{\alpha_{2,mid}})/\overline{\alpha_{2,mid}}$ for each tip gap height tested.

with the PV compared to the flat and squealer tips at low Re_x . At higher tip gap heights, the same distributions are reported at $Re_x = 100'000$ and $Re_x = 200'000$ for the squealer and squealer–winglet tips combined with a high coolant mass flow ratio. At $\tau/C_x = 1\%$, it can be observed that a decrease in Reynolds number, while increasing losses in the wake region, leaves the PV loss region essentially unchanged. Interestingly, the TLV losses are even reduced, particularly for the squealer–winglet tip, for which the $\overline{C_{pt}}$ curve does not appear to shift rightward. At $\tau/C_x = 3 - 5\%$, an increase in $\overline{C_{pt}}$ is again observed in the wake region, but there is a substantial reduction of TLV losses when moving to a lower Reynolds number. This is probably linked to Reynolds-dependent effects such as recirculations and/or separation occurring inside the cavity tip at high coolant flow rates. It is also worth noting that the squealer tip at low Re_x further accentuates the PV loss region. Nevertheless, no significant effect is visible on the exit flow yaw angle distortion for any of

the tip gap heights investigated. Only a slight increase in the over-turning can be observed for both tip geometries at the lowest Reynolds number in the cases of $\tau/C_x = 3 - 5\%$.

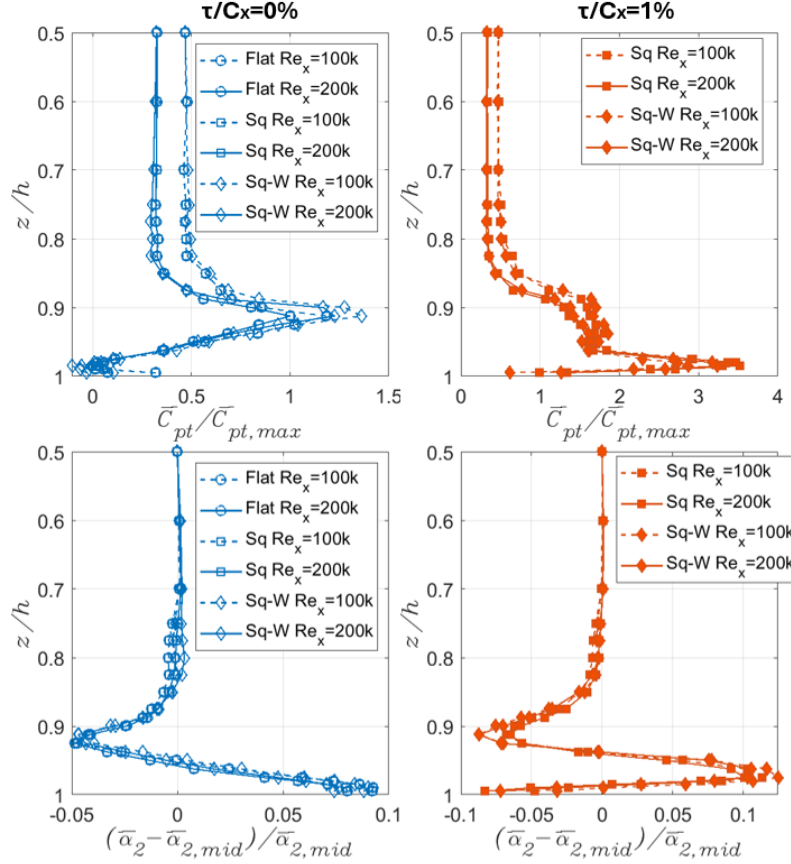


Figure 5.45: Reynolds number variation effect on the different tip configurations in terms of spanwise distributions of (top) normalized total pressure loss coefficient $\overline{C}_{pt}/\overline{C}_{pt,max}$ and (bottom) exit flow yaw angle deviation $(\overline{\alpha}_2 - \overline{\alpha}_{2,mid})/\overline{\alpha}_{2,mid}$ for $\tau/C_x = 0 - 1\%$.

For a more quantitative and concise comparison of all the conditions observed and discussed, Figure 5.47 reports the integrated mass-averaged secondary losses measured for all investigated cases. Solid black lines are added to highlight the effectiveness of each tested tip geometry configuration as a function of the tip gap height, while dotted black lines are instead used to indicate the effectiveness of the different tip geometries with coolant ejected from the tip, again as a function of tip gap. This distribution of integrated losses, in agreement with what was observed in the maps and spanwise distributions of C_{pt} , shows that, except for the no-gap case (where flat and squealer tips exhibit the same losses and the squealer–winglet tip results in slightly higher losses) an increase in tip gap generally leads to higher losses, but moving from flat to squealer to

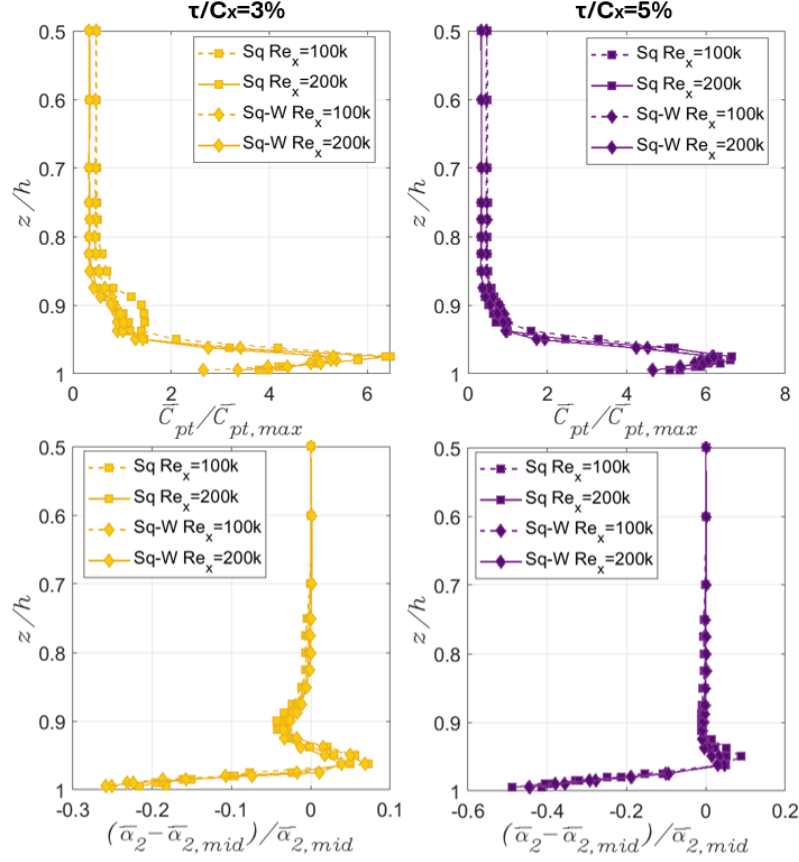


Figure 5.46: Reynolds number variation effect on the different tip configurations in terms of spanwise distributions of (top) normalized total pressure loss coefficient $\bar{C}_{pt}/\bar{C}_{pt,max}$ and (bottom) exit flow yaw angle deviation $(\bar{\alpha}_2 - \bar{\alpha}_{2,mid})/\bar{\alpha}_{2,mid}$ for $\tau/C_x = 3 - 5\%$.

squealer–winglet tips, the losses progressively decrease. Specifically, at the maximum tip gap height, the squealer reduces losses by approximately 20% compared to the flat-tip configuration, while the squealer–winglet achieves a reduction of about 38%. Observing the effect of coolant flow, it can be seen that at a low mass flow ratio, the secondary losses do not undergo significant variations for the squealer tip. For the squealer–winglet tip, slightly higher losses are observed at $\tau/C_x = 1\%$ and $\tau/C_x = 5\%$ compared to the squealer–winglet case without coolant, but they still remain considerably lower than the corresponding losses for the squealer tip at the same coolant flow rate. At the highest coolant flow rate, the losses for the squealer tip decrease only for $\tau/C_x = 3-5\%$ compared to the no-coolant condition. Conversely, for the squealer–winglet tip, secondary losses increase at every tip gap height, but they remain lower than the corresponding losses for the squealer tip. However, when also accounting for the energy spent for the coolant ($\omega_{t,c}^{(MA)}$)

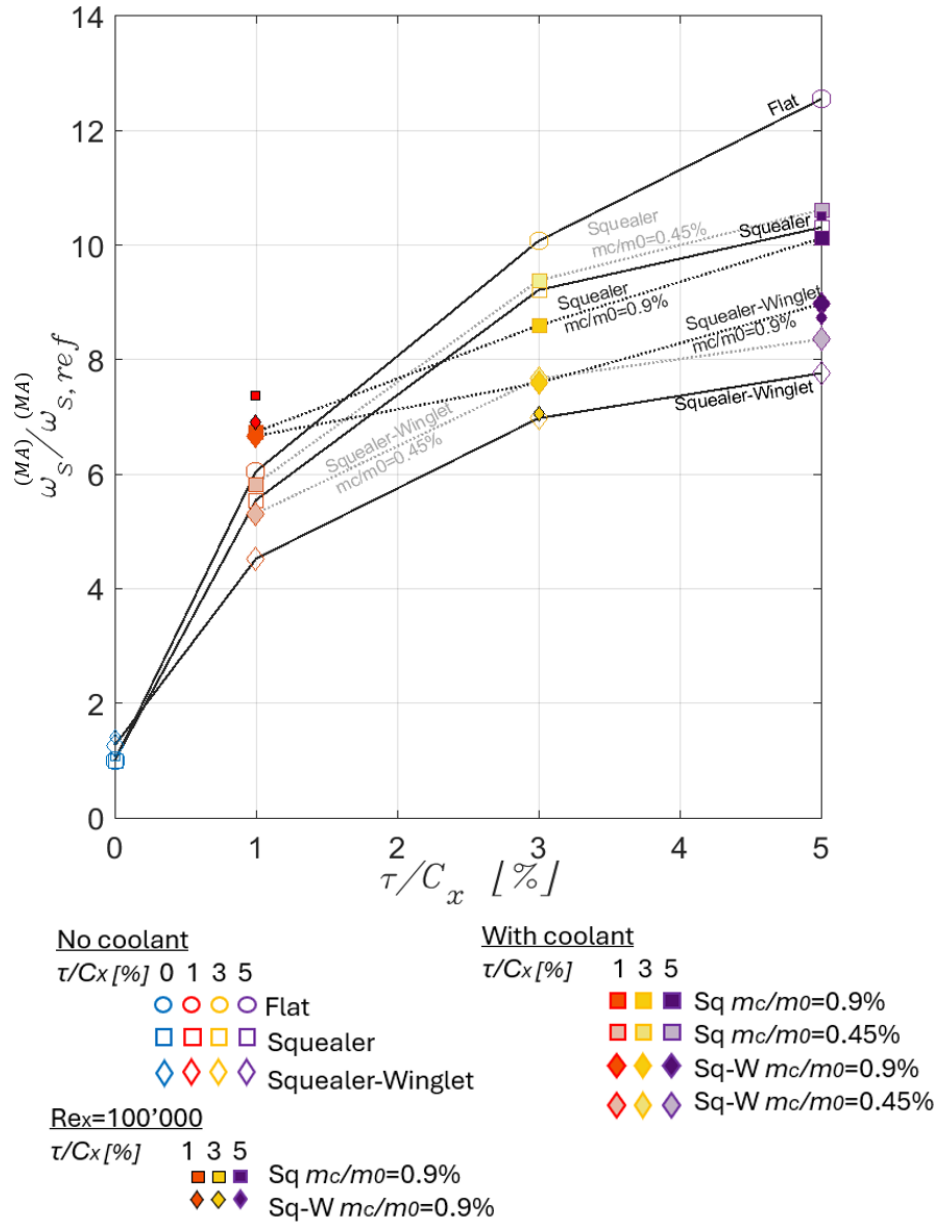


Figure 5.47: Normalized secondary losses as a function of tip gap height for each tip geometry, coolant mass flow rate, and operating condition tested.

higher loss values are obtained, especially at high coolant mass flow rates, in agreement with Rao and Camci [114]. Finally, considering the effect of Reynolds number, it is interesting to note that, for the same coolant mass flow ratio exiting from both squealer and squealer–winglet tips at the two Re_x tested, the secondary losses increase for the squealer

tip, whereas they decrease for the squealer–winglet configuration, demonstrating the effectiveness of this device even under varying operating conditions.

In Figure 5.48, an analogous plot showing the measured flow yaw distortion coefficients for all investigated conditions is reported. Here, it is clearly visible that the normal-

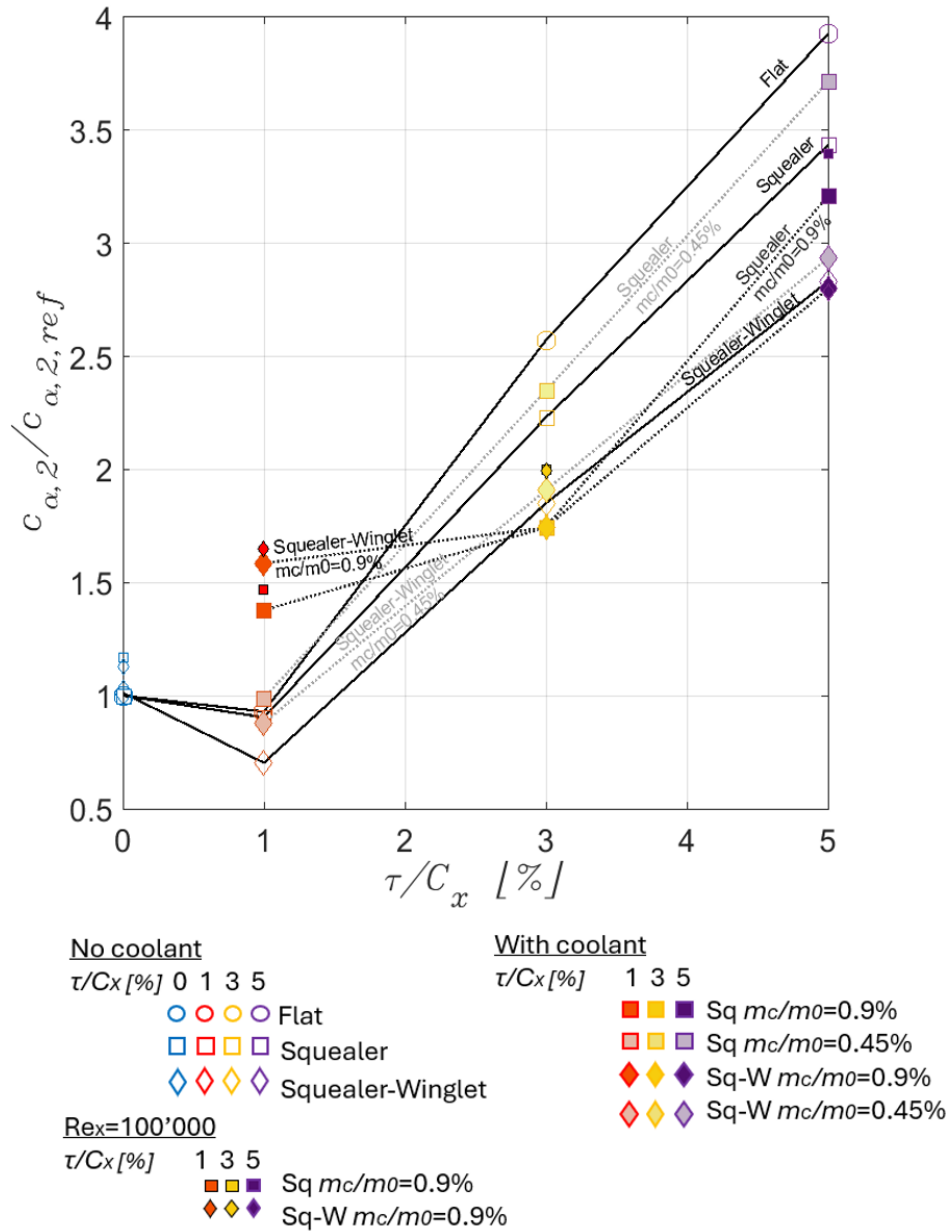


Figure 5.48: Normalized exit flow yaw angle distortion coefficient as a function of tip gap height for each tip geometry, coolant mass flow rate, and operating condition tested.

ized mean flow angle deviation at the cascade exit for the LPT configurations equipped with flat, squealer, and squealer–winglet tips is essentially the same as in the no-gap condition. Then, a slight decrease in $c_{\alpha,2}$ can be observed for the flat and squealer tips at $\tau/C_x = 1\%$, while a pronounced reduction is obtained with the squealer–winglet tip. For larger tip gap heights, an increase in the distortion coefficient is visible, although it progressively decreases when moving from flat to squealer and further to squealer–winglet tips. At the maximum tip gap height, the exit flow yaw angle distortion is reduced by 13% with the squealer configuration and by 28% with the squealer–winglet, relative to the flat tip. Considering the low coolant mass flow rate, it can be noted that at $\tau/C_x = 1\%$, the exit flow yaw angle distortion does not increase significantly, except for the squealer–winglet case, where $c_{\alpha,2}$ aligns with the value obtained for the same condition with the squealer tip. Conversely, at higher tip gap heights, $c_{\alpha,2}$ is higher than the respective no-coolant cases, although for the squealer–winglet case it still remains lower than for the squealer tip with coolant. Considering the maximum coolant mass flow rate, it can be observed that at $\tau/C_x = 1\%$, the distortion increases strongly (due to an excess of coolant in the small gap, which induces a blockage effect, increasing the PV, and amplifies the mass leakage flow, thereby enhancing the TLV). At larger tip gaps, however, the distortion is significantly reduced compared to the no-coolant cases. The squealer–winglet tip with the higher coolant mass flow ratio is the most effective in terms of reduced exit flow distortion, i.e., providing a more uniform flow entering the downstream cascade. Interestingly, at $\tau/C_x = 3\%$, the squealer tip with high coolant flow also achieves good flow uniformization, comparable to the squealer–winglet tip. Finally, considering the Reynolds number effect, an almost equivalent increase in the exit flow yaw distortion coefficient is observed at no-gap, $\tau/C_x = 1\%$, and $\tau/C_x = 3\%$ for both squealer and squealer–winglet tip configurations with high coolant mass flow. Interestingly, decreasing the Reynolds number at higher tip gap heights results in an increase of the distortion coefficient only for the squealer tip, while the squealer–winglet configuration appears unaffected by changes in Reynolds number.

5.2.4 Tip Loss Correlation

The experimental campaign enabled the collection of a comprehensive test matrix, which was subsequently used to calibrate a mass-averaged tip loss model as a function of the investigated parameters, hence also accounting for a broad range of tip designs. Before developing the new model, however, several well-established correlations from the literature were assessed against the experimental data obtained from the present flat tip cascade configuration. Table 5.1 summarizes the models, their formulations, and the mean percentage errors computed for the current dataset. The Ainley and Mathieson correlation [92] systematically overpredicts losses by approximately 50%. The Dunham and Came model [94] exhibits even larger errors, despite introducing a non-linear dependence on

gap size. Their prediction appears significantly out of scale, consistent with the observations of Matsunuma [77], likely due to an incorrect proportionality coefficient. More accurate estimates are instead provided by the models of Lampart [60], Yaras and Sjolander [97], and Denton [37], which rely on more realistic approximations of blade loading distributions (v_{PS} and v_{SS} correspond to the velocities on the pressure and suction sides, respectively) and incorporate the dependence on the actual leakage mass flow (via the discharge coefficient c_D), thereby accounting for additional physical mechanisms such as the corner separation that forms with a flat tip. Overall, the correlation by Yaras and Sjolander [97] provides the most accurate predictions, with a mean deviation of approximately 15%.

Table 5.1: Tip loss correlations from literature and corresponding errors computed for the present dataset.

Model	Formulation	$\overline{E(Y_{tip})}$
Ainley-Mathieson [92]	$0.5 \left(\frac{\tau}{h}\right) \left(\frac{C}{s}\right)^2 C_L^2 \frac{\cos^2(\alpha_2)}{\cos^3(\alpha_m)}$	50%
Dunham-Came [94]	$0.47 \left(\frac{C}{h}\right) \left(\frac{\tau}{C}\right)^{0.78} \left(\frac{C}{s}\right)^2 C_L^2 \frac{\cos^2(\alpha_2)}{\cos^3(\alpha_m)}$	>100%
Lampart [60]	$2c_D \left(\frac{\tau}{h}\right) \int_{tg(\alpha_1)}^{tg(\alpha_2)} \sqrt{\left(1 - \frac{(-\sqrt{c_1/3c_2 + \sqrt{1+x^2}})^2}{(\sqrt{c_1/3c_2 + \sqrt{1+x^2}})^2}\right)^2} dx$	45%
Yaras-Sjolander [97]	$2K_E c_D \left(\frac{\tau}{h}\right) \left(\frac{C}{s}\right) C_L^{1.5} \frac{\cos^2(\alpha_2)}{\cos^3(\alpha_m)}$	15%
Denton [37]	$2c_D \left(\frac{\tau}{h}\right) \left(\frac{C}{s}\right) \frac{1}{\cos(\alpha_2)} \int_0^1 \left(\frac{v_{SS}}{v_2}\right)^3 \left(1 - \frac{v_{PS}}{v_{SS}}\right) \sqrt{1 - \left(\frac{v_{PS}}{v_{SS}}\right)^2} \frac{dx}{C}$	30%

The present loss model, defined in Eqs. 5.2, 5.3, and 5.4, is an extended version of literature models designed to broaden its applicability to a wider range of tip designs, including tips equipped with TLV control and mitigation devices such as squealers, squealer-winglet, and tip cooling systems. Mass-averaged tip losses are considered additive to mass-averaged profile $\omega_p^{(MA)}$ and secondary (gap-free) $\omega_{s,ng}^{(MA)}$ losses. The model is based on a Yaras and Sjolander-like formulation. The main dependence on blade loading is expressed as a function of the Zweifel number, which replaces the lift coefficient of the original formulation. The coefficient a_0 , tuned with the present data, can also be expressed as a function of the mean flow angle α_m , the exit flow angle α_2 , the solidity, and the coefficient K_E to recover the original Yaras and Sjolander model, hence preserving the energy loss based calculation. The variation with tip gap height is non-linearly defined by the exponent a_1 , accounting for the interaction between the TLV and the passage vortex. Losses are linearly scaled with the blade aspect ratio. The discharge coefficient is used to account for the actual versus ideal leakage mass flow rate. In the absence of specific leakage mass flow measurements, $c_D = 0.8$ is suggested for the flat tip (see also [299]),

while $c_D = 0.72$ is recommended for tips equipped with a squealer cavity. These values were determined through numerical simulations that account for the leakage mass flow reduction induced by the squealer. The influence of the winglet is introduced via the effective Zweifel number at the blade tip, Zw_{eff} , reduced relative to the baseline Zweifel by a coefficient $k_{winglet} = 0.15$ for the present configuration. The term $c_{cooling}$ represents an additive contribution to the total pressure loss due to coolant ejection and depends quadratically on the coolant mass flow ratio, as previously observed. The proportionality coefficient a_2 has been tuned separately for squealer and squealer–winglet tips. Figures 5.49 and 5.50 show the prediction capability of the present loss model using the suggested coefficients for flat, squealer, and squealer–winglet tips across various tip clearance sizes (without and with coolant).

$$\omega_t^{(MA)} = \omega_p^{(MA)} + \omega_{s,ng}^{(MA)} + \omega_{tip}^{(MA)} \quad (5.2)$$

$$\omega_{tip}^{(MA)} = a_0 c_D Zw_{eff}^{1.5} \frac{C}{h} \left(\frac{\tau}{C} \right)^{a_1} + c_{cooling} \quad (5.3)$$

$$\left\{ \begin{array}{l} a_0 = a'_0 \frac{K_E (C/s)}{\cos(\alpha_m)^{1.5} \cos(\alpha_2)}; \quad \left\{ \begin{array}{l} a_0 = 1.00 \\ a'_0 = 0.28 \end{array} \right. \\ a_1 = 0.61 \\ c_D = \begin{cases} 0.80 & \text{for flat tip} \\ 0.72 & \text{for squealer and sq-winglet tips} \end{cases} \\ Zw_{eff} = Zw(1 - k_{winglet}); \quad k_{winglet} = 0.15; \\ c_{cooling} = a_2 \left(\frac{\dot{m}_c}{\dot{m}_0} \right)^2; \quad \left\{ \begin{array}{l} a_2 = 9.05 \cdot 10^4 \quad \text{for squealer tip} \\ a_2 = 12.10 \cdot 10^4 \quad \text{for sq-winglet tip} \end{array} \right. \end{array} \right. \quad (5.4)$$

The present loss model, once properly scaled, is generalizable to other cascade geometries. Within the coefficient a_0 , the effects of different stagger angles ($\propto \alpha_m$) and peak velocity positions ($\propto K_E$) are enclosed, as for the Yaras and Sjolander model [97]. Different exponents a_1 may express different interaction mechanisms between tip and main flow vortical structures, being in general function of the inlet boundary layer and main parameters usually affecting the gap-free secondary flow scenario. The discharge coefficient c_D may include the effect of the blade thickness, as documented by Yaras and Sjolander [98], where c_D was expressed as a function of the clearance size relative to the blade maximum thickness, and especially, of the blade rotation that acts reducing the leakage mass flow. No correction is provided here for the Reynolds number variation. The present correlations were developed based on data at $Re_x = 200,000$; therefore, their reliability might not be guaranteed when extrapolated to different Reynolds numbers. However, it should be noted that Re has been shown to not significantly affect tip losses for both squealer and squealer–winglet tips, as reported in [77] and [223]. Nevertheless, limitations regarding

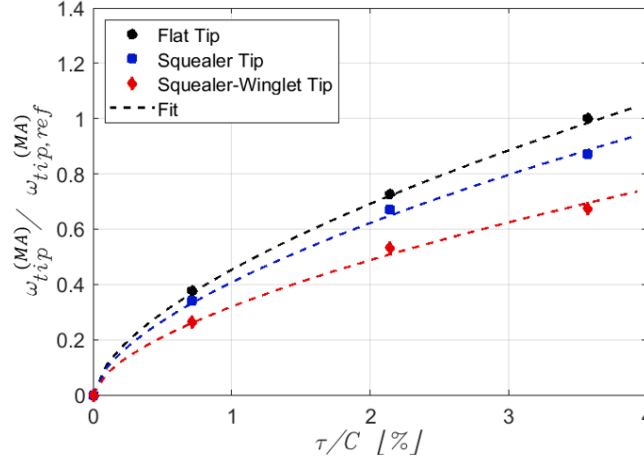


Figure 5.49: Prediction curves from current loss model for flat, squealer and squealer-winglet tips (without tip cooling).

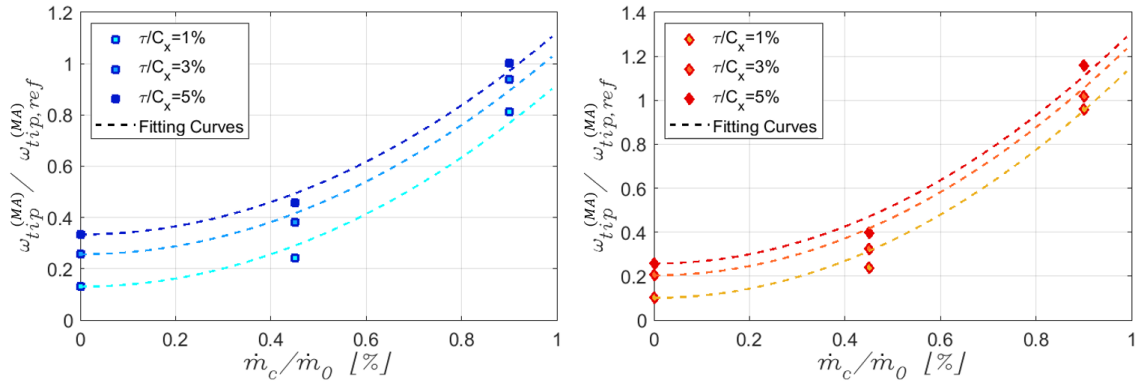


Figure 5.50: Prediction curves from current loss model for (left) squealer and (right) squealer-winglet tips with tip cooling.

extrapolation remain for other parameters. Specifically, the accuracy of the results may not be guaranteed outside the tested ranges for tip gap height and coolant mass flow ratio (i.e., for $\tau/C_x > 5\%$ and $\dot{m}_c / \dot{m}_0 > 0.9\%$), nor for tip geometries that differ significantly from those investigated in this study.

5.2.5 Further analysis with RANS and LES

To better understand the mechanisms by which the squealer and squealer-winglet tips, combined with coolant mass flow, act to control and mitigate secondary flows, it is necessary to consider the numerical simulations conducted on the same cascade configurations by Morfo Design, using the experimentally measured and rescaled boundary conditions,

following [290]. See Appendix 2 for further details on the numerical setup. After an initial comparison between experimental and numerical results to validate both RANS simulations (used during the design phase of the tip geometries) and LES simulations, the latter were employed to investigate how the modified tips alter the flow field in the tip regions. Only the cases at $Re_x = 200'000$ have been simulated.

The comparison between experimental results and numerical simulations is carried out by analyzing the spanwise distributions of the total pressure loss coefficient, focusing on the region affected by the leakage flow ($0.7 < z/h < 1$). Figure 5.51 reports the pitchwise-averaged $\overline{C}_{pt}$ distributions for the different tip geometries and tip gap heights investigated. Both experimental and RANS results are reported, normalized for confidentiality reasons. At the smallest gap height ($\tau/C_x = 1\%$, left), RANS simulations well reproduce the experimental trends, clearly capturing the two loss peaks associated with the passage vortex and tip leakage vortex. The spanwise position and intensity of the PV peak are in good agreement with experiments, while minor discrepancies are observed in TLV penetration and magnitude, with a slight overprediction of profile losses in the wake region. At $\tau/C_x = 3\%$ (middle), RANS maintains good agreement with experiments, correctly predicting the reduced PV and TLV intensity and the lower TLV penetration observed for the squealer–winglet configuration. At $\tau/C_x = 5\%$ (right), RANS simulations successfully capture the effect of changing tip geometry from flat to squealer to squealer–winglet, showing reduced TLV losses and increased confinement near the casing, although peak magnitude and penetration are slightly overestimated. For this condition, LES results are also available and show a markedly improved agreement with experiments.

A quantitative comparison is provided in Figure 5.52, where the normalized mass-averaged tip losses $\omega_{tip}^{(MA)}$ are reported. RANS simulations correctly reproduce the mono-

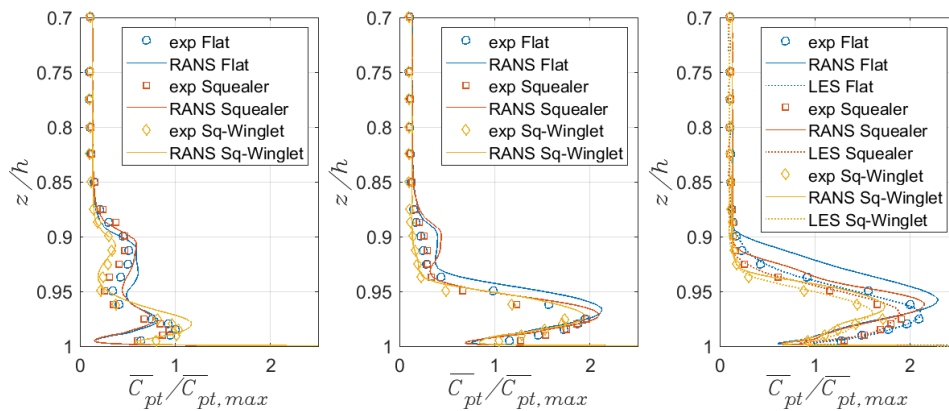


Figure 5.51: Normalized spanwise loss distributions from experiments and RANS simulations, comparing flat, squealer, and squealer–winglet tips at (left) $\tau/C_x = 1\%$, (middle) $\tau/C_x = 3\%$, and (right) $\tau/C_x = 5\%$ [300].

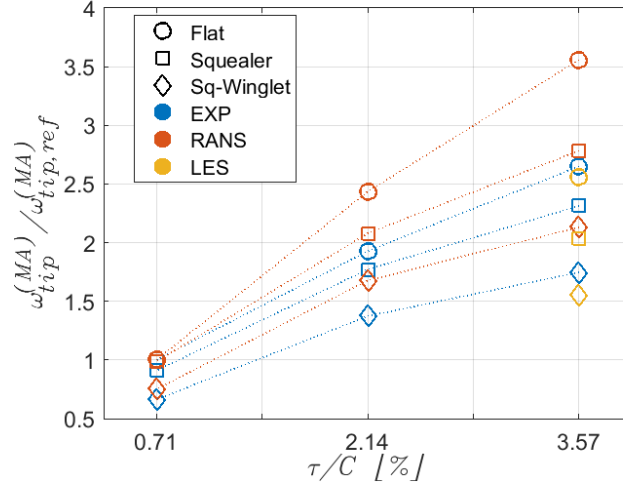


Figure 5.52: Experimental and numerical normalized tip losses for flat, squealer, and squealer–winglet tips at the different tip gap heights tested [300].

tonic increase in losses with tip gap height and the systematic loss reduction achieved by the squealer and squealer–winglet tips, although absolute values are overpredicted. LES results at the largest gap height show significantly improved agreement, with only a slight underprediction of losses.

The effect of coolant mass flow rate is illustrated in Figures 5.53 and 5.54, which compare experimental and RANS $\overline{C}_{pt}$ distributions for the squealer and squealer–winglet tips at different coolant flow ratios and gap heights. For the squealer tip (Figure 5.53), RANS captures the main experimental trends, including the negligible effect of low coolant flow and the increase in losses associated with high coolant injection, particularly at small

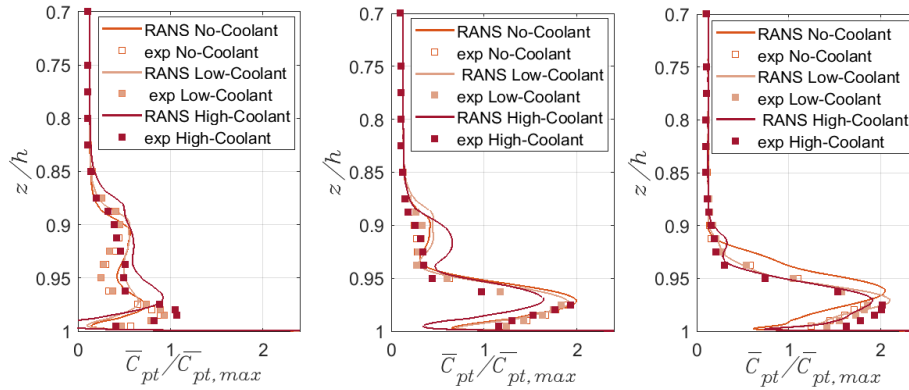


Figure 5.53: Experimental and RANS spanwise distributions of the normalized total pressure loss coefficient for the squealer tip at different coolant flow ratios for tip gap heights: (left) $\tau/C_x = 1\%$, (middle) $\tau/C_x = 3\%$ and (right) $\tau/C_x = 5\%$ [301].

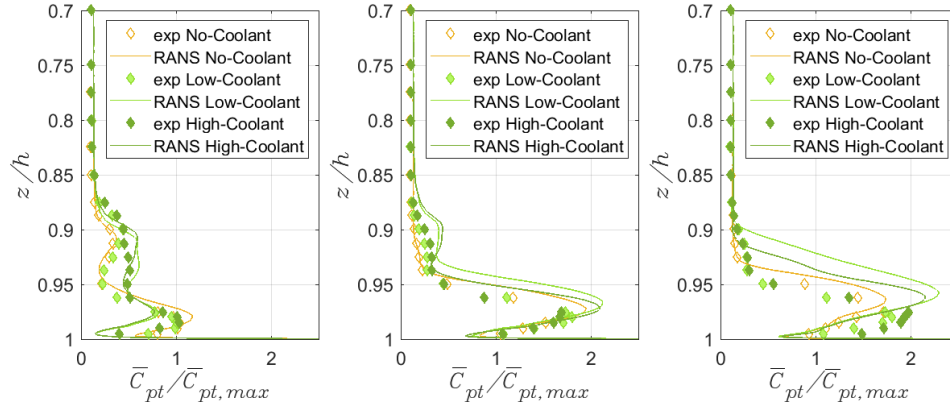


Figure 5.54: Experimental and RANS spanwise distributions of the normalized total pressure loss coefficient for the squealer-winglet tip at different coolant flow ratios for tip gap heights: (left) $\tau/C_x = 1\%$, (middle) $\tau/C_x = 3\%$ and (right) $\tau/C_x = 5\%$ [300].

clearance. Some discrepancies emerge at larger gaps, where simulations tend to predict more complex TLV structures. For the squealer-winglet tip (Figure 5.54), RANS reproduces the experimental behavior across all gap heights, capturing the expansion of the PV influence region with increasing coolant mass flow. Slight overprediction of losses and deeper TLV penetration are observed at larger clearances, but the overall trends and the relative impact of coolant injection are well predicted.

Given the good agreement between experimental measurements and numerical simulations in terms of loss distributions, which further supports the use of RANS in the design phase, the LES carried out by Morfo Design can be reliably employed to gain deeper insight into the flow developing within the tip clearance for the three investigated tip geometries. To this end, Figure 5.55 illustrates, in the top row, the mechanism of TLV formation for the flat (left), squealer (middle), and squealer-winglet (right) tips at $\tau/C_x = 5\%$, using streamline visualizations. The streamlines are colored according to chordwise vorticity, and a corresponding vorticity contour is shown on a plane orthogonal to the camber line. Conversely, the contribution of the leakage flow to loss generation is presented in terms of C_{pt} on a tangential plane downstream of the blades, reported in the bottom row for improved clarity and comparability. Additionally, the blade surfaces are colored according to the pressure coefficient distribution at the tip. The comparison shows that, for each geometry, the flow leaks through the tip clearance due to the pressure difference between the pressure and suction sides, emerging on the SS where it interacts with the main passage flow to form the TLV. This vortex, characterized by positive chordwise vorticity, produces a region of high total pressure losses in the tip region on the downstream plane. The plots also highlight how the squealer and squealer-winglet geometries significantly modify the leakage flow behavior observed for the flat tip. For the flat tip, the leakage flow moves

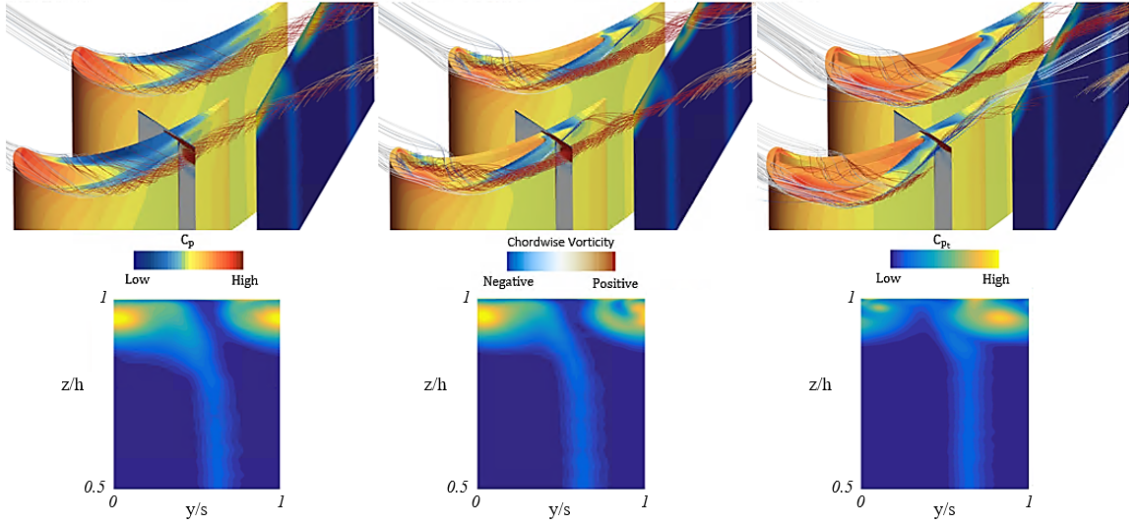


Figure 5.55: 3D visualizations of the flow streamlines for the (left) flat, (middle) squealer, and (right) squealer-winglet tip geometries at $\tau/C_x = 5\%$, obtained from LES. The streamlines and the contour plot along the blade are colored according to the chordwise vorticity. The blade tip surface is colored according to c_p . A downstream plane, also shown in the bottom row, reports the contour map of C_{pt} [300].

directly from the PS to the SS due to the large c_p variation, wrapping into a strong vortical structure along most of the SS tip edge. Downstream, an intense and extended loss core develops on the blade SS due to the TLV. Moreover, a low pressure area is clearly visible on the blade tip surface (blue region), indicating the presence of a separated flow region. When the squealer tip is employed, the c_p variation is reduced, the low pressure region is confined to the rear portion of the blade tip, and the flow develops in a more axial direction. The leakage flow streamlines are forced to recirculate and accumulate within the cavity, exiting through a more confined region of the SS rim. Consequently, the TLV becomes more compact, as indicated by the reduced extension of the downstream loss core. In the squealer-winglet configuration, the flow streamlines over the blade tip become even more axial compared to the squealer case. This behavior is enabled by the more uniform c_p distribution on the winglet surface, directly induced by its geometry. Although a smaller portion of the leakage flow recirculates within the cavity relative to the squealer tip, the flow exits the SS winglet in a more organized manner, producing a weaker vortical structure at the winglet extremity. The resulting TLV is substantially reduced in both size and intensity compared with the flat and squealer tips. Moreover, it is located closer to the PS of the adjacent blade, as shown by the C_{pt} contours, due to the tangential shift induced by the winglet. Interestingly, LES also reveals the presence of minor secondary structures for both squealer and squealer-winglet tips, likely associated with the recirculating flow inside the cavity, which, after developing streamwise within

the recess, experiences an increased pressure gradient near the TE tip surface, crosses the SS rim or winglet, and merges with the main TLV loss core. For a more detailed visualization of the flow within the tip clearance and cavity, Figure 5.56 reports the C_{pt} contours on planes orthogonal to the camber line at $x/C_x = 0.5$ (top) and $x/C_x = 0.75$ (bottom) for the three tip configurations investigated. For the flat tip (left plots), the separation bubble previously identified by the low pressure region on the blade tip is clearly visible. Above this bubble, the flow crosses from the PS to the SS in an almost isentropic leakage jet (see, e.g., [58]), forming an intense TLV on the suction side. With the squealer tip (middle plots), the leakage flow predominantly recirculates between the two rims, generating vortical structures and high mixing losses within the cavity. This reduces the amount of flow leaking across the tip. As a result, the TLV is smaller in spatial extent compared with the flat tip (even at $x/C_x = 0.5$), although it remains relatively intense. The addition of the winglet (right plots) further promotes recirculation within the cavity, providing more effective tip clearance sealing. This results in a negligible TLV at $x/C_x = 0.5$ and a weaker vortical structure—both in size and intensity—at $x/C_x = 0.75$. This behavior can be explained, as already observed from the 3D streamline visualizations, by the reduced pressure coefficient at the blade tip. Indeed, for unshrouded blades such as those investigated here, the TLV intensity is largely governed by the tangential flow leaking across the blade tip from PS to SS, which is itself driven by the strong pressure gradient imposed by the blade in the tip region. To further investigate this aspect, Figure 5.57 reports the c_p distributions for the flat, squealer, and squealer-winglet geometries at the blade heights $z/h = 0.8$ (left), $z/h = 0.9$ (middle), and $z/h = 0.95$ (right). In accordance with Coull *et al.* [261], it is evident that the superior effectiveness of the squealer-winglet in mitigating the TLV—relative to both flat and squealer tips—can be attributed to the reduced aerodynamic loading in the vicinity of the clearance. While at $z/h = 0.8$ and $z/h = 0.9$ the

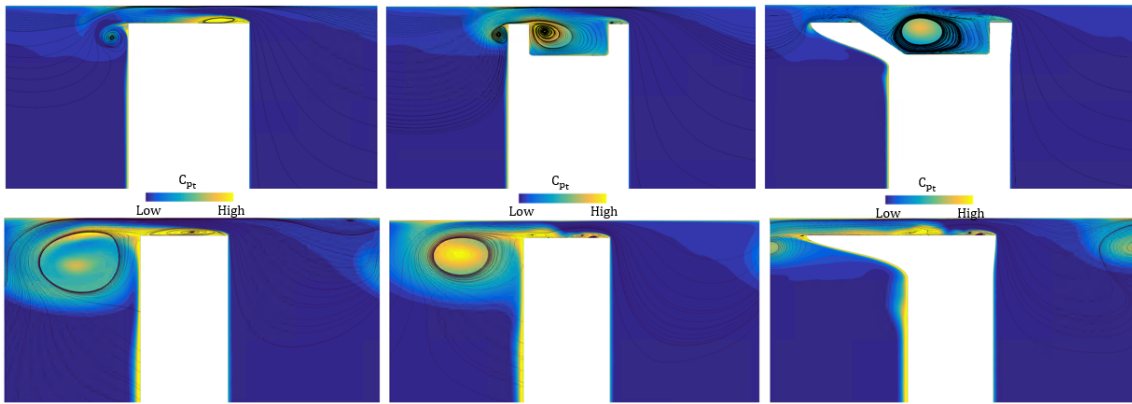


Figure 5.56: C_{pt} contour maps on orthogonal planes at (top) $x/C_x = 0.5$ and (bottom) $x/C_x = 0.75$ for the (left) flat, (middle) squealer, and (right) squealer-winglet tip geometries [300].

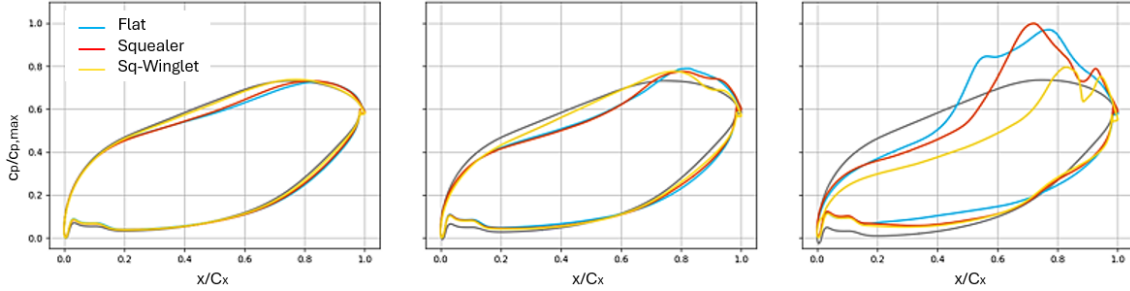


Figure 5.57: Normalized pressure coefficient along the normalized axial coordinate for (left) $z/h = 0.8$, (middle) $z/h = 0.9$, and (right) $z/h = 0.95$ for the flat (blue), squealer (red), and squealer-winglet (yellow) tip geometries [300].

squealer-winglet exhibits slightly higher aerodynamic loading compared with the flat and squealer cases, this loading is substantially reduced at the blade tip, almost halved relative to the flat configuration. Interestingly, the squealer geometry exhibits a narrow region on the blade SS, just upstream of the blade passage throat, where the aerodynamic loading is higher than in the flat case, albeit less extended in the axial direction. Consequently, the leakage flow crosses the blade tip through a more confined area, consistent with the previous observations that the resulting TLV is therefore more compact.

Since the LES were conducted for the three tip geometries only at the maximum tip gap height, the analysis of the effects of tip gap variation and coolant flow rate on the TLV is carried out using 3D streamline visualizations obtained from the RANS simulations. First, the same conditions shown in Figure 5.55 (where the LES results were presented) are reproduced in Figure 5.58 using the RANS data. This allows a detailed investigation of why RANS overpredict the TLV loss intensity and penetration for all tip configurations compared with LES, as shown in Figure 5.51 (right plot). In Figure 5.58, it is clearly visible that the flow streamlines are forced to leak through the tip clearance and, together with a probable recirculating flow that remains attached to the SS edge up to the TE for all the three tip configurations, contribute to the formation of the TLV at the blade SS tip. This results in a more intense and extended loss core on the downstream plane (bottom plots) compared with LES. Additionally, a portion of the streamlines exhibits negative chordwise vorticity, likely induced by the TLV rotation, producing a slightly more pronounced region of opposite-signed vorticity relative to the TLV than in the LES cases. However, it is important to note that the mechanism through which the squealer and squealer-winglet geometries alter the development of the leakage flow on the blade tip and mitigate the TLV is well captured by RANS. Indeed, it is clearly observed that the strong leakage flow typical of the flat tip becomes more axial for the squealer and squealer-winglet configurations. For these two cases, RANS also correctly reproduces the strong recirculation inside the cavity to which the leakage flow is subjected. This behavior results

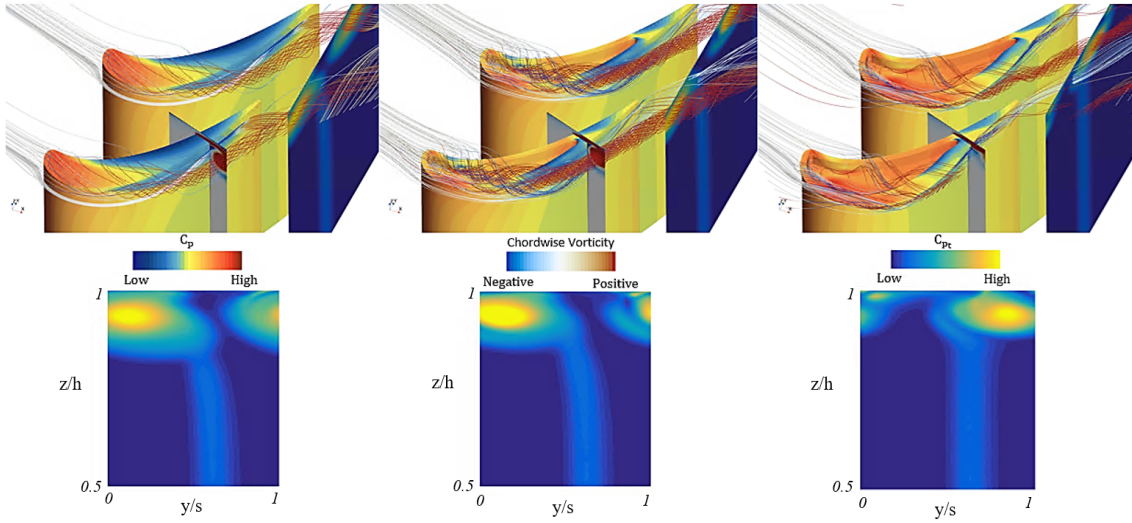


Figure 5.58: 3D visualizations of the flow streamlines for the (left) flat, (middle) squealer, and (right) squealer-winglet tip geometries at $\tau/C_x = 5\%$, obtained from RANS [300].

from the increasingly uniform pressure coefficient distribution on the blade tip, which is also captured by RANS simulations, and consequently leads to a TLV which—although overpredicted in absolute terms—is reduced in size when moving from flat to squealer, and further reduced in the squealer-winglet configuration. Moreover, the TLV intensity is diminished and the loss core is located closer to the blade PS, in agreement with the LES observations. To further compare RANS and LES simulations, the mass-averaged total losses are reported in Figure 5.59 as a function of the leakage velocity v_{leak} , following the approach proposed by [302]. Both quantities are normalized by their corresponding values at $\tau/C_x = 1\%$. Considering the results at the maximum tip gap, it is evident that, although RANS overpredict the losses for all tip configurations, the leakage flow velocity is well captured, being only slightly lower than the LES prediction. This indicates that the increased losses observed in the RANS simulations are primarily associated with the mixed losses generated by the interaction between the leakage flow exiting the tip region and the main stream. Moreover, the decreasing loss trend from flat to squealer to squealer-winglet, corresponding to decreasing leakage flow velocity, is consistently reproduced by both simulations. This not only further validates the low-fidelity simulation predictions but also highlights that the higher effectiveness of the squealer, and especially of the squealer-winglet, relative to the flat tip, is partly attributable to the lower leakage flow velocity in the tip region.

The effect of tip gap height variation on both total losses and leakage flow velocity can be further analyzed using Figure 5.59. It can be observed that as the tip gap decreases, the reduction in losses is accompanied by a corresponding decrease in leakage flow velocity. Furthermore, the figure shows that the difference in leakage velocity between the flat

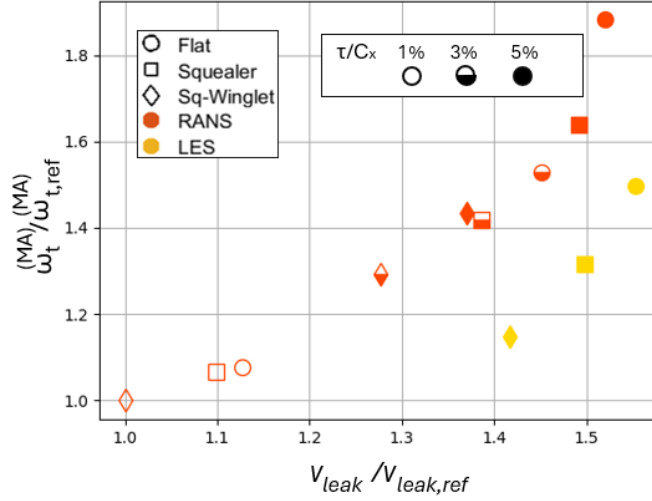


Figure 5.59: Normalized mass-averaged total losses as a function of the normalized leakage flow velocity at the exit of the tip region for all conditions investigated using both RANS (red) and LES (yellow). All tip configurations and tip gap heights are shown [300].

and squealer tips, as well as between the squealer and squealer-winglet configurations, diminishes with decreasing tip gap height. Similarly, the differences in total losses among the different tip geometries become progressively smaller as the tip gap is reduced.

To better understand how the loss generation mechanisms evolve with variations in tip gap height, the 3D streamline visualizations in Figure 5.60 are considered. Only the squealer-winglet configuration is shown for the various gap sizes (increasing from left to right). At the smallest tip gap ($\tau/C_x = 1\%$), it is clearly visible that, in addition to the main vortical structure generated by the leakage flow over the winglet at the blade tip, a secondary structure with negative vorticity develops slightly deeper along the blade span. As discussed in [133], this structure corresponds to the passage vortex, which originates primarily in the aft portion of the blade suction side. It arises from streamlines that, near the leading edge of the adjacent blade pressure side, are influenced by the strong pressure gradient imposed by the blade and migrate across the inter blade channel toward the neighboring blade. These counter-rotating structures subsequently give rise to two distinct loss cores, clearly visible in the downstream C_{pt} map. The TLV, located close to the casing, appears stronger and more extended than the PV, which is likely partially dissipated by the TLV. Increasing the tip gap height to $\tau/C_x = 3\%$ generates a larger pressure coefficient gradient on the blade tip surface, which drives a greater portion of the mainstream to leak from the pressure side toward the suction side. At the same time, the boundary layer developing along the casing becomes less influenced by the blade induced pressure difference due to the wider gap, resulting in an almost negligible PV. Nevertheless, a region

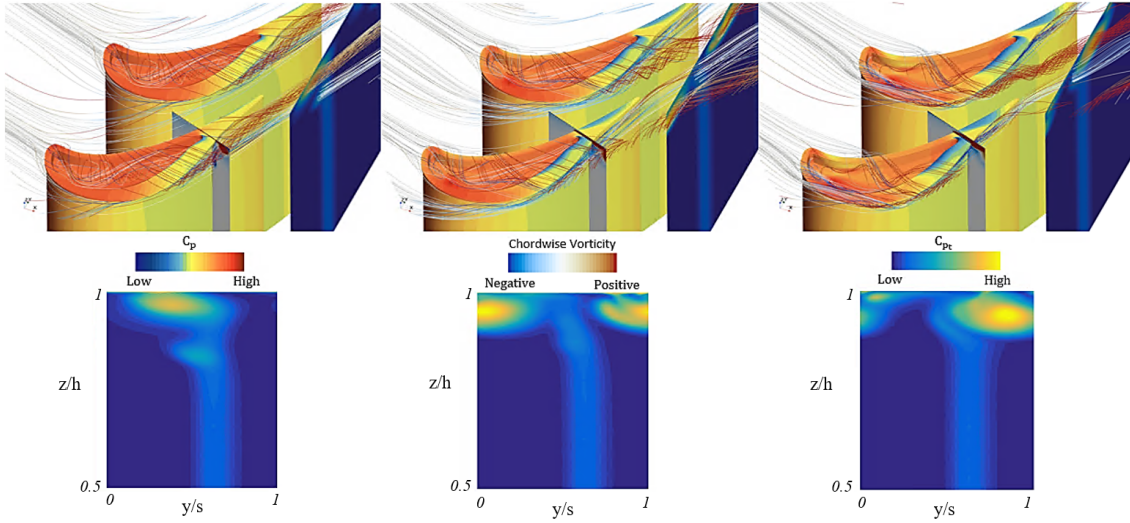


Figure 5.60: 3D visualizations of the flow streamlines for the squealer-winglet tip geometry at (left) $\tau/C_x = 1\%$, (middle) $\tau/C_x = 3\%$ and (right) $\tau/C_x = 5\%$, obtained from RANS [300].

of negative vorticity persists, although weaker and less extended, likely induced by the rotation of the TLV, and remains closely attached to the blade suction side. Conversely, a stronger and more extended TLV develops at the blade tip and shifts toward the mid channel region compared with the $\tau/C_x = 1\%$ case. Furthermore, the negative vorticity region appears to be entrained around the TLV, causing a slight displacement toward the casing, where the two counter-rotating vortices seem to interact and partially dissipate each other—consistently with the smearing of the total pressure loss contours observed near the blade tip pressure side. At the maximum tip gap height ($\tau/C_x = 5\%$), the tip surface experiences an even steeper pressure coefficient gradient, leading to a more intense TLV that is further displaced toward the pressure side of the adjacent blade. The negative vorticity region, on the other hand, becomes significantly weaker and remains confined to the blade tip suction side, as indicated by the local smearing of the loss contours. Interestingly, as the tip gap increases, the recirculating flow between the PS rim and the SS winglet fills a smaller portion of the cavity, exiting from a narrower section of the suction side tip edge and thereby forming a more compact vortical structure.

The modification of the pressure field and tip flow streamlines due to coolant ejection from the squealer tip cavity is illustrated in Figure 5.61, which reports C_{pt} contours at $x/C_x = 50\%$ (top) and $x/C_x = 75\%$ (bottom) for the low (left) and high coolant (right) cases. At $x/C_x = 50\%$, approximately corresponding to the location of the third cooling hole, the plot clearly shows the ejected coolant jet. The jet is primarily deflected toward the suction side, but part of it recirculates, generating additional losses (yellow region). The contours at $x/C_x = 75\%$ reveal the formed TLV, which appears nearly the same size

as the core observed in the no-cooling case in Figure 5.56. The main difference lies in the near-isentropic jet from PS to SS: while it is clearly visible in the flat tip configuration and still recognizable (though reduced) in the squealer tip without cooling, in the present case the loss free jet core cannot be clearly identified due to the vortical structures induced by the cooling flow. For the high-coolant case, the scenario changes further. At $x/C_x = 50\%$, a strong jet ejected from the cooling hole is observed. The flow wraps around its sides, forming two vortices that contribute to loss generation, particularly near the suction side rim. The TLV, visible at $x/C_x = 75\%$, appears displaced farther from the suction side and is primarily formed by the coolant mass flow. Moreover, the TLV is less extended in the spanwise direction compared to the other cases, consistently with the experimental and numerical observations at the cascade outlet.

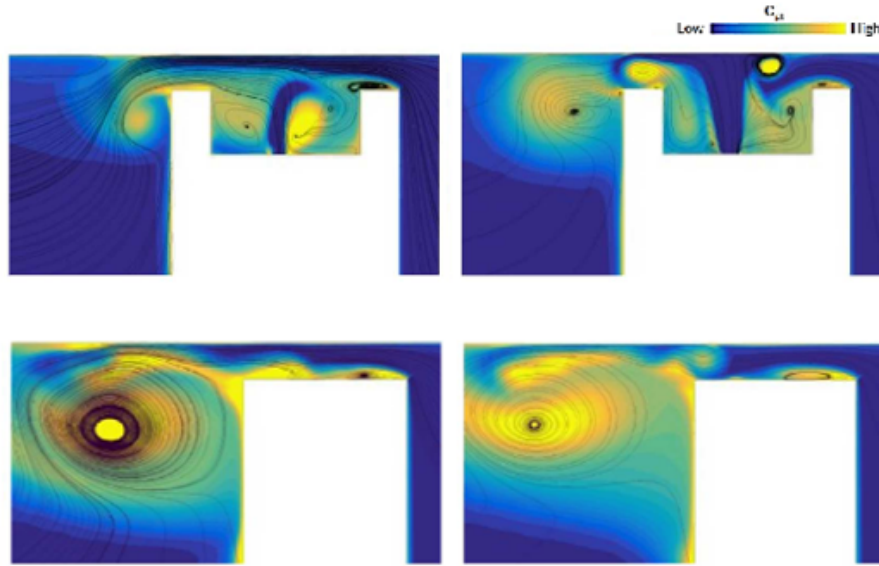


Figure 5.61: C_{pt} contour maps on orthogonal planes at (top) $x/C_x = 50\%$ and (bottom) $x/C_x = 75\%$ for the squealer tip with (left) low- and (right) high-cooling flow rates at $\tau/C_x = 5\%$, with superimposed streamlines from secondary velocity vectors [301].

Finally, Figure 5.62 presents 3D streamline visualizations illustrating the effect of varying the tip cooling mass flow rate on the tip leakage vortex developing over the squealer–winglet configuration at $\tau/C_x = 5\%$. The influence of increasing the coolant mass flow from $\dot{m}_c/\dot{m}_0 = 0$ to 0.45% and 0.9% is shown from left to right. It can be observed that the coolant mass flow rate does not significantly alter the pressure coefficient gradient on the blade tip surface, which remains high regardless of the injection rate. A slightly more uniform c_p distribution inside the cavity is noticeable at higher flow rates, potentially providing some thermal benefit. The aerodynamic impact of the coolant can instead be interpreted in terms of the blockage effect induced by the jet exiting the cavity,

which tends to seal the tip clearance and thereby limit the leakage flow from pressure to suction side. At the lowest coolant flow rate, this effect is minor: the main observable feature is that the leakage flow exits through a more confined region along the suction side tip edge. However, this does not significantly modify the downstream C_{pt} field, which remains intense and extended, still showing probable interactions between regions of opposite chordwise vorticity. At the highest coolant mass flow rate, fewer streamlines are able to leak along the suction side. While the TLV develops weaker, a much more extended and intense region of negative vorticity emerges, altering the structure of the downstream loss cores. This results in a slight reduction of the tip-loss region compared with the intermediate coolant case ($\dot{m}_c/\dot{m}_0 = 0.45\%$), but not with respect to the no-coolant condition. Despite the good agreement between RANS and experimental results (Figure 5.54), only high-fidelity simulations can more accurately and reliably capture the detailed interaction between the cooling mass flow and the tip leakage flow, as well as the subsequent mixing with the main stream that ultimately governs tip-loss generation.

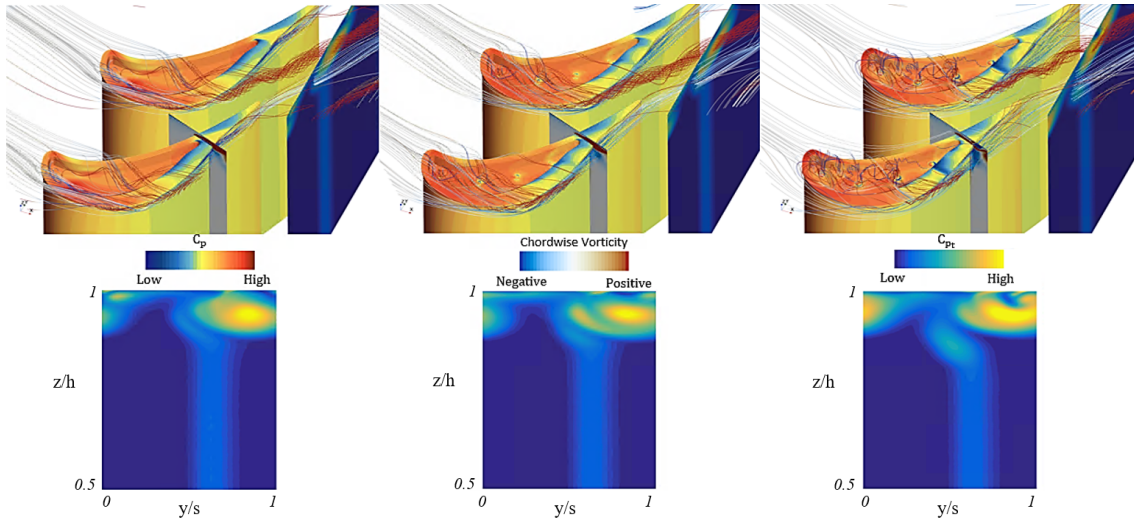


Figure 5.62: 3D visualizations of the flow streamlines for the squealer-winglet tip geometry at (left) $\dot{m}_c/\dot{m}_0 = 0$, (middle) $\dot{m}_c/\dot{m}_0 = 0.45\%$ and (right) $\dot{m}_c/\dot{m}_0 = 0.9\%$, obtained from RANS [300].

Additional simulations with a rotating endwall were carried out by Morfo Design for the squealer tip under no-coolant conditions and for various tip gap heights, to assess the effect of relative motion between the blade and casing. The rotation was simulated by imposing a moving endwall with a fixed tangential velocity to match the target flow coefficient of the current application ($\phi = 1.05$). Results in terms of spanwise loss distributions are reported in Figure 5.63, where the previous $\overline{C_{pt}}$ distributions for a fixed endwall (solid lines) are compared with those obtained under rotation (dotted lines) at tip gaps $\tau/C_x = 1\%, 3\%$ and 5% . The plot is zoomed in the range $0.7 < z/h < 1$ to highlight

effects in the near wall region. It can be observed that rotation slightly increases the secondary loss peak associated with the passage vortex at all tip gaps, while it reduces the TLV loss core at $\tau/C_x = 1\%$ and $\tau/C_x = 3\%$. This trend is consistent with the expected effects of rotation discussed in the literature by Rahman *et al.* [71], Yaras and Sjolander [98], and Yang *et al.* [303]. The only exception occurs at $\tau/C_x = 5\%$, where the TLV loss peak predicted by RANS slightly increases. Overall, these variations are limited, and the impact on total losses is approximately -10% , 0% , and $+5\%$ of the total loss coefficient at $\tau/C_x = 1\%$, 3% , and 5% , respectively. Compared with standard HPT designs, the effect of rotation in the present application is relatively small, mainly due to the relatively high target flow coefficient. Therefore, the results presented and the effects of parameter variations discussed previously are expected to be only marginally affected by blade rotation.

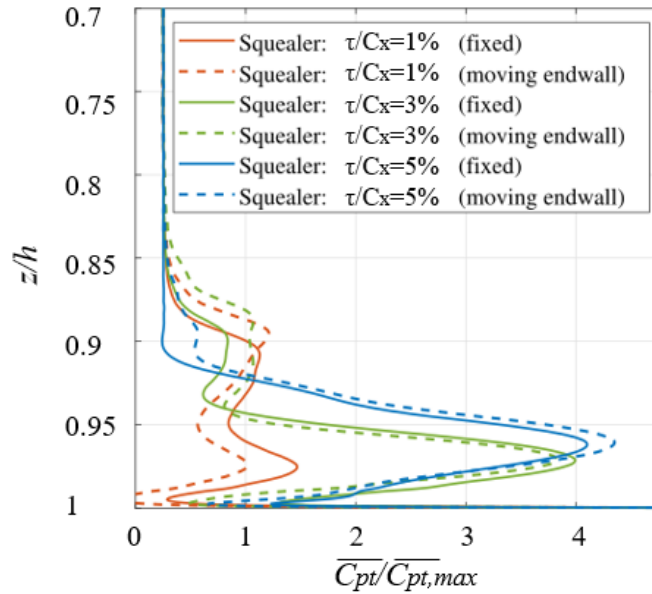


Figure 5.63: Normalized RANS $\overline{C_{pt}}$ distributions for the squealer tip under no-cooling conditions, comparing fixed (solid lines) and rotating (dotted lines) endwalls at different tip gap heights [301].

To conclude, the comparison between numerical and experimental results validated both low- and high-fidelity simulations. This demonstrated that low-cost approaches are capable of capturing the impact of active and passive systems on tip leakage flow development, proving to be reliable tools for the design of advanced tip and cooling configurations. For a closer agreement with the experimental data, LES provides superior accuracy, offering valuable insights into the mechanisms responsible for mitigating the tip leakage vortex. Importantly, both simulations and experiments showed that squealer and, more

effectively, squealer–winglet tip configurations reduce both secondary and tip losses, as well as the flow distortions induced by vortical structures at the cascade exit, compared with the flat tip configuration. This suggests a likely improvement also in the performance of the downstream cascade.

All these analyses are presented in several of the author’s works: see [287] for the squealer study and [300] for the squealer–winglet investigation. Other complementary studies include [301] and [304].

Chapter 6

Conclusions

This study focused on the experimental investigation of different secondary flow control devices aimed at reducing secondary flows in modern, compact, and high specific thrust aero-engines. Fences and endwall ridging designs were tested within a low-aspect ratio nozzle guide vane cascade installed in a diverging meridional channel, with the objective of assessing their capability to mitigate the passage vortex losses, comparable in magnitude to profile ones. In parallel, squealer and squealer-winglet tip configurations, also tested with different coolant mass flow rates, were evaluated as replacements for a flat tip on unshrouded low pressure turbine blades, to determine their effectiveness in reducing the tip leakage vortex under various tip gap sizes. To assess the effectiveness of the devices, two extensive experimental campaigns were conducted on both NGV and LPT cascades, in baseline and controlled configurations. In addition to the experiments, numerical simulations were conducted by Morfo Design and validated against the measurements. Both low- and high-fidelity simulations provided detailed insights into the flow fields of all configurations, enabling a deeper understanding of the mechanisms by which the different devices alter the flow and mitigate secondary losses.

For the NGV cascade, the controlled configurations included: fences installed on the blade suction side, conventional ridges applied to the divergent endwall, a combined configuration with fences and conventional ridges, and an optimized endwall ridging design. The results demonstrated that fences effectively reduce the penetration of the passage vortex, confining it closer to the endwall. Endwall ridges (especially the optimized design) were shown to mitigate the passage vortex by reducing both its intensity and size. The combined fence-ridge configuration exhibited cumulative effects, simultaneously mitigating secondary vortices and limiting PV loss core penetration. Total pressure losses remained approximately constant or slightly increased when fences, conventional ridges, or their combination were applied; however, a slight reduction was observed with the optimized ridging, which more effectively counteracts cross-passage secondary flows. Most importantly, all these device configurations significantly improved flow uniformity at the NGV exit. This improvement produces a beneficial carry-over effect, enhancing

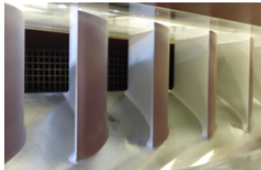
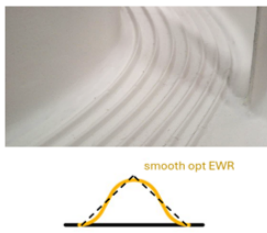
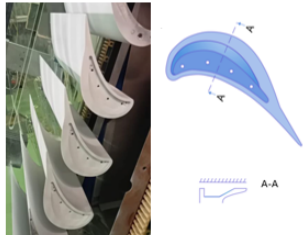
the downstream rotor performance and increasing the turbine stage efficiency by approximately 0.2–0.25% (as estimated from LES). These findings indicate that flow uniformity downstream of the NGV is the main driver behind the improved stage performance provided by the different devices; by reducing secondary flow penetration, they positively affect both flow uniformity and overall stage losses (particularly the optimized endwall ridging design).

Regarding the LPT cascade, in the absence of coolant injection, the squealer tip reduced tip losses by approximately 20% relative to the flat tip, while the squealer-winglet achieved a reduction of about 38%. Both configurations promote a more axial alignment of the leakage flow in the tip region compared with the flat tip. This improved behavior results from a more uniform pressure-coefficient distribution along the tip surface, which decreases the amount of flow leaking from the pressure to the suction side and forces it to exit the cavity through a smaller portion of the suction-side rim or winglet. As a consequence, a weaker and more compact TLV loss core develops, enhancing flow uniformity downstream of the cascade and reducing secondary vortex penetration particularly in the squealer-winglet configuration. The analysis of coolant flow rate effects showed that increasing coolant injection deteriorates the aerodynamic performance of both squealer and squealer-winglet tips, although the latter remains more effective than the squealer for low coolant mass flow rates. At a coolant mass flow ratio of approximately 0.9% of the main flow, however, its aerodynamic performance nearly deteriorates to that of the squealer case. Nonetheless, coolant injection, especially at higher flow rates, promotes a more uniform pressure distribution within the cavity, which is expected to provide thermal benefits. All collected LPT data were subsequently used to update the tip-loss correlations available in the literature, extending their applicability to configurations featuring different tip geometries and various coolant mass-flow rates.

Thus overall, this work provides new insights into the aerodynamic effectiveness of a broad range of secondary flow control and mitigation strategies for NGVs and LPT blades of modern aero-engines. The most effective device for each cascade, its quantitative impact on the primary metrics, and the corresponding operating conditions are summarized in Table 6.1, providing a blueprint for the selection of the most promising flow-control architectures.

However, the authors wish to emphasize that while the underlying aerodynamic concepts and the identified flow-control features can be generalized, the specific geometric parameters presented herein should not be directly extrapolated. To ensure optimal performance, a dedicated design optimization is recommended, tailored to the specific application—particularly when operating conditions (e.g., channel and blade geometry, tip gap height, coolant mass flow ratio, and Reynolds number) fall outside the ranges investigated in this study.

Table 6.1: Summary of the best performing devices, operating conditions, and performance improvements for the NGV and unshrouded LPT cascades.

Nozzle Guide Vane		Operating Conditions	Optimized Endwall Ridging	Performance improvement
		$Re_x=400'000$		ω_s -2% $C_{a,2}$ -32% Relative to the baseline configuration
Unshrouded Low Pressure Turbine		Operating Conditions	Squealer-Winglet Tip	Performance improvement
		$Re_x=200'000$ No coolant At all tip gap heights		At $\tau/C_x=5\%$ ω_s -38% $C_{a,2}$ -28% Relative to the flat-tip configuration

Appendix 1

A brief focus on the uncertainty and repeatability of experimental measurements is reported in the following.

Uncertainty of Experimental Measurements

Table A1.1 provides a summary of the instrumentation, including probe types, calibration requirements, acquisition systems, and the relative uncertainty for each primary variable.

Table A1.1: Instrumentation summary and measurement specifications.

PROBE	CALIBRATION	ACQUISITION SYSTEM	SPECIFICATIONS	UNCERTAINTY
Hot Wire Anemometer 55P14	☑	Constant Temperature Anemometer Module with DAQ-NI acquisition system	400'000 samples 40 kHz sampling frequency - Overheat ratio=0.8 20 kHz anti-aliasing LP filter	Velocity uncertainty < 1% (Propagation Analysis 1)
Static pressure taps (int. Ø 0.6 mm)	☒	Scanivalve Digital Sensor Array with 16 differential pressure transducers	250 samples per tap 500 Hz sampling frequency - Pressure range 2500 Pa - Accuracy ± 0.08% full scale	c_p uncertainty < 1.5% (Propagation Analysis 2)
Kiel (head Ø 1.5 mm)	☒	SETRA differential transducers	5000 samples 1 kHz sampling frequency - Pressure range ± 620 Pa - Accuracy ± 0.075% full scale	C_{pt} uncertainty ≈ 2% (Propagation Analysis 3)
Five-hole probe (head Ø 2.5 mm)	☑			α_2 uncertainty ≈ 0.2°; For $C_{\alpha 2}$ ≈ 0.5% (Propagation Analysis 4)

Propagation Analysis 1: Flow Velocity (v) The flow velocity v is derived from the anemometer output voltage E and the zero-flow voltage E_0 (measured before each test) using King's Law:

$$v = \left(\frac{E^2 - E_0^2}{B} \right)^{1/n}$$

The total uncertainty δv accounts for the following sources:

- Voltage Measurement ($\delta E, \delta E_0$): Derived from the NI-DAQ resolution and electronic noise, estimated at $\pm 1 \times 10^{-4}$ V.
- Calibration Constant (δB) and Exponent (δn): Derived from the non-linear least-squares fit. They account for both the regression residuals and the uncertainty of the reference velocity source (pressure transducer and ambient conditions) used during the calibration procedure ($B = 2.5923 \pm 0.0122$ and $n = 0.4894 \pm 0.0019$).

The relative uncertainty $\frac{\delta v}{v}$ is calculated through logarithmic sensitivity propagation:

$$\frac{\delta v}{v} = \frac{1}{n} \sqrt{\left(\frac{\delta(E^2 - E_0^2)}{E^2 - E_0^2} \right)^2 + \left(\frac{\delta B}{B} \right)^2 + (\ln(v) \cdot \delta n)^2}$$

where $\delta(E^2 - E_0^2) = \sqrt{(2E \cdot \delta E)^2 + (2E_0 \cdot \delta E_0)^2}$. For the investigated velocity range, the uncertainty remains below 1%.

Propagation Analysis 2: Pressure Coefficient (c_p) The pressure terms in c_p (Eq. 4.12) are measured via the Scanivalve DSA system. Assuming independent measurements with an absolute uncertainty δp , the relative uncertainty $\delta c_p / c_p$ is:

$$\frac{\delta c_p}{c_p} = \sqrt{\left(\frac{\sqrt{\delta p_t^2 + \delta p^2}}{p_t - p} \right)^2 + \left(\frac{\sqrt{\delta p_t^2 + \delta p^2}}{p_t - p_2} \right)^2}$$

For the operational range, the cumulative uncertainty is lower than 1.5%.

Propagation Analysis 3: Total Pressure Coefficient (C_{pt}) and Loss (ω_t) For C_{pt} (Eq. 4.14), pressures are measured by SETRA transducers (absolute error δp). The relative uncertainty of C_{pt} is propagated as:

$$\frac{\delta C_{pt}}{C_{pt}} = \sqrt{\frac{\delta p_{t,1}^2 + \delta p_{t,2}^2}{(p_{t,1} - p_{t,2})^2} + \frac{\delta p_{t,1}^2 + \delta p_2^2}{(p_{t,1} - p_2)^2}}$$

The resulting error is approximately 2%. Regarding the total loss coefficient ω_t , the integration process minimizes random fluctuations, leaving only the systematic transducer uncertainty:

$$\delta \omega_t \approx \delta C_{pt} \approx 2\%$$

Propagation Analysis 4: Yaw Angle (α) and Angular Distortion ($c_{\alpha,2}$) The uncertainty of the flow yaw angle $\delta\alpha$ combines mechanical positioning errors and pressure-induced sensitivity related to the coefficient CP_{32} obtained through the five-hole probe calibration:

$$\delta\alpha = \sqrt{\delta\alpha_{mech}^2 + \left(\frac{\partial\alpha}{\partial CP_{32}} \delta CP_{32} \right)^2}$$

where $\frac{\partial\alpha}{\partial CP_{32}}$ represents the local sensitivity derived from the calibration curve slope. Given the SETRA transducers accuracy and a mechanical positioning resolution of $\pm 0.05^\circ$, the cumulative uncertainty for the yaw angle is approximately 0.2° within the calibrated range of $\pm 30^\circ$.

Regarding the angular distortion coefficient $c_{\alpha,2}$, the uncertainty is further mitigated by the large sample size N used for averaging ($\delta\bar{\alpha} = \delta\alpha/\sqrt{N}$):

$$\frac{\delta c_{\alpha,2}}{c_{\alpha,2}} = \sqrt{\frac{\delta\bar{\alpha}_2^2 + \delta\bar{\alpha}_{2,mid}^2}{(\bar{\alpha}_2(z) - \bar{\alpha}_{2,mid})^2} + \left(\frac{\delta\bar{\alpha}_{2,mid}}{\bar{\alpha}_{2,mid}} \right)^2}$$

Due to this statistical reduction and the operational range of the tested angles, the cumulative relative uncertainty for $c_{\alpha,2}$ is estimated at approximately 0.5%.

Repeatability of Experimental Measurements

For completeness, the repeatability of the experimental measurements, specifically regarding the pitchwise-averaged C_{pt} , is shown in Figure A1.1 for (a) the NGV cascade and (b) the LPT cascade. The experimental campaigns demonstrated excellent consistency: for both NGV and LPT configurations, variations in secondary losses ω_s were found to be within 0.2%, confirming the reliability of the measurement chain and the stability of the test conditions.

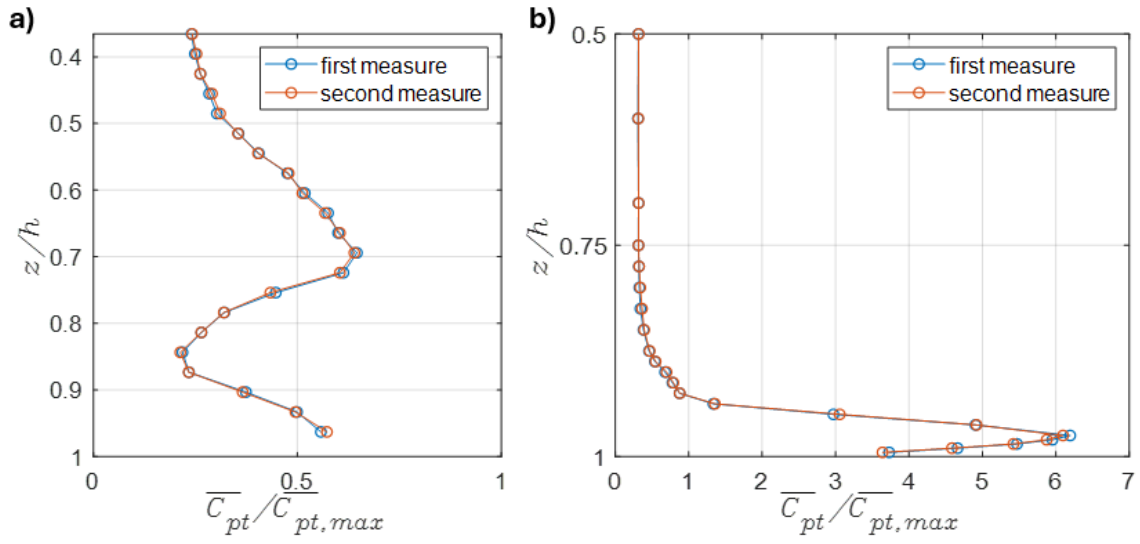


Figure A1.1: Repeatability of the $\overline{C}_{pt}$ distributions for (a) NGV and (b) LPT cascades.

Appendix 2

The main characteristics of the numerical simulation setup conducted by Morfo Design on both NGV and LPT cascades are reported in the following.

Numerical Setup for NGV Cascade Configurations

CFD has been extensively employed in the development of fences and both EWR versions for the NGV cascade, playing a key role in the design, optimization, and performance assessment phases. In particular, RANS simulations were used as an effective tool for device design, leveraging prior experience where they proved to be successful in developing solutions aimed at mitigating secondary flows [22], [305]. Additionally, CFD was applied in the post-test phase on the experimental validation rig, where both RANS and LES analyses were performed. RANS simulations were primarily aimed at verifying the accuracy of low-fidelity approaches and assessing their applicability during the design phase. In contrast, LES, after initial validation against experimental data, were employed to gain a deeper physical understanding of the effects of the different devices on the flow.

Both LES and RANS calculations were carried out using the finite volume software STAR-CCM+. All RANS simulations, including those for both stage optimization and rig conditions, employed the SST $k - \omega$ turbulence model proposed by Menter [306], enhanced with Durbin's realizability constraint [307], and coupled with the $\gamma - Re_\theta$ transition model [308] to capture transitional flows. The optimization was conducted considering a full turbine stage and realistic Reynolds and Mach numbers. Boundary conditions were defined in terms of total pressure and temperature at the NGV inlet, and static pressure at the rotor outlet. A turbulent boundary layer with a thickness of 1% of the span was imposed at the NGV inlet. Periodic boundary conditions were applied in the pitchwise direction, while no-slip conditions were enforced on the endwalls and blade surfaces. A preliminary sensitivity analysis was performed with mesh sizes of 4M, 6M, and 9M cells to determine the optimal resolution. For all mesh levels, Δn^+ was kept below 1, with local refinement near the endwall to accurately capture secondary flows and the presence of both fences and ridges. Moving from one mesh level to the next, refinement was applied with a target reduction factor of 0.8 for the polyhedral cell size. The 6M-cell mesh used

for optimization was unstructured, featuring polyhedral cells in the freestream region and prism layers near the walls. For comparison with experimental data, cascade simulations were performed. The computational domain extended from one axial chord upstream of the leading edge to 1.5 axial chords downstream of the trailing edge. Periodic conditions were imposed in the pitchwise direction, while no-slip conditions were enforced on the endwalls and blade surfaces. An incompressible segregated flow approach was adopted to match the low-speed conditions of the test rig, ensuring second-order accuracy in both spatial and temporal domains, since the flow within the cascade is governed by the incompressible Navier–Stokes equations due to the low Mach number conditions (the exit Mach number for $Re_x = 400'000$ is approximately 0.08). The inlet boundary conditions, including velocity profiles, turbulence intensity, and length scales, were carefully chosen to replicate the conditions measured at plane D_1 , ensuring that RANS reproduces the experimental conditions as described in [290]. The cascade numerical domain and setup used for RANS simulations were the same as those employed for LES. Specifically, for LES, the governing equations were discretized adopting the WALE subgrid-scale model [309]. At the inlet, velocity profiles were prescribed, and freestream turbulence was generated using the Synthetic Eddy Method (SEM) [310]. For grid generation, polyhedral cells were employed in the freestream region, while prism cells were used near wall boundaries. On the blade surface, the cell sizes were selected to achieve a mesh resolution along the blade curvilinear abscissa Δs^+ and along the blade span Δz^+ of approximately 15–25, while in the direction normal to the blade surface Δn^+ remained below 0.5, as shown in Figure A2.1.a-b. All reported parameters are made non-dimensional using the inner-unit turbulence length. In the freestream region, the cell size was approximately ten times the local

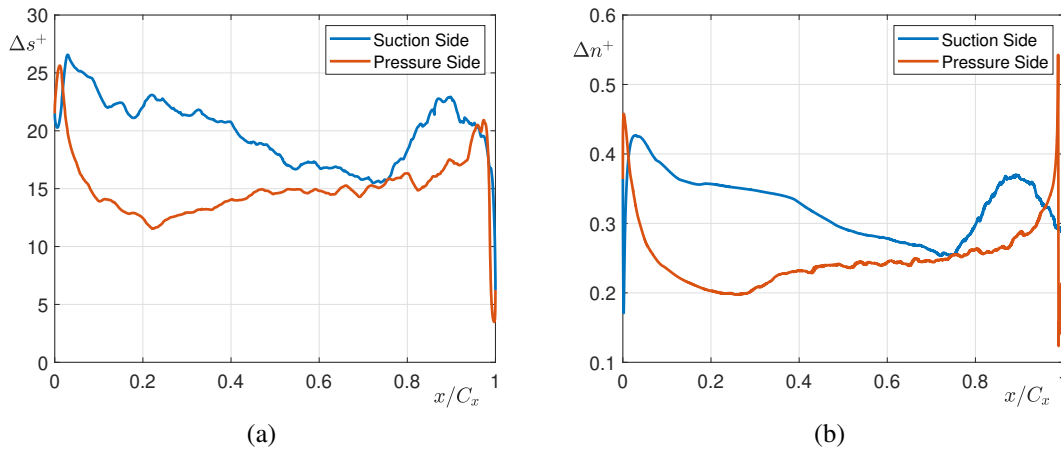


Figure A2.1: Non-dimensional mesh resolution along the blade surface at midspan. (a) Streamwise direction. (b) Wall-normal direction.

Kolmogorov length scale, which was estimated based on prior RANS calculations. The final computational grid consisted of approximately 140 million cells. Simulations were conducted ensuring that a Courant number (CFL) close to unity was maintained around the blade. The LES was initialized using the RANS solution. An initial transient phase, corresponding to four flow-through times (FTT), was run to allow the flow to fully develop and eliminate any transient effects that could influence the final results. Following this phase, the primary flow quantities were averaged over additional four FTT to ensure adequate convergence. As an example, Figure A2.2.a, and particularly the zoomed portion in Figure A2.2.b, show the normalized exit yaw angle computed using different averaging durations (2 FTT, 3 FTT, and 4 FTT), along with an estimate of the convergence error calculated following the procedure suggested by M. Bergmann et al. [311].

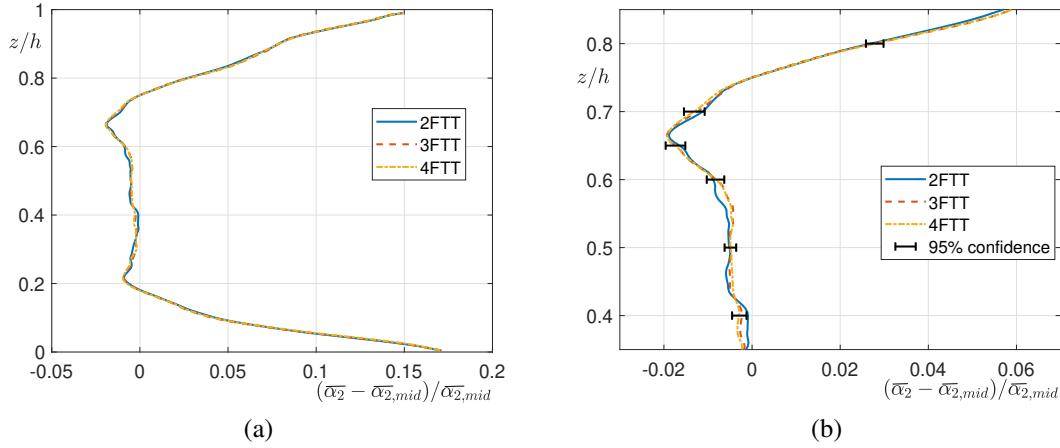


Figure A2.2: Normalized exit flow angle computed for different average times and estimation of convergence error. On the right a zoomed-in view of the figure to better highlight the distributions.

Numerical Setup for LPT Cascade Configurations

The numerical campaign conducted on the LPT cascade was carried out through a synergic approach, integrating both LES and RANS methods. LES were specifically employed to demonstrate the aerodynamic benefits of the different tip designs. Following initial validation against experimental data, LES were used to gain deeper insight into the physical mechanisms by which the squealer and squealer-winglet configurations modify the flow fields. In parallel, the RANS simulations allowed assessment of the low-fidelity approach in reproducing experimental trends, evaluating not only the effects of different tip geometries, but also the influence of tip clearance height and cooling mass flow ratio on the TLV.

This is particularly relevant, as RANS represents the primary tool used during the design phase.

Both RANS and LES simulations were performed using the incompressible finite solver STAR-CCM+. RANS calculations employed the $k - \omega$ SST turbulence model [306], enhanced with Durbin’s realizability constraint [307], and coupled with the $\gamma - Re_{\Theta}$ transition model. The simulations were carried out on an unstructured polyhedral grid, as shown in Figure A2.3. Thirty prism layers were used along the blade and endwall surfaces to accurately resolve the boundary layer, with the height of the first cell chosen to ensure $y^+ < 1$. The resulting total grid consisted of 8.5 million cells. Cell sizes on the blade surfaces, including the tip, within the tip clearance, and in the wake region were parametrized. This allowed a mesh sensitivity study by varying the total number of cells between 5 and 11 million. The selected mesh provided the best compromise, as negligible differences were observed between the baseline and refined grids, whereas the coarse mesh exhibited minor but noticeable variations in the pressure loss distribution. LES sim-

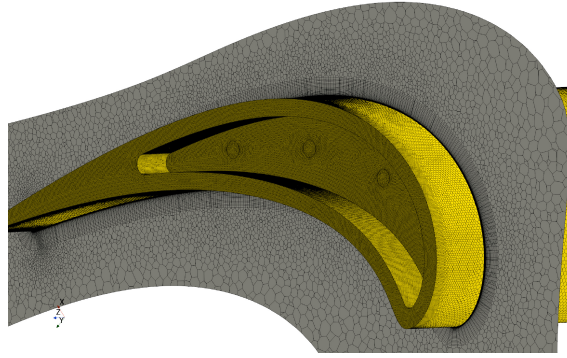


Figure A2.3: Details of the numerical grid.

ulations employed the WALE subgrid scale model [309] coupled with the Synthetic Eddy Method (SEM) [310] for turbulence generation at the cascade inlet. The computational mesh was fully unstructured with polyhedral cells, while a structured prism layer was generated around the blade. Also in this case, the first cell height was prescribed to ensure $y^+ < 1$ over the entire blade surface. In addition to controlling the wall-normal spacing, the streamwise and spanwise resolutions along the blade were set to achieve $x^+, z^+ < 25$, providing the near wall resolution required for wall resolved LES. In the freestream region, the mesh resolution was based on eight times the Kolmogorov length scale estimated from a precursor RANS computation. Time integration was performed using a timestep of 4.0×10^6 s, ensuring a CFL number below 1 throughout the domain. Statistical convergence was achieved by running an initial 4 FTT to allow the flow to fully develop, followed by at least 4 additional FTT for the computation of time-averaged quantities, consistent with the procedure used for the NGV cascade simulations.

RANS and LES simulations share the same numerical domain, which is schematically

shown in Figure A2.4. The computational domain extends for $3.5 C_x$ in the streamwise direction, with the inlet positioned one axial chord upstream of the blade leading edge. At the inlet, the experimentally measured total pressure profile and flow angle are prescribed, while the static pressure is imposed at the outlet. No-slip boundary conditions are enforced on the blade and casing surfaces. A symmetry plane is applied on the opposite boundary, allowing the domain height to be reduced to half the hub-to-casing distance without affecting the results, as the mean flow is largely two-dimensional over the blade span. Periodic boundary conditions are imposed in the pitchwise direction. For the cases with tip injection, the exit velocity through each hole (with area A_c) is uniform and normal to the cavity floor. The velocities v_{c_i} at each hole are determined by solving:

$$\begin{cases} p_{c,0} + 0.5\rho v_{c_i}^2 = p_{c_i} \\ 4\rho A_c \sum_{i=1}^4 v_{c_i} = \dot{m}_c \end{cases}$$

where $p_{c,0}$ is the static pressure at the hole location without coolant flow, $p_{c,i}$ is the local pressure at the same location when the coolant is injected, and $\dot{m}_c$ is the desired coolant mass flow rate.

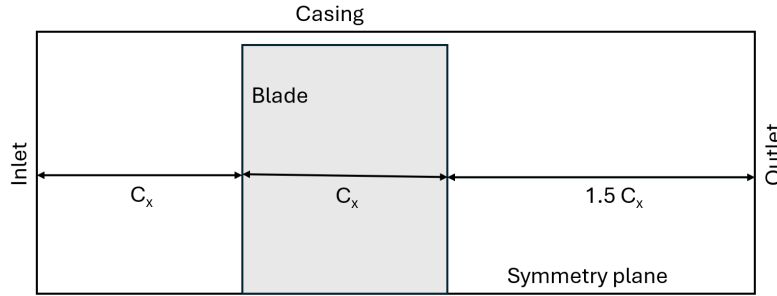


Figure A2.4: Sketch of the numerical domain

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