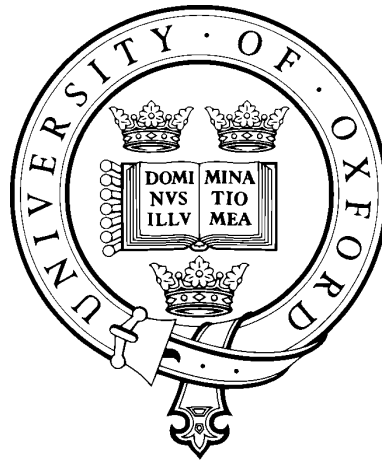


Aero-Thermal Performance and Enhanced Internal Cooling of Unshrouded Turbine Blade Tips



Amandeep S. Virdi
Lady Margaret Hall
University of Oxford

A thesis submitted in partial fulfilment of the requirements for the degree of

Doctor of Philosophy

Hilary Term 2015

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Abstract

The tips of unshrouded, high-pressure turbine blades are prone to significantly high heat loads. The gap between the tip and over-tip casing is the root cause of undesirable over-tip leakage flow that is directly responsible for high thermal material degradation and is a major source of aerodynamic loss within a turbine. Both must be minimised for the safe working and improved performance of future gas-turbines.

A joint experimental and numerical study is presented to understand and characterise the heat transfer and aerodynamics of unshrouded blade tips. The investigation is undertaken with the use of a squealer or cavity tip design, known for offering the best overall compromise between the tip aerodynamics, heat transfer and mechanical stress. Since there is a lack of understanding of these tips at engine-realistic conditions, the present study comprises of a detailed analysis using a high-speed linear cascade and computational simulations. The aero-thermal performance is studied to provide a better insight into the behaviour of squealer tips, the effects of casing movement and tip cooling. The linear cascade environment has proved beneficial for its offering of spatially-resolved data maps and its ability to validate computational results. Due to the unknown tip gap height within an entire engine cycle, the effects of gap height are assessed. The squealer's aero-thermal performance has been shown to be linked with the gap height, and qualitative different trends in heat transfer are established between low-speed and high-speed tip flow regimes.

To the author's knowledge, the present work is the first of its kind, providing comprehensive aero-thermal experimental research and a dataset for a squealer tip at engine-representative transonic conditions. It is also unique in terms of conducting direct and systematic validations of a major industrial computational fluid dynamics method for aero-thermal performance prediction of squealer tips at engine-representative transonic conditions.

Finally, after recognising the highest heat loads are found on the squealer rims, a novel shaped squealer tip has been investigated to help improve the thermal performance of the squealer with a goal to improve its durability. It has been discovered that a seven percent reduction in tip temperature can be achieved through incorporating a shaped squealer and maximising the internal cooling performance.

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Nomenclature

List of symbols

Symbol	Description	Units
A	Area	[m ²]
Bi	Biot number	[-]
C_β	Calibration coefficient for yaw angle	
C_M	Calibration coefficient for Mach number	
Cp_0	Total Pressure Loss Coefficient (compressible flow)	[-]
Cp_b	Base Pressure Coefficient	[-]
C_T	Calibration coefficient for total pressure	
Cx	Blade axial chord	[m]
E	Total energy	[J]
F	Force	[N]
$\vec{F}$	Flux vector	
$\vec{F}^*$	Time-averaged flux vector	
G	Greyscale value	
H	Cavity/rim height	[m]
I	Turbulence intensity	
L	Length scale	[m]
M	Mach number	[-]
	<i>or</i> Mass flux	[kgm ⁻² s ⁻¹]
N_F	Number of control faces	
P	Blade pitch	[m]
Q	Source term	
$\bar{Q}^n$	Cell-averaged source term at time $n\Delta t$	
$\bar{Q}^*$	Cell and time-averaged source term	
R	Residual	
	<i>or</i> Universal Gas Constant	[Jkg ⁻¹ K ⁻¹]
	<i>or</i> Span-wise distance from blade root	[m]
Re	Reynolds number	[-]
S	Surface	
	<i>or</i> Blade span	[m]
	<i>or</i> Total curve length of suction side	[m]
T	Temperature	[°C] or [K]
U	Flow velocity	[ms ⁻¹]
V	Volume	[m ³]
V_x/U	Flow coefficient	[-]
W	Conservation variable	
	<i>or</i> Rim width	[m]
$\bar{W}^n$	Cell-averaged conservation variable at time $n\Delta t$	
Y	Stagnation Pressure Loss Coefficient	[-]
a	Acceleration	[ms ⁻²]

c_p	Specific heat capacity for constant pressure	$[\text{Jkg}^{-1}\text{K}^{-1}]$
g	Tip gap	$[\text{m}]$
$\vec{g}$	Gravity acceleration	$[\text{ms}^{-2}]$
h	Heat transfer coefficient or Impulse response or Step size	$[\text{Wm}^{-2}\text{K}^{-1}]$
k	Turbulent kinetic energy or Thermal conductivity or Runge-Kutta coefficient or Summation variable	$[\text{Jkg}^{-1}]$ $[\text{Wm}^1\text{K}^1]$
l	Length scale	$[\text{m}]$
m	Mass or Cell face identity	$[\text{kg}]$
$\dot{m}$	Mass flow rate	$[\text{kgs}^{-1}]$
n	Numerical integration variable or Sample number	
p	Pressure	$[\text{Pa}]$
$\dot{q}$	Heat flux	$[\text{Wm}^{-2}]$
s	Entropy or Local position on suction side	$[\text{Jkg}^{-1}\text{K}^{-1}]$ $[\text{m}]$
t	Time	$[\text{s}]$
u, v, w	Velocity vector components	$[\text{ms}^{-1}]$
$\vec{v}$	Velocity vector	$[\text{ms}^{-1}]$
x, y, z	Cartesian coordinates or Axial, pitch, radial positions	$[\text{m}]$
y^+	Non-dimensional wall distance	$[-]$

Greek

Symbol	Description	Units
Δ	Difference	
Ω	Finite control volume	
α	Thermal diffusivity	$[\text{m}^2\text{s}^{-1}]$
β	Yaw angle	Degrees
γ	Ratio of specific heats	$[-]$
ϵ	Turbulent kinetic energy dissipation rate	$[\text{Jkg}^{-1}\text{s}^{-1}]$
ζ	Kinetic energy loss coefficient	$[-]$
η	Film cooling effectiveness	$[-]$
θ	Rim inclination angle	Degrees
ν_t	Turbulent kinematic eddy viscosity	$[\text{m}^2\text{s}^{-1}]$
ρ	Density	$[\text{kgm}^{-3}]$
τ	Dummy time variable	$[\text{s}]$
$\bar{\tau}$	Stress tensor	$[\text{Pa}]$
ω	Turbulent dissipation rate	$[\text{s}^{-1}]$

Subscripts

Symbol	Description
0	Stagnation
1	Inlet
∞	Mainstream
A, B, C	Left, center and right three-hole probe holes
I	Nodal point index
ad	Adiabatic
avg	Average of three-hole probes channels A and C
c	Coolant temperature or cooled
$exit$	Exit condition
f	Fluid
h	Horizontal (x-y) plane
i	Mesh point location in x direction <i>or</i> initial time <i>or</i> internal
$in/inlet$	Inlet
j	Mesh point location in y direction
k	Mesh point location in z direction
m	Mainstream
n	Normal
rec	Recovery temperature
s	Solid
ss	Suction side
w	Wall

Abbreviations

1D	One-dimensional
2D	Two-dimensional
3D	Three-dimensional
CAD	Computer Aided Design
CFD	Computational Fluid Dynamics
CHT	Conjugate heat transfer
DC	Direct current
DES	Detatched Eddy Simulation
DNS	Direct Numerical Simulation
EFE	Environmentally Friendly Engine
GDP	Gross Domestic Product
HSLC	High-Speed Linear Cacade
HTC	Heat transfer coefficient
LES	Large Eddy Simulation
OTL	Over-tip leakage

PID	Proportional-Integral-Derivative
RAF	Royal Air Force
RANS	Reynolds-averaged Navier Stokes
RNG	Renormalisation Group
RSM	Reynolds Stress Model
SA	Spalart-Allmaras
SFC	Specific fuel consumption
SST	Shear Stress Transport
TET	Turbine entry temperature

Chapter 1

Introduction

This thesis documents an experimental and numerical study of unshrouded gas turbine blade tips with an assessment of the aerodynamic and thermal performance. The main goal of this thesis is to develop a tip design concept to reduce the high thermal load associated with a widely-established unshrouded tip configuration, known as a squealer tip. The tip itself is prone to very high heat loads on its rim seals, but is often a preferred choice for its best overall performance. Without greatly compromising the aerodynamic benefit of a squealer tip, the goal is to reduce the tip heat transfer and to extend its life in the harsh environment it lives in. The thesis takes these aspects into consideration, but more importantly under engine-realistic aerodynamic flow conditions that are often neglected.

1.1 General Background

Born in Coventry, England, Sir Frank Whittle registered a patent in 1930 for the first turbojet engine whilst a German-born engineer, Dr. Hans von Ohain, was granted a similar patent for turbojet engine theory in 1936. Both are known as co-inventors of the jet engine, even though Whittle's patent was granted first. On 27th August 1939, Heinkel He 178 claimed the first flight using von Ohain's design, taking off from the Baltic Coast. This was shortly followed by flight of the Gloster E.28/29 on 15th May

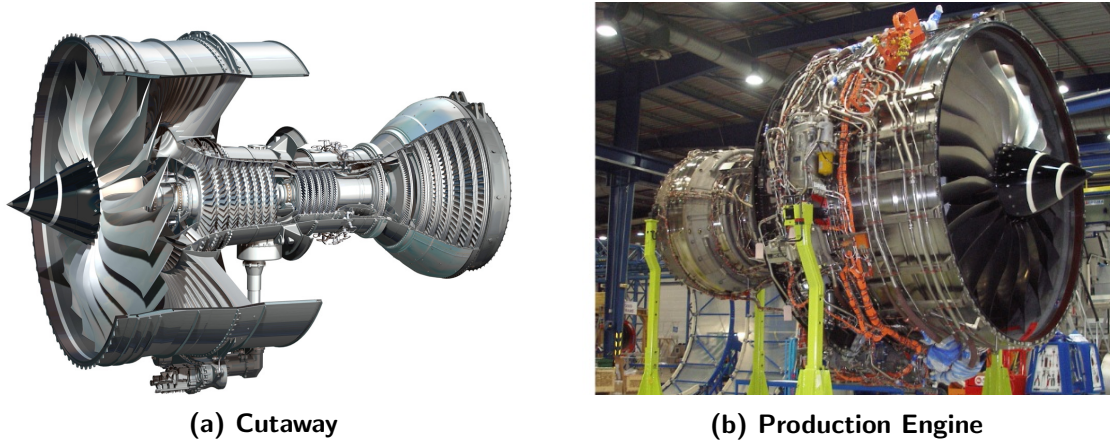


Figure 1.1: Rolls-Royce Trent XWB Turbofan Engine
(Images reproduced from (Nathan, 2011; Norris, 2013))

1941 from RAF Cranwell, testing Whittle's invention.

Since then, the revolution of air transportation lay with the gas turbine, inevitably based on the Whittle engine. Since the turn of the war, the growth and demands of civil air travel have been astonishing and have heavily relied on advancements in science and technology to improve the power and efficiency, and to reduce the carbon footprint of the modern gas turbine. In 2014, the world's busiest airports were Dubai and London Heathrow reporting in excess of 68.1 million passengers travelling long-haul (Anderson, 2015), signifying the importance and requirement of cleaner, quieter, more efficient and more durable gas turbine engines. Providing 3.6% of the UK GDP through the aviation industry (Economic Benefits from Air Transport in the UK, 2011) and the proposed plan to expand London Heathrow by introducing a third runway shows an economic reliance on improved and advanced turbomachinery.

A modern, high-bypass ratio gas turbine engine is shown in Figure 1.1. A high bypass ratio helps to increase the thrust and reduce the specific fuel consumption (SFC), and are designed to meet requirements of new engines and produce a lower noise output than low or zero-bypass designs. The low-pressure turbine stage at the exit of the engine core provides power to the front fan. To further increase the power supplied to this fan, the power generated in the engine core has to be increased and is

achieved either by increased turbine entry temperatures or increased mass flow rates to the core. However, to increase the mass flow rate, the size and weight of the core have to be increased and poses a limiting factor.

Given the state-of-the-art, in order to meet the requirements of increased power and efficiency of new aero-engines, attention has been drawn for mechanisms to increase the rotational speed of the turbine stage and to increase the turbine entry temperature (TET). Naturally, the considered choices are plausible, but from a design point of view, they must be well balanced with the imposed thermal and mechanical stresses.

Within the core of a modern engine, the high-pressure turbine typically uses one of two types of rotor blade. These come in the form of either shrouded or unshrouded blades (Figure 1.2). A shrouded blade tip is one which carries a shroud which forms a link with its neighbouring blades and is therefore, part of a continuous ring. The main objective of a shrouded blade tip is to act as a seal to limit the leakage flow in the radial direction. Since the shroud contributes to a significant portion of the total mass of each blade, it is subject to high centrifugal stresses. This therefore limits the rotational speed of shrouded rotor blades.

Eliminating the shroud eliminates these limitations and can result in faster rotational speeds. However, the removal of the shroud is at the expense of fluid leakage and thus reduced turbine efficiency. Shrouded blades are equipped with fins and fences that help provide blockage. Without these, the gap between the blade tip and the casing wall gives rise to increased fluid leakage over the tip due to the pressure difference between the two aerofoil surfaces. Mixing losses, as well as fluid interaction with the endwall flow, all attribute to the losses generated over the tip region. The greater the distance of the blade tip to the casing wall, the greater the loss.

Sealing techniques are often employed to minimise the unavoidable leakage losses. As previously described, one method is by the use of a shroud. However, for un-



(a) Shrouded Tip: Rolls-Royce Trent 800 High-Pressure Turbine Blade After 8299 hours (cleaned)



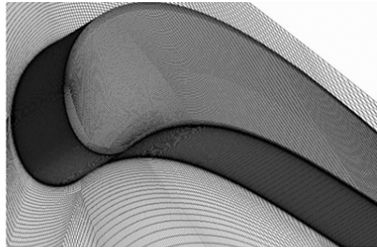
(b) Unshrouded Tip: GE High-Pressure Turbine Blade with Squealer Tip

Figure 1.2: Examples of shrouded and unshrouded turbine blade tips in operation (Reproduced from Bunker (2013))

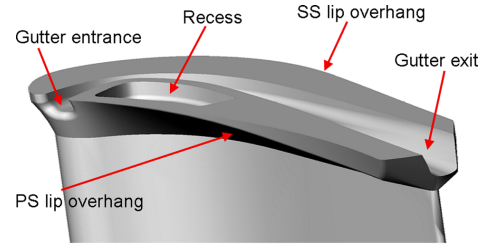
shrouded blades, techniques and research have frequently included the use of flat (plain), squealer (cavity), winglet and winglet-squealer (squealet) tips. Such designs are shown in Figure 1.3.

The choice of tip is subject to numerous advantages and disadvantages. A squealer tip acts as a double seal and the cavity vortex structures help to reduce the losses generated. Furthermore, it is a suitable tip choice since the contact area with the casing when under thermal expansion is much less than other tip contenders. However, the difficulty of cooling the thin rim walls of a squealer design are a cause for concern in terms of its life and durability. Tip choices such as the winglet were sought mostly for their aerodynamic benefit. The pressure difference between the pressure side and suction side surface can be reduced, thus reducing the leakage flow. However, the benefits are largely offset by the increased tip area exposed to the high temperature mainstream flow and also the difficulty of effectively cooling the tip.

The key to improving aero-engines is by increasing the TET, but it is also a function of the overall pressure ratio of the compression system. The major benefit



(a) Flat Tip (Reproduced from Zhang & He (2014))



(b) Winglet Tip (Reproduced from O'Dowd et al. (2013))



(c) Squealer Tip (Reproduced from Bunker (2013))



(d) Winglet-Squealer Tip (Reproduced from Schabowski et al. (2014a))

Figure 1.3: Various unshrouded tip geometries (Reproduced from a multiple sources)

provided by a higher TET is the increase in power generated (specific work), but is also capable of reducing the SFC; an important criterion for civil aero-engines, as shown in Figures 1.4a and 1.4b. Reduced or lower SFC not only results in reduced fuel costs and running costs of an aero-engine, but can also target environmental sustainability through lower carbon dioxide (CO_2) emissions.

Bearing in mind that in modern engines the TET is often hundreds of degrees Kelvin higher than the melting temperature threshold of the materials used, blade cooling is crucial. The survival of the blades is achieved through the desired material properties, thermal barrier coatings and by actively cooling components with air bled from the compressor. However, use of the cooling air is disadvantageous since it reduces the overall efficiency of the engine. In order to maximise the gain in efficiency through increased TET, the advancement of material properties such as superalloys

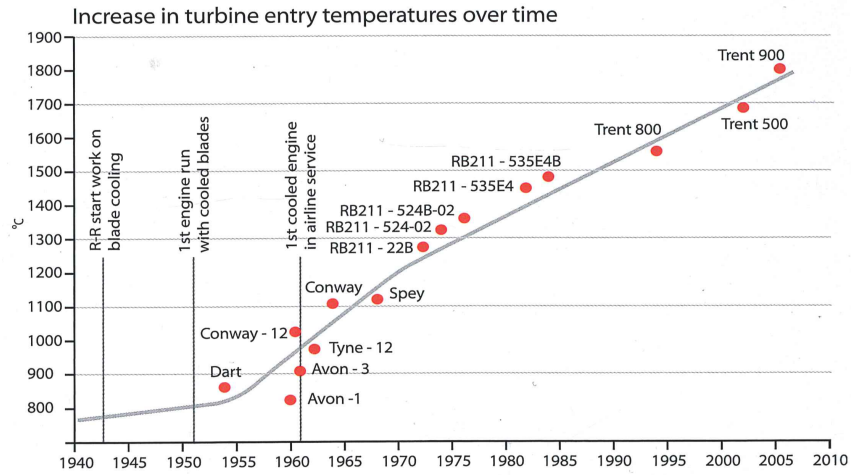
(Pollock & Tin, 2006) and advanced cooling schemes must be well addressed. As described by Friedrichs (1997), the life expectancy of a rotor blade can be doubled by reducing the mean temperature by only 15 K.

1.2 Implications of Over-Tip Leakage Flow

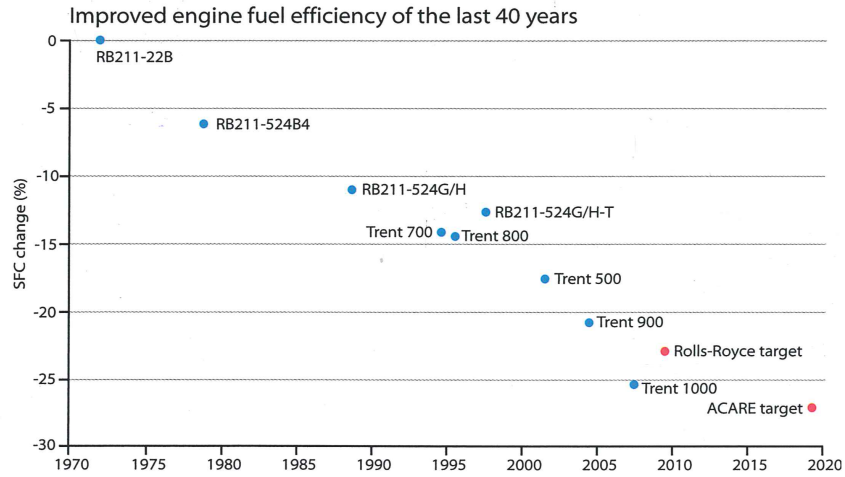
The challenges faced within the gas turbine industry to improve efficiency and performance, whilst keeping the costs and weight minimal can only feasibly be achieved by the processes described in Section 1.1. In order to effectively do so, the use of unshrouded tips is paramount since the more traditional shrouded tip approach introduces high stresses to the blade, tip, root and disc. Shrouds also increase weight and require a significantly larger amount of cooling to avoid damage. Transitioning to unshrouded tips means that tip clearance control becomes a fundamental issue. A clearance gap, or a tip gap, must exist to prevent rubbing with the casing endwall, but must also be small enough to prevent a high efficiency penalty.

The setting of the high-pressure turbine means that high temperature gas leaks over the tip because of the driving pressure potential between the pressure side and the suction side. This is known as over-tip leakage (OTL) flow. The loss caused by the OTL flow may be as high as one-third of the overall turbine stage losses (Booth, 1985). The tip flow itself reduces the extracted work done on the rotor since the fluid is not able to turn the blade. The losses themselves are caused by boundary layer viscous losses in the gap and mixing within and past the gap.

On top of the losses are the heat transfer issues associated with OTL flow. It is evident that the tip experiences high heat transfer rates because the flow is of a high temperature and has a high velocity. Additionally, since the tip is difficult to cool, the problem is exacerbated and is twofold. With high temperatures and a lack of effective cooling, burn-out, creep and structural failures are all accelerated and can



(a) Trend in TET over time



(b) Trend in SFC over time

Figure 1.4: Trends in Rolls-Royce aero-engines TET and SFC over time (Reproduced from Rolls-Royce (2005))

be collateral if parts of the tip break free.

Therefore, it is important that both the aerodynamic losses and heat transfer are minimised. One of the most effective methods to minimise the losses can be achieved through improved aerodynamic tip geometries, however this often has adverse effects in terms of heat transfer, as described previously. The present research focuses on these aspects. Firstly, to understand the effects of tip clearance on a reputable geometry, the squealer tip, with and without casing motion effects. Secondly, to understand

the effects of tip cooling on the squealer and examine the predictability of it through computational fluid dynamics, and thirdly, to address the high heat transfer issue without compromising the aerodynamic performance a squealer tip provides. Most importantly, the research is conducted under engine-realistic conditions since it has been recognised that low-speed studies do not accurately translate to those that are high-speed. Often high-pressure turbines experience transonic OTL flow and it is important that high-speed research is conducted and engine conditions are matched to better understand the OTL flow and heat transfer effects.

1.3 Motivation for Current Research

The motivation of the current research is to improve high-pressure turbine blades based on the aerodynamic and thermal constraints. It is of high interest to understand the aero-thermal performance of unshrouded tips to help improve current designs and meet the demands placed on aero-engine designers. The major motivation of this thesis is to improve the durability of high-pressure turbine blade tips through both aerodynamic and thermal perspectives.

Recent aero-thermal research of high-speed, high-pressure turbine blade tips has shown qualitatively different flow features to those seen in low-speed environments and simulations. For designers, it is important that the flow physics are well understood before an appropriate and optimal design can be achieved. This means that conducting tests at engine-realistic conditions is crucial. Perhaps from a practicality perspective, the research conducted by Moore *et al.* (1989) and Moore & Elward (1993) were overshadowed. Their early work on turbine blade tips had shown compressible effects and tip flow that was of a transonic nature. The formation of shocks and high heat transfer regions were reported and gave an indication of different flow physics found under a high-speed regime. The bulk of the research that followed

was mostly under low-speed conditions and therefore unable to capture any compressibility effects that would be seen in a real engine. Following the comparison of low-speed and high-speed flows using modern, advanced computational fluid dynamics methods, as documented by Wheeler *et al.* (2011), the significance of engine-realistic conditions became apparent. The experimental research by O’Dowd (2010) revealed a reversal in the heat transfer effects when comparing high and low-speed flows. Furthermore, compressible effects at a transonic condition similar to Moore *et al.* (1989) and Moore & Elward (1993) were discovered by Zhang *et al.* (2011) and its impact on the aerodynamic performance (Zhang & He, 2011). Following this, various research has acknowledged the importance of high-speed testing and analysis (Coull & Atkins, 2015; Lavagnoli *et al.*, 2013; Li *et al.*, 2014), in particular for blade tip heat transfer.

As described in Section 1.2, it is extremely important to minimise the amount of flow leaking over the tip region, both in terms of efficiency and durability. By addressing the aerodynamic performance only, designs such the winglet are highly effective. This geometry is most capable of reducing the pressure gradient that drives the tip leakage flow. However, research conducted by O’Dowd *et al.* (2013) revealed high levels of heat transfer mainly due to the increase in tip area. Although purposefully done to reduce the tip leakage and allow for extra film cooling holes, the durability issue is not truly accounted for. Therefore, the current research is motivated by the coupled aerodynamic and heat transfer effects and uses a practical tip design which offers high performance (a squealer tip). However, as will be outlined in Chapter 2, there are a lack of studies and detailed analyses on squealer tips under engine-realistic conditions, and to the author’s knowledge, none of the existing research has addressed the effect of tip clearance on heat transfer or addresses the durability issue under transonic conditions. For a squealer tip in particular, Figure 1.5 shows that the degradation of in-service blade tips. The lifespan of blade tips used in-service is high on the agenda along with methods to improve the tip cooling.



Figure 1.5: Distressed unshrouded turbine blade tips (Reproduced from Bunker (2013))

Lastly, it is recognised that true engine-realistic conditions are those that involve rotation. These offer the highest fidelity in terms of replicating a real engine. However, there are only limited facilities offering these conditions and often only capable of providing limited data. Therefore, the final motivation of this study aims to provide highly detailed aerodynamic loss and heat transfer maps to help thoroughly understand the aero-thermal performance under engine-realistic conditions. These results are also much better suited to validating computational studies (computational fluid dynamics) and should be of high value when progressing to higher fidelity testing and simulations to help aid the understanding of the physical flow features present in an engine-realistic setup.

1.4 Aims and Objectives

The following study is entirely focused on unshrouded turbine blade tips and draws attention to the favourable squealer tip. The current challenge is brought about by a lack of unshrouded engine experience for larger civil aircraft. The challenge to improve gas turbine performance through increasing TET and blade rotation speeds requires in-depth studies of both the heat transfer and the aerodynamics to help assist in the development of unshrouded blade tips. Therefore, the study aims to achieve the following:

1. Obtain both aerodynamic loss and heat transfer measurements on a squealer tip

at engine-representative conditions in a linear cascade (matching Mach numbers and Reynolds numbers)

2. Examine the effect of tip clearance on the aerodynamic loss and heat transfer using a squealer geometry in a linear cascade at engine-representative conditions
3. Obtain aerodynamic loss and heat transfer measurements on a cooled squealer tip at engine-representative conditions (limited to mainstream flow only) and examine the effect of tip clearance
4. Conduct Computational Fluid Dynamics simulations and validate Rolls-Royce's solver which is a major aerodynamic design tool and the Ansys CFX solver which is chosen for further conjugate heat transfer analyses of a novel shaped tip configuration
5. Based on validated Computational Fluid Dynamics, understand the effect of relative casing movement and examine tip clearance and geometrical effects
6. Develop an unshrouded tip to address the durability issue and further consolidate this with a conjugate heat transfer study

The main objective for the research is to provide detailed aerodynamic loss and heat transfer measurements at engine-realistic conditions using a given tip geometry to help develop durable unshrouded turbine tips. The study aims to prove a concept of a novel tip geometry that can maintain high aerodynamic performance whilst minimising the tip heat transfer. This is principally done through maximising the internal cooling capability for the near-tip region.

1.5 Thesis Structure

The structure of the thesis is as follows:

Chapter 2 is a review of the published literature outlining a detailed, but non-exhaustive overview of the relevant research on the aerodynamics and heat transfer of turbine blade tips.

Chapter 3 describes the theory and implementation of the computational methods used for the analyses of the tip aerodynamics and heat transfer. The chapter provides details of numerical solvers and performance calculations used.

Chapter 4 introduces the experimental facility and provides details of the Oxford University High-Speed Linear Cascade used to obtain the near-tip aerodynamic loss and tip heat transfer measurements. The chapter also describes the apparatus and measurement techniques used.

Chapter 5 presents the experimental measurements obtained using the High-Speed Linear Cascade using a squealer tip. Computational results are also presented and compared directly to the experimental measurements. A tip gap survey is conducted for both cases.

Chapter 6 continues the computational study on an uncooled squealer tip and presents results with relative casing movement. The chapter includes a tip gap survey and a comparison to a plain tip.

Chapter 7 presents the experimental measurements obtained using a cooled squealer tip in the High-Speed Linear Cascade. Results of a simple computational

study of the cooled squealer are also presented. A tip gap survey is also conducted for both cases.

Chapter 8 introduces theory behind a proposed concept of a shaped tip and provides experimental and computational results to serve as proof of the concept.

Chapter 9 follows the concept-proof and presents computational, conjugate heat transfer results on a shaped tip designed to enhance the cooling performance of unshrouded turbine tips to increase its in-service life.

Chapter 10 provides a summary of the research conducted, outlines the major conclusions and suggestions for future study.

The research presented in Chapter 5 has been presented at the ASME Turbine Blade Tip Symposium, in Hamburg, Germany in 2013 and has subsequently been published in the American Institute of Aeronautics and Astronautics in the Journal of Propulsion and Power (Virdi *et al.*, 2015). The concept tip studied in Chapters 8 and 9 has been patented by He *et al.* (2013) assigned to Rolls-Royce plc.

Chapter 2

Literature Review

2.1 Introduction

For over five decades, a vast amount of research has been undertaken to understand and develop efficient and durable high-pressure turbine blade tips, yet the design of unshrouded blades for large aero-engines remains a challenge. Due to the vulnerability and concern of the lifespan of these tips, the field has grown in terms of importance and popularity. The complex process of tip design involves minimising aerodynamic losses and reducing the heat transfer from the hot gas to the tip surface. Linking research knowledge and findings with real engine experience has become an increasingly vital aspect to not only meet expected service hours, but to extend the material durability. This is done by carefully considering the design choice and by implementing advanced cooling methods as proposed in this study. Given that recent studies have shown important compressibility effects, the objective becomes significantly more challenging in regards to engineering high performance, high-pressure turbine blades.

The majority of unshrouded tip research has focused mostly on low-speed flows with the goal of reducing the aerodynamic losses associated to the OTL flow. Thus, in order to minimise the losses, two widely established tip designs have been explored;

the squealer and the winglet. The squealer is essentially a plain or flat tip with a cavity or recess, producing a rim that acts as a fluid sealing mechanism. The winglet is, in essence, formed of pitchwise extensions of the blade at the tip region similar to those found on aircraft wings (such as on the Boeing 737-700) used to reduce the drag component. Tip designs have generally concentrated on reducing the aerodynamic losses and then estimating the heat load so that protective cooling can be applied. The awareness of the tip durability is acknowledged, however remains as one of the biggest challenges.

This chapter provides a review of relevant literature related to the tip leakage flow phenomenon and heat transfer studies of unshrouded turbine blades, and summarises the current state-of-the-art in tip aero-thermal design and performance.

2.2 Aerodynamics and Tip Leakage Flow

2.2.1 High-Fidelity Rotor Tests

The research into the fluid dynamics and thermodynamics of gas turbines has been carried out in numerous ways including both experimentally and numerically, both of which have relied on sound theory to understand the aero-thermal performance. In terms of being able to best replicate a real engine, high-fidelity rotational rigs are superior. They are considered to be the next best alternative and often replicate a complete stage of an engine. Their size, complexity, running costs and ability of achieving well-resolved measurements are factors to why linear cascades are more favourable for rotor studies. Often, additional vanes and hardware are installed downstream of the rotor, limiting access for loss measurements. These technicalities are also made more difficult when rotors are spun at high-speed. Therefore, many loss results obtained have been under low-speed, scaled-up rigs. A number of rotor loss measurement studies have been undertaken, namely, the work of Dring & Joslyn

(1981), Yamamoto *et al.* (1994), Xiao *et al.* (2001) and McCarter *et al.* (2001).

In addition to these, Rao *et al.* (2006) researched casing roughness and its effect on rotor loss in an axial turbine facility. The addition of roughness was seen to alter the flow physics and reduce tip leakage mass flow rates. Other ways of reducing the tip leakage losses have been studied and the use of casing coolant injection was one means of doing so. Such methods have been reported experimentally and computationally by Behr *et al.* (2008) and Mischo *et al.* (2007). A method of tip coolant injection was used by Dey & Camci (2000) to try to limit the amount of flow entering the tip region. They showed that it was possible to reduce the tip leakage losses, but at the expense of a relatively large amount of coolant injection to make any significant improvements.

Chana & Jones (2003) obtained static pressure measurements over the casing wall and noted that the difference between a non-uniform and uniform inlet showed very little differences. An alternative method of improving the aerodynamic losses through unshrouded tips were sought by Dey & Camci (2001) and Camci *et al.* (2003). Results showed that a suction side extension was unable to reduce the losses, but a pressure side extension was able to reduce the mass flow rate and momentum of flow reaching the suction side. The latter study focused on numerous partial squealer tips and found that they were capable of reducing losses with suction side squealers providing better performance than channel arranged types.

Rotational experiments examining aerodynamic loss of unshrouded tips conducted at high-speed are few and far between. Those that investigate both the heat transfer and aerodynamic loss are addressed in Section 2.3.1. Focusing exclusively on aerodynamic loss, Payne *et al.* (2003) obtained measurements downstream of a rotor at engine-representative conditions. They presented measured and predicted results of entropy at the stage exit, 15% downstream of the rotor trailing edge. Their results show distinctive loss features and gave insight into the impact on the stage efficiency.

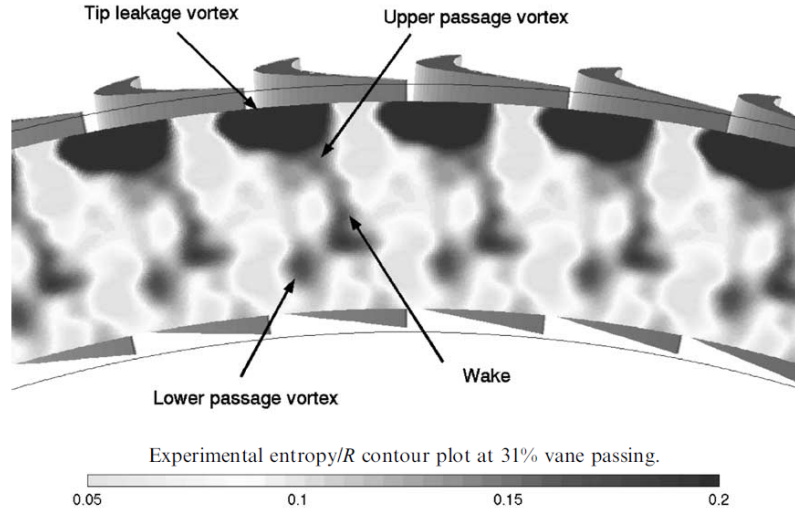


Figure 2.1: Measured loss structure downstream of a rotor (Reproduced from Payne et al. (2003))

The tip leakage vortex was clearly identified and attributed to a 3.3% drop in efficiency. Additional loss features including upper and lower passage vortices and a wake were also found (Figure 2.1).

Annular cascades are similar to rotational rigs, but lack the relative motion of the rotor. They are often used to reduce the complexity and costs of rotational tests and consist of a scaled-model blade row; they are also used to recreate the flow as close to what is experienced in a real engine. Annular cascades are generally used for nozzle guide vanes, but are also considered fit for aerodynamic loss studies for rotor blades. Some studies using annular cascades include those by Matsunuma (2006) who explored the effects of tip clearance with varying Reynolds numbers and turbulence intensities. Other studies include Morphis & Bindon (1988) and Bindon (1980).

2.2.2 Simplified Models

Prior to the high-fidelity studies of high-pressure turbine rotor blades, the early work using simplified models provided the foundations of tip aerodynamic flows. These simplified models are beneficial since they can offer a high spatial resolution, unlike the

high-fidelity tests. Rains (1954) conducted a study based on rotor blade tips in axial compressors and pumps using a conceptual model describing the flow within the tip gap. Rains (1954) detailed that a vortex is formed by the gap flow and its interaction with the passage flow. Smoke visualisations were used by Allen & Kofskey (1955) to understand secondary flows in the tip region of a low-speed turbine, also giving an early indication of the flow features produced over a rotor blade tip. Following the water analogy study by Rains (1954), Booth *et al.* (1982) and Wadia & Booth (1982) conducted water-based cascade studies and addressed how to minimise tip leakage flow and reduce the efficiency penalty by adjusting the blade tip geometry. They showed that each percent of increase of tip clearance produced a 1.5 percent loss in efficiency.

Using a simple model, accurate predictions of the flow structure within the tip gap were made by Yaras *et al.* (1989) who presented visualisations of the pressure and velocity fields within the tip gap. They noticed separation bubbles forming at the blade tip pressure side corner that were of the form of chord-wise vortices especially towards the rear of the blade. These were also reported by Sjolander & Cao (1995) who noticed an attachment area near the pressure side corner due to a counter-rotating vortex that is created by the oncoming shear flow through the gap.

The use of idealised techniques are continued to be used in the present day. Krishnababu *et al.* (2009) investigated leakage flows using a simple tip model to understand the flow within the tip gap. The results showed the effects of separation and reattachment and compared results to a numerical analysis.

2.2.3 Low-Speed Linear Cascades

Although simplified studies can provide highly-resolved results due to the well controlled and conditioned environments, these studies are unable to capture the complex, three-dimensional flow features found in or near the tip gap region. Nowadays,

it is more common for experimental studies to be performed on linear or annular cascades. As described, rotational rigs are also in operation in order to best represent a real-engine, but are less frequently built due to cost, complexity and time constraints.

On the other hand, linear cascades are particularly well suited environments for capturing the flow physics around the tip region because of their relatively simple design and practicality. They are attractive as they offer ease in adjustment and the ability to capture a wide range of measurements. For instance, subsonic linear cascades are able to run for long durations enabling very high spatial resolutions for detailed traverse measurements.

Studies by Moore & Tilton (1988), Yaras *et al.* (1989) and Heyes *et al.* (1992) used linear cascades to examine the tip gap flow and had shown that simple models were able to predict the flow structures noticed. These studies all confirmed the formation of separation bubbles at the pressure side corner of the tip gap. The distinctive features spotted by Moore & Tilton (1988) and Heyes *et al.* (1992) were the acceleration of flow into the tip gap and formation of a *vena contracta*. Subsequent mixing and diffusive processes as flow approached the suction side were also described.

Bindon (1989) outlined that a tip separation bubble is an underlying feature of loss associated with tip leakage flow. He described how it appeared to be responsible for most of the internal gap shear and core losses within the mixing region. These were illustrated by Figure 2.2. He was first to quantify the breakdown of losses (48% from mixing, 39% from internal gap shear and 13% from endwall or secondary flow) and developed a conceptual model for tip clearance loss formation. Further to this, Dishart & Moore (1990) measured losses in a linear cascade and reported that the tip leakage loss contributes to approximately 17% of the total loss. Also in contrast to Bindon, Yaras & Sjolander (1992b) concluded that the tip gap losses are relatively small in comparison to the overall loss.

The flow structure within the tip gap was broken down in more detail by Denton

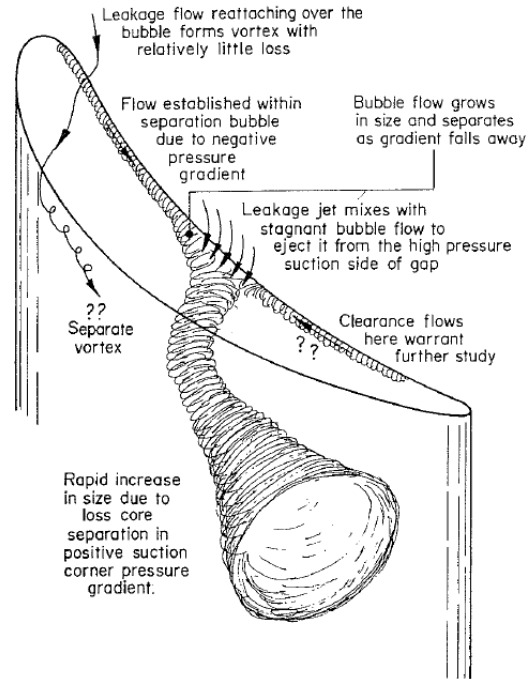


Figure 2.2: Formation of tip leakage loss (Reproduced from Bindon (1989))

(1993) who also described the effects of blade thickness. Thin blades attributed to flow exiting in a jet-like manner, whilst thicker blades, as typically found in axial turbines, show tip leakage entering at the pressure side that first separates and then reattaches. The leakage then flows downstream within the tip gap. Denton's paper provides a comprehensive review of the loss mechanisms in turbines where he also includes much of his own analysis. He refers to tip leakage flow as entropy generation within the tip gap, where it is associated with the mixing process between the tip leakage flow and the mainstream flow. He also explains tip leakage losses attribute to one third of the total loss and the three components of loss are comparable in magnitude to each other, differing to those presented by Bindon (1989) and Dishart & Moore (1990).

Tip clearance studies conducted by Yamamoto (1989) indicated a strong dependence of tip gap size and flow incidence on the tip leakage flow through detailed measurements of velocities vectors and streaklines. This was also demonstrated by

Bindon (1989) who suggested a linear variation of tip clearance loss with gap clearance. Furthermore using computation fluid dynamics, Ameri *et al.* (1998a) reported a similar linear relationship between efficiency and tip gap height.

Meanwhile, after gaining a good understanding of tip leakage flows and losses due to tip clearance, a fair amount of research has been conducted using moving casing endwalls. It is important to note that a limitation of linear and annular cascades is that they are unable to produce rotational effects as associated with rotor blades in an axial turbine. Their main advantages however, are that they are able to provide optical access to the rotor blades and tips and do not restrict instrumentation for detailed loss measurements. In some circumstances they are able to replicate relative motion between the blade and casing by employing a moving endwall, for example, with moving belts and rotating discs. Yaras & Sjolander (1992a) and Yaras *et al.* (1992b) used a moving belt to simulate the effects of relative casing movement on the tip gap flow. Their results show that moving casing decreased the tip gap flow rate, and downstream, it reduced the size and strength of the tip vortex whilst enhancing the passage vortex.

Using a large-scale linear cascade with a moving belt, Palafox *et al.* (2008) took *Particle Image Velocimetry* measurements and showed that the moving casing significantly effected the size and shape of the separation bubble. This was reduced by approximately a half in size with the belt on. They also noticed that the tip leakage vortex and passage vortex were distorted and shifted closer to the blade suction side. It was also seen that the upstream velocity magnitudes were reduced with the moving casing, however, less affected near the trailing edge.

Computational studies examining the aerodynamic tip leakage loss in a low-speed linear cascade were performed by Tallman & Lakshminarayana (2001a) and Tallman & Lakshminarayana (2001b). These simulations were based on a 3D Navier-Stokes, pressure-correction based CFD code. Their results reported an increased size of tip

leakage vortex with increased tip clearances. They also showed a decrease in the size of the vortex when a moving wall was simulated. They reported that the vortex reduced by a third of the size compared to the fixed wall case, but showed a slightly higher increase in the overall loss where flow became entrained in a strong secondary vortex beneath the tip leakage vortex.

It is clear that the aerodynamics and loss generation of high-pressure turbine tips in linear cascades has been well researched and documented, although, less researched are the effects of compressibility and the understanding of the tip leakage flow under engine-representative conditions. The majority of the early work conducted in low-speed environments has encouraged the continual research for designing efficient and durable gas turbines, and remain of high value. However, recent studies have proved that the flow physics under a high-speed regime are essential for high-pressure turbine tip design and that transonic effects must not be discounted. A study by Zhang & He (2011) explains that at engine-realistic conditions, the flow over the tip is choked and sets a limiter on the local tip leakage flow. These features show the conventional pressure driven mechanism found at subsonic conditions is broken down. These contrasting effects including those on the heat transfer will be discussed in more detail in Section 2.4.

2.3 Tip Heat Transfer

Modern day trends in gas turbine design are driven by the desire to increase TET as one of the means to increase the specific work done, however this comes with the risk of reducing blade life. When blades exceed a safe temperature limit, the blade must be actively or passively cooled to protect the material and structure. The difficulty of truly understanding the thermal performance of real engines is limited due to the lack of optical access to the tip surface. Understanding turbine tip heat transfer is

a significantly important aspect of turbine tip design and research has focused on a variety of test conditions varying from low-speed linear cascades giving a detailed picture of tip heat transfer to near engine-representative rotational rigs providing information at discrete locations at best.

One of the earliest heat transfer investigations was conducted by Mayle & Metzger (1982) who suggested that the average tip heat transfer was largely independent of rotational effects. Metzger *et al.* (1989) presented experimental and numerical results on a grooved tip noticing that the inclusion of a cavity reduced heat transfer levels on the floor, but these were largely offset by the additional interior walls. They noticed deeper cavity floors were able to reduce heat transfer further, up to a depth-to-width cavity ratio of 0.2. These results were reinforced by Chyu *et al.* (1989) where a moving shroud was secondary in terms of tip heat transfer. They also found that cavity based tips were able to reduce heat transfer, but overly deep cavities (depth-to-width ratios greater than 0.5) were undesirable because of their overall heat transfer levels. The idealised experimental and numerical calculations made by Krishnababu *et al.* (2009) on flat, cavity and suction side squealer tips had shown that the cavity tip heat transfer is worse than the flat tip, but single squealers offered an improvement. From a thermal aspect and ignoring the effects of cooling, the cavity tip showed large peaks of Nusselt number, but aerodynamically it was shown to be superior.

Notably, these findings have been based on simplified configurations and have stimulated the ongoing research on tip heat transfer. To thoroughly understand the tip flow and its relation to tip heat transfer, realistic geometries and experimental setups must be considered and the remainder of the chapter will address this requirement. Some of the heat transfer research of high-pressure and high temperature turbine blade tips is presented in a review by Bunker (2001).

2.3.1 High-Fidelity Rotor Tests

In terms of assessing the tip heat transfer, attempts have been made in engine-realistic environments that can provide a better representation of the 3D geometry, flow-field and rotational speeds. Dunn *et al.* (1984a) and Dunn *et al.* (1984b) obtained heat transfer data in a full rotating turbine rig. The use of thin film gauges were located at a number of vane and rotor locations. At discrete, limited locations on the rotor tip they measured an increase of heat flux moving from 10-30% of the axial chord. These were comparable in magnitude to those on the vane leading edge indicating the high magnitude of thermal stresses on the tip. Metzger *et al.* (1991) compared these results using a one-dimensional model and found remarkably good agreement.

Measurements in short duration wind tunnels at engine-representative Mach numbers were carried on the over-tip casing and tip by Chana & Jones (2003). They noticed peaks in heat transfer rates on the casing wall as the blade passed; a minimum near the suction side which increased and reached a maximum when the pressure side passed, caused by the tip leakage flow. They also used a simple one-dimensional model to estimate the tip and casing heat transfer and found reasonable agreement. Similarly, Thorpe *et al.* (2004a) and Thorpe *et al.* (2004b) found that the variation of heat transfer rate was strong on the casing and that the heat transfer gradually dropped 36% from the rotor inlet position to 80% of the axial chord. Further studies were conducted by Thorpe *et al.* (2005) who investigated tip heat transfer and measured heat transfer rates along a camberline of a transonic rotor blade. Although in-line with early linear cascade tests (Section 2.3.2), noticeable differences to low-speed work were spotted. Atkins *et al.* (2012) built on these experiments and through validation of steady-state results based on Wheeler *et al.* (2011), unsteady effects due to the rotor and stator interaction were found to have a significant effect on the instantaneous tip heat load. The leading edge tip heat flux was also affected by passing the vane trailing edge. Other tip heat flux measurements were obtained

by Dunn & Haldeman (2000) using a recessed tip. They discovered very high heat transfer (higher than the blade stagnation value) near leading edge cavity floor and on the suction side rim. The heat transfer was also found to decrease as the distance increased along the cavity.

Tallman *et al.* (2009) combined numerical and experimental studies on tip heat transfer in a $1\frac{1}{2}$ stage turbine, providing some valuable results alongside CFD predictions. Tip heat fluxes were obtained at 4 tip locations, but these only offered fair agreement as the CFD over-predicted the heat transfer by 27% on average compared to experimental results. Similar measurements have been obtained by Molter *et al.* (2006).

It is recognised that these high-speed full rotating rig tests provide the highest experimental fidelity in a research environment and they will continue to provide an important basis for new design validation and advances in computational methods development. For tip research though, challenges remain in providing easily accessible data from a high-speed rotating rig at least on two counts: a) the challenge in getting detailed spatially resolved instrumentation and data, and b) providing an easily definable and adjustable inflow condition for a rotor row so that different factors (for example, inflow turbulence and temperature) for the blade tip can be examined in isolation before putting everything together in a complex unsteady inflow environment such as a full transonic stage. Also, it might be argued that the value of these high-speed transonic rigs may not be truly appreciated unless the necessity of a transonic flow condition is appreciated. The dominance of low-speed flow data in influencing the conventional wisdom indicates that the role of a transonic flow played in blade tip heat transfer had not been quite appreciated until recently.

This appreciation of transonic flow effects on blade tip heat transfer has significantly been enhanced by some recent findings, revealing distinctive aero-thermal behavioural changes in a transonic tip compared to a low-speed tip, therefore, play-

ing an important role in high-pressure turbine blade tip research and development. However, low-speed research should not be overlooked, so an account of low-speed linear cascades is made first.

2.3.2 Low-Speed Linear Cascades

One of the first sets of heat transfer experiments carried out were by Yang & Diller (1995) who measured the heat transfer coefficients at a single point on a recessed tip (notably in a transonic linear cascade). The results stated that the heat transfer was not dependent on the tip gap height or local Mach number. However, these results are based on a single heat flux gauge measurement. Bunker *et al.* (2000) then published the first set of detailed tip heat transfer measurements in a low-speed linear cascade with two flow passages. The thickest portion of blade, near the leading edge, had shown a low heat transfer region which they termed the “sweet spot”. Results reported that heat transfer was increased with tip gap height and the turbulence intensity. This work was also simulated by Ameri & Bunker (2000) providing generally good agreement with the experimental data, however, they incurred errors of up to 30% associated within the area of their “sweet spot”. Furthermore, the transonic rotor study by Thorpe *et al.* (2005) had identified the sweet spot possibly due to laminar flow effects as shown by Wheeler *et al.* (2011), but found it to be smaller in size and magnitude as previously suggested. Interestingly, the recent findings of over-tip heat transfer at transonic conditions (to be detailed in Section 2.4), have inferred a different notion on the “sweet spot”. It was remarked by He (2013) that a transonic tip effectively has a “sweet region” in the rear, high-speed portion.

Continuing with subsonic flows, Azad *et al.* (2000) experimentally tested turbine blade tips in a linear cascade taking near-tip pressure and heat transfer measurements. Their results agreed with conclusions made by Bunker *et al.* (2000). They showed that high heat transfer regions were linked to leakage flow paths such as near the

pressure side leading edge tip region. They depicted heat transfer coefficient contours using thermochronic liquid crystals for different tip gaps and were able to show a clear increase in heat transfer with increased tip gap size.

Nasir *et al.* (2004a) also performed experimental tests to see the effects of varying tip gaps, as well as altering tip geometries. The flat tip heat transfer results were similar to those seen previously and sweet spots were also visible. They found that a squealer tip was able to reduce leakage flow and overall heat transfer coefficients in comparison to the flat tip. A reduction of the tip gap also reduced heat transfer.

Newton *et al.* (2006) observed the separation bubble, as discussed earlier, and found that the highest heat transfer was observed where flow reattaches. The highest heat transfer was found near the pressure side corner where flow enters the tip gap and reattaches after initially separating at the corner. Squealer tips were able to reduce high heat transfer areas as seen with the flat tip. However, high local heat transfer regions were seen, where flow impinges onto the surface. Their cooling method was capable of reducing the high heat transfer regions past the separation bubble by injecting cooling 5 mm away from the pressure side corner and into the separation bubble.

Krishnababu *et al.* (2007a), Krishnababu *et al.* (2007b) and Newton *et al.* (2007) numerically investigated the effect of tip geometry, tip clearance, relative casing motion and tip cooling effects on heat transfer. This was validated by the experimental work of Newton *et al.* (2006) (with their thermal predictions approximately 5-10% lower than those experimentally calculated). They found that squealer tips reduced leakage flow and as a result had lower heat transfer levels. The cavity squealer was shown to be advantageous in comparison to both a flat tip and a suction side squealer as it offered benefits thermally and aerodynamically. They also consolidated reports that both leakage flow and heat transfer increased with an increase of tip gap. This ranking was also deemed independent of casing motion, but there was a reduction in

heat transfer even though it was only marginal for the suction side squealer.

Palafox *et al.* (2012) utilised a very large scale linear cascade that enabled them to obtain detailed heat transfer measurements using infrared thermography. The common heat transfer trends were also reported in this study. However, casing movement altered the locations of high heat transfer regions. The inclusion of a suction side squealer reduced heat transfer by 32% compared to their flat tip and the relative casing movement reduced heat transfer by approximately 13%, differing to the findings of Krishnababu *et al.* (2007b) and early idealised studies. Tang *et al.* (2010) computationally examined this experimental work and reported Nusselt number discrepancies of 5%. However, depending on the choice of turbulence model, up to 35% under-predicted values were calculated. Their predictions of relative casing motion qualitatively showed similar results.

Focusing directly on the effects of endwall motion on tip heat transfer, Srinivasan & Goldstein (2003) used a rubber belt and a naphthalene sublimation technique that is based on a mass and heat transfer analogy to deduce heat transfer. Again showing contrast to Mayle & Metzger (1982), their results show that at certain locations that the heat/mass transfer is reduced by 9% for a small tip clearance of 0.6% of tip to chord length, but this difference diminished when increasing the tip gap.

Numerical studies based on moving casing effects were examined by Yang *et al.* (2006) and found that the casing movement only slightly affected tip heat transfer on flat and squealer tips. They did however notice an increase in the casing heat transfer when subjected to movement. Studies by Zhang *et al.* (2009) and Zhou *et al.* (2012) focused on moving casing effects with film cooling. Contrasting effects were noticed for the types of tip used. Zhang *et al.* (2009) found that a moving casing decreased film cooling effectiveness and increased averaged heat transfer for flat tips and squealer tips, however the opposite was seen with a winglet tip (Zhou *et al.*, 2012). Yang *et al.* (2010) examined moving casing effects on flat and squealer tips

and also noticed an increase in tip heat transfer. Coriolis and centrifugal effects were shown to be limited on the squealer tip, whilst on a flat tip they were capable of reducing the heat transfer.

These research pieces outline the popularity of low-speed tests and the above accounts for only a few salient articles published. Notably, the validity of the low-speed research has recently been questioned since the requirement of engine-representative flow conditions is not met. On one hand, the physical understanding attained is substantial and the research documented is valid for those subsonic flow regimes. On the other hand, recent work has shown contrasting flow behaviour at high-speed conditions and has introduced newly shed light on the aero-thermal behaviour of unshrouded blades.

2.4 Importance of High-Speed Conditioning

The role of high-speed test conditions has gained a great deal of attention in recent years. At engine-representative conditions, tip flow is largely supersonic and the flow physics are significantly changed compared to low-speed flows. In this sense, compressibility effects must be accounted for when it comes to high-pressure turbine tip design. A pivotal study by Wheeler *et al.* (2011) examined the effects of high-speed flow and heat transfer in comparison to low-speed flow. The distinctive nature of findings associated with transonic blade tips have had a substantial impact and that the flow physics related tip leakage flow, under engine-representative conditions, may not have been fully understood.

Compressible effects were seen in early tip studies by Moore *et al.* (1989). Based on the hydraulic analogy, water table flow simulations gave insight into a shock structure within their idealised 2D tip gap. They also noticed that at higher Mach numbers, the shock structure was quite complex noticing interacting oblique shocks with multiple

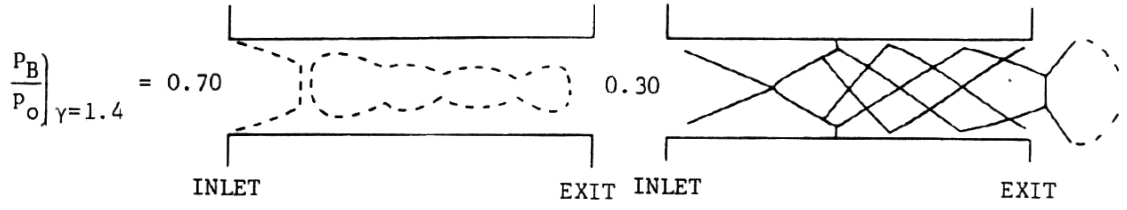


Figure 2.3: Shock structures found using water table simulations (Reproduced from Moore et al. (1989))

reflections as shown in Figure 2.3. Moore & Elward (1993) assessed the shock formation using the hydraulic analogy and found an oblique shock formed in an area with high heat transfer. Their findings based on the hydraulic analogy have not caught much attention, perhaps due to the indirectness of the method. Their quantitative or even qualitative aerodynamic equivalence to modern, advanced high-pressure turbine blading operated in fluid with large density changes, have not entirely been followed up. More specifically, the implication on blade tip heat transfer had not been known at all.

Figure 2.4 shows some of the significant findings of Wheeler *et al.* (2011) who analysed the effects of high-speed tip flow. A validated CFD study using the results in a transonic rotor were used to address the heat transfer in low and high-speed cascades. The tip heat transfer validation was done using the results of Thorpe *et al.* (2005). The simulated results show that at high-speed conditions (exit Mach number of 0.98), the flow over the tip was predominantly transonic and most significantly, the tip heat load decreased by 60%. Turbulent mixing in the high-speed flow region was found to reduce significantly as a result of increased acceleration of the mainstream flow over the separation bubble, of which was also reduced by a factor of 2 when increasing the exit Mach number from 0.1 to 1.0. Furthermore, Wheeler & Sandberg (2013) conducted Direct Numerical Simulations (DNS) with a high-order accurate CFD solver over a modelled turbine blade tip at engine-scale Mach and Reynolds numbers. The results found suggest that the tip heat transfer is unaffected by the freestream turbulence level, unless significantly high (over 10%). Also, they found

that for transonic tips, there was little advantage using complex turbulence models such as $k-\omega$ SST and found that the Spalart-Allmaras model actually performs much better when flow reattachment is concerned.

The most important contribution of the work by Wheeler *et al.* (2011) was the quantitative and qualitative differences between low-speed and high-speed flows (Figure 2.4). Tip heat fluxes are shown to differ greatly when compared to incompressible flows. The results show a reversal of heat flux trends. At high-speeds, there is a clear change in the heat transfer pattern. The leading edge tip region shows high heat transfer, in contrast to the sweet spot as commonly cited, and at the trailing edge a significant area of low heat transfer exists, termed the “sweet region” (He, 2013). At high Mach numbers, a similar shock structure to Moore *et al.* (1989) is observed; oblique shocks formed and reflected from the casing and tip, causing thickening and thinning of boundary layers with regions of accelerating and decelerating flow.

The findings by Wheeler *et al.* (2011) were experimentally confirmed by Zhang *et al.* (2011) and also previously validated by Thorpe *et al.* (2005), researching transonic flow and the over-tip shock structure with its direct impact on heat transfer. Experimental work conducted using the Oxford High-Speed Linear Cascade at engine-representative conditions was also supported by a computational analysis using the Rolls-Royce HYDRA solver. Tip flow peaked at Mach numbers of 1.8 over the tip region and the shock structure was clearly identifiable; a heat transfer distribution in the form of stripes were visible, a feature that was not previously seen. Using a virtual Schlieren visualisation, they linked the striping feature with the shock structure over the tip. The striping effect was described by increases and decreases in turbulence production. After the pressure side separation, flow reattachment causes high heat transfer because the separation promotes turbulence production. This is followed by supersonic flow that is accelerating, causing a reduction in turbulence and reducing heat transfer. Underneath the reflected oblique shock, the boundary layer

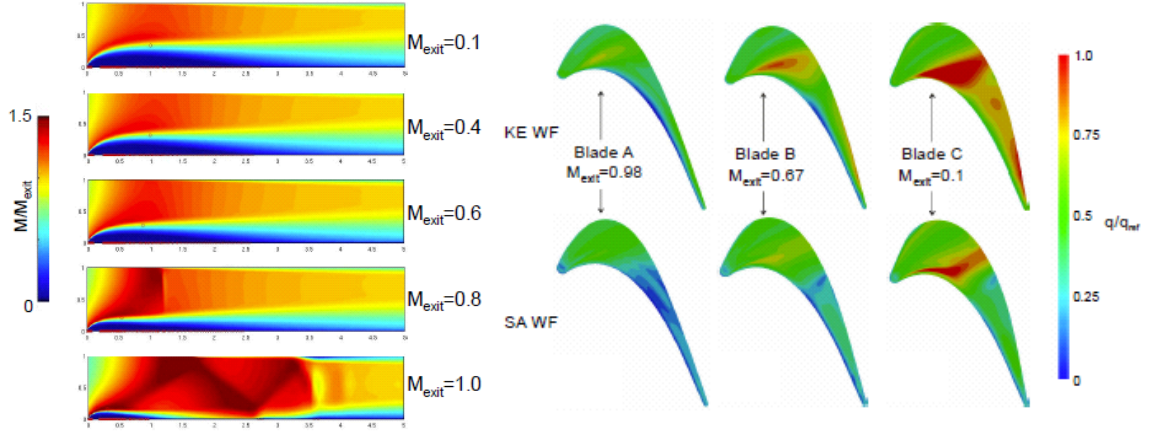


Figure 2.4: Low-speed and high-speed flow over a flat tip (Reproduced from Wheeler et al. (2011))

sees a large adverse pressure gradient and thus a rapid rise in turbulence promoting high heat transfer.

Of particular interest was the comparison to low-speed heat transfer results. It was shown that at transonic conditions, rather than a low region of heat transfer over the thickest portion of the tip, termed the sweet spot, this region was now an area of the highest heat transfer coefficients. The research by O'Dowd (2010) revealed the striking contrasting behaviour, as shown in Figure 2.5. However, similar to low-speed studies, the pressure side separation bubble reattachment region remained an area of high heat transfer (but not the highest as found by Newton *et al.* (2006)). It has become apparent that the tip heat transfer patterns are qualitatively different depending on the whether the tip is transonic or subsonic. Although the trends are clearly different, it should be noted that there is a large change in Reynolds numbers between Figure 2.5a and Figure 2.5b and will have some influence on the local heat transfer patterns due to the variation of Nusselt number with Reynolds number.

The effects of tip gap height on the tip and near-tip heat transfer were investigated by Zhang *et al.* (2010a), again under transonic flow conditions using the High-Speed Linear Cascade. A numerical analysis using HYDRA consolidated this work. The

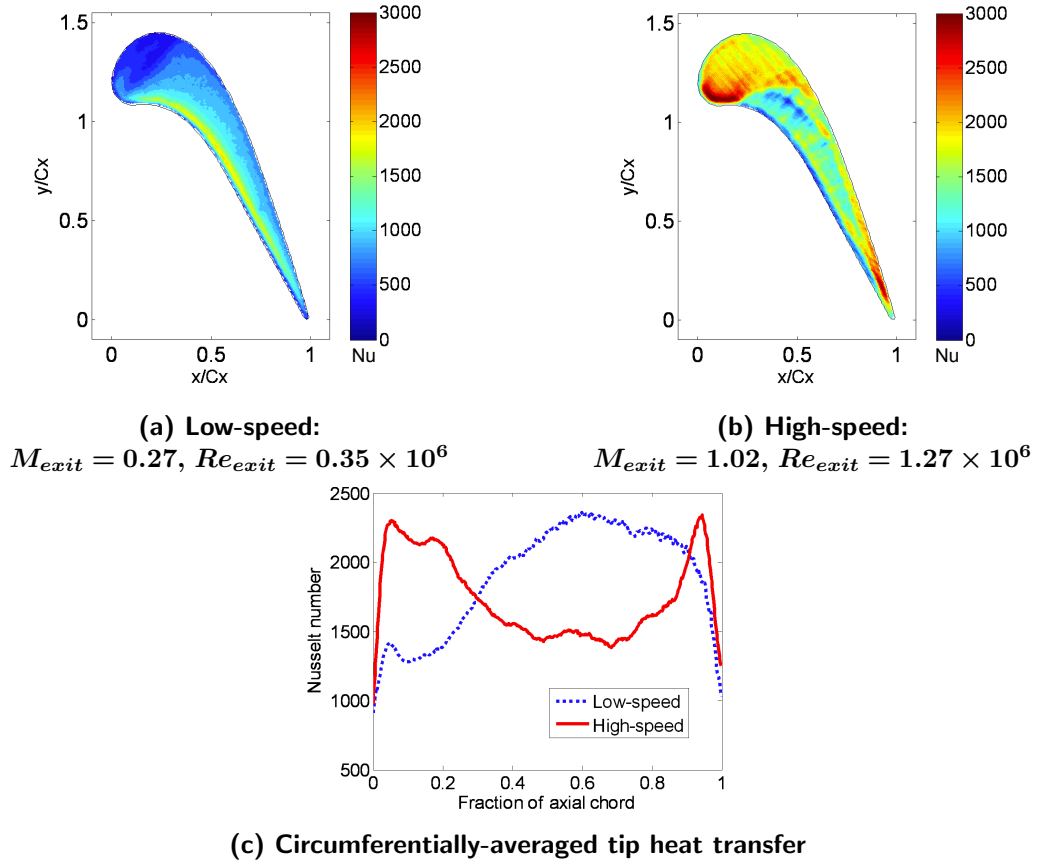


Figure 2.5: Effects of engine-representative flow conditioning on tip heat transfer (Reproduced from O'Dowd (2010))

results showed that the solver was highly capable of predicting the heat transfer patterns for all tip gaps. The tip gap effect was clear, showing a reduction of heat transfer over the frontal regions with reduced clearance, however towards the trailing edge, the opposite trend was observed. The high heat transfer regions were thought to be due to enhanced turbulent thermal diffusion. Supersonic flow was apparent over the tip in all three cases. A computational moving casing study was also included and it was shown that the tip flow remains partially supersonic with higher heat transfer towards the trailing edge.

The aerodynamic performance of transonic blade tips was addressed by Zhang & He (2011). By conducting a numerical assessment, they found that aerodynamic choking effects set a limiter on tip leakage flow and is understandably different to

low-speed flow. They identified a decoupling between blade loading and over-tip leakage mass flow suggesting the pressure-driven mechanism can be broken down. It was also shown that a highly loaded blade was able to considerably reduce the tip losses. Thus, differences in flow structure and tip losses were found with high-speed conditions. Comparable findings about the tip being choked were also presented by Shyam *et al.* (2012).

An investigation of inlet boundary layers and turbulence were conducted by Zhang *et al.* (2014) under high-speed test conditions using the Oxford High-Speed Linear Cascade for a flat tip. The results show that firstly, high turbulence intensity levels (7-9%) do not affect the tip heat transfer compared to low turbulence levels (1%), and HYDRA predictions showed similar results. The turbulence effect was diminished due to strong acceleration of flow over the tip. Secondly, different boundary layer thicknesses were also studied and no impact was seen on tip heat transfer and tip leakage mass flow rate. The main differences were seen on the blade suction side. However, more of an impact was observed by Coull & Atkins (2015) when considering both casing motion and inlet boundary layers simultaneously. For moderate turbulence levels (approximately 5%), Wheeler & Sandberg (2013) found that tip heat transfer was shown to be insensitive to free-stream turbulence.

2.5 Conventional Unshrouded Tip Designs

Having discovered the role of high-speed flow conditions and its impact on the aero-thermal behaviour, it is now of interest to outline the major research findings of unshrouded tip designs. It is important to note that there has been no set strategy for unshrouded tip design and geometries have generally focused on limiting the over-tip leakage flow prior to reducing the tip heat transfer. The following section introduces the main contenders for unshrouded tip design and are considered as conventional

choices.

2.5.1 Flat Tips

The majority of high-pressure turbine tip literature focuses on plain, flat tips. The flat tip is generally used as a baseline case enabling aero-thermal comparisons to other tip geometries such as squealers and winglets. The flat tip, as already discussed, often proves inferior to design strategies such as the squealer. Much of the early, low-speed linear cascade studies were based on the flat tip and so do recent studies, including the present work. One of the first 3D computational studies on flat tips was the work of Ameri & Steinthorsson (1995) who provided detailed predictions of the flow structure over a flat tip used in a space shuttle engine. The separation bubble, reattachment and tip leakage vortex were all visible.

Geometry plays an important role in the design of turbine blade tips and many studies looking at geometric effects compare to this flat tip. Ameri & Bunker (2000) studied the effects of edge radiusing and found that the separation bubble on the pressure side edge was completely eliminated. Their radiused edge simulations agreed better to measurements compared to the sharp edged tips. The results obtained by Wheeler *et al.* (2011) and Atkins *et al.* (2012) also had correct corner radiuses in their CFD studies. One common geometric parameter used in the design process is the inclusion of a cavity. These tips are often referred to as squealer tips; these are flat tips that have a cavity or recess. Investigations show that although there are high heat transfer coefficients associated with impingement flow and at the trailing edge, the double squealer tip geometry is one of the best aero-thermally performing blade tips.

2.5.2 Squealer Tips

Squealer tips represent the majority of in-service, first-stage, high-pressure turbine blades. The tip itself is similar to a labyrinth seal with two teeth that are positioned radially upwards towards the casing. The geometry is favourable as it offers the best compromise between most design aspects: aerodynamics, heat transfer and cooling, mechanical stress and cycle time between repair. Often variants are used including single suction side or pressure side rims. The squealer tip also reduces the impact of tip rubs relative to other tip designs.

Research on squealer tips has often involved investigations of cavity depth, rim thickness and single/partial rimmed squealers, however the bulk of studies are under a low-speed flow regime, and to the author's knowledge, there are currently no published, detailed thermal measurements performed in a linear cascade under transonic conditions. In regards to cavity depth, Bunker & Bailey (2001) showed that deeper cavities can produce overall lower heat transfer coefficients, offering up to a 50% reduction of heat load compared to a flat tip. Interestingly, the removal of rim material on the pressure side also helped reduce heat transfer. Loss generation was examined by Heyes *et al.* (1992) and found that squealers were superior to a flat tip, in particular suction side squealers. Similar findings were seen by Booth *et al.* (1982).

A computational analysis of squealer tips was conducted by Ameri *et al.* (1998b) on a GE-E³ first-stage gas turbine blade. Tip leakage losses of a double squealer were reduced by 10% and 14% for 2% and 3% squealer depth cavity tips due to increased blockage by a cavity vortex. This however, did not effect the overall row efficiency or the high heat transfer rates noticed on the cavity floor. Azad *et al.* (2000b) also studied the GE-E³ blade using a transient liquid crystal method to detail tip heat transfer and found that squealers provide an overall lower heat transfer coefficient compared to the flat tip case. The squealer rims were however areas of very high heat transfer. Yang *et al.* (2002) used this experimental data for numerical validation

purposes. They were able to show that the squealer tip could reduce the leakage flow over the tip, both velocities and flow rate were reduced. The over-tip leakage vortex was also weakened by the inclusion of a cavity. Similar heat transfer observations were seen by Newton *et al.* (2006) and on the double squealer arrangements by Acharya *et al.* (2003) and Mischo *et al.* (2008). As mentioned previously, the aero-thermal benefit of a squealer tip was noticed by Krishnababu *et al.* (2007a), and Palafox *et al.* (2012) showed promise through a suction side squealer.

A variety of squealer tip variants and rim arrangements have been investigated to seek aero-thermal improvements. Ameri (2001) applied a camberline strip with a radiused edge and found it was able to reduce the tip leakage compared to a radiused edge flat tip, but total pressure losses were largely unchanged. Acharya *et al.* (2003) simulated a number of squealer arrangements and evaluated a tip desensitization strategy. The best performing tips were suction side and double squealers. The use of a ribbed tip was also capable of reducing tip leakage and tip heat transfer, but they concluded that the suction side squealer offered the most reductions. Azad *et al.* (2002) experimentally tested different squealer arrangements and found that for single squealers, a suction side squealer gave the lowest heat transfer. For double squealers, there was no consistent pattern when comparing different arrangements. Kwak *et al.* (2003) also showed lower heat transfer with the squealer tip and benefits of using a suction side squealer. Their results indicated that tip heat transfer increased with larger tip clearances. Saxena & Ekkad (2004) found that a double squealer offered the best aero-thermal performance in comparison to several other squealer tip arrangements through measurements in a linear cascade.

Ledezma *et al.* (2013) attempted to close the loop on squealer tip designs and found that by conducting measurements in an annular sector cascade, a double squealer had an equivalent aerodynamic performance compared to one with no pressure side rim. These results were backed up by a numerical analysis. Overall pressure ratios were

however notably higher for the suction side squealer. Lomakin *et al.* (2013) examined closed and cut-back squealers and noted an aerodynamic benefit of having a rear suction side cut-back squealer in comparison to a flat tip. A double squealer showed similar reductions at a tip gap of 1.36% of the span. At even smaller tip clearances, the suction side cut-back had the far greater advantage showing 25.5% reduction compared to the flat tip. The thickness of squealer rims was examined numerically by El-Ghandour *et al.* (2012) and found a double squealer performed better than those with thick pressure side and suction side rims.

Flow velocities (tangential, radial and axial) were measured and calculated for a squealer tip in a low-speed, 1 and $\frac{1}{2}$ stage rotating rig using Doppler global velocimetry by Fischer *et al.* (2013). The detailed images shed light onto the tip flow surrounding a squealer tip. Most noticeable were the fluctuations over the cavity, showing two regions of minimum tangential flow velocity believed to be the result of entering tip flow over the leading edge and pressure side and their interaction with each other.

Several of the studies above described a hot spot region near the leading edge of the squealer tip. This was caused by the impingement of tip leakage flow onto the cavity floor. Mischo *et al.* (2008) attempted to eliminate this by re-profiling the cavity shape. The re-profiled squealer was able to change the tip leakage flow physics and structure, providing a reduction in heat load by 7% compared to the baseline squealer and 13% compared to the flat tip. They also reported an improvement of turbine efficiency by 0.2% compared to the flat tip determined using a 1 and $\frac{1}{2}$ stage turbine facility.

Investigation of tip leakage losses at high-speed is very limited. Focusing on tip design, Key & Arts (2006) compared leakage flow for flat and squealer tips at high-speed test conditions. They noticed that squealer tips were fairly insensitive to increased Reynolds number in comparison to the flat tip and generated lower loss. This was related to the squealer having lower flow velocities within the tip gap and

thus lower mixing losses. Wheeler *et al.* (2013) observed that under transonic flow tip leakage flows differ greatly to subsonic flows. Single and double squealers generally outperformed the flat tip, but the suction side and pressure side squealer at high pressure-ratios performed poorly due to a reduction of the separation bubble and reattachment on the squealer rims. Unlike the majority of low-speed investigations (Acharya *et al.*, 2003; Kwak *et al.*, 2003; Ledezma *et al.*, 2013), high-speed results suggest the double squealer is superior. Li *et al.* (2014) also provided impacts of transonic flow on squealer tip design using CFD. The flow features were analysed and compared to a flat tip. The findings stated that the double squealer is effective at both low and high-speed, and cavity depth was sensitive to flow regime. At transonic regions, cavity depth had less impact on tip leakage flow. At high pressure-ratios, cavity depths were analysed by da Silva *et al.* (2014). Deeper cavities, with the largest at a cavity depth-to-rim ratio of 7.5, provided efficiency gains at a range of different operation conditions.

De Maesschalck *et al.* (2015) numerically compared a contoured tip to a double squealer and found the squealer offered enhanced aerodynamic performance, but the thermal aspect was the main concern. The tip heat transfer was increased by 42% compared to the flat tip. In spite of the higher heat transfer of a squealer tip than a flat tip, the aerodynamic performance was regarded as very high for an unshrouded tip design. It is also preferred to a flat tip in case of a tip rub, since the squealer offers less contact area and therefore less likelihood of mechanical damage.

2.5.3 Winglet Tips

Aero-thermal tip clearance control has also been explored with the use of partial shrouds and winglets. Winglet tips have small extensions in the pitchwise direction and often extend on both the pressure and suction sides of the tip. The major advantage of using a winglet is to reduce the pressure gradient over the tip and thus

reduce the tip leakage flows, however the major disadvantages include the additional manufacturing costs and increased surface area to be cooled.

Patel (1980) first examined a double-sided winglet and found a stage efficiency gain of 1.2% compared to a plain tip. Booth *et al.* (1982) and Wadia & Booth (1982) experiments examined winglets along side squealers and flat tips and found winglets offered superior performance. Yaras & Sjolander (1991) studied three types of winglet design (pressure-side, suction-side and double-sided) in a low-speed linear cascade and found that all winglets were capable of reducing the tip leakage losses and the highest loss reduction was achieved using a double-sided winglet. In a rotational rig, Dey & Camci (2001) showed that a pressure side extension could reduce the strength of a tip leakage vortex, and the suction side extension was ineffective. A continuation of this was done by Shavalikul & Camci (2008) and examined numerous pressure side extensions. Their numerical analysis showed that a modified bump (“*Modified Bump No. 2*”) configuration offered the most aerodynamic loss improvement compared to a baseline tip, reducing the mass flow rate over the tip and the total pressure loss by 12% and 10%, respectively.

A theoretical and CFD based study on winglet tips was done by Harvey & Ramsden (2001) for application in high-pressure turbines that can withstand the high stresses and temperatures. They indicated a potential of 1.2-1.8% stage efficiency improvement at a tip gap of 2% of the blade span, but this study was lacking an appropriate validation for the CFD predictions. Further investigations were conducted in a rotating rig at high-speed by Harvey *et al.* (2006) and found it comparable in reducing the tip leakage loss to a shrouded tip with two fins. Despite being more efficient than a flat tip, the stage efficiency was 1% lower than a fully shrouded tip with 2 fins and 2 fences at a representative engine tip gap height. Aerodynamic loss was also measured and computed for winglet-squealer tips in a low-speed linear cascade by Schabowski & Hodson (2014). They saw a 22% reduction in tip gap exchange

rate compared to a flat tip. Also in comparison to a squealer, it was acknowledged that the additional cooling requirement may not be decreased due to the increased surface area, even though they believed the heat transfer levels would be lower. An aero-mechanical optimisation of the winglet-squealer was conducted by Schabowski *et al.* (2014b) and further improvements were noticed.

None of the above studies address tip heat transfer and relatively few have been conducted. Papa *et al.* (2002) studied a combined winglet-squealer tip using a mass transfer analogy in a low-speed linear cascade. The results presented show that a squealer tip had higher overall mass transfer coefficients than the winglet-squealer, however they were comparable at their smallest tip gap, 0.6% of the blade chord. A numerical aerodynamic loss and heat transfer analysis on different types of winglet-squealers and squealer tips was performed by Saha *et al.* (2006). They found the winglet did not reduce either the pressure loss coefficient or the average heat transfer coefficient compared to a two-sided squealer. A pressure side winglet extension added to a suction-side squealer only showed minor improvements in the aero-thermal performance. The only improvements they found were by desensitizing a suction-side squealer tip by attaching a constant width pressure-sided extension, heat transfer reduced by 5.5% and the loss coefficient was 26% lower than the suction-side squealer.

Following on from Harvey *et al.* (2006), another derivative of a winglet design was tested by O'Dowd *et al.* (2011a). Experimental and numerical heat transfer and aerodynamic loss data for an Environmentally-Friendly-Engine (EFE) blade design were obtained. They presented spatially resolved tip heat transfer results and clearly showed a large portion of the tip flow was supersonic, peaking at a Mach number of 1.8. Mainly due to the winglet's larger surface area, it gave rise to 2.7 times higher heat load than its equivalent flat tip.

Coull *et al.* (2014) continued a study on winglets. An optimisation study found that full-perimetered overhangs gave higher efficiencies. However, the thermal aspects

were negative showing that the inclusion of overhangs gave higher heat loads. Building on this, Coull *et al.* (2014b) found that the inclusion of a cavity on a winglet improved efficiency but was unable to reduce heat loads relative to flat and normal squealer tips, as was shown by O'Dowd *et al.* (2011a).

2.6 Over-Tip Shaping

Exploration of blade tip design was done by Zhang & He (2013) using the knowledge gained about transonic tip flows and following a concept instigated by He *et al.* (2010). The same experimental and numerical set up was used as in Zhang *et al.* (2011) and Zhang *et al.* (2010a). A method of Over-Tip Shaping (based on a flat tip) was sought by observing lower heat transfer levels in high-speed flow regions. The design is based on accelerating the tip flow in order to reduce turbulence levels, in turn reducing heat transfer as previously found. The results presented show that the shaping concept was able to effectively reduce high heat transfer regions near the leading edge simply by accelerating the flow using a converging-diverging duct geometry.

In addition to this idea, tip-shaping trials were made by Shyam & Ameri (2011a). The goal was to try to manipulate the shock-boundary layer interaction. Success was achieved by using a contoured tip with a hole that gave a reduction in aerodynamic losses and tip heat transfer. The purpose of the hole was used in an attempt to add blockage in the tip gap. In a real engine, this is infeasible and the equivalent tip without the hole was able to reduce heat transfer, but increased the overall loss. Further tip shaping ideas were extensively tested in a low-speed linear cascade by Nho *et al.* (2012). The results showed that suitable tips for the best aerodynamic performance were the double squealer tip and a grooved pressure side tip. It was suggested that an optimised dimpled tip could perform better than these tips due to the high thermal stresses associated with squealer tips.

De Maesschalck *et al.* (2013) presented a numerical tip shaping optimisation for aero-thermal performance improvements. They found that tip shapes that perform well at subsonic conditions are not optimal at supersonic conditions. It was shown that appropriate tip shapes could reduce losses by 20% and reduce heat transfer by 40% compared to a flat tip. The results are again achieved by locally accelerating the local flow and controlling the state of the boundary layer. The design strategy used was an attempt to use contouring to control the flow and to provide some useful guidelines for full 3D tip design. An effort to locally contour a tip by De Maesschalck *et al.* (2015) show a possible means to reduce the tip heat load and performed particularly well against a squealer tip. However, the squealer tip performed better aerodynamically.

Currently, tip-shaping is being researched in order to improve its associated aero-thermal performance. It has been shown that a more basic tip, a cavity or squealer tip, offers outstanding aerodynamic performance and good thermal performance in comparison to other tip variants. The main concern however, is the high thermal load on the squealer rims and difficulty of cooling them. Designers tend to begin with an aerodynamically optimised tip and then consider the thermal aspects and consider protecting the tip from the hot gas by using engine coolant. Ideally, tip design should consider the interaction between the two, both aerodynamics and thermal management. The current research aims to address this issue.

2.7 Film Cooling

Since the squealer tip geometry performs well both aerodynamically and thermally, a great deal of attention has been paid to film cooling effects on this tip geometry. Studies include Kwak & Han (2003) and Ahn *et al.* (2005). The two studies found that although increasing blowing ratios of coolant from holes on the tip did increase

effectiveness, it also increased heat transfer coefficients. These results were numerically simulated by Yang *et al.* (2004). The results generally over-predicted the levels of cooling effectiveness, but the trends were fairly well captured. Additional numerical work was also conducted by Acharya *et al.* (2002).

Mhetras *et al.* (2005) continued these studies by combining an experimental and numerical investigation of shaped and angled cooling holes to target the pressure side rim that is most exposed high temperature gases. The ability to cool the pressure side rim was achieved, however, film cooling effectiveness levels were consistently low, averaging approximately no higher than 0.25. Linked to this work, Gao *et al.* (2009) studied the effect of flow incidence angles testing at angles of -5° , 0° and 5° . The results showed that at all blowing ratios used, the overall cooling effectiveness were mostly unchanged. However changing the incidence angles distorted the coolant jet flow directions. Park *et al.* (2014) measured cooling effectiveness for different hole arrangements on a squealer tip and found that the rim surfaces could be protected by locating some holes nearer the pressure side and some nearer the suction side.

Mhetras *et al.* (2008) examined cut-back squealers, by essentially creating an open cavity squealer tip towards the trailing edge. The cut-back squealer, with the help of cooling holes inclined towards the rear of tip, was able to increase cooling effectiveness near the trailing edge. A cooled cut-back squealer was also simulated by Acharya *et al.* (2010).

An experimental study by Nasir *et al.* (2004b) reported that at small tip clearances (0.5% blade span), there was no effectiveness on a squealer tip at all, including increased blowing ratio cases and were likely to be jetting straight onto the shroud wall due to low local flow velocities compared to a plain tip, however this was inconclusive as no measurements were obtained for the shroud. Nasir *et al.* (2007) found that combining pressure side and tip coolant injection for a tip gap of 1% blade span gave better effectiveness results than separate cases. Pressure side injection for a

squealer tip was also considered to be beneficial in protecting the pressure side rim. This study also found that at high blowing ratios of 3.0, effectiveness was decreased. However, it increased when ratios increased from 1.0 to 2.0. Similar impacts were found in Acharya *et al.* (2010). They found that using blowing ratios over 1.8 caused detachment and were not beneficial in cooling the high heat transfer regions that the holes were intended for.

The influence of cooling in a transonic linear cascade on a squealer and partial squealer tip was presented by Hofer & Arts (2009). The double squealer was found to be most influenced by cooling injection, accounting for more flow reattachment. Losses were also only marginally affected. However, the losses were 3% higher for the partial squealer compared to the full squealer. Also under transonic flow conditions, Wheeler & Saleh (2013) found potential to reduce tip leakage losses for flat and squealer tips studied. A winglet tip studied by O'Dowd *et al.* (2013) under transonic conditions showed better film cooling effectiveness at smaller tip clearances. Other effects such as increased tip Nusselt numbers and reduced losses were also results with coolant injection over a winglet tip. The loss reduction was believed to be due to aerodynamic tip sealing that was likely to have had a blockage effect over the tip.

At low-speed, flat and squealer tips were assessed aerodynamically by Zhou & Hodson (2011). It was noticed that a flat tip performed better than a squealer at small tip clearances, but at larger tip clearances the trend was reversed. A further study on a cooled winglet was conducted by Zhou *et al.* (2013). Slight deviations in the downstream flow angles were found and the reported losses were shown to decrease slightly with a low coolant mass flow rate, but found areas of both increased and decreased loss with an engine-representative coolant mass flow rate. Although the results do not compare the heat load with an uncooled tip, they do show that as the coolant mass flow increased, the cooling effectiveness increased and the heat load decreased.

Tang (2011) found that film cooling from the pressure side near-tip region was able to provide bulk coverage of the blade tip, in contrast to holes located on the suction side near-tip. The tip geometry was a combined squealer-winglet tip as an approach to combine the aero-thermal benefits of the two individual designs. At high blowing ratios, Narzary *et al.* (2014) experimentally investigated a similar idea by locating 17 cooling holes near the pressure side tip region and near-tip leading edge for a cut-back squealer and suction side squealer. Film cooling effectiveness increased with decreasing tip gaps and it increased with higher blowing ratios. However, effectiveness levels were reported at around 0.1 which are quite low and most likely due to the lack of tip cooling holes.

Lu *et al.* (2013) addressed relative motion between the plain and squealer tips with the casing wall and found distinct differences when numerically examining film cooling. The plain tip showed higher overall effectiveness, but the squealer showed better coverage of coolant. It was also shown that the leading and trailing edge tip regions were difficult regions to cool for both configurations. Acharya & Moreaux (2014) investigated the role of a relative moving casing and blade rotational effects on a low-speed, cooled squealer tip and found rotational effects were secondary with the relative shroud motion being dominant in terms of heat transfer and effectiveness. The majority of the tip features were similar for both cases, but noticeable differences were found on the suction side surface. It was also apparent that at a small tip clearance, the tip and pressure side cooling holes had more of an impact on the suction side rim. Overall cooling effectiveness values were not provided, but it is clearly identifiable that the effectiveness was low. Endwall motion was also studied on a winglet tip by Zhou *et al.* (2013) providing an increase of cooling effectiveness by 9% in comparison to the stationary case.

2.8 Conjugate Tip Heat Transfer

Modern developments in CFD have meant that estimates of material temperatures can be obtained through coupling the fluid and solid solutions, this means that solid conduction is also being modelled rather than being obtained through a separate finite element analysis. According to research by Starke *et al.* (2008), the comparison of a conventional thermal analysis of a rotor blade with a squealer to a coupled approach showed that the tip temperature predictions were lower through the coupled solver. It is likely that the fully coupled approach is more accurate as the solution is obtained simultaneously, rather through iterative coupling of fluid and solid solvers. Although research is growing in terms of conjugate heat transfer for turbine vanes and rotors, far fewer reports have been made in terms of the internal cooling effects for turbine tip heat transfer. A modelling study by Vogel *et al.* (2009) used a conjugate heat transfer analysis in order to understand tip metal temperatures. The study found that tighter clearances gave lower tip temperatures, but this could be counter-balanced by delivering more coolant to the tip especially from the pressure side tip. The results showed that a 1% increase in tip clearance led to 1.5% increase in tip metal temperatures.

Using a conjugate analysis on a squealer tip, Wang *et al.* (2015) improved the cooling ability by partially replacing the rear squealer cavity (under a transonic regime) with a flat geometry. Moreover, this did not contribute to an aerodynamic penalty. Although considered a benefit, moving to a flat tip based geometry has a larger risk when considering tip rubbing. Using a quasi-3D analysis, it was shown than a cooled flat tip aerodynamically outperformed a cooled squealer tip as the blockage effect was stronger. Additional conjugate solutions were presented by Teloxa *et al.* (2013). The effects of coolant Reynolds number were analysed jointly with gap variation and found improved film coverage with higher Reynolds numbers and a smaller tip gap.

To the author's knowledge, there is no published work regarding conjugate heat

transfer analyses on internal cooling designs, in particular for turbine rotor blades with a squealer tip.

2.9 Summary

To summarise, this chapter presented an overview of the existing research in the open literature into unshrouded turbine blade tip aerodynamics and tip heat transfer. It can be seen that the research for a considerably small region of the blade is extensive and is continually growing, showing that the tip region still faces a challenge for designers and manufacturers to build and equip engines with better performing and longer lasting high-pressure turbine blades.

As discovered, the bulk understanding and wisdom of the aero-thermal performance of high-pressure turbine blade tips have been based upon low-speed studies. However, recent research has shown qualitatively different patterns in heat transfer at a high-speed regime when matching engine-conditions. Therefore, the focus has shifted in terms of design and presented a new challenge, such that many of the topics covered at low-speed, all have needed to be revisited and perhaps restarted to gain a thorough understanding of unshrouded tip behaviour at engine-realistic conditions. Designs such as the flat tip and winglet have been exposed, however far less attention has been drawn to the squealer at high-speed conditions.

More specifically, the contrasting differences between the squealer and winglet have shifted attention to durability and lifespan of blade tips. However, the challenge is intensified whilst aiming to maximise engine performance and minimising costs and weight at the same time. When considering an appropriate tip, it must perform well both aerodynamically and thermally. Of these tips, the squealer has shown that it can perform well under both aspects (with exception to the rims that are typically thin and difficult to cool). The winglet itself possesses a larger surface area exposed

to hot gas, it has a high heat load and is prone to higher mechanical stress, therefore the squealer is considered to be the main contender. For these reasons, the squealer forms the basis of this study and importantly employs high-speed flow conditions. Moreover, the present study aims to address the rim heat transfer by introducing a shaped squealer concept to improve the internal cooling capability. This concept is also supported by experimental measurements under engine-representative conditions.

Chapter 3

Computational Methods for Tip Aero-Thermal Analyses

3.1 Introduction

This chapter introduces the Computational Fluid Dynamics setup and the procedure used. The main aims of the CFD solvers used in the present study are to calculate aerodynamic losses and tip heat transfer using either a wall thermal boundary condition for heat transfer coefficient predictions or using a coupled solid-fluid conjugate heat transfer method using an internally cooled geometry.

The initial aim is to validate the Rolls-Royce HYDRA solver for an open cavity squealer tip at a range of tip gaps; 1.0, 2.0 and 3.0% of blade chord. Once complete, these results can help provide a better understanding of the flow physics of a squealer tip under transonic conditions and to assess the effects of casing movement. These simulations are done using a structured, multiblock grid; one with hexahedral elements that is built using a commercial solver Pointwise. A revised double squealer is then assessed in a similar manner using HYDRA for the 2% blade chord tip gap. These results are compared to shaped squealer geometries and assessed accordingly.

In terms of addressing the impact of internal cooling, an additional solver is used due to HYDRA’s current lack of capability in solving the coupled fluid dynamics solution. For this, the commercial Ansys CFX solver is used. The external fluid domain remains as a structured grid whilst the internal is comprised of pyramids and tetrahedra, and therefore becomes a hybrid grid.

3.2 Aero-Thermal Performance Parameters

3.2.1 Aerodynamic Loss Definition

Understanding the complex flow field downstream of a turbine blade row is very important in turbine blade design. One of the ultimate goals is being able to reduce the losses that may be caused due to growth and separation of boundary layers on blade surfaces or due to the mixing of blade wakes or shock wave generation. These contributions were quantified by Mee *et al.* (1992) in a transonic linear cascade for a range of different flow regimes. A comprehensive review of losses were also presented by Denton (1993) where losses were placed into categories of endwall, profile and leakage losses. A definition of loss was also provided and provides a measure of loss downstream of a blade row. By taking local measurements of the total pressure, contour maps can be produced that help to identify flow structures. Results for loss are typically defined by the stagnation pressure loss coefficient, Y , for a turbine blade as stated by Denton (1993):

$$Y = \frac{P_{total,inlet} - P_{total,exit}}{P_{total,inlet} - P_{static,exit}} \quad (3.1)$$

The numerator represents the local total pressure loss and is normalised by the exit dynamic head which should be carefully defined. In the present study, the exit dynamic head is mass-averaged and obtained at the mid-span level. For low-speed

studies, Equation 3.1 is often used. For compressible flow, using an isentropic stagnation property, the definition becomes:

$$Cp_0 = \frac{P_{total,inlet} - P_{total,exit}}{\frac{1}{2}\gamma P_{static,exit} M^2} \quad (3.2)$$

Equation 3.2 cannot be used for cooling loss studies as it neglects contributions of the coolant pressure and temperature. Based on Denton (1993), the loss was described as a rise in entropy. This approach was taken to account for the coolant pressure. Thus, the following definition of the kinetic energy loss coefficient has been used by Day *et al.* (1999), Hofer & Arts (2009) and O'Dowd *et al.* (2013):

$$\zeta = 1 - \frac{[\dot{m}_c + \dot{m}_m] \left[1 - \left(\frac{P_{static,local,exit}}{P_{total,local,exit}} \right)^{\left(\frac{\gamma-1}{\gamma} \right)} \right]}{\dot{m}_m \left[1 - \left(\frac{P_{static,mid-span}}{P_{total,inlet}} \right)^{\left(\frac{\gamma-1}{\gamma} \right)_m} \right] + \dot{m}_c \left[1 - \left(\frac{P_{static,mid-span}}{P_{total,coolant}} \right)^{\left(\frac{\gamma-1}{\gamma} \right)_c} \right]} \quad (3.3)$$

The numerator uses local values of static and total pressure and the denominator uses mass-averaged exit static pressures at the mid-span and the average mainstream and coolant total pressures.

Lastly, if the flow field has a moving casing then the loss definition must be adapted to account for the additional work done by the casing on the fluid. Using the knowledge that losses can be quantified as a rise in entropy (Δs), an entropy rise function is also used to calculate the overall losses. The entropy is defined as:

$$\Delta s = c_p \ln \left(\frac{T_{total,exit}}{T_{total,inlet}} \right) - R \ln \left(\frac{P_{total,exit}}{P_{total,inlet}} \right) \quad (3.4)$$

Equation 3.4 is then rearranged to give:

$$\exp \left(\frac{-\Delta s}{R} \right) = \frac{P_{total,exit}/P_{total,inlet}}{(T_{total,exit}/T_{total,inlet})^{\frac{\gamma}{\gamma-1}}} \quad (3.5)$$

Thus, an entropy rise function can be expressed as:

$$1 - \exp\left(\frac{-\Delta s}{R}\right) = 1 - \frac{P_{total,exit}/P_{total,inlet}}{(T_{total,exit}/T_{total,inlet})^{\frac{\gamma}{\gamma-1}}} \quad (3.6)$$

3.2.2 Heat Transfer Coefficient Calculation

Convective heat transfer is energy transport due to bulk fluid motion and occurs in a thin layer of fluid close to a surface. The heat transfer coefficient, h , can be described by Newton's Law of Cooling:

$$\dot{q} = h(T_w - T_{ad}) \quad (3.7)$$

Conventional design is still largely based on an uncoupled approach so that heat transfer is determined by the aerodynamics. For small differences in T_w and assuming a linear relationship between $\dot{q}$ and T_w , h may be approximated by:

$$h = \frac{\partial \dot{q}}{\partial T_w} \quad (3.8)$$

So, the value of h can be calculated by:

$$h = \frac{\dot{q}_2 - \dot{q}_1}{T_{w2} - T_{w1}} \quad (3.9)$$

For the present study, two wall temperatures, T_{w1} and T_{w2} , are chosen at a difference of 20 K apart which is assumed sufficiently small for the approximation to remain valid.

3.3 Computational Fluid Dynamics

3.3.1 The Role of Computational Fluid Dynamics

Since the turn of the century, computational costs have greatly reduced and our ability to perform vast amounts of numerical calculations has increased. Computational Fluid Dynamics, or as better known as CFD, provides a third methodology to theoretical and experimental work. Today, CFD places itself as a tool to help interpret and understand the theoretical and experimental work. The role CFD plays in the modern day is to predict fluid flow in practically any environment given that accurate flow and boundary conditions are provided, this includes a well designed numerical grid for calculations. CFD tools help to simulate flow, heat transfer and any other chemical reactions that might occur. The prediction capability of CFD is now heavily used as a design tool that aids research and design. The wide range of applicability of CFD has made it extremely popular in aeronautical, medical, environmental and sports engineering. For example, the difficulties of experimental testing for hypersonic flight can be overcome with accurate CFD prediction, however the validity of the model remains fundamental.

For turbomachinery and in particular the turbine, CFD helps to predict flow patterns and assist with predicting heat transfer; it helps us to interpret the physical environment. Specifying flow conditions and the correct boundary conditions can provide simulations that agree with realistic scenarios and allow visualisations of flow structures. However, there are limitations which must be taken into account. Much of the CFD work today cannot be accepted without accurate modelling of the physical features present. These include turbulence models, compressibility and boundary conditions. Furthermore, numerical calculations are affected by errors, these include approximation errors in rounding and truncation errors. The latter case can be reduced by grid refinement, tending to zero as the grid is refined.

3.3.2 Governing Equations

When describing viscous, compressible and often unsteady fluid flow, differential equations in the form of the Navier-Stokes equations are used. The strategy of CFD is to replace a continuous domain problem with a discrete domain problem. Numerical methods are used to produce approximate solutions at discrete locations over a chosen domain. These solutions are calculated by creating volumes in the fluid domain. The size and type of volumes used determine the accuracy, as well as time taken to solve the flow simulation. Modelling boundary layers and turbulence models also play an important role when solving these equations.

The governing equations of fluid dynamics are based on three physical principles:

1. Conservation of mass or the continuity equation
2. Conservation of momentum or Newton's second law of motion, $F = ma$
3. Conservation of energy or the first principle of Thermodynamics.

In three dimensions, these equations account for five equations since momentum must consider the velocity vector in the three components of space. When the equations are applied to a viscous fluid, these equations are referred to as the Navier-Stokes equations. If the fluid is considered perfect, or one that is inviscid, they are known as the Euler equations. Thus, the general flow equations for any Newtonian compressible fluid based on the conservation laws are:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0 \quad (3.10)$$

$$\frac{\partial(\rho \vec{v})}{\partial t} + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot (\bar{\tau}) + \rho \vec{g} \quad (3.11)$$

$$\frac{\partial(\rho E)}{\partial t} + \nabla \cdot (\rho \vec{v} E) + \nabla \cdot (p \vec{v}) = \nabla \cdot (k \nabla \cdot T) + \nabla \cdot (\bar{\tau} \vec{v}) \quad (3.12)$$

3.3.3 Solution of the Governing Equations

Following the attainment of the Navier-Stokes/Euler equations, a system of equations can be built. For instance,

$$\frac{\partial W}{\partial t} + \nabla \cdot \vec{F} = Q \quad (3.13)$$

The vector, W , is the conservative variables, $\vec{F}$, is the total flux vector and Q represents source terms that express the effects of external forces, heat sources and work done by the external forces.

Having defined a system of equations in the form of Equation 3.13, they are ready to be solved by a vast amount of solution methodologies. The space, of which the flow is computed, is divided into a large amount of geometrical elements usually called grid or mesh cells. This process is known as grid generation or meshing. Often a grid can be imagined as a space filled with grid points (nodes) and then joined using straight lines (gridlines). A grid may be designed in 2D as triangles or quadrilaterals, or in 3D as tetrahedra, hexahedra, prisms, pyramids or other polyhedra.

3.3.3.1 Spatial Discretisation

Spatial discretisation of the Navier-Stokes equations is essentially a numerical approximation of the fluxes and source term. Different methodologies of solving the governing equations (Equation 3.13) include the finite difference method, finite volume method and finite element analysis, all relying on grid generation to discretise the equations.

The types of grids in existence include structured and unstructured grids. A structured form allows for quicker and easier access to neighbouring grid points (using addition and subtraction) and each node is represented by uniquely identifiable indices (i, j, k) with corresponding Cartesian coordinates $(x_{i,j,k}, y_{i,j,k}, z_{i,j,k})$. On the other

hand, unstructured grids cannot be uniquely identifiable, but offer advantages such that less time is required to build a grid, often necessary for complex geometries. Unstructured grids can also be easily automated. The disadvantages include not resolving boundary layers properly and the generation of unnecessarily large sized grids. These are usually overcome by employing hybrid grids comprising of both structured and unstructured cells. Once a grid has been generated, one of the solution methodologies can be chosen for evaluation.

3.3.3.2 Finite Difference Method

The finite difference method was developed as a means to numerically solving differential equations. The method is based on a Taylor series expansion for the derivatives of the flow variables. For instance, computing the first derivative of a function $u(x)$ at a point x :

$$u_x = \frac{\partial u}{\partial x} = \lim_{\Delta x \rightarrow 0} \frac{u(x + \Delta x) - u(x)}{\Delta x} \quad (3.14)$$

Removing the limit in the equation (Equation 3.14) gives a finite difference. Given a small, finite value of Δx , the right hand side gives an approximation to u_x . However, for any finite value of Δx a truncation error is introduced. This naturally reduces to zero as Δx tends to zero. The order of accuracy can be obtained from the Taylor series. For example, the Taylor series expansion of $u(x + \Delta x)$ about a point x_0 is

$$u(x_0 + \Delta x) = u(x_0) + \Delta x \left(\frac{\partial u}{\partial x} \right)_{x_0} + \frac{\Delta x^2}{2} \left(\frac{\partial^2 u}{\partial x^2} \right)_{x_0} + \dots \quad (3.15)$$

Therefore,

$$\left(\frac{\partial u}{\partial x} \right)_{x_0} = \frac{u(x_0 + \Delta x) - u(x_0)}{\Delta x} - \frac{\Delta x}{2} \left(\frac{\partial^2 u}{\partial x^2} \right)_{x_0} + \dots \quad (3.16)$$

The first term on the right hand side represents the finite difference of the partial

derivative and the following terms represent the truncation error (usually given as $O(\Delta x)$). The first term of the truncation error gives the accuracy of the solution, thus Equation 3.16 is defined as first-order accurate. For most CFD applications, a minimum of second order accuracy is required and a similar technique can be applied to obtain more accurate difference formulae or higher-order derivative approximations.

The simple approach of the finite difference method and the attainment of higher order approximations are advantageous, however the approach is restricted to structured domains. The finite difference method is considered to be the foundation of partial differential equation discretisation and when considered in integral form, finite volume approach is developed.

3.3.3.3 Finite Volume Method

The finite volume method uses an integral formulation of the conservation laws. The method discretises the governing equations by dividing the physical space into control volumes. These volumes are associated to each grid point and the integral conservation laws are applied to each local volume.

Considering a finite control volume Ω_I , bounded by a surface, S_I associated to a mesh point I , that does not change in time, the conservation law becomes:

$$\frac{\partial}{\partial t} \int_{\Omega_I} W d\Omega + \oint_{S_I} \vec{F} \cdot d\vec{S} = \int_{\Omega_I} Q d\Omega \quad (3.17)$$

Equation 3.17 is then replaced by its discrete form. The volume integrals are expressed as average values over the cell and the surface integral is replaced by a summation over all bounding faces of the volume.

$$\frac{\partial}{\partial t} (W_I \Omega_I) + \sum_{m=1}^{N_F} \vec{F} \cdot \Delta \vec{S} = Q_I \Omega_I \quad (3.18)$$

where N_F is the number of control volume faces. Using Figure 3.1, U_I can be iden-

tified, where for example, cell 1(i, j) would have an area of ABCD and the flux terms would be summed over AB, BC, CD and DA. Unlike the finite difference method, Equation 3.18 can also be applied to unstructured grids and cell-vertex configurations, where the unknowns are defined at the corners of the grid rather than in the centers of the grid.

Integrating from $t = n\Delta t$ to $(n+1)\Delta t$ for a control volume Ω_I associated to a cell I , Equation 3.17 becomes

$$\left. \int_{\Omega_I} W d\Omega \right|^{n+1} = \left. \int_{\Omega_I} W d\Omega \right|^n - \sum_{m=1}^{N_F} \int_n^{n+1} (\vec{F} \cdot \Delta \vec{S}) dt + \int_n^{n+1} dt \int_{\Omega_I} Q d\Omega \quad (3.19)$$

By cell-averaging W_I and Q_I at time $n\Delta t$, the conservation variables become $\bar{U}_I^n$ and $\bar{Q}_I^n$, respectively. The cell and time-averaged numerical flux over each side, $\vec{F}^*$ and sources, $\bar{Q}_I^*$ are given by:

$$\vec{F}^* \cdot \Delta \vec{S} = \frac{1}{\Delta t} \int_n^{n+1} \vec{F} \cdot \Delta \vec{S} dt \quad (3.20)$$

$$\bar{Q}_I^* = \frac{1}{\Delta t} \int_n^{n+1} \bar{Q}_I dt \quad (3.21)$$

Which then gives the conservative discretisation of the form:

$$[\bar{U}_I \Omega_I]^{n+1} = [\bar{U}_I \Omega_I]^n - \Delta t \sum \vec{F}^* \cdot \Delta \vec{S} + \Delta t \bar{Q}_I^* \Omega_I \quad (3.22)$$

A general numerical scheme defined as a system of ordinary differential equations in time can be given by:

$$\frac{d}{dt} [\bar{U}_I \Omega_I] = - \sum \vec{F}^* \cdot \Delta \vec{S} + \bar{Q}_I^* \Omega_I \quad (3.23)$$

In Equation 3.25, the balance of fluxes and source terms over the cell, I , are

accounted for by a residual, R_I , where:

$$-R_I \equiv -\sum \vec{F}^* \cdot \Delta \vec{S} + \bar{Q}_I^* \Omega_I \quad (3.24)$$

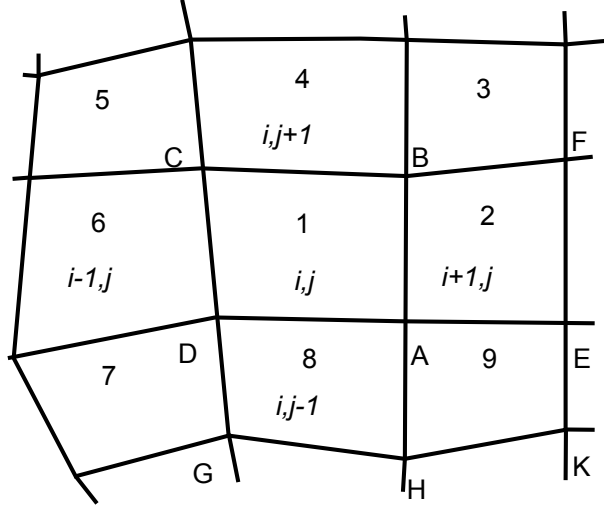


Figure 3.1: Structured cell centered finite volume grid

Cell face values for discretisation can be obtained by either a central scheme or an upwind scheme. Central schemes are based on central difference formulae that average the conservative variables to the left and to the right of a given control volume to calculate the flux. Upwind schemes are more advanced as they distinguish between upstream and downstream information. These methods provide information from upstream information only and are often used for their physical reliability.

3.3.4 Temporal Discretisation

Temporal discretisation is performed when solving the Euler or Navier-Stokes equations separate to the space discretisation. When applied to the governing equations (Equation 3.17) for each control volume, a system of coupled ordinary differential equations in time are developed. The time derivative can be approximated with the objective to iterate to a converged, steady-state solution by minimising the residual. This approximation can be either explicit or implicit. An explicit scheme uses

known flow quantities and is approximated by a forward difference with the residual evaluated at current time levels only. Direct computations of dependant variables made with known quantities only are said to be explicit. Popular methods include multistage schemes such as Runge-Kutta methods.

Runge-Kutta methods offer a high order approximation and well-behaved algorithms for temporal discretisation of ordinary differential equations and can be applied using several stages. They are preferred to Euler methods as the truncation error per step is smaller and they are generally more stable.

Consider an ordinary differential equation:

$$y' = f(x, y), \quad y(x_0) = y_0 \quad (3.25)$$

For a given value of h , $x_n = x_0 + nh$. At x_n , $y(x)$ can be approximated by y_n . Equation 3.25 becomes:

$$y'_n = f(x_n, y_n) \quad (3.26)$$

The simplest integration scheme is known as the Forward Euler method, where:

$$y_{n+1} = y_n + y'_n h \quad (3.27)$$

Advanced methods such as fourth-order Runge-Kutta are much preferred due to their increase in accuracy with low computational costs. A standard fourth-order Runge-Kutta method that is widely used, takes the form:

$$k_1 = hf(x_n, y_n), \quad (3.28)$$

$$k_2 = hf(x_n + h/2, y_n + k_1/2), \quad (3.29)$$

$$k_3 = hf(x_n + h/2, y_n + k_2/2), \quad (3.30)$$

$$k_4 = hf(x_n + h, y_n + k_3), \quad (3.31)$$

$$y_{n+1} = y_n + \frac{k_1}{6} + \frac{k_2}{3} + \frac{k_3}{3} + \frac{k_4}{6} + O(h^5) \quad (3.32)$$

The implementation of these explicit methods can be used with any spatial discretisation scheme and are computationally inexpensive. The downside is that the time-steps are restricted due to stability limitations and can often take time to reach steady state. Local time-stepping by advancing the solution in each volume by the maximum time-step achievable can often help to accelerate the solution, as well as multigrid methods that implement coarser grids to help dampen high-frequency errors. These are then fed into the finest grid to update the solution. Implicit schemes may be used since they offer better stability and can have much larger time-steps. An implicit method is one when dependant variables are defined by coupled sets of equations and iterative or matrix inversions are required to obtain the solution. For example a backward Euler method is implicit:

$$y_{n+1} = y_n + y'_{n+1}h \quad (3.33)$$

Notably, the right hand side of Equation 3.33 contains information at the next level. As mentioned, implicit equations can handle larger step sizes of h and can better handle “stiff” or numerically unstable differential equations.

3.3.5 Turbulence Closure

In many applications fluid flows are turbulent and enter with a form of instabilities. These instabilities occur when a critical value of the Reynolds number is reached, this is where the ratio of inertial to viscous forces exceed the critical value. Representing turbulence can be directly calculated based on the time-dependent Navier-Stokes equations. These simulations are termed as Direct Numerical Simulation (DNS) that are very computationally expensive and less suitable for research and industrial applications. Many CFD applications resort to approximations of the turbulent fluctuations. A high order approach is known as Large Eddy Simulation (LES) that directly simulates large scale fluctuations and model small scale ones. Beyond that, the next approach widely used is the Reynolds Averaged Navier-Stokes (RANS) model which averages the turbulence fluctuations. The RANS equations are based on ensemble-averaged flow quantities that are additional terms included to represent the effects of turbulence. These are such that solution variables are decomposed into mean and fluctuating components. For example, velocity components would be represented as $u = \bar{u} + u'$, where $\bar{u}$ and u' are the respective mean and fluctuating velocities. These are substituted into the continuity and momentum equations. The resulting Reynolds stresses of the form, $-\rho\overline{u'u'}$, are added in order to close the equations.

3.4 Boundary Conditions and Turbulence Modelling

The boundary conditions used for this study are engine-representative and are based on the experimental cascade data that will be described in more detail in Chapter 4. A constant, uniform total pressure boundary condition is set at the inlet and a constant static boundary condition is defined at the outlet. The turbulence model for HYDRA simulations was Spalart-Allmaras (SA) and both SA and $k - \omega$ Shear

Stress Transport (SST) were mainly used for Ansys CFX simulations, all with no wall functions. Both solvers will be introduced in detail in Section 3.4.1. Viscous (no-slip) wall conditions are applied to the blade, tip and casing. Initial CFD runs were based on half the blade span height using an inviscid wall condition on the hub region, until it was discovered that the blade loading at this location was changed when a moving casing condition was applied. Subsequent runs used the full blade span in order to match the blade loading for all cases. As the study focused on the tips aero-thermal performance, the inviscid hub wall condition was maintained. Fluid total temperature was also kept similar to the experimental cascade and chosen to be at 315 K. Translational periodic boundary conditions were applied to the two remaining mid-passage walls.

The SA turbulence model is a one equation model and solves for the kinematic eddy viscosity. Generally, it has been used mainly for aerospace applications and is growing in popularity for turbomachinery applications. Recent turbomachinery studies under high-speed conditions reported in the literature review use the same model and has shown to agree well with experimental results. For this reason in particular, it is also chosen for the CFD study. However, since Ansys CFX has a wider variety of turbulence models, they will be implemented to assess their performance in predicting high-speed tip flow.

The usage of such turbulence models in RANS predictions requires an estimation of the inlet turbulence. Often this is an educated guess, but this can be overcome by measurements using hot-wire anemometry. Such techniques are described in detail by Jørgensen (2001). The present study utilised this approach and as an outcome, Zhang *et al.* (2014) reported the inlet turbulence length scale and intensity. As the same turbulence grid was used in the present study, these variables provide a boundary condition for the incoming flow. The results indicated that the turbulence intensity (I) was between 7-9% and the length scale l was approximately 15 mm. As

mentioned, the SA model uses the turbulent kinematic eddy viscosity, ν_t , and was calculated using Equation 3.34. For other models the turbulent energy, k , and the turbulence dissipation rate, ω were calculated by Equation 3.35 and Equation 3.36 accordingly.

$$\nu_t = \sqrt{3/2 UI l} \quad (3.34)$$

$$k = 3/2 (UI)^2 \quad (3.35)$$

$$\omega = \frac{\sqrt{k}}{l} \quad (3.36)$$

The effect of relative moving casing was also examined for HYDRA simulations only. This casing condition applies a velocity in a direction from the blade suction side to the blade pressure side. The scaled flow coefficient, V_x/U , of 0.31 is used. For all other simulations, a stationary wall is used.

Another modelling aspect addressed in the present study is the inclusion of film cooling, an essential barrier for turbine blades protecting them from the hot gas. Modelling film cooling is a gruelling task due to the complex geometrical features and computational costs. Source terms are often used in design to provide a simple and effective way to model cooling flows in a gas turbine. Source terms appear in the conservation laws that cannot be accommodated into the transient, convective or diffusive terms; they are additional terms. They are mostly used for internal generation processes such as chemical reactions or heat generation, such as through electrical currents. The reversal is also valid where negative quantities are classified as sink terms. In this instance, the included source terms are declared for film cooling purposes and are accommodated by HYDRA and utilised in the present study.

The aerodynamic data for the film cooling sources are based on the HSLC pressure

and temperature measurements recorded from the coolant plenum assuming that there is no temperature difference from cooling plenum to each cooling hole. The mass flow exiting each cooling hole is based on the isentropic Mach number at each cooling hole using the total pressure within the plenum and a local static pressure value at each cooling hole location numerically calculated through HYDRA. This decision was made to account for the differences in the near-tip pressure loading that varies per tip gap. As a result, the coolant mass flow ratios differ for each tip gap and are equivalent to 1.33%, 1.32% and 1.31% of the mainstream mass flow for tip gaps 0.5%, 1% and 1.5% of gap to engine-equivalent span, respectively. Coolant flow angles are prescribed and are based on the cooling hole orientations. Each cooling hole is modelled as a separate source, rather than a strip of cooling holes. The temperature of coolant is also defined and based approximately on the HSLC coolant temperature at 280 K. The model assumes that the static pressure distribution near the hole exits agrees with the experimental setup and that the temperature does not differ from cooling plenum to hole exit.

3.4.1 Computational Flow Solvers

3.4.1.1 HYDRA

The HYDRA flow solver is an in-house CFD solver initially developed by Professor Mike Giles at the University of Oxford and further developed by Rolls-Royce plc (Moinier & Giles, 1998). It is a density based solver (where density is obtained by the continuity equation and the pressure is obtained by the equation of state) and uses an explicit solution method. The HYDRA suite comprises of non-linear, linear, unsteady and adjoint solvers and its general usage is for 3D computational fluid flows which uses an edge based data structure. The solver is used for RANS calculations and the core of the software is a preconditioned, time-marching solver that is iterated towards steady state using a 5 stage Runge-Kutta scheme. The equations are discretised in

space using a 2nd order finite volume scheme. Further convergence is achieved using a multi-grid method.

The turbulence models available in HYDRA are SA and the standard $k-\epsilon$. Higher fidelity methodologies exist such as LES and detached eddy simulations (DES). All HYDRA calculations presented in this study use the SA turbulence model without wall functions and are steady state.

3.4.1.2 Ansys CFX

The second flow solver used in the present study is Ansys CFX version 14.5. The Ansys CFX flow solver is a commercial code that is second order accurate in time and space and uses a cell-vertex finite volume method. Ansys CFX is generally used for complex three-dimensional fluid flow problems. It uses an implicit, pressure-based algorithm (usually obtained through the continuity and momentum equations to obtain pressure) and the equation system is fully coupled so that all the hydrodynamic equations are solved as a single system or in other words, the pressure and velocity are solved in a single step. The solver also uses an algebraic multi-grid technique.

A wide range of turbulence models are implemented in Ansys CFX and includes LES and DES capabilities. The turbulence models used for the present study include both SA and $k-\omega$ SST models. All CFD solutions are run at steady state.

3.5 Chosen Blade Tip Geometries

The starting basis of this study is to understand the potential of a squealer tip that is subject to engine-representative conditions. The squealer tip is an extension of a flat tip such that rails or rims are added to the tip in a radial direction and can either be open or closed, forming a continuous ring. The appeal of the chosen geometry is that in comparison to flat tips, they are lighter and perform better both aerodynamically

and thermally. With less risk of failure, squealer tips may be exposed to higher temperatures relative to other tip designs such as flat or winglet tips. Whatever design choice is considered, the different rates of thermal expansion between the casing and blade tips means that there is a high chance of a tip rub. As squealer tips have a reduction in exposed area, they are less prone to catastrophic damage.

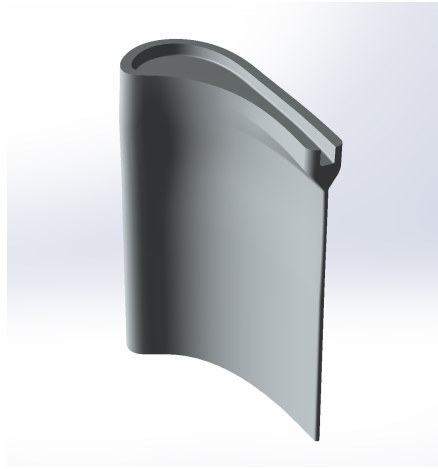
The first squealer design tested is an open cavity squealer of varying cavity depth and was designed by Rolls-Royce plc. The cavity depth-to-squealer rim ratio varies from 1.10 to 1.58. The geometries used include both uncooled and cooled designs and are shown in Figures 3.2a and 3.2b. This design was compared to flat tip (Figure 3.2c) created from an extrusion of the squealer’s near-tip profile at 90% of its span.

The second design uses a simplified squealer that is closed at the trailing edge and has a fixed cavity depth. The cavity depth-to-rim ratio is 1.2. This type of squealer is more commonly studied and allows for a better scientific control of the study and isolation of the variable studied. This geometry is displayed in Figure 3.2d.

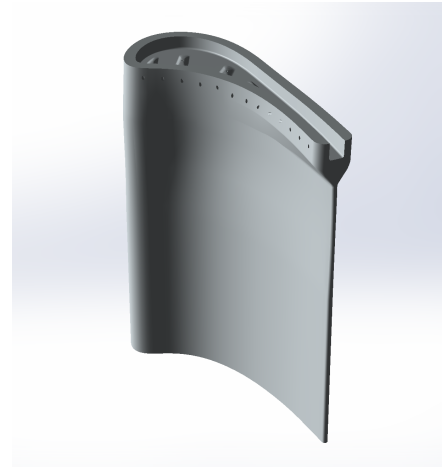
3.6 Computational Domain and Mesh Set Up

For all CFD simulations, the domain of the fluid flow is based on a single blade passage with periodic boundary conditions. The domain inlet plane was chosen to be at the measurement plane of pressures and temperature, one axial chord upstream of the blade leading edge. Likewise, the outlet plane was placed one axial chord downstream of the blade trailing edge chosen at the same plane of the total pressure measurements. The computational domain and blocking structure are shown in Figure 3.3.

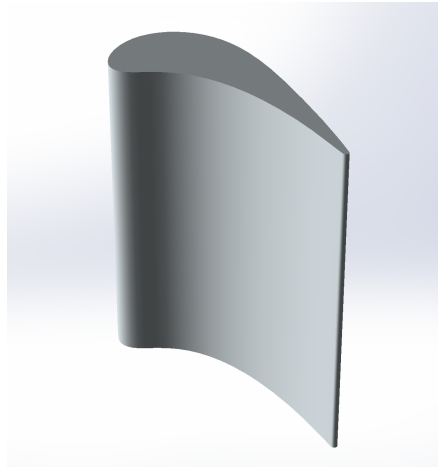
The meshing strategy for all fluid domains was of a structured, multiblock type. For conjugate heat transfer simulations in Ansys CFX, the solid material was meshed using either structured or an unstructured grid comprising of tetrahedra and pyramids giving a hybrid mesh. The unstructured mesh offered the best compromise



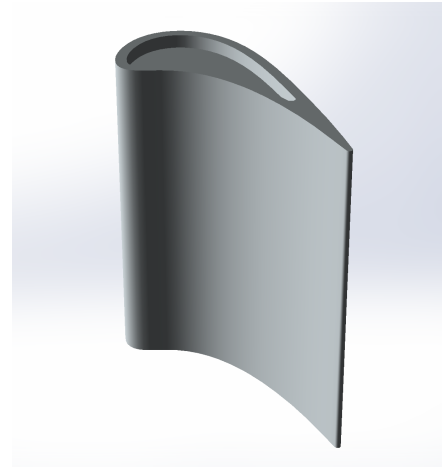
(a) Uncooled Open Cavity Squealer



(b) Cooled Open Cavity Squealer



(c) Flat Tip



(d) Closed Cavity Squealer

Figure 3.2: Geometries used in present study

between geometry complexity and solution required; due to a linear heat transfer equation being solved, the mesh type and quality are less important. All meshing was done using Pointwise. Using a structured mesh for fluid regions helped to ensure good mesh quality, generally over 40° minimum cell angle with an average of cells at approximately 70° . Growth of cells were generally kept to no larger than 1.2 and in terms of obtaining accurate heat transfer solutions, wall-adjacent cell heights should be as low as possible. Blade and tip surfaces had y^+ values mostly around 1.0. A few of the meshes used in the present study are displayed in Figure 3.4.

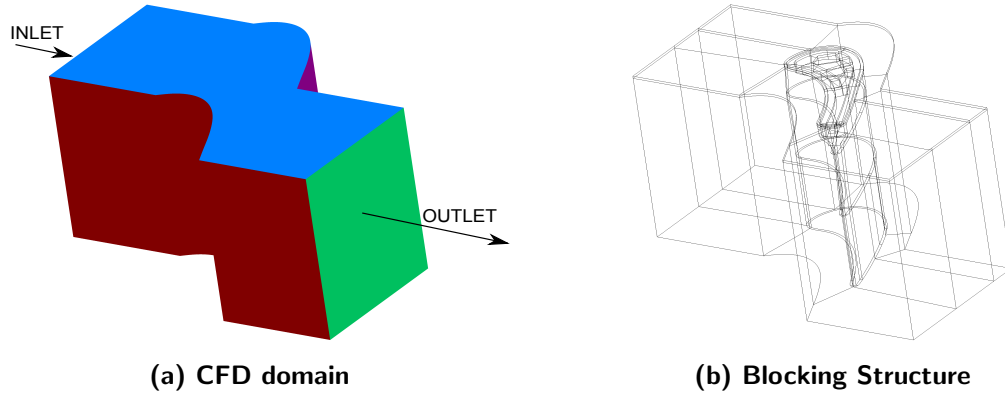


Figure 3.3: CFD Domain

The fluid mesh has a significant impact on the rate of convergence of solutions (or lack of convergence), solution accuracy and CPU time required. A good quality mesh can be attributed by the grid density, adjacent cell lengths/volume ratios, skewness and a well resolved boundary layer mesh.

3.7 Solution Convergence

Lastly, the convergence of numerical solutions were judged by assessing the flow residuals for each equation solved. Unlike most commercial solvers, HYDRA combines the five flow residuals for each Navier-Stokes equation into one residual. Ansys CFX outputs each residual independently. Convergence histories for two HYDRA cases are shown in Figure 3.5. These show two contrasting cases of flow residuals, one with an oscillating behaviour (squealer tips) and one that converges very well (flat tips). A typical convergence history for the CFX solver is shown in Figure 3.6. All values plotted are root mean square residuals. The criteria used for the simulations were that they were either flat; with or without oscillations around a local value or that they dropped at least three orders of magnitude. A final convergence check was done assessing the mass flow rate percentage error and convergence was ensured by requiring this error to be significantly lower than 0.1%, typically achieving between

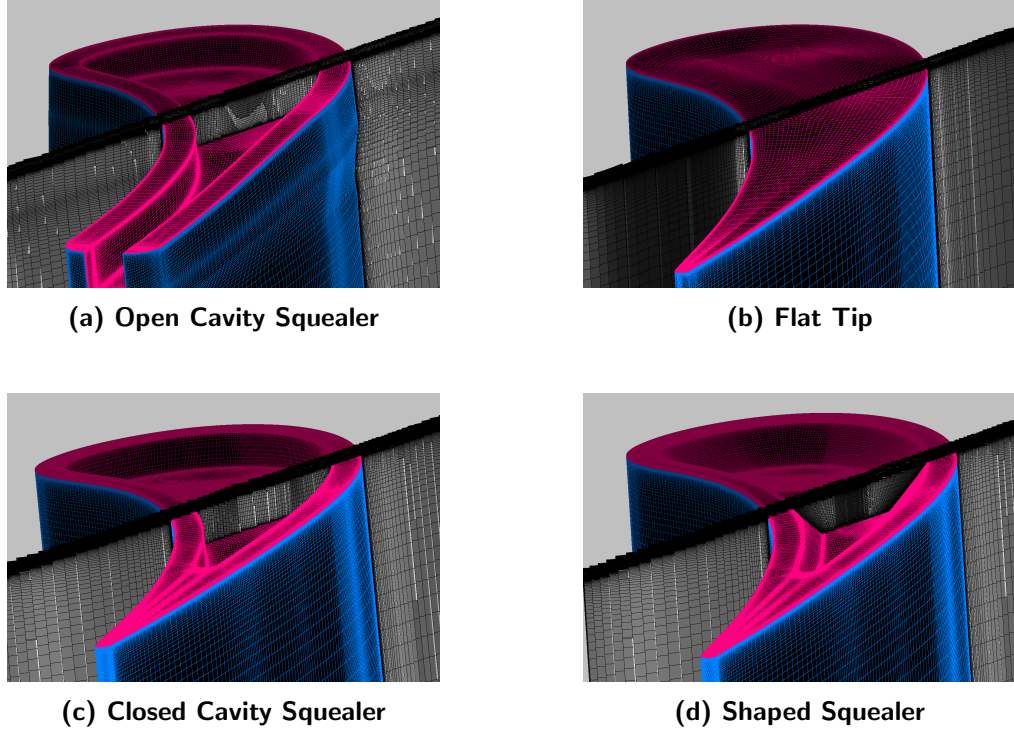


Figure 3.4: Meshes used in present study

0.01-0.05%.

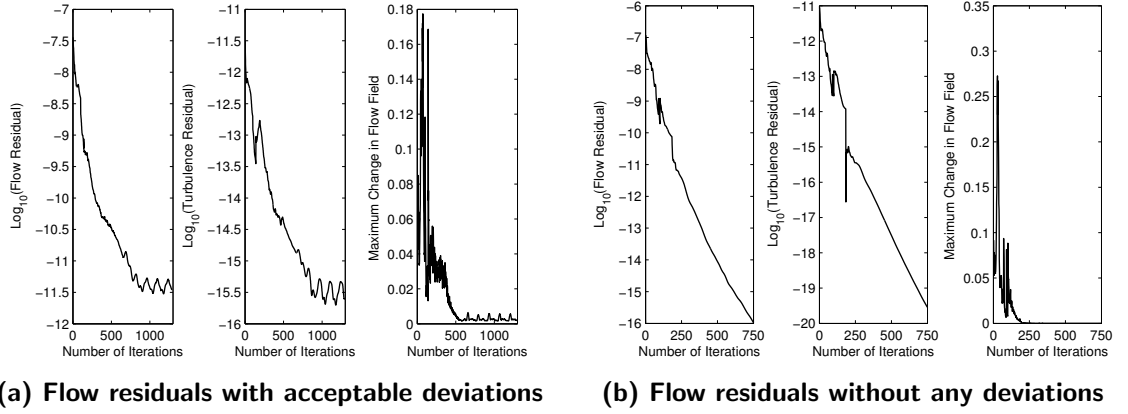


Figure 3.5: Convergence history examples for HYDRA simulations

3.8 Mesh Dependency Tests

In order to rule out any effects of the mesh on the CFD solution, a mesh dependency study was performed to determine the minimum number of cells required for changes

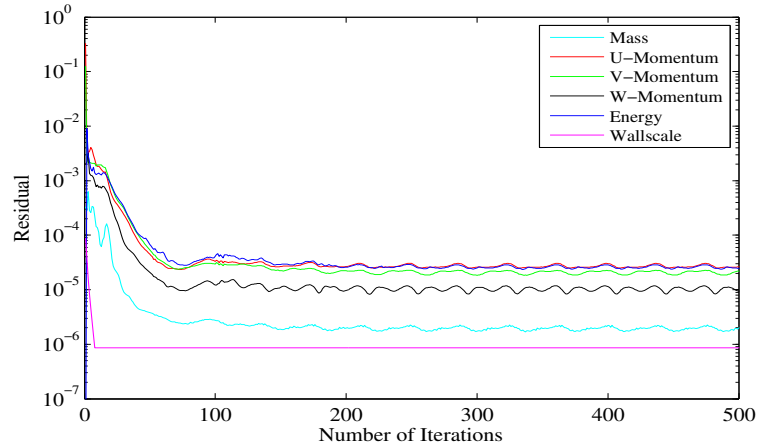


Figure 3.6: Convergence history example for Ansys CFX simulations

in the solution to be negligible.

Two studies were performed, one based on the open cavity squealer and one based on the closed cavity squealer. Refinements for both cases involved starting from a coarse mesh and increasing the density until little or no change is noticed in the solution. Refinements were global and included increasing the number of cells in all directions (axial, pitchwise and spanwise). The number of nodes within the tip gap per case were also increased and the minimum wall distance was reduced per case as mesh density increased.

Analysing the tip surface heat transfer using calculated heat fluxes, it was established that the open cavity squealer requires a minimum of 7.7 million cells to achieve a mesh independent solution. Local tip heat fluxes are shown in Figure 3.7 and averaged heat fluxes over the tip surface are shown in Figure 3.8b. It must be noted that for this geometry, the leading edge tip region shows irregular heat transfer patterns where flow impinges on the cavity floor. Due to this phenomenon, it is difficult to achieve true mesh independency where the changes are negligible. However, comparing the finer meshes to the coarse mesh and the mid-axial chord to trailing edge region, it is easily identifiable that the solution becomes mesh independent at approximately 7.7 million cells. This mesh has 50 grid lines within the tip gap and

blade and tip surface y^+ values are largely between 1 and 3. The difference in overall heat flux for the 7.7 million cell mesh compared to the 8.5 million cell mesh is 1.8% and mostly due to the fluctuating heat flux at the leading edge.

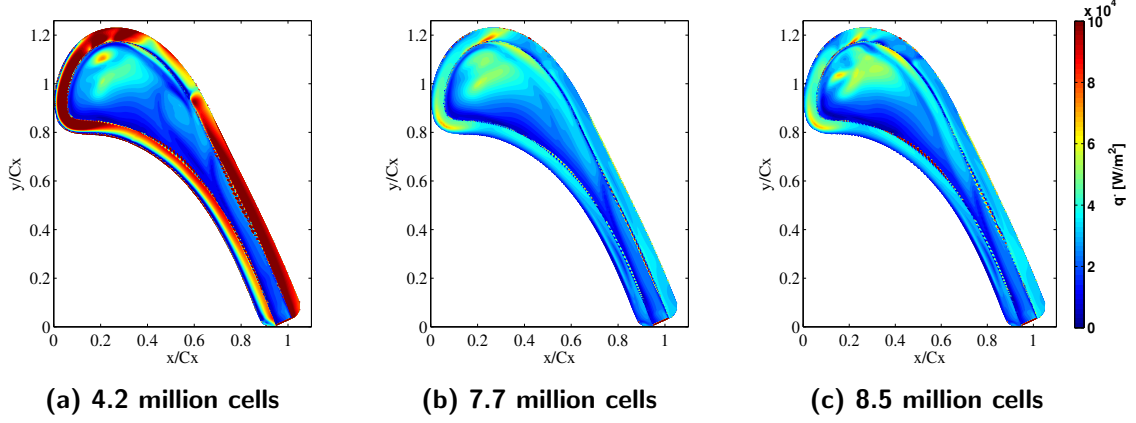


Figure 3.7: Tip surface heat fluxes (W/m^2)

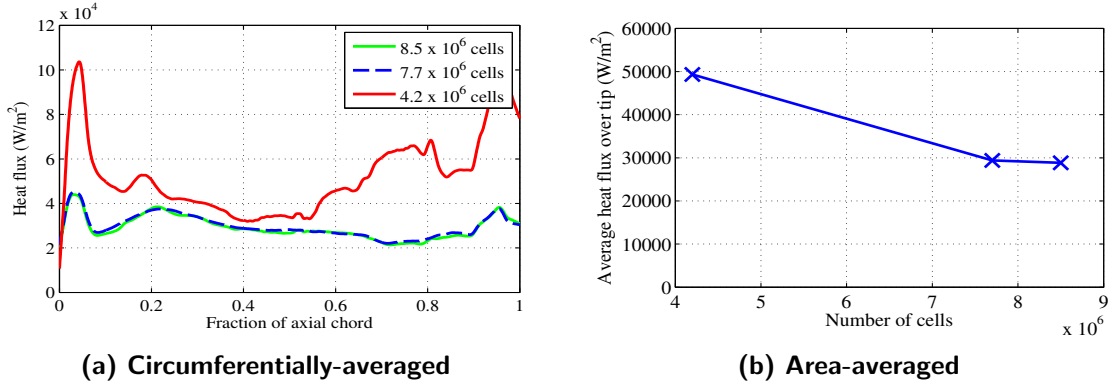


Figure 3.8: Averaged tip surface heat fluxes (W/m^2) for open squealer (HYDRA)

Similarly, the closed cavity squealer was also assessed for mesh independency. This geometry required a similar number of grid cells and it can be seen from Figures 3.9a and 3.9b that 8.4 million cells are required to achieve a grid independent solution. The difference in heat flux for the two finest cases is 0.9%. For this case, the higher wall temperature condition is used and explains the difference in the magnitude of calculated heat fluxes. The number of grid points within the tip gap is 60 and blade and tip surface y^+ values are also between 1 and 3 and for the two cases discussed, are shown in Figure 3.10.

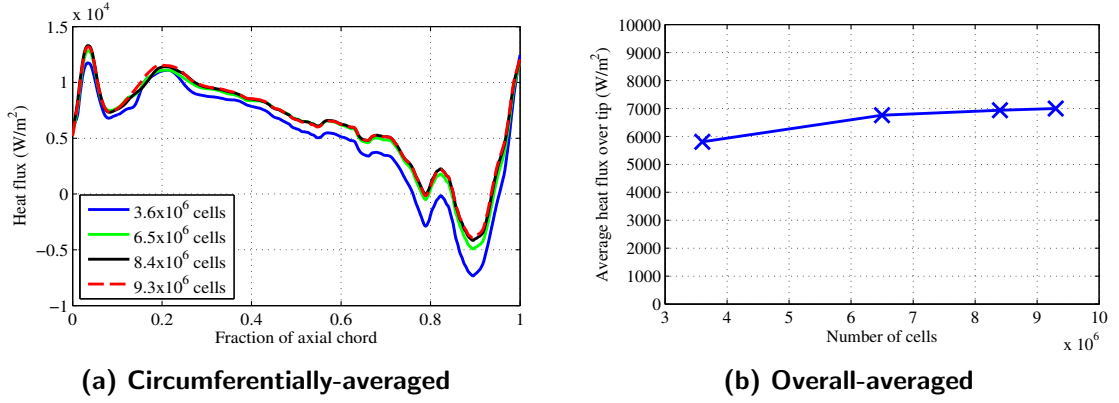


Figure 3.9: Averaged tip surface heat fluxes (W/m^2) for closed squealer (HYDRA)

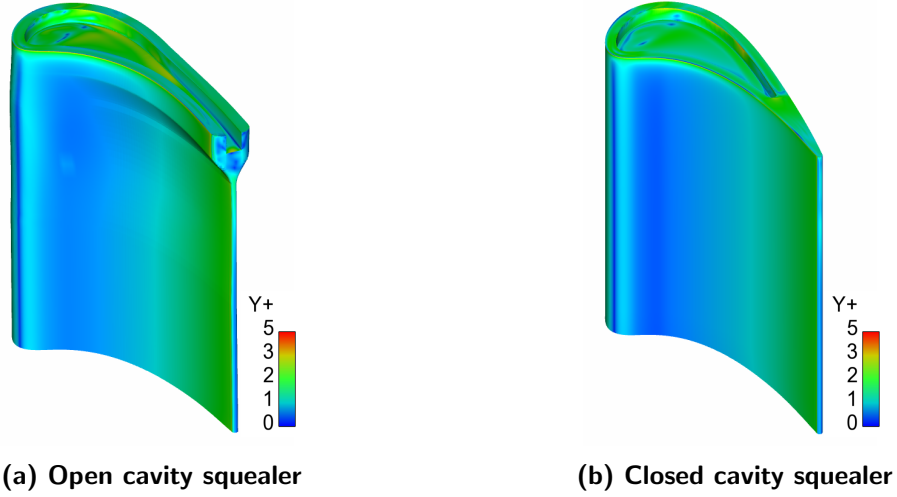


Figure 3.10: Wall-adjacent cell y^+ distributions over blade and tip surfaces

As previously mentioned, the computational set up is based on the HSLC. Therefore, CFD results are validated using data obtained for the open cavity squealer comprising of both aerodynamic and thermal studies. The closed squealer and its variations are validated using aerodynamic studies conducted also in the HSLC. Thermal data is limited to the open cavity squealer due to heater mesh and power supply failures. Since the frontal region of the closed squealer is that of the open squealer, the thermal data obtained can be compared accordingly.

3.8.1 Mesh Dependency Ansys CFX

Both non-conjugate and conjugate meshes used in Ansys CFX were also tested for mesh dependency. The mesh descriptions and mesh dependency for the conjugate study are detailed in Section 9.4. The results of tip heat flux for the two finest meshes as used in HYDRA are shown in Figure 3.11. The mesh was also judged as mesh independent at 7.7 million cells as the differences of local heat fluxes and is small and overall heat flux percentage change between the two meshes is 0.03%.

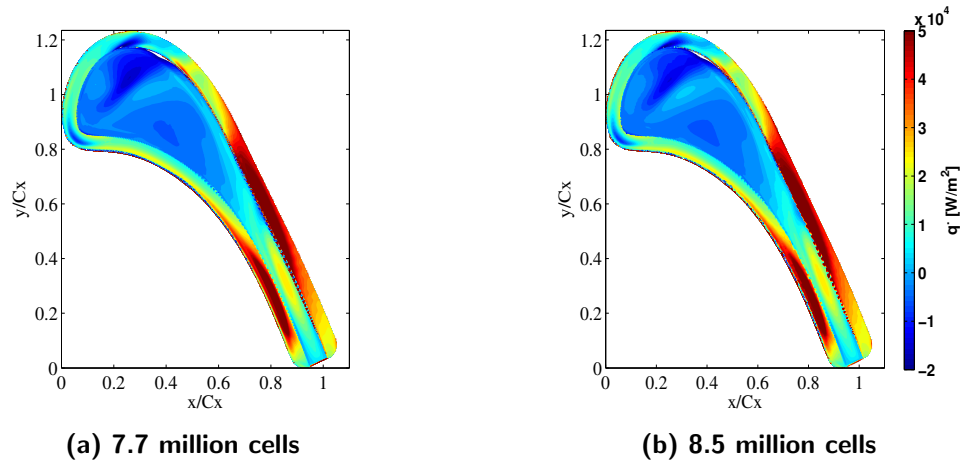


Figure 3.11: Tip surface heat fluxes (W/m^2) for open squealer (Ansys CFX)

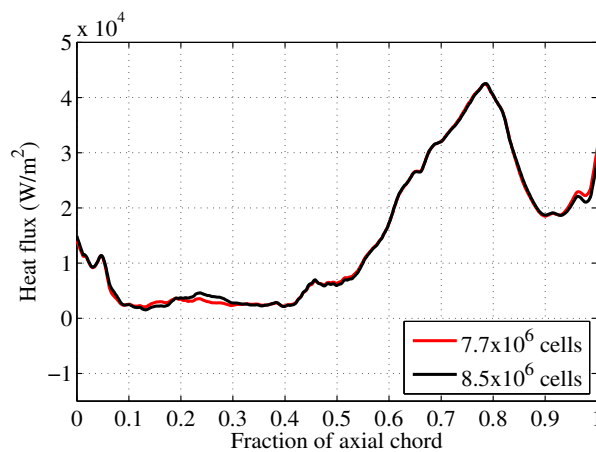


Figure 3.12: Circumferentially-averaged heat flux (W/m^2) for open squealer (Ansys CFX)

3.9 Summary

In this chapter, the aero-thermal parameters and calculation methods were introduced for assessing turbine blade tip performance. The computational methods were then introduced along with a brief introduction to the theory behind CFD. Described in this chapter were the flow solvers, fluid domain and the boundary conditions required to obtain solutions. Information was provided on the governing fluid dynamics equations and how to numerically solve the partial differential equations. Following this, the two CFD solvers used for the study were introduced with details of their solving methodologies.

The base geometries used for the computational (and experimental) analyses were provided. Using the performance parameters, the grids were tested for grid independence using both CFD solvers to ensure that the computational results are acceptable following convergence of the solution variables. The implicit solving technique used in Ansys CFX allows for quicker convergence of the solution, whilst both Rolls-Royce's HYDRA and Ansys CFX show similar sensitivities to the grid size and the solutions dependency on the grid.

Overall, for all geometries studied, the results obtained were shown to have converged well and have negligible dependency on the final grids generated.

Chapter 4

High-Speed Linear Cascade Set Up and Instrumentation

4.1 Introduction

This chapter describes the experimental element of the research conducted using the High-Speed Linear Cascade (HSLC) test facility at the Osney Thermo-Fluids Laboratory, Oxford University. This chapter provides an overview of the facility, instrumentation and measurement techniques. A new test section was designed and installed to facilitate a new squealer blade design. The tested blades were manufactured using a stereolithography technique. Aerodynamic and heat transfer measurements are taken at tip gaps of 0.5 mm, 1.0 mm and 1.5 mm (0.5%, 1% and 1.5% of the engine-equivalent span or 1%, 2% and 3% of the blade chord) for both uncooled and cooled blades for an open squealer tip, and at the intermediate tip gap (1% engine-equivalent span) for the closed double squealer and its variants. Aerodynamic measurements are taken using a three-hole probe and a single-hole probe for near casing measurements. Downstream pressures are obtained using a 2-dimensional traverse system. Heat transfer measurements are taken using a transient, infrared thermography approach.

4.2 HSLC Facility Overview

The HSLC facility is a blow-down type facility that operates under transonic conditions. The facility is supplied by compressed air that is regulated using a series of valves and a pressure regulator. The mainstream supply pressure of air from storage tanks is 2750 kPa. A Spirax-Scarco control valve is used in conjunction with a proportional-integral-derivative process controller that is used to maintain a set pressure as the tank pressure drops. A photograph and a schematic of the test facility are shown in Figures 4.1 and 4.2. The development and installation of the testing facility was done by Dr. Qiang Zhang. Commissioning, instrumentation and measurement techniques were then conducted by Dr. Devin O'Dowd whose contributions are gratefully acknowledged. A more detailed account of the HSLC facility is documented by O'Dowd (2010), therefore, only a concise overview is presented.



Figure 4.1: Oxford University High-Speed Linear Cascade facility

The HSLC is instrumented to measure both pressures and temperatures. An inlet pitot-static probe and inlet thermocouple are located one axial chord upstream of the blade row. A turbulence grid is also placed at the inlet of the cascade test section.

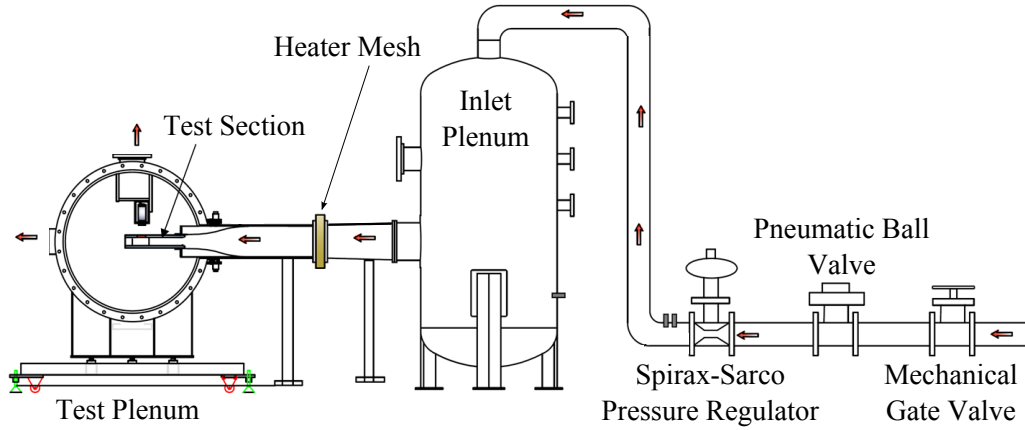
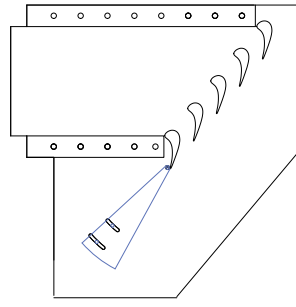


Figure 4.2: Schematic of the Oxford High-Speed Linear Cascade facility (CAD courtesy of Zhang (2012))

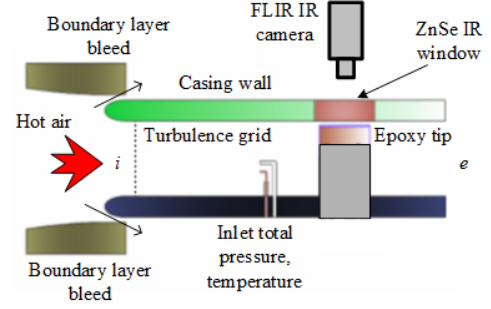
Measurements were made with a manually traversed hot-wire probe in assistance with Dr. Qiang Zhang whose results have been presented in Zhang *et al.* (2014). A turbulence intensity was recorded at approximately 7%. Static pressure measurements are made on both hub and casing walls, one axial chord downstream of the blade row. A PC logs all pressures and temperatures data and LabVIEW 8.6 is used for data acquisition purposes. A GE Druck DPI520 pressure controller is used for calibration of the 32-channel transducer system used for pressure measurements.

The test section and instrumentation is shown in Figure 4.3. The section consists of 5 blades and 4 passages with an adjustable tip gap and boundary layer bleeds. Measurements are taken at tip gaps of 0.5 mm, 1.0 mm and 1.5 mm which equate to 0.5%, 1% and 1.5% of the engine-equivalent span or 1%, 2% and 3% of the blade chord for both uncooled and cooled blade tips for an open cavity squealer tip. The intermediate tip gap (1% engine-equivalent span) is then used for the closed double squealer and its variants. Two of the blades are used to represent the sidewalls consisting of a suction side and a pressure side. The scaling factor of the test section is 1.8. Details of geometry parameters are given in Table 4.1.

Blade loading test data was supplied by Rolls-Royce for the XWB blade. The HSLC was operated at a range of inlet conditions and the measured inlet total pressure



(a) Test Section



(b) Instrumentation



(c) Photo of Test Section

Figure 4.3: HSLC Test Section and Instrumentation

Inlet angle	40.11°
Exit angle	64.23°
Axial chord	49.14 mm
Blade pitch	66.46 mm
Test Section Height	71.84 mm
Engine-equivalent blade span	101.78 mm

Table 4.1: HSLC Geometry Parameters

and outlet static pressures were fed into 2D Ansys Fluent simulations conducted by Dr. Qiang Zhang (Zhang, 2012). The chosen operating conditions for the HSLC were based on the results with the closest agreement with the Rolls-Royce data. The results of blade loading, represented as isentropic Mach number, for the test blade are shown in Figure 4.4. The HSLC nominal flow conditions used throughout the study are given in Table 4.2.

The test blade tips for the open squealer both uncooled and cooled were manufactured by Rolls-Royce using stereolithography. The material is an epoxy resin

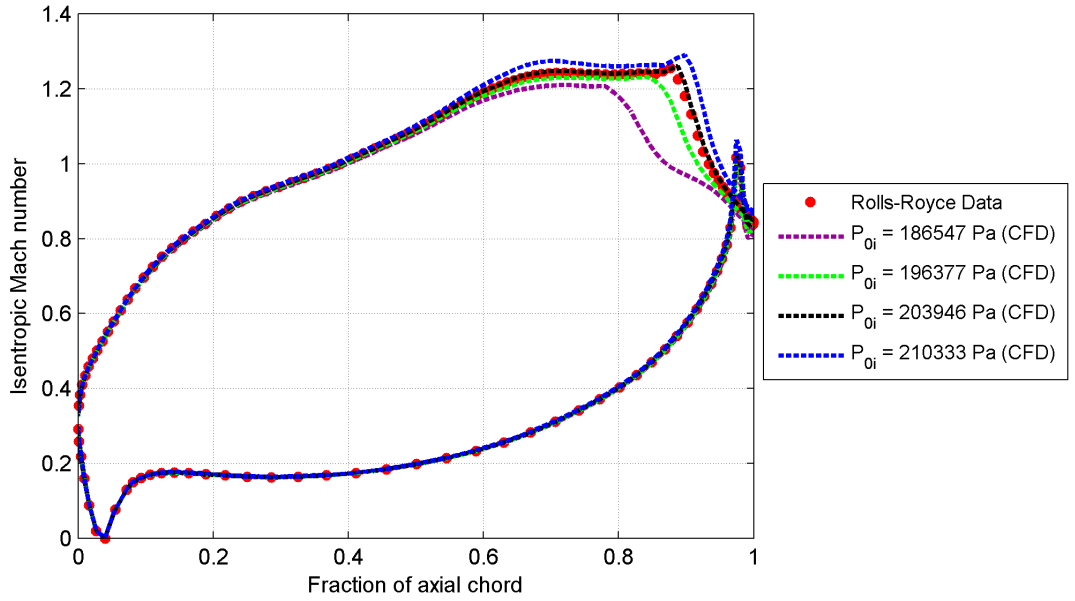


Figure 4.4: Mid-span isentropic Mach number distributions (CFD courtesy of Zhang (2012))

Inlet total pressure	204 kPa
Inlet static pressure	192 kPa
Inlet Mach number	0.3
Inlet velocity	101.3 ms^{-1}
Inlet Reynolds number (based on axial chord)	0.66×10^6
Exit static pressure	107 kPa
Isentropic exit Mach number	1.01
Exit Reynolds number	1.24×10^6

Table 4.2: HSLC Flow Conditions

using DSM Somos GP Plus 14122 with a low thermal conductivity. The closed cavity squealer was manufactured by Gemini Prototyping Ltd. using DSM Somos Water-Shed XC 11122. These blades have a short height (30 mm) and are bolted onto a base blade made of steel. They are designed so that they are rigid under the prescribed aerodynamic load and can be interchanged with ease. A pin is also fixed near the trailing edge region to reduce deformation.

4.3 Aerodynamic Loss Measurements

Aerodynamic loss measurements are acquired using a two-dimensional traverse system that helps obtain an evaluation of the downstream total pressures. These measurements are taken at two locations, at half and one axial chord downstream of the blade trailing edges. A three-hole and single-hole probe are used to obtain the total pressures. These probes are constructed from stainless steel and are 0.7 mm in diameter. A sting of 7 mm diameter is used to support the probes. The traverse system itself is driven by stepper motors and was controlled by LabVIEW 8.6. A calibration of the three-hole probe was done by O'Dowd (2010). Due to the three-hole probes thickness, the probe could only take measurements 4 mm away from the casing wall. This was addressed by using a single-hole probe that is much thinner and able to measure up to 0.5 mm away from the casing wall. This was mechanically rotated by $\pm 5^\circ$ to act as a three-hole probe. The total pressure near the casing was then obtained by interpolation to obtain the maximum total pressure. Initially, for the open squealer, the data acquired from the single-hole probe was disregarded as results were not well captured. It was later discovered that the probe was blocked. Once this issue was resolved, all remaining aerodynamic loss data used the addition of a single-hole probe, namely for measurements of the closed squealer tip and its variants.

The traverse system was used to take measurements at uniform pitchwise and spanwise locations. All measurements are taken from the same initial pitchwise position at each radial position. Measurements at the exit planes are taken at 27 equally spaced pitch-wise locations and 6 spanwise locations, covering 77% to 96% of the engine equivalent blade span covering one blade pitch in total. For the closed squealer tip cases, the measured data included the upper 25% of the engine equivalent span. Measurements were taken at 22 equally spaced pitchwise locations and at 9 spanwise locations, of which 2 were used for single-hole data acquisition. Figure 4.5 shows data obtained at 8 mm away from the casing wall detailing the measured total pressure

loss coefficient and the wake generated at the exit of the cascade (one axial chord downstream).

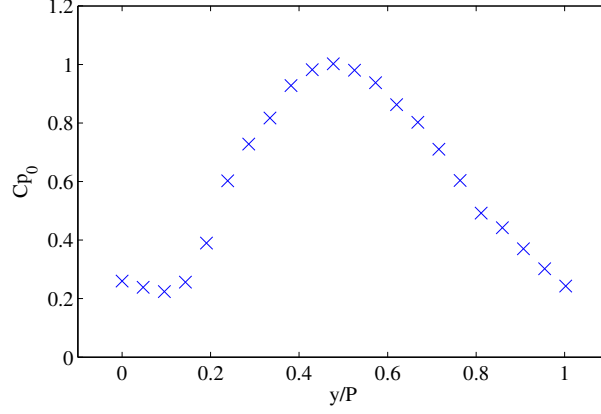


Figure 4.5: Pitchwise locations of measured total pressure loss, C_{p0}

4.3.1 Three-hole Probe Calibration

The calibration of the three-hole probe was conducted in a transonic tunnel at the University of Oxford and covered a range of $-20 \leq \beta \leq 20$ and $0.6 \leq M \leq 1.3$. The design of the probe and calibration is based on Povey *et al.* (2008) and was conducted by O'Dowd (2010) and also presented in O'Dowd *et al.* (2010).

Generally, the three pressures measured by the probe are P_A , P_B and P_C which are functions of β , M , P_0 and Reynolds number. The subscripts A, B and C define the left, center and right ports of the probe. The design of the probe ensured insensitivity to the Reynolds number, so the calibration only depends on β , M and P_0 . The calibration coefficients for yaw (C_β), total pressure (C_T) and Mach number (C_M) are defined as:

$$C_\beta = \frac{P_A - P_C}{P_B - P_{avg}} \quad (4.1)$$

$$C_T = \frac{P_0 - P_B}{P_B - P_{avg}} \quad (4.2)$$

$$C_M = \sqrt{\frac{2}{\gamma - 1}} \left[\left(\frac{P_{avg}}{P_B} \right)^{\frac{1-\gamma}{\gamma}} - 1 \right] \quad (4.3)$$

$$P_{avg} = \frac{P_A + P_C}{2} \quad (4.4)$$

As described by Main *et al.* (1996), the calibration data acquired from the tunnel runs are put into a probe calibration matrix and transformed to application matrices. For example, the calibration matrix for total pressure, $C_T(M, \beta)$ and application matrix, $C_T(C_\beta, C_M)$ maps are shown in Figure 4.6. The total pressure, yaw angle and Mach numbers can then be obtained through a lookup table using a linear interpolation.

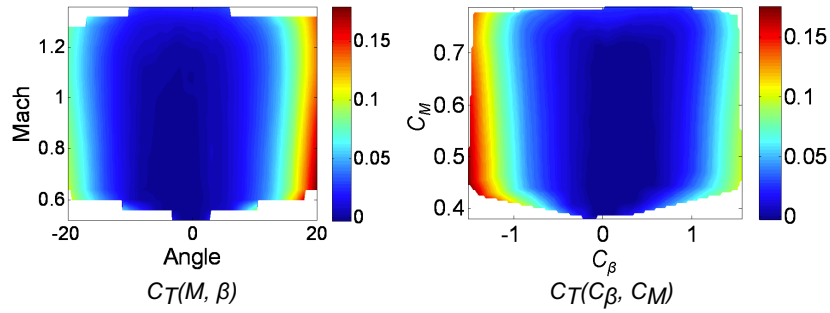


Figure 4.6: Three-hole probe calibration and application matrix for C_T (Reproduced from O'Dowd et al. (2010))

4.4 Heat Transfer Measurements

Heat transfer measurements are taken on the tip of the centrally located blade and were obtained using a transient, infra-red thermography method to determine local heat transfer coefficients. A heater mesh is located upstream of the test section and provides a step-wise increase in mainstream flow temperatures of approximately 20°C. An example of the temperature histories measured during the thermal tests is given in Figure 4.7 where the slow response of the tip surface can be clearly identified. A

preheating compression effect due the opening of the air supply valves can also be seen which takes approximately 15 seconds to meet the desired test conditions, this is marked as “A”. However, this effect is not problematic and is accounted for using the heat transfer process technique used in the present study. At time “B”, the heater mesh is activated and the time period “C” is the useful range for the conduction analysis, equating to approximately 2 seconds.

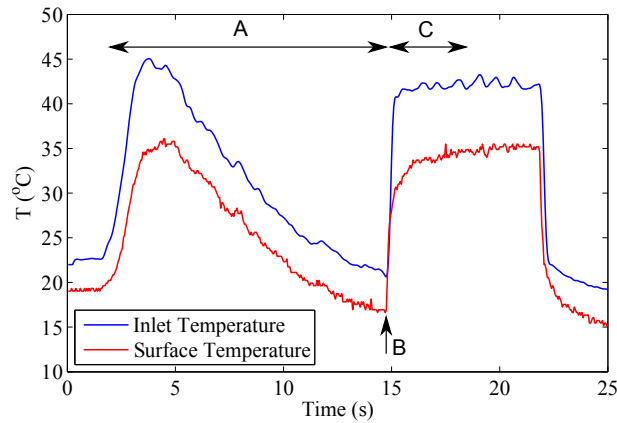


Figure 4.7: Temperature histories showing the mainstream and the tip response temperatures

The heater mesh is powered by a Drake System DC power supply rated at 100 V and 1.5 kA. A 25 μm diameter, fast response thermocouple is located on the tip and used for in-situ calibration of the infrared images. The calibration of recorded greyscale values to measured tip surface temperatures is shown in Figure 4.8. Heat transfer is modelled using a semi-infinite, solid conduction analysis. Since the thermal conductivity of the epoxy resin is low, the semi-infinite, one dimensional conduction analysis can be assumed. An infrared transmissive viewing window made of Zinc Selenide is placed within the casing wall to provide optical access to the tip. All blade tips used for thermal measurements are painted black. A FLIR A20M infrared camera is used to record spatially resolved surface temperature measurements into a digital file simultaneously with tip surface temperature readings. The camera is equipped with a FireWire 400 data cable and the camera supplies an 8-bit greyscale output at a resolution of 160 x 120 at 30 frames per second. A linear temperature

range of 0-50°C is used. The camera is housed in an anti-vibration casing, vertically above the blade tip allowing a clear view of the tip surface.

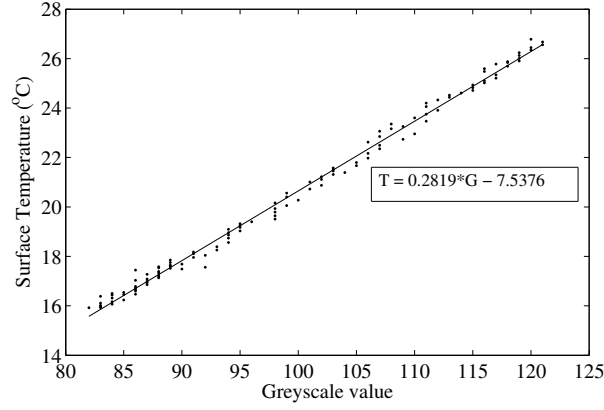


Figure 4.8: Infrared in-situ calibration slope

4.4.1 Heat Transfer Data Processing

Based on the one-dimensional transient conduction equation, heat transfer can be described by Fourier's law:

$$k \frac{\partial^2 T}{\partial x^2} = \rho C_p \frac{\partial T}{\partial t} \quad (4.5)$$

where T is the temperature, x is the depth into the solid, normal to the surface, k is the thermal conductivity, ρ is the density of the solid, C_p is the specific heat and t is the time. Equation 4.5 is then solved for the given initial boundary and convective boundary conditions:

$$at \quad t = 0, \quad T = T_i \quad (4.6)$$

$$as \quad x \rightarrow \infty, \quad T = T_i \quad (4.7)$$

$$at \quad x = 0, \quad -k \frac{\partial T}{\partial x} = h(T_{ad} - T_w) \quad (4.8)$$

Assuming the solid conduction penetrates to infinity in all directions but one, the solution of Equation 4.5 with the given initial and boundary conditions is:

$$\frac{T_w - T_i}{T_\infty - T_i} = \operatorname{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right) - \left[\exp\left(\frac{hx}{k} + \frac{h^2 \alpha t}{k^2}\right)\right] \left[\operatorname{erfc}\left(\frac{x}{2\sqrt{\alpha t}} + \frac{h\sqrt{\alpha t}}{k}\right)\right] \quad (4.9)$$

The solution as well as other complex conduction problems has been shown by Schneider (1955). Knowledge of the surface wall temperature, T_w , in Equation 4.9 enables the evaluation of the only unknown, that is h , the heat transfer coefficient. The equation itself is a function of the initial temperature T_i , the mainstream temperature, T_∞ which also requires a stepwise change after the initial time, and the solid thermal characteristics, α , where $\alpha = k/\rho C_p$.

Since a large number of data locations and high sampling rate is required, the computational cost of an idealised analysis is high. Furthermore, the lack of a clean step increase in temperature is also problematic. In terms of reducing this cost and accounting for any preheating effects, a heat flux reconstruction is done. For a small number of data locations and low sampling rates, a finite difference heat flux reconstruction can be done using the surface temperature time history (Oldfield *et al.*, 1978). This approach was then further developed to improve the efficiency and provide highly resolved results. This method is called the ‘‘Impulse Response Method’’ developed by Oldfield (2008). Rather than using a finite difference approach, the Impulse Response Method uses a frequency domain and discrete deconvolution to derive impulse responses. The technique is based on knowing a pair of analytical solutions for the heat transfer equations that are sampled and used by the convolution to provide an impulse response. The response method assumes a linear time invariant

system where the data is initially steady. The basis of the method works by generating an impulse response, $h(t)$, using a temperature input to generate a heat flux through a convolution of the temperature and the impulse response, as shown in Figure 4.9.

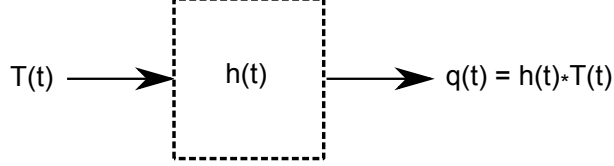


Figure 4.9: Basis of the impulse response method

Using the impulse response, the output can be computed using a convolution integral:

$$q(t) = h(t) * T(t) = \int_{-\infty}^{\infty} h(\tau) T(t - \tau) d\tau \quad (4.10)$$

As the signal being recorded is discrete and being sampled, Equation 4.10 can be replaced by a discrete convolution sum:

$$q[n] = \sum_{k=0}^{N-1} h[k] \cdot T[n - k] \quad (4.11)$$

This convolution is implemented using a Fast Fourier Transform based filter. The heat flux reconstruction is done at each measured pixel location. Due to the increased speed, efficiency and elegance of the Impulse Response Method, as described by O'Dowd *et al.* (2011b), it is the chosen heat transfer measurement technique used for this study. Following this, the local heat transfer coefficients can be determined due to the relationship between the heat flux and wall temperature during each blow-down run determined by Equation 4.12:

$$\dot{q} = h(T_{ad} - T_w) \quad (4.12)$$

where $\dot{q}$ is the heat flux, T_w is the wall temperature and T_{ad} is the adiabatic wall

temperature.

Thus, as a linear relationship is expected, the heat transfer coefficient, h , can be calculated by the gradient of the transient heat flux data to the temperature data. The adiabatic wall temperature, T_{ad} , can be calculated by extrapolating the temperature for zero heat flux. For each transient test, local h values are obtained using this technique as shown in Figure 4.10. The uncertainty of this method of heat transfer coefficient calculation has been completed by O'Dowd *et al.* (2011b) and is directly applicable to the present study also. An uncertainty of 9.5% was estimated in heat transfer coefficient and 1.2 K in adiabatic wall temperature. The thermal product, $\sqrt{\rho C_p k}$, of the DSM Somos GP Plus 14122 material was supplied by the manufacturers at $644 \text{ J s}^{-\frac{1}{2}} \text{ m}^{-2} \text{ K}^{-1}$.

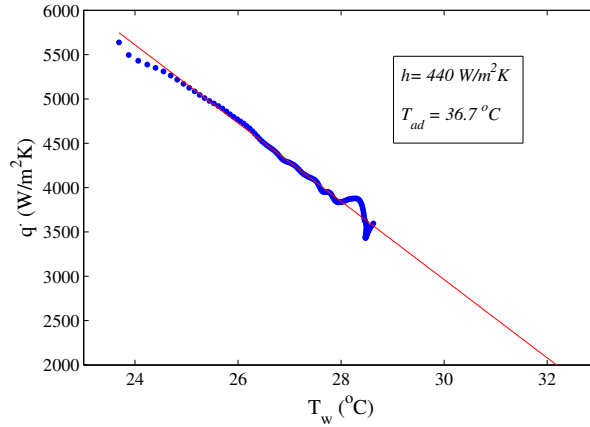


Figure 4.10: Heat flux versus tip surface temperature history

4.5 Film Cooling Setup and Measurement

The three central blades are all replaced with the cooled versions of the open cavity squealer tip to provide an assessment of the tip film cooling at high-speed test conditions. The coolant feed system is driven by ambient temperature air at 650 kPa linked to three connecting vortex tubes. The use of vortex tubes enables a reduction of air

temperature in relation to the mainstream air supply to provide an adequate coolant temperature. The vortex tubes are connected to the test blade, whilst the neighbouring blades are supplied with uncooled air to maintain periodicity. The vortex tubes, cooling feed and controls are shown in Figure 4.11.

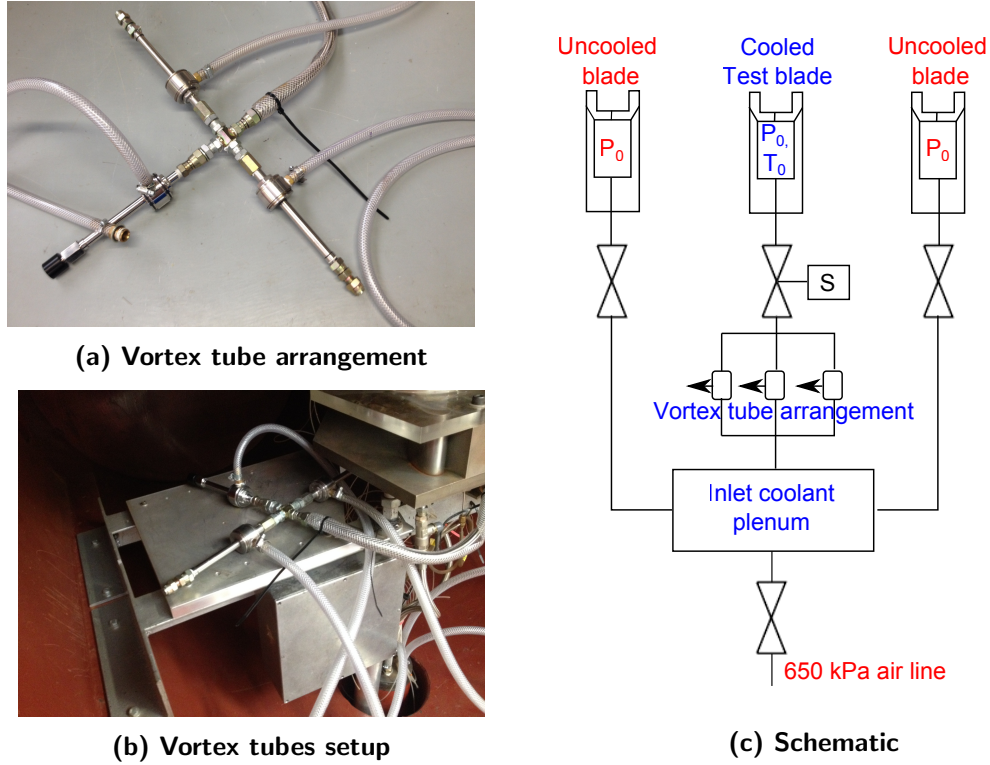


Figure 4.11: Coolant feed system

The vortex tubes are able to drop the temperature to approximately 7-10°C, providing a mainstream-to-coolant temperature ratio of 1.1. This ratio is under the target ratio of 2 which is typical of an operational engine. However, the power supply needed to raise the mainstream gas temperature to required levels is expensive and restricted by the current setup. Furthermore, it is limited by the thermal constraints of the epoxy material. Providing a colder coolant temperature is also troublesome for the epoxy resin as the holes have a tendency to freeze if driven much lower than already prescribed.

The coolant feed system must be operational for at least 30 minutes prior to

running tests in order to establish a stable temperature. Coolant pipes are also insulated to help reduce heat transfer from the surrounding ambient air. A solenoid switch is used to ensure the blade is not pre-cooled for the test runs and is directed towards the internal cooling plenum momentarily before the heater mesh is engaged. This eradicates any fluctuations the coolant temperature. The coolant-to-mainstream total pressure ratio is 1.1 to prevent ingestion of mainstream flow into the blade coolant reservoir and is held constant for all tip gap configurations. Through the use of a sonic orifice plate, the coolant mass flow rate was measured to be 1.20% of the HSLC mainstream mass flow rate at a tip gap of 1.5% of the engine-equivalent span.

To quantify the film cooling performance, heat transfer coefficients may be used to assess the tip film cooling, but a more useful technique that is widely established is to calculate the film cooling effectiveness, η :

$$\eta = \frac{T_{rec} - T_{ad}}{T_{rec} - T_c} \quad (4.13)$$

Equation 4.13 uses the adiabatic wall temperature that is calculated during the process of calculating the heat transfer coefficient. In this equation, T_{rec} is the recovery temperature, T_{ad} is the adiabatic wall temperature of the cooled blade and T_c is the coolant temperature. T_c does not use local values as these are difficult to measure, so a single temperature of the internal chamber is assumed. T_{rec} is the adiabatic wall temperature calculated when no cooling is present. Rather than plugging the holes, the method of replacing the cooled blade with the uncooled equivalent was used to calculate T_{rec} .

4.6 Uncertainty Analysis

The measurement uncertainty for the methods described above have been conducted by O'Dowd *et al.* (2011b) for heat transfer and O'Dowd (2010) for the aerodynamic

pressure measurements. These uncertainties measured are also applicable to the present study as a continuation of the study and measurement techniques used.

The heat transfer uncertainties were determined by a combination of partial derivatives as given by Kline & McClintock (1953), a jitter analysis as described by Moffat (1988) and the standard error of estimate as described by Coleman & Steele (1999). For the Impulse Response Method, the measurement uncertainties for heat transfer are given in Table 4.3.

	Heat transfer coefficient	Adiabatic wall temperature
Material property	6.4%	negligible
Measured wall temperature	7.1%	1.2 K
Mainstream temperature	negligible	negligible
Overall uncertainty	9.5%	1.2 K

Table 4.3: Measurement uncertainties for heat transfer using the Impulse Response Method

The three-hole probe aerodynamic uncertainties were also conducted by O’Dowd *et al.* (2010). These were determined at each point by substituting the calibration data into the application matrix and computing the total pressure, yaw angle, and Mach number and are based on root mean square differences over the whole range. Using the method described by Kline & McClintock (1953), the uncertainties are determined and given in Table 4.4. Measurements of the true total pressure within the over-tip leakage vortex differ by 2.8% at a yaw angle of 20° and indicates an overestimate of the maximum error induced.

Measurement	Uncertainty
True total pressure, P_0	0.96%
Yaw angle	2.2°
Mach number	0.081

Table 4.4: Measurement uncertainties for the three-hole probe

4.7 Summary

In this chapter, the transonic flow experimental facility has been described with details of the flow conditions and measurement techniques used. The use of the High-Speed Linear Cascade serves fittingly and is capable of providing engine-representative test conditions (exit Mach number of 1.0 and exit Reynolds number of 1.23×10^6) and well resolved data. The facility itself has accounted for a wealth of tip research on flat and winglet tips. The techniques described in this chapter have enabled the acquisition of spatially-resolved heat transfer and aerodynamic loss measurements to provide a better physical understanding of a squealer at high-speed test conditions.

For the heat transfer, a technique was described that enables the acquisition of detailed, spatial thermal measurements. The method uses transient infrared thermography with the Impulse Response Method to reconstruct the variation of heat flux with surface temperature measurements. With use of both methods and the one-dimensional, semi-infinite conduction analysis, a map of surface heat transfer on the squealer tip can be obtained with a measurement uncertainty previously determined to be less than 10%.

For the aerodynamics, a traversed three-hole and single-hole pressure probe measurement technique was described and is able to provide a detailed map of the losses created by the squealer tip at set locations downstream of the blade row. The three-hole probe has been calibrated and shows low uncertainty, no greater than 2.8% from the true total pressure and is able to provide exit yaw angles and Mach numbers.

Both aerodynamic and heat transfer measurement techniques can also be applied to a cooled setup and the details of the experimental conditions capable by the facility and cooling performance parameters were also given.

Chapter 5

Transonic Squealer Tip

5.1 Introduction

The contents of this chapter include the experimental results obtained using the High-Speed Linear Cascade (HSLC) facility at the University of Oxford, Thermo-Fluids Laboratory and provide a basis for validating CFD code. The main aims are to understand the aerodynamic and thermal performance of a squealer tip at engine-representative conditions. The chosen squealer tip is an open cavity variant of a standard squealer and was described in Chapter 3. Detailed, spatially-resolved heat transfer measurements are taken on blades manufactured using stereolithography and done using a transient, infrared thermography technique described in Chapter 4. Aerodynamic loss measurements are taken at two exit planes, 50% and 100% of the axial chord downstream of the blade row, normal to the axial axis. The measurements are obtained using a three hole probe. Both studies use an uncooled squealer tip and all measurements are taken at three different tip gaps (0.5%, 1% and 1.5% of the blade span which is equivalent to 1%, 2% and 3% of the blade axial chord). Total pressure loss is examined at both exit planes and Mach number and yaw angle are limited to the furthest exit plane. CFD predictions are obtained using HYDRA.

5.2 Tip Heat Transfer

This section examines the uncooled open cavity squealer heat transfer at three tip clearances, 0.5%, 1% and 1.5% of the blade span. Measured tip heat transfer coefficients are presented in contour plots in Figure 5.1. It is well established based on previous literature that the heat transfer increases with larger tip gaps and the present results also confirm this.

The general flow physics are such that a strong reattachment is present within the frontal cavity region of which follows a large area of flow separation created by the pressure side rim. The latter is also more pronounced for smaller tip gaps. For all cases, the leading edge rim has the highest heat transfer coefficient and it is easily identifiable that the rims in particular are most severely affected. This is also true for the smallest tip gap, but in contrast, low levels of heat transfer within the cavity are well observed. In regards to the rims, it is expected that there are some 3D conduction effects that are not taken into account. Nevertheless, the rims and leading edge cavity are regions that are the most at risk to the hot gas.

Heat transfer coefficient predictions using HYDRA are shown in Figure 5.2. Firstly, the computational results capture the majority of the trends found with the HSLC tests. The tip gap survey shows increasing heat transfer coefficient with increased tip gap. The leading cavity hot spot and suction side rim both show this trend well. Secondly, the magnitudes of heat transfer coefficient are well predicted for most regions, in particular for the two larger tip gaps. Finally, the spatial distributions of heat transfer are detected well. For example, the leading edge rim and cavity region of high heat transfer coefficient, and the low region adjacent to the pressure side rim. The overall averaged tip heat transfer coefficients are given in Table 5.1.

The circumferentially-averaged heat transfer coefficients for the three tip gaps are shown in Figure 7.13. Results show both measured and predicted values. It is clear that increasing the tip gap results in higher heat transfer coefficients. The trends

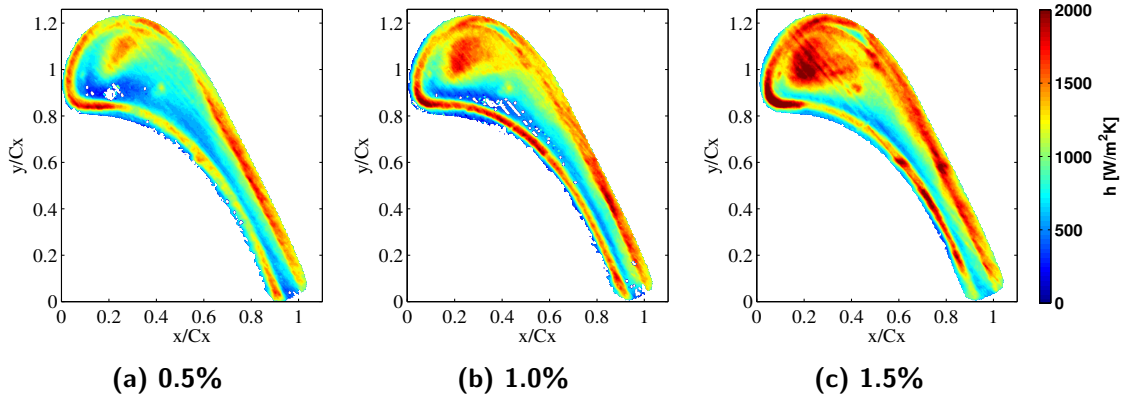


Figure 5.1: Measured squealer tip surface heat transfer coefficients (W/m^2K) for 0.5%, 1% and 1.5% tip gaps (HSLC)

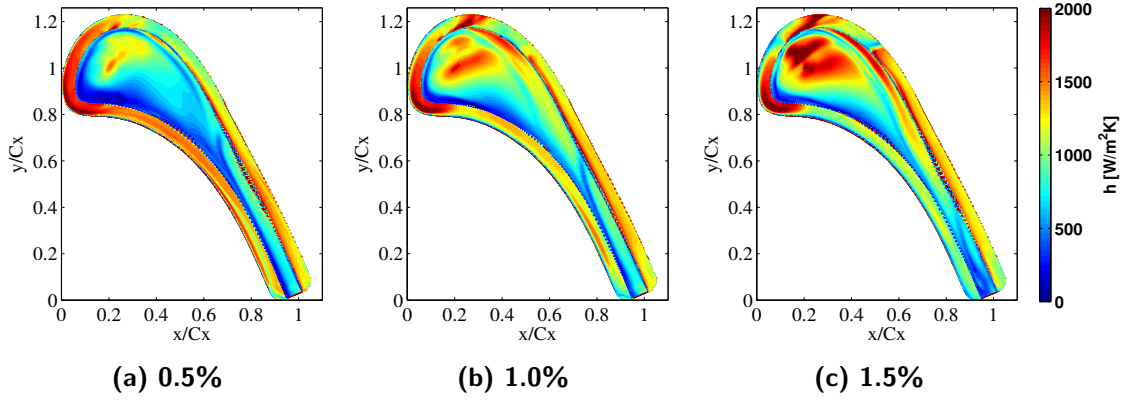


Figure 5.2: Predicted squealer tip surface heat transfer coefficients (W/m^2K) for 0.5%, 1% and 1.5% tip gaps (HYDRA)

shown suggest that local heat transfer is largely affected by tip clearance size when the local flow is subsonic. In this region (0-60% axial chord), h decreases with a reduction in tip gap. The different flow regimes for the three tip gaps can be seen in Figure 5.4 showing the flow velocity at mid-tip gap planes. Interestingly, where the flow is transonic, changes are less obvious. For regions near the trailing edge, an opposite trend is observed; there is a cross-over point in the heat transfer coefficient with tip gap variation at around 80% axial chord. This is more apparent with the HYDRA results. Similar cross-over behaviour was reported for a flat tip by Zhang *et al.* (2010a).

Referring to Figure 7.13, noticeable differences are identified. The leading edge

Tip gap (g/S)	HSLC h [W/m ² K]	HYDRA h [W/m ² K]
0.5%	1012.6	1025.4
1%	1122.0	1107.2
1.5%	1228.0	1136.6

Table 5.1: Mean tip heat transfer coefficients

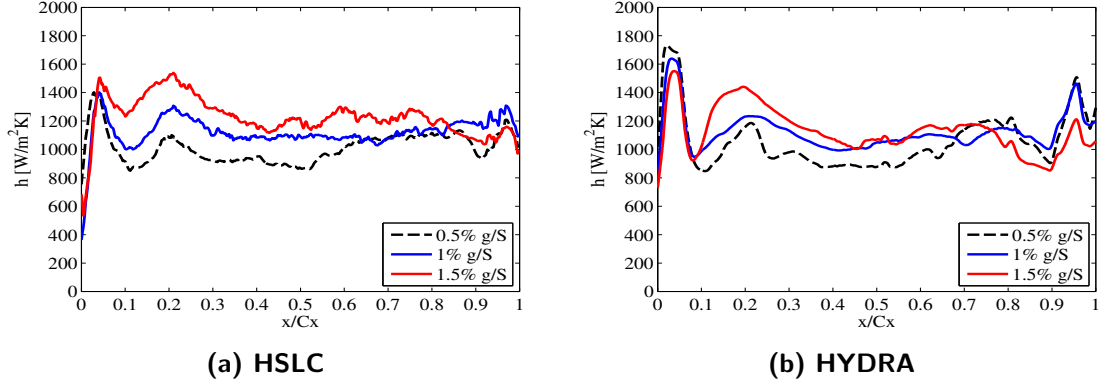


Figure 5.3: Circumferentially-averaged tip heat transfer coefficient (W/m²K)

heat transfer follows a different trend and HYDRA shows the peak h at the leading edge (0-10% axial chord). This discrepancy is mostly associated with the smallest tip gap, where in practice, accurately positioning the blades and measuring a tight tip clearance of 0.5% has a significant source of error associated with it. Large discrepancies are also seen towards the trailing edge of the blade where three-dimensional conduction effects most likely dominate.

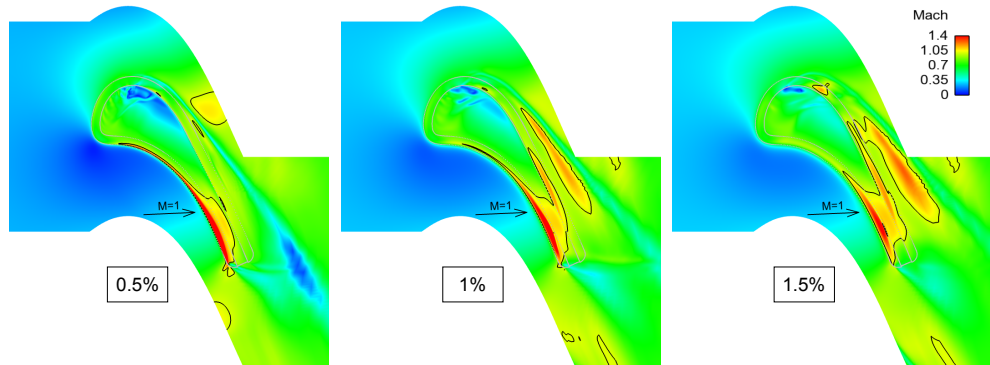


Figure 5.4: Local mid-gap Mach number distributions (solid line indicates Mach=1) over squealer tip (HYDRA)

Overall, the mean heat transfer coefficients give percentage errors between the measured and predicted results of 1.3%, 1.3% and 7.4% for the 0.5%, 1% and 1.5% tip gaps respectively. The majority of features are detected by HYDRA for this geometry and the trends associated with the tip gap survey are very consistent and compare well with the experimental data, falling within the quoted uncertainty bounds of $\pm 9.5\%$.

5.3 Aerodynamic Loss

Total pressure loss is examined at two exit planes of the domain. These planes are located at half an axial chord and one axial chord downstream of the blades trailing edges. Exit Mach number and yaw angle is examined at the one axial chord plane only. Attention is focused to the tip region, thus measurements and calculations are made on well defined planes covering approximately the upper quarters of each domain. The two domains are illustrated in Figure 5.5.

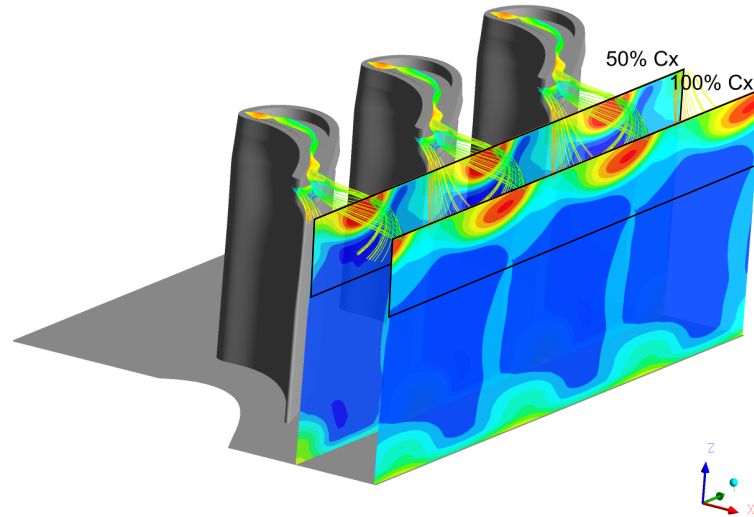


Figure 5.5: Locations of exit planes used in the present study (Ansys CFX)

5.3.1 Aerodynamic Loss: One Axial Chord Downstream

5.3.1.1 Total Pressure Loss

Firstly, the measured total pressure loss coefficients, C_{p_0} , for the open cavity squealer tip at one axial chord downstream of the blade row are presented in Figure 5.6. The trailing edges of the blade row are located at positions $y/P = 0, 1, 2$. The results presented illustrate typical over-tip leakage flow features. The main observation is the larger the tip gap, the larger the over-tip leakage vortex. Higher losses are created with the growth of these vortices. Secondly, an additional development is the shifting of the leakage vortex as the gap increases. This is clearly identifiable by the contour maps. As the tip gap becomes larger, the vortex cores shift away from the blade suction surfaces. This is noticed by the centres of the cores moving towards the right. Furthermore, the smallest tip gap shows the formation of a passage vortex and a faint indication observed with the middle gap (between $0.5 < y/P < 1$).

Numerically predicted total pressure loss coefficient distributions for the three tip gaps determined by HYDRA are shown in Figure 5.7. The comparison with the experimental results shows good agreement, both in terms of the OTL vortex size as well as the pitchwise shifting of these vortices. The weak passage vortex is detected at the small tip gap whilst at the middle tip gap, the results are less pronounced and likely to be caused by numerical dissipation errors with the solver.

Pitchwise, mass-averaged C_{p_0} plots are presented in Figure 5.8 for the three tip gaps. Figure 5.9 shows overall mass-averaged total pressure losses¹. The formation of loss is well predicted by HYDRA. Overall, the measured and calculated results agree fairly well. The overall mass-averaged aerodynamic loss percentage errors are 5.8%, 7.3%, 5.2% for the 0.5%, 1% and 1.5% tip gaps, respectively.

¹Overall values for open cavity squealer exclude the uppermost (4% span) data after establishing that the initial data from using the single hole probe were incorrect due to being blocked

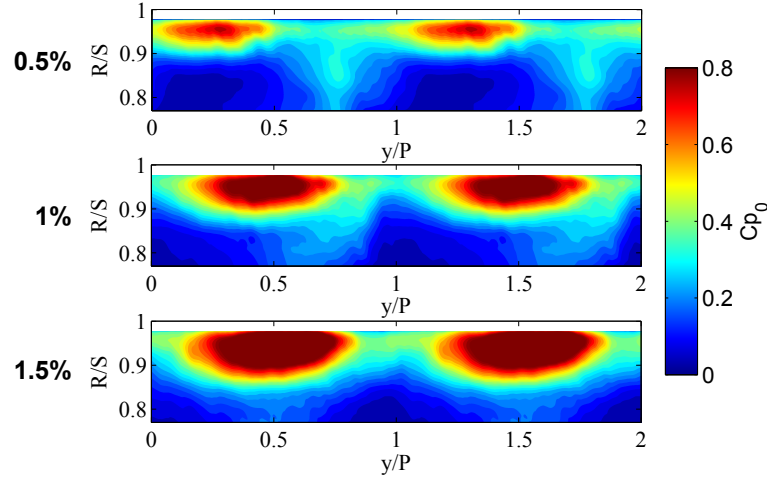


Figure 5.6: Experimentally measured total pressure loss coefficient, C_{p0} , contours for open cavity squealer tip for 0.5%, 1% and 1.5% tip gaps at 100% C_x downstream (HSLC)

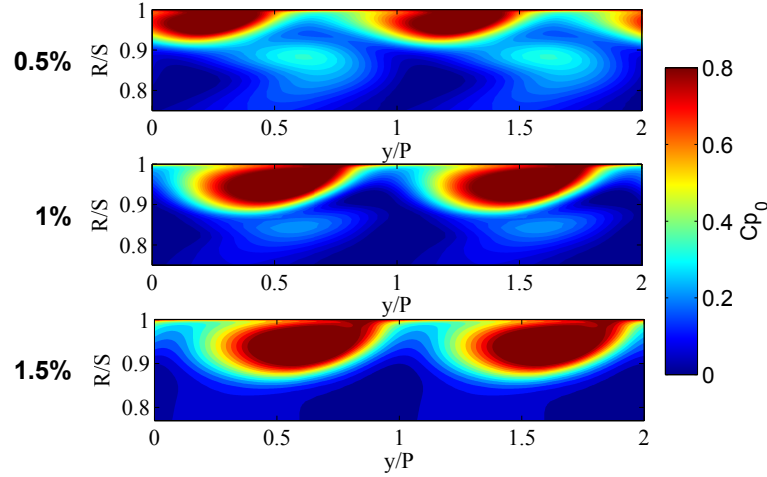


Figure 5.7: Numerically calculated total pressure loss coefficient, C_{p0} , contours for open cavity squealer tip for 0.5%, 1% and 1.5% tip gaps at 100% C_x (HYDRA)

5.3.1.2 Exit Mach number and Flow Angle

The transonic nature of the flow under the prescribed conditions can be identified by downstream Mach numbers. The measured and predicted Mach numbers, at the one axial chord downstream, are presented in Figure 5.10. Typical features are found including low velocity flow in the vicinity of the tip leakage vortex and high velocities elsewhere where losses are minimal. The peak measured values of Mach number calculated using the three hole probe is 1.05 for the three tip gaps. Notably, previous HSLC acquired data showed larger discrepancies with recorded Mach numbers

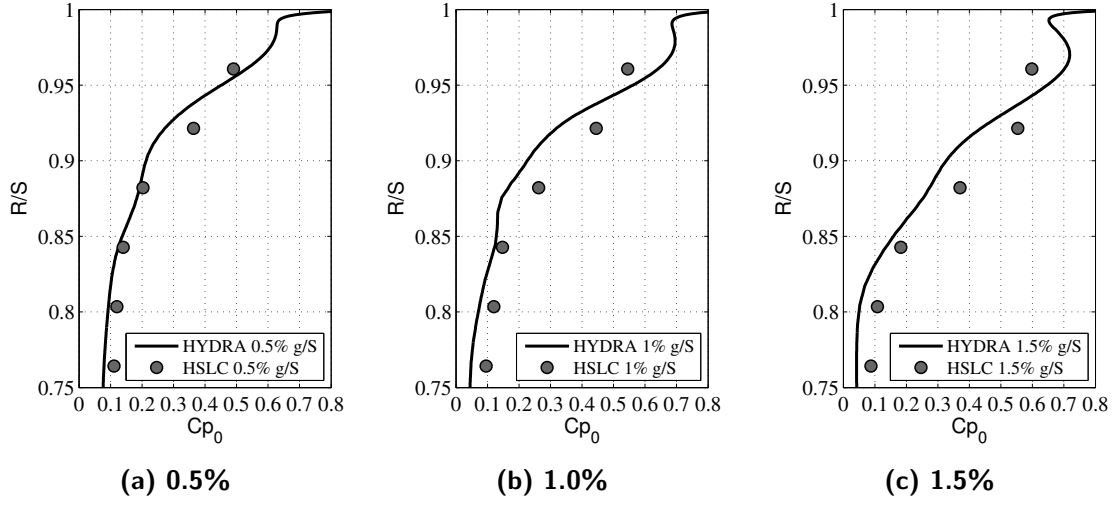


Figure 5.8: Open cavity squealer tip pitchwise-averaged total pressure loss C_{p0} for 0.5%, 1% and 1.5% tip gaps at 100% C_x (HSLC and HYDRA)

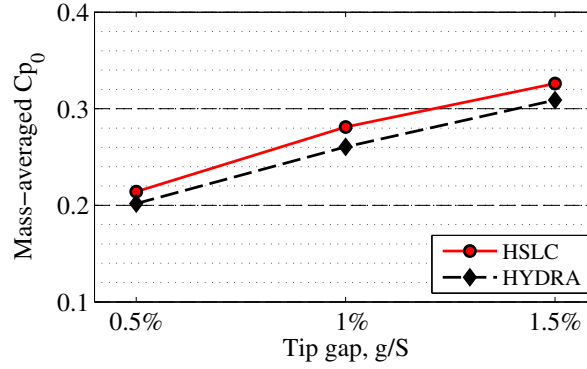


Figure 5.9: Mass-averaged total pressure loss variations for 0.5%, 1% and 1.5% tip gaps (HSLC and HYDRA)

between 1.2 and 1.3 (Tang, 2011; O'Dowd, 2010). The averaged predicted peak Mach number for the three tip gaps is 1.00. The HYDRA calculations show reasonable agreement to those measured, however large discrepancies are found especially in locations close to the vortex cores.

The final set of data acquired using the three hole probe is the exit yaw (β) angle, giving an indication of the turning of flow within a turbine and is often used as a measure of aerodynamic loss in gas turbine design. This parameter is used as a performance indicator as it determines the work transfer within a turbine stage. Figure 5.11 shows results of exit yaw flow angle that are relative to the blade exit

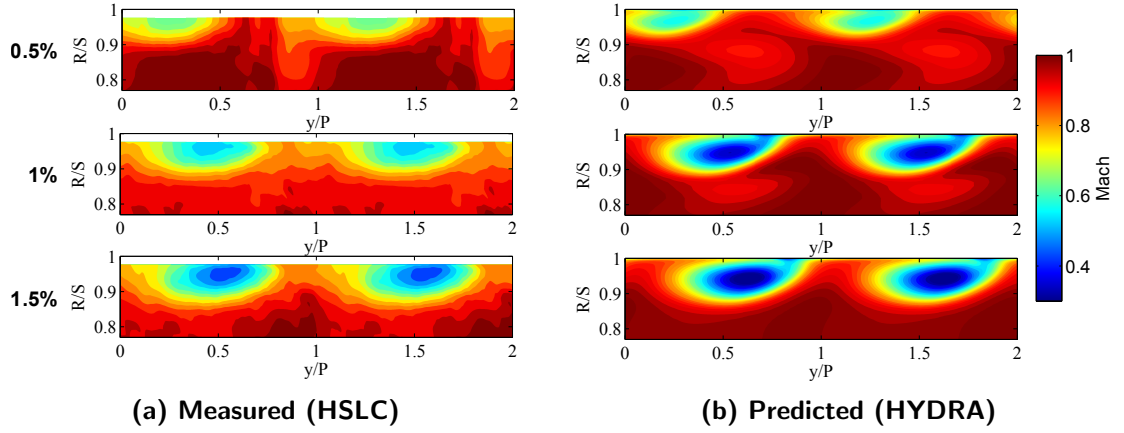


Figure 5.10: Exit Mach number distributions

metal angle (67°). The plots show the strengths of the secondary flows near the endwall. Relative to the blade exit metal angle, the results show the under-turning (positive angles) and over-turning (negative angles) of flow whose magnitudes increase with increased tip leakage vortex. The yaw angles show only fair agreement for the range measured especially near the casing wall (around 96% R/S) and for the smallest tip gap.

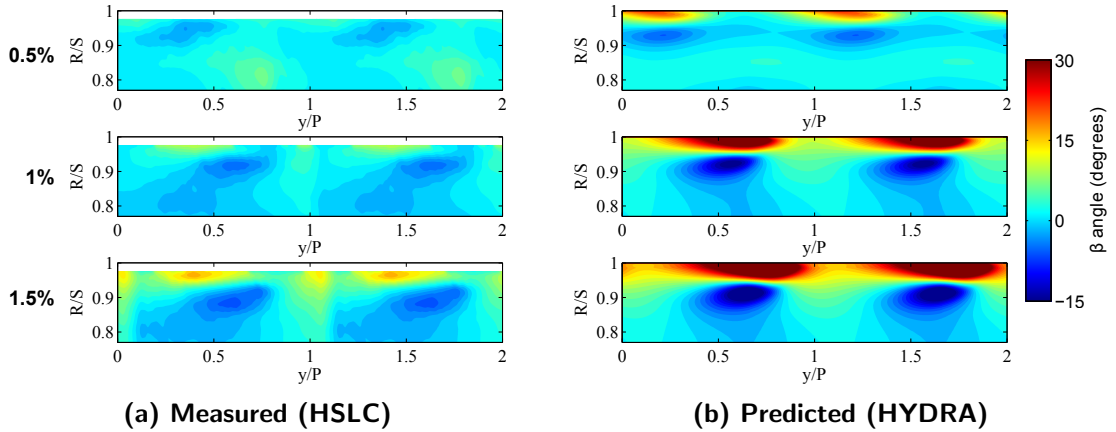


Figure 5.11: Exit yaw flow angle distributions relative to blade exit metal angle

The mass-averaged, pitchwise-averaged yaw angles for the three tip gaps are plotted in Figure 5.12. The majority of the predicted yaw angles are within 1-2 degrees of those measured, with exception to those closer to the casing wall. These differences can be accounted for by the range at which the probe was calibrated over

($-20 \leq \beta \leq 20$). This calibration is reported in O'Dowd (2010) and based on previous work by Povey *et al.* (2008).

The yaw angles show large under-turning of fluid occurring when the tip gap is increased, which correlates well with the growth of tip leakage loss. Interestingly, when the gap is reduced there is a slight increase in yaw angle in comparison to the two larger tip gaps, from regions $0.75 < R/S < 0.9$, this is linked to the passage vortex seen in Figures 5.6 and 5.7.

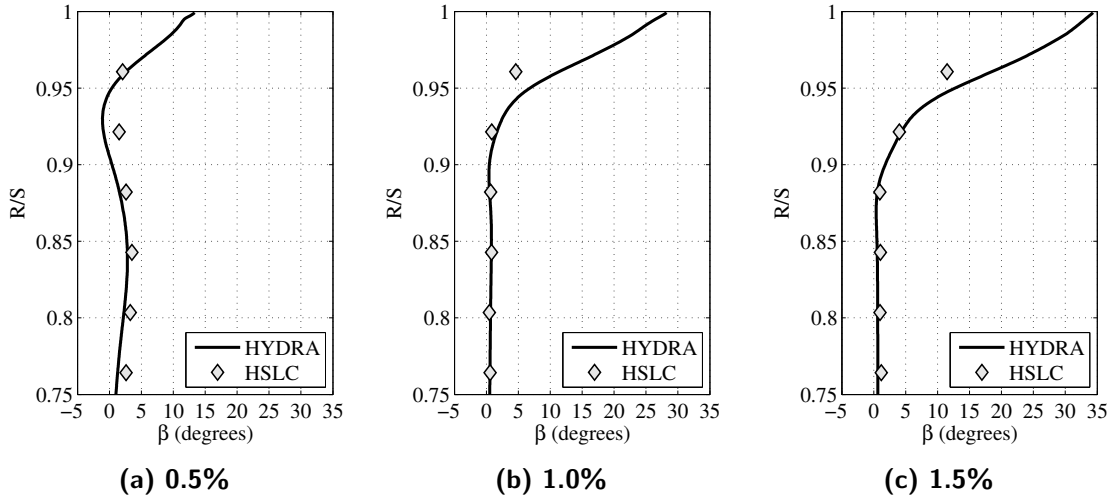


Figure 5.12: Open cavity squealer tip pitchwise-averaged yaw angle relative to blade exit metal angle, for 0.5%, 1% and 1.5% tip gaps at 100% C_x (HSLC and HYDRA)

5.3.2 Aerodynamic Loss: Half an Axial Chord Downstream

The second set of aerodynamic loss measurements are obtained at the exit plane closer to the blade row, half an axial chord downstream. Results presented are limited to total pressure losses and these are shown in Figure 5.13.

The features seen with the loss generation at the previous exit plane are also found at this exit plane, namely, the increase of loss with larger tip gaps and the shifting of vortices away from the suction side blade surface. Noticeably, an additional feature is observed. For all tip gaps, a region of high loss is seen in close proximity to the tip leakage vortex. The loss tends to increase near the casing as the gap increases,

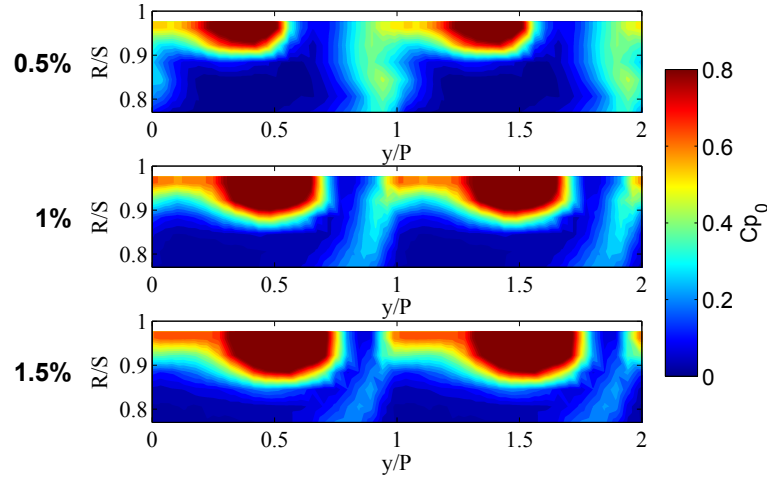


Figure 5.13: Experimentally measured total pressure loss coefficient, C_{p0} , contours of open cavity squealer tip for 0.5%, 1% and 1.5% tip gaps at 50% C_x downstream (HSLC)

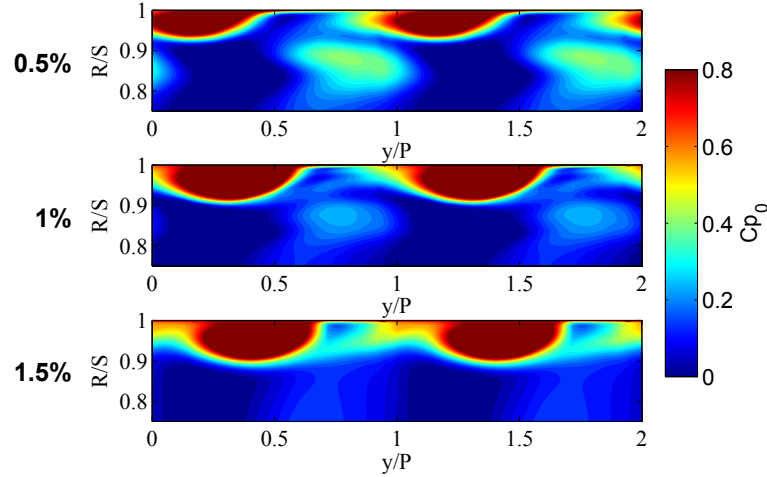


Figure 5.14: Numerically calculated total pressure loss coefficient, C_{p0} , contours of open cavity squealer tip for 0.5%, 1% and 1.5% tip gaps at 50% C_x (HYDRA)

however travelling towards the mid-span, the loss increases as the gap decreases. The latter feature is the occurrence of an upper passage vortex that is clearly identifiable at the smallest tip gap.

Numerical calculations of the total pressure loss at the half axial chord exit plane are presented in Figure 5.14. Again, HYDRA detects the majority of loss features found experimentally. Utilising a key component of CFD, these features can be better understood by applying flow visualisation. Streamlines are released from near the tip leading edge to help identify and classify the over-tip leakage vortices that are created.

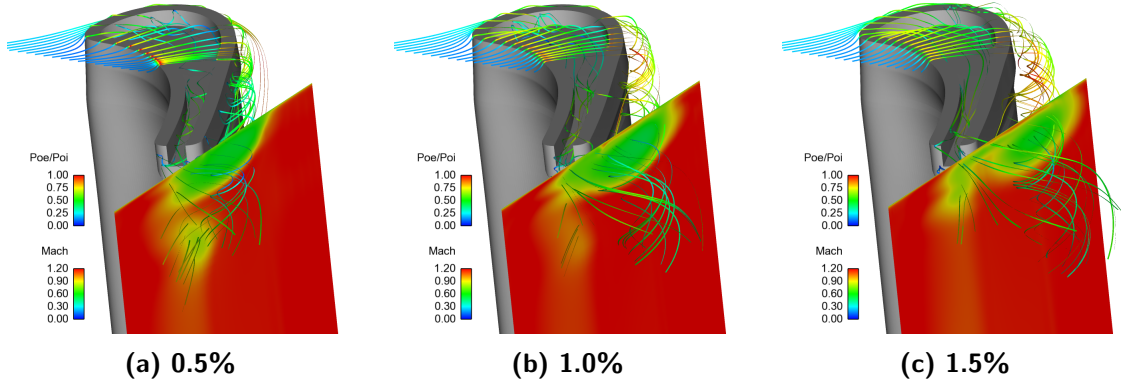


Figure 5.15: Flow visualisation showing over-tip leakage flow streamlines and cut planes with contours of total pressure ratio for tip gaps 0.5%, 1% and 1.5% tip gaps (HYDRA)

Figure 5.15 shows over-tip leakage streamlines that can be clearly seen rolling into the main tip leakage vortex. Taking a cut plane near the trailing edge helps to link the results together and shows how the vortices develop including the generation of passage vortices as the tip gaps are reduced.

Although the trends are well observed, the finer details show that the loss predictions for the 1% and 1.5% tip gaps are generally lower than measured. These discrepancies can be observed using the pitchwise-averaged total pressure coefficient plots shown in Figure 5.16. A factor that may attribute to these discrepancies is due to the complicated flow structure and the interaction of flow features. An over-tip leakage vortex, a passage vortex and loss due to the tip's blunt trailing edge are all present in the plane examined. On the other hand, the traverse resolution may also be an obvious factor. It is thus recommended that any future experiments conducted at exit planes close to the trailing edge, have much finer resolution. In particular, in the spanwise direction as only 6 positions were used for the present study.

Lastly, the overall, mass-averaged, total pressure loss coefficients for the upper 23% span are presented in Figure 5.17. They show that the trends are well captured, but the absolute values are almost uniformly offset. The average percentage error between the measured and the numerical results is 17% (15.5%, 18.2% and 16.1% for the 0.5%, 1% and 1.5% tip gaps, respectively). This result may suggest that the

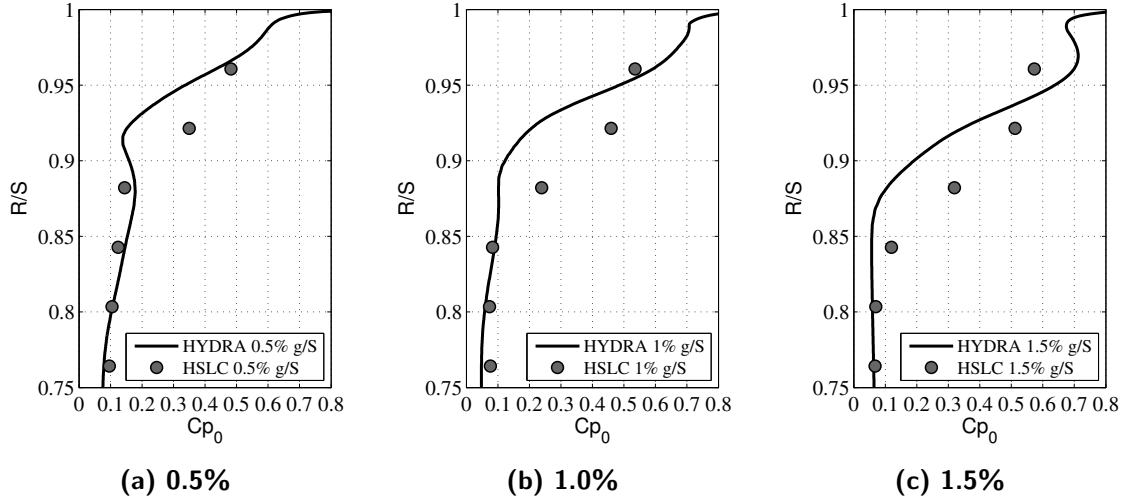


Figure 5.16: Open cavity squealer tip pitchwise-averaged total pressure loss C_{p0} for 0.5%, 1% and 1.5% tip gaps at 50% C_x (HSLC and HYDRA)

positioning of the traversing apparatus and the set distance of the measurement probe was off. This error becomes more important at closer measurement locations, thus greater care should be taken for future experiments and the probe locations should be carefully placed and checked. Alternatively, the calculated loss plane may need to be closer to the blade row in the event of high numerical dissipation.

Although the results tend to show less agreement, the data is valuable as it identifies the complicated flow structure associated with a tip geometry of the present kind. It is clear that the numerical predictions are in better agreement at the furthest exit plane, but the qualitative trends are still well predicted at both planes. The ability of CFD predicting absolute values remains questionable, but as a tool for research and development, it has extremely high value.

Based on the comparisons of experimentally and numerically calculated heat transfer and aerodynamic loss, the results show that the HYDRA solver is well capable of predicting the aero-thermal behaviour of a squealer tip under high-speed conditions. However, the results at the closer exit domain have proven to be challenging. Nevertheless, the correct trends are predicted. The usage of the high-speed linear cascade remains beneficial as a testing vehicle. Having validated the HYDRA code, attention

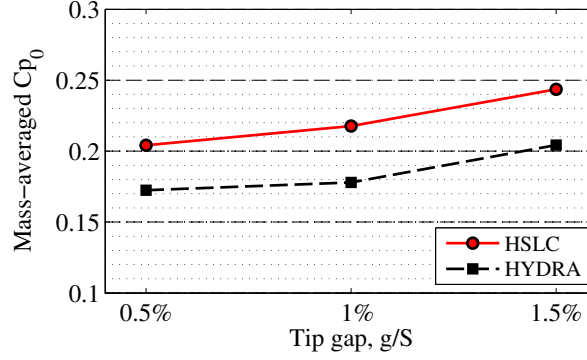


Figure 5.17: Mass-averaged total pressure loss variations for 0.5%, 1% and 1.5% tip gaps at 50% C_x (HSLC and HYDRA)

can be drawn to other important aspects of turbine tip design and performance.

5.4 Trailing Edge Design

The design of the tested squealer design has a distinctive blunt trailing edge to accommodate for an opening in the cavity believed to improve the over-tip flow aerodynamics. A CFD analysis is conducted to examine and understand the influence of the cavity opening and the impact of the gap height on the trailing edge aerodynamics near the tip region.

The near-tip pressure loading (in terms of isentropic Mach number) at 95% of the blade span is shown in Figure 5.18. The pressure side distribution is mostly unaffected by the change in tip gap height and this is largely true on the suction side. The shock region at approximately 70% axial chord is consistent for all gaps and collapse on one-another. However, the influence of the external flow field and the shock vary per tip gap. It is clear that at the smallest gap, the reduction in tip leakage flow means the shock is more penetrating than for the two larger gaps. For the larger tip gaps, the shock is less penetrating and influenced by the increase in tip leakage over the suction side.

Drawing attention to the trailing edge shows the influence of the tip gap on the

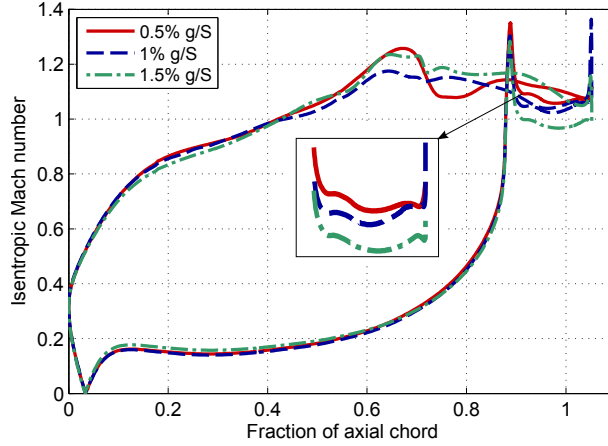


Figure 5.18: Near-tip isentropic Mach number (95% span)

pressure difference at the trailing edge. The isentropic Mach numbers of interest are the lower curves of the loop for each tip gap. Noticeably, the results show a trend with tip gap; as the tip gap increases, the local surface back pressure is increased. This directly influences the outflow from the cavity at the trailing edge exit. This in turn affects the heat transfer and can impact the cooling approach.

Examining the cavity outflow shows how the local pressure distribution has a strong influence on the local flow field. Figure 5.19 shows the mass flux (normal to examined plane) leaving the cavity with velocity arrows and blade surface static pressure contours. At smaller tip gaps, the mass flux is mostly uniform across the passage with a vortex generated as can be seen by the associated velocity vectors. As the tip gap increases, the vortex motion reduces and the fluid is drawn upwards. The change of the local pressure is a driving mechanism for these features and also influences the mass flow rate exiting the cavity as well as over the suction side surface. The suction side over-tip leakage flow increases linearly with tip gap as conventionally understood, but the cavity outflow shows no trend and is mostly affected by the presence of a large tip gap, as shown in Figure 5.20. Furthermore, linking this to the heat transfer (Section 5.2), the increased exit cavity mass flow rate at smaller tip gaps also shows higher trailing edge heat transfer on the cavity surface, thus outlining a compromise between the aerodynamic loss and the heat transfer.

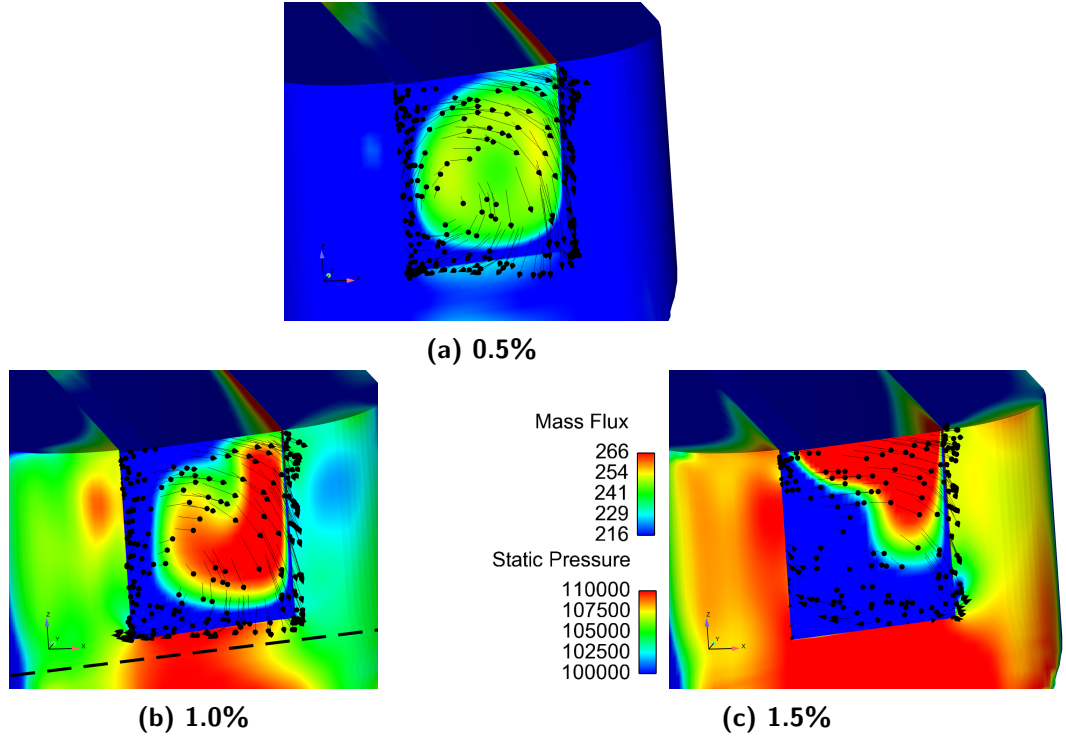


Figure 5.19: Mass flux, ρv_n (kg/m²s), contours on a cut plane through the cavity exit with velocity vector arrows superimposed. Static pressure (Pa) contours also shown on the trailing edge blade surface (HYDRA)

Another feature that affects the loss behind a bluff body or in this instance the blunt trailing edge is the base pressure deficit. This is described by Denton (1993) as the base pressure coefficient, Cp_b , defined in this case using Equation 5.1. Typically, it is used to assess the losses at the trailing edge and is important for design purposes as it directly impacts the vortex shedding. In this case, the base pressure coefficient is used to determine the performance of the trailing edge with respect to the different tip clearances prescribed. Also, it should be noted that in this definition, Equation 5.1 means that a higher Cp_b means lower base pressure. Physically, this would relate to higher trailing edge losses.

$$Cp_b = \frac{P_{static,exit} - P_{static,base}}{\frac{1}{2}\rho U_{exit}^2} \quad (5.1)$$

Given in Figure 5.21 are the near-tip trailing edge base pressures at 95% span

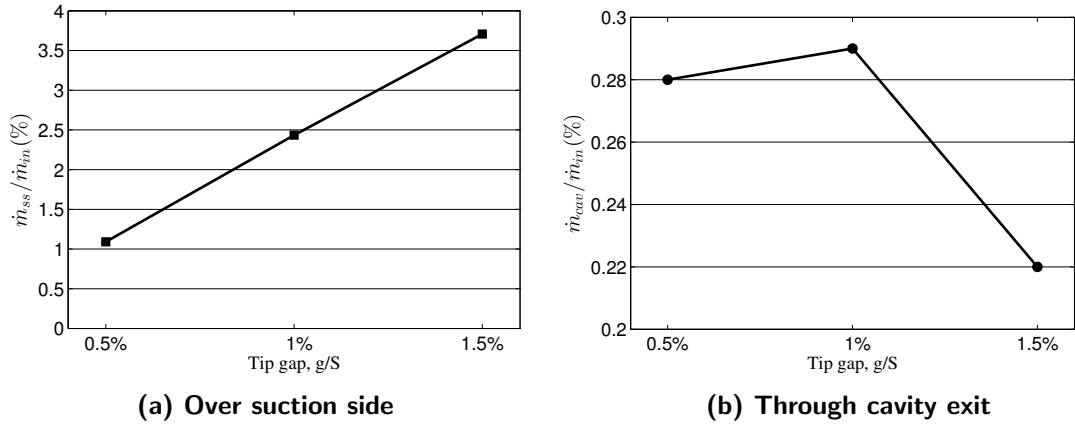


Figure 5.20: Total tip leakage mass flow rates normalised by the inlet mass flow rate for tip gaps 0.5%, 1% and 1.5% (HYDRA)

(as also indicated by the dashed line in Figure 5.19b). The results are interesting given that a higher base pressure coefficient is found for the smallest tip gap and this reduces non-uniformly per tip gap. A higher base pressure coefficient for the present case would suggest higher mixing losses, whilst the largest tip would suggest the reverse. Although the smallest tip gap is able to reduce the tip leakage losses, the base pressure results indicate that a blunt trailing edge may be restrictive in terms of the aerodynamic performance at small tip clearances. Therefore, if there is a requirement of having an open cavity tip, a sharp trailing edge with a cut-back may be an option. Average base pressure coefficients for each tip gap are given in Table 5.2

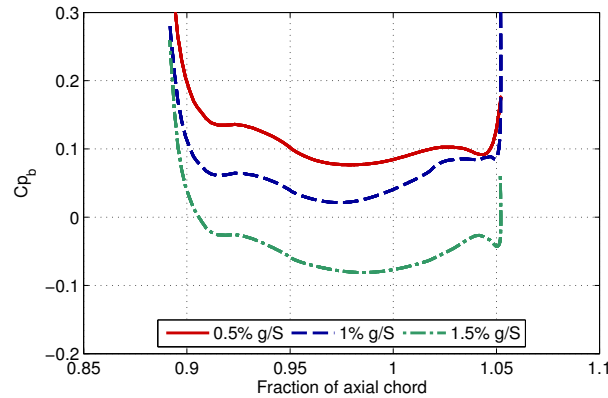


Figure 5.21: Near-tip base pressure coefficient at trailing edge at 95% span (HYDRA)

Tip gap (g/S)	Base Pressure Coefficient, C_{p_b}
0.5%	0.10
1%	0.05
1.5%	-0.06

Table 5.2: Average base pressure coefficient at near-tip trailing edge at 95% span (HYDRA)

5.5 Turbulence Modelling Sensitivity

The choice of turbulence model usage is often up to the user and whilst certain models are better suited to specific scenarios, there tends to be debates over which turbulence model is best suited for a particular application. The following section addresses the sensitivity to the turbulence prediction methods that are implemented and widely used in computational fluid dynamics, and to understand which models are more capable of predicting tip heat transfer and aerodynamics. Currently, there are a lack of turbulence models available in HYDRA (Version 6.1.16), so the Ansys CFX solver was used for the following study.

The dependence of turbulence modelling on the blade tip aero-thermal performance has been investigated at both low-speed and high-speed. Under low-speed conditions, it has been reported that tip heat transfer is sensitive to the turbulence model used and there is also geometric dependence on the model used. Tang *et al.* (2010) showed that for a flat tip, the $k - \omega$ model should be trialled first. This was also the case for Krishnababu *et al.* (2007a), showing that the $k - \omega$ and $k - \omega$ SST produced between 5-10% difference with their experimentally determined heat transfer coefficient results. The $k - \epsilon$ model performed poorly with over-predictions of 50%. For a squealer tip, (Yang *et al.*, 2002) two equation models gave contrasting results, the standard $k - \epsilon$ over-predicted heat transfer and the Renormalisation Group (RNG) $k - \epsilon$ model under-predicted heat transfer. They showed that the expensive Reynolds Stress Model (RSM) gave the best prediction of heat transfer although, low values were significantly under-predicted. For high-speed research, there are nu-

merous sources of published data that recommend the usage of the Spalart-Allmaras model. (Zhang *et al.*, 2011, 2010a; O'Dowd *et al.*, 2011a). These results suggest simpler models, for a flat tip, are better suited to high-speed simulations and was described in greater detail by Wheeler *et al.* (2011). Furthermore for a flat tip, Wheeler & Sandberg (2013) indicated that the freestream turbulence has little dependence on transonic tip flow and that the Spalart-Allmaras model is highly capable for tip heat transfer prediction. There have since then been few studies that assess the sensitivity of the turbulence modelling on the aero-thermal performance of blade tips at high-speed ranging from a simple model to more complex models. In particular, none of the previous results have addressed this for squealers at a transonic condition.

In order to understand the dependence of the turbulence prediction, it is important that the turbulence boundary conditions are kept consistent for each model used. These conditions are based on the experimental setup. Namely, the measured turbulence intensity and length scale of the turbulence generated by the grid installed upstream of the HSLC test section. These are used for each of the models associated turbulence parameters. Specifically, the turbulent kinematic eddy viscosity parameter for Spalart-Allmaras one equation model and turbulent kinetic energy and turbulent or specific dissipation rates for two equation models. For the Reynolds Stress Model, each component is estimated through the solver and based on the turbulent kinetic energy and turbulent dissipation.

5.5.1 Tip Heat Transfer

Firstly, the dependence of turbulence modelling on tip heat transfer is assessed. Shown in Figure 5.22 are contour maps of heat transfer coefficient on the open cavity squealer tip for a chosen group of turbulence models that vary between low and high complexity. Note that due to the large variation observed with the heat transfer coefficient calculations, the contour range has been increased for Figures 5.22c - 5.22f.

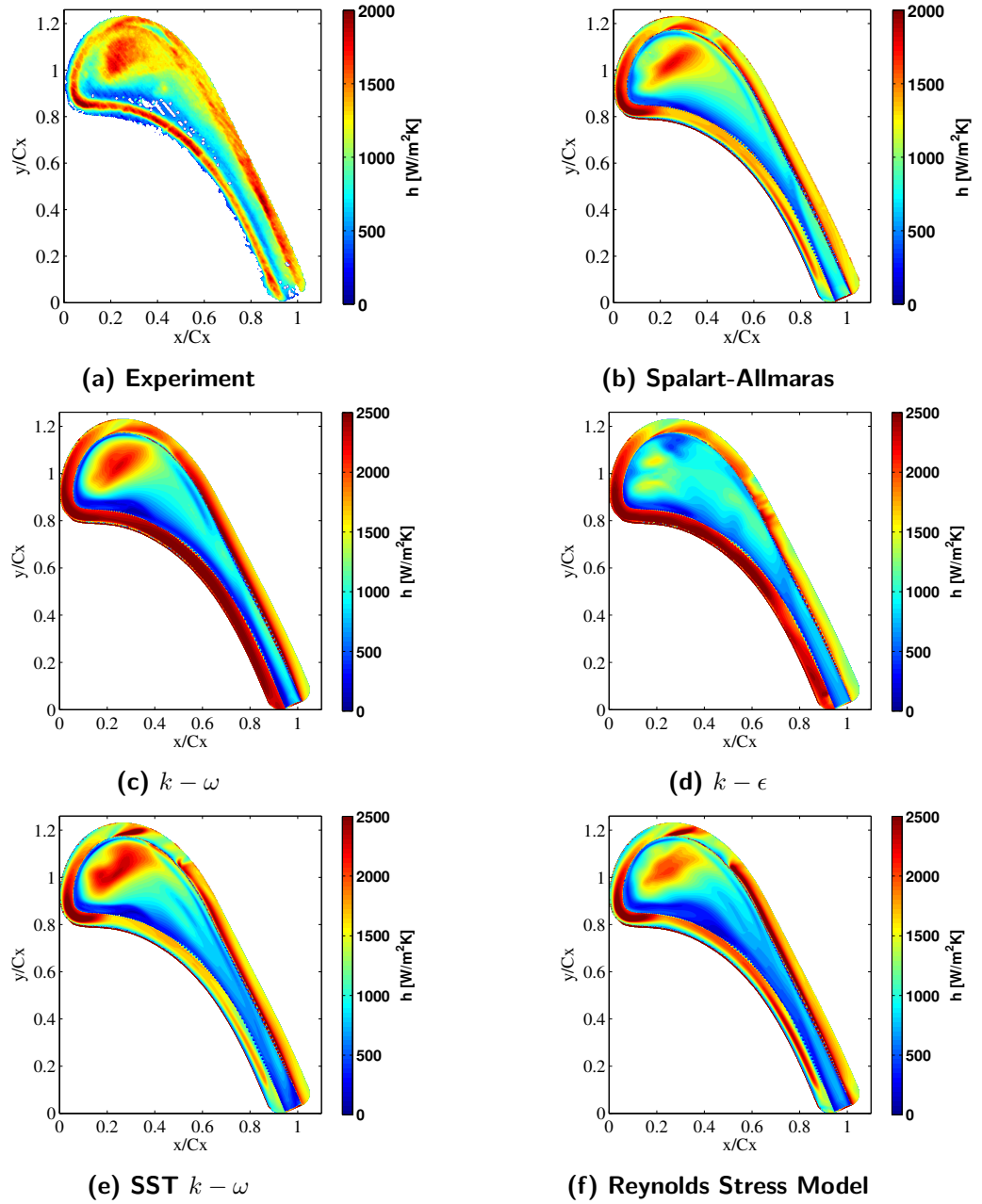


Figure 5.22: Comparison of tip heat transfer coefficients ($\text{W}/\text{m}^2\text{K}$) with 5 different turbulence model predictions with 1% tip gap (Ansys CFX)

The spatially resolved maps reveal a clear dependence on the turbulence model used for heat transfer predictions. The results suggest that more complex turbulence models over-predict the tip heat transfer, especially the hotspot region within the leading edge cavity (10-40% axial chord) and the rim regions. However, the remainder of the cavity region is often better predicted. In contrast to Yang *et al.* (2002), the

Reynolds Stress Model also over-predicts the heat transfer, however, more in-line with Wheeler *et al.* (2011) and Wheeler & Sandberg (2013) showing that two equation models produce a higher heat load in comparison to the SA model as also found here. It is worthwhile noting that apart from the $k - \epsilon$ model, the trends are well captured for all models. These trends are as follows:

- high heat transfer levels on both rims, suction side and pressure side
- high heat transfer levels at the leading edge 'hot spot' cavity region
- relatively large area of low heat transfer within cavity regions away from the leading edge 'hot spot'

In order to help understand the predictions of heat transfer coefficient with the various turbulence models, the eddy viscosity is analysed through the mid-tip gap region and at sliced region through the leading edge cavity region. The former mid-gap eddy viscosity contour plots for the five models assessed are shown in Figure 5.23 and at a cut plane through the leading edge cavity in Figure 5.24. At the mid-tip gap plane, high levels of turbulence are being generated with the RSM, $k - \omega$ and $k - \epsilon$ models. The band of high turbulence over the pressure side rim attribute to the high heat transfer found. These results show the opposite trend for the SA and SST $k - \omega$ models. By linking back to the heat transfer results, these models show comparatively lower heat transfer coefficients.

Figure 5.24 shows the eddy viscosity at a cut plane over the tip leading edge at 15% axial chord. Significantly higher eddy viscosity is generated over the pressure side rim for the RSM, $k - \omega$ and $k - \epsilon$ models in comparison to the other two. Only the SA and SST $k - \omega$ models predict lower turbulence, the others are under-predicting the turbulence dissipation through the contracting and the flow acceleration region over the pressure side.

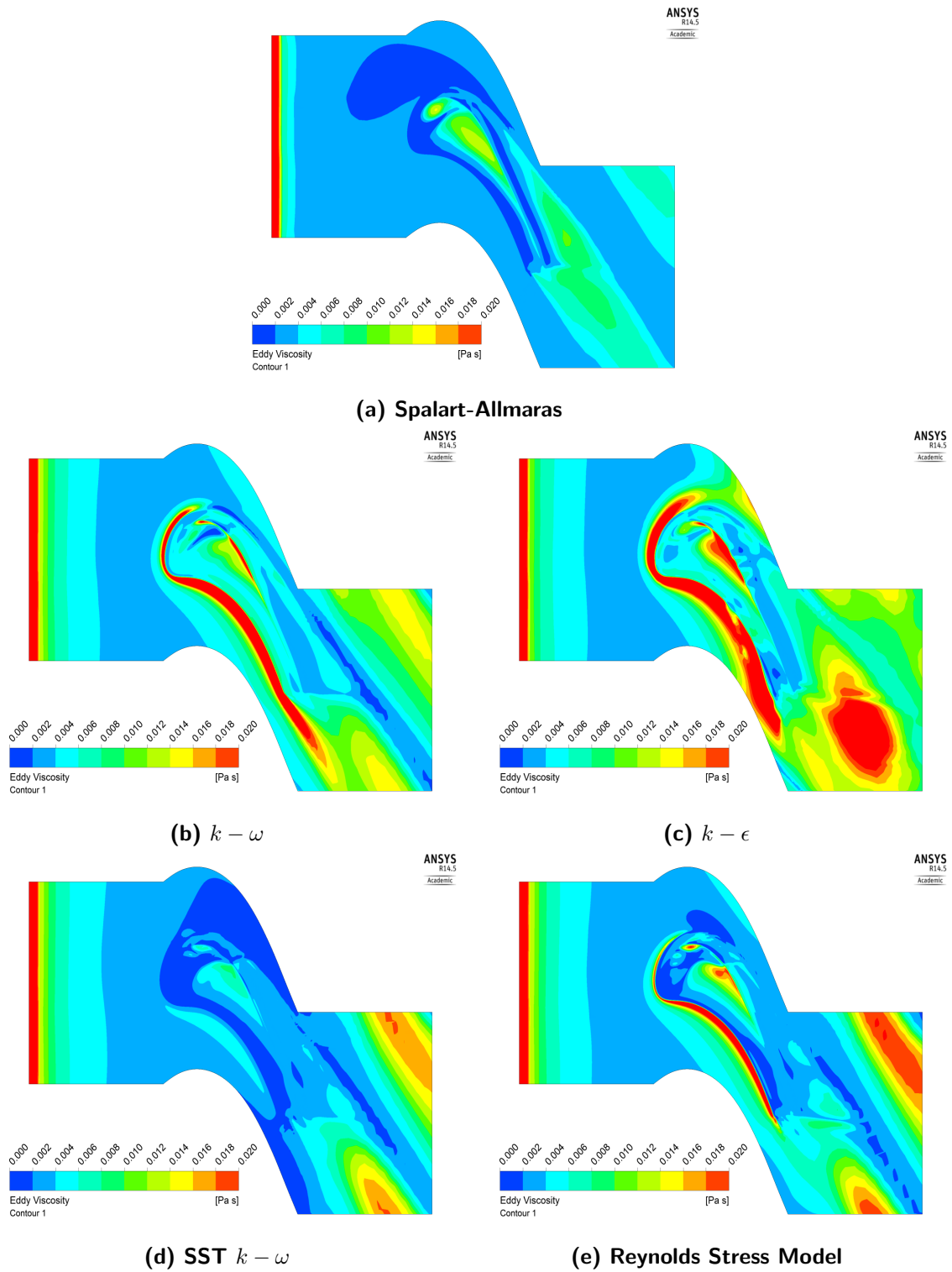


Figure 5.23: Eddy viscosity contours through the mid-tip gap plane over the squealer tip with 1% tip gap (Ansys CFX)

Focusing on the cavity flow region, the SA model produces a thicker region of low eddy viscosity near the cavity floor in comparison to all other models and accounts for the major difference in the predicted heat transfer results. Studying the flow velocities at the same cut plane can also help in understanding the various results found. The Mach number contours with velocity vectors overlaid are presented in Figure 5.25. It is again apparent that the flow separation shows contrasting results with the $k - \omega$ and $k - \epsilon$ models. The two most similar cases are the RSM and SA model, however the details show much larger flow impingement with the RSM result attributing to the very high heat transfer hot spot. This is also true for the SST $k - \omega$ model which has a large stagnation region associated with it, forcing high heat transfer.

The relative performance of the turbulence models can be quantified by the overall heat transfer coefficient percentage errors between the measured and predicted results. These are presented in Figure 5.26. It can be seen that the Spalart-Allmaras model is best suited to tip heat transfer predictions for a high-speed squealer tip and this has previously been shown for both flat tips and winglet tips tested under a high-speed regime as described in Section 2.4. Other models such as $k - \omega$ SST and RSM, also show good results qualitatively. Quantitatively, these two models in particular give overall percentage error differences of 11.4% and 10.8% which is in fact only slightly outside the given range of expected experimental uncertainty at 9.5% in heat transfer coefficient calculated by O'Dowd (2010).

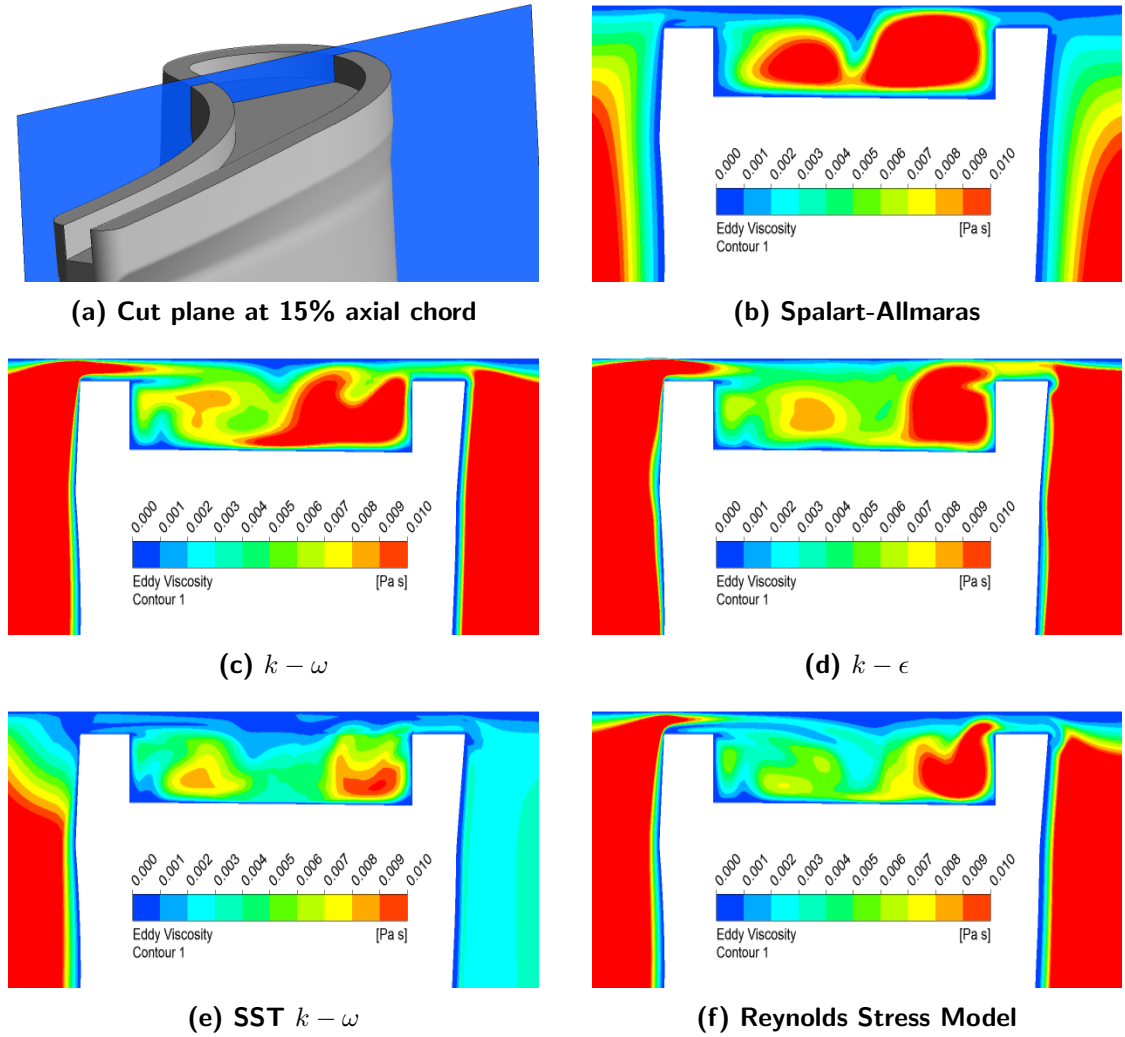


Figure 5.24: Eddy viscosity contours through a cut plane over the leading edge of the squealer tip with 1% tip gap (Ansys CFX)

5.5.2 Aerodynamic Loss

Lastly, the dependence of turbulence modelling on aerodynamic loss is assessed. Total pressure loss is examined at the furthest exit plane, one axial chord downstream, taking into consideration the upper 23% of the blade span area, as previously examined. Contour maps of the loss generated with different turbulence models are shown in Figure 5.27. Analysing the contour maps, most of the turbulence models perform well aerodynamically, apart from the $k - \epsilon$ model which shows poor agreement. Drawing closer attention to the spatially resolved results it is observed that the SST $k - \omega$ and

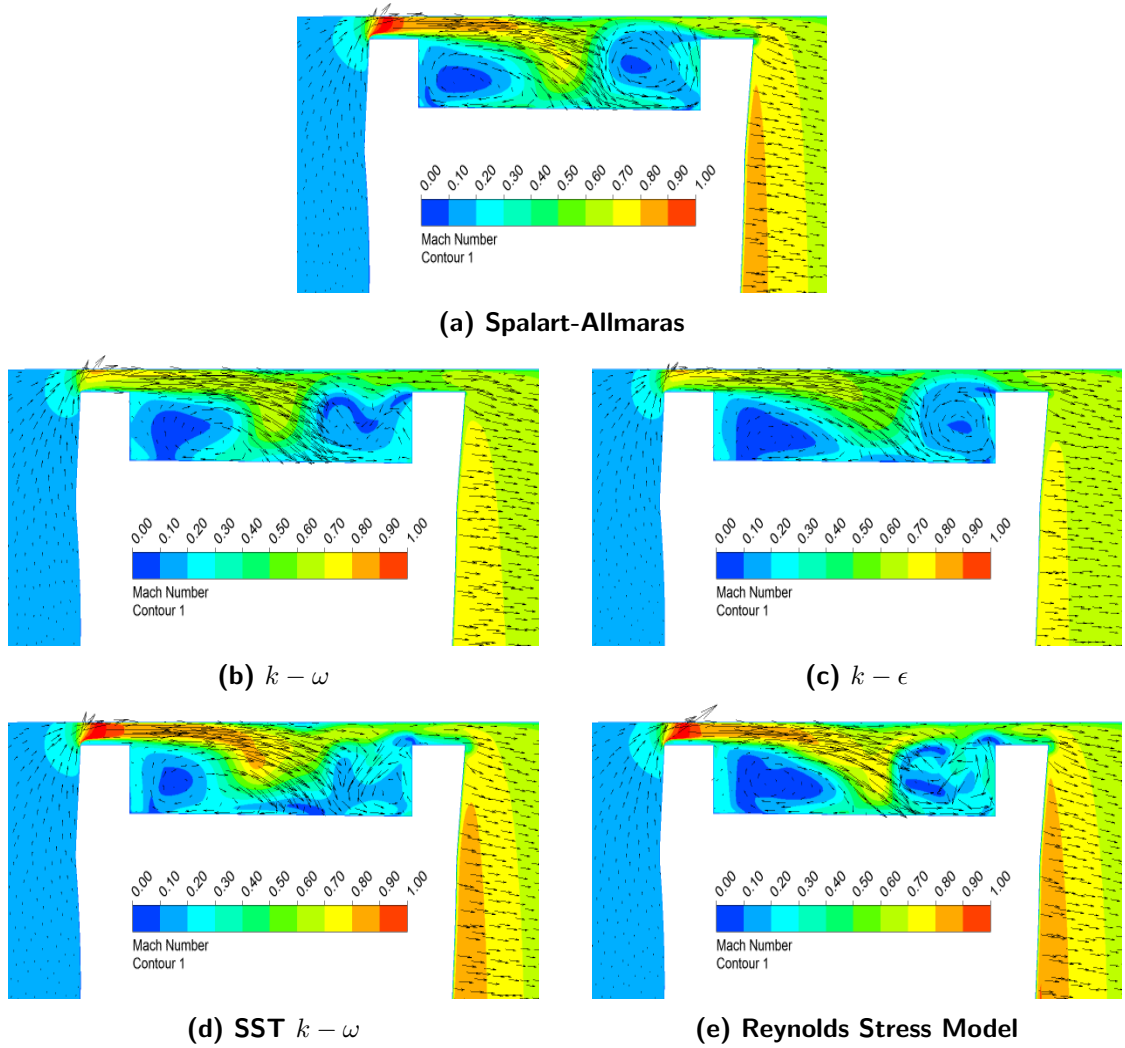


Figure 5.25: Mach number contours with velocity vectors through a cut plane over the leading edge of the squealer tip (Ansys CFX)

RSM slightly over-predict the size of the vortex.

The pitchwise-averaged total pressure losses are plotted in Figure 5.28. The results generally show little variation, however, it can be seen that the $k - \omega$ shows the most promise in detecting the lossy region between $0.77 < R/S < 0.9$. The majority of the other models under-predict this feature. Near the casing at $R/S = 0.96$, the experimental data is likely to be anomalous and none of the models are comparable.

Finally, the overall, mass-averaged total pressure loss errors between the experiment and computational results are given in Figure 5.29. It can be seen that the

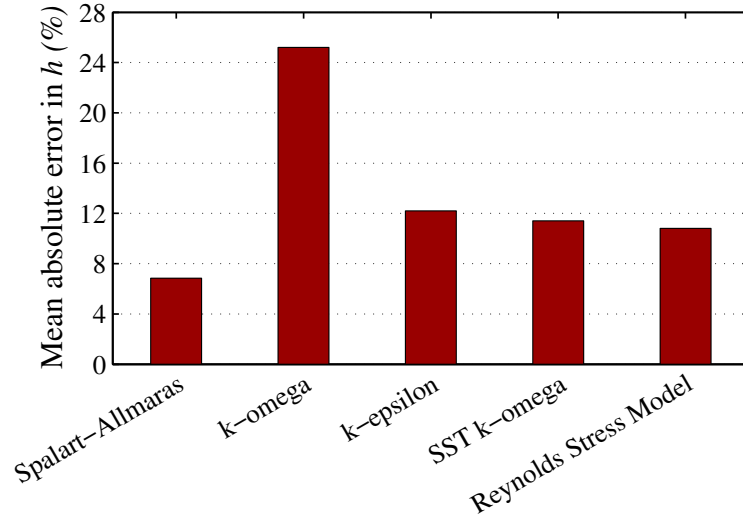


Figure 5.26: Mean absolute error in tip heat transfer coefficient at 1% tip gap (Ansys CFX)

closest agreement is found using the RSM, then the SST $k-\omega$ and then the SA model. It is evident that the $k-\epsilon$ model performs worst. Unlike the heat transfer, the overall losses suggest the more complex turbulence models are better suited to aerodynamic flow prediction. However, based on the contour maps, the pitchwise-averages and the overall losses as a whole, it is thus deemed inconclusive for which turbulence model is best suited for aerodynamic loss production for an open cavity squealer tip under a high-speed regime. Altogether, the combined aero-thermal performance predictions suggest that three of the five turbulence models tested prove capable (SA, SST, RSM), given the experimental induced uncertainties.

In all, it is apparent that turbulence modelling shows a significant effect on the tip aero-thermal simulations. Generally, the basic two-equation models such as $k-\omega$ and $k-\epsilon$ are not well suited, however improvements are shown by Menter's modification to the original $k-\omega$ model (Menter *et al.*, 2003) and more complex turbulence models (RSM). The best agreement thermally, has been found using the Spalart-Allmaras model and aerodynamically, it also ranks highly. Although the model is much simpler than others tested, it is highly capable of predicting flow within a high-speed turbine and has consistently performed well regardless of tip geometry, whether a flat tip,

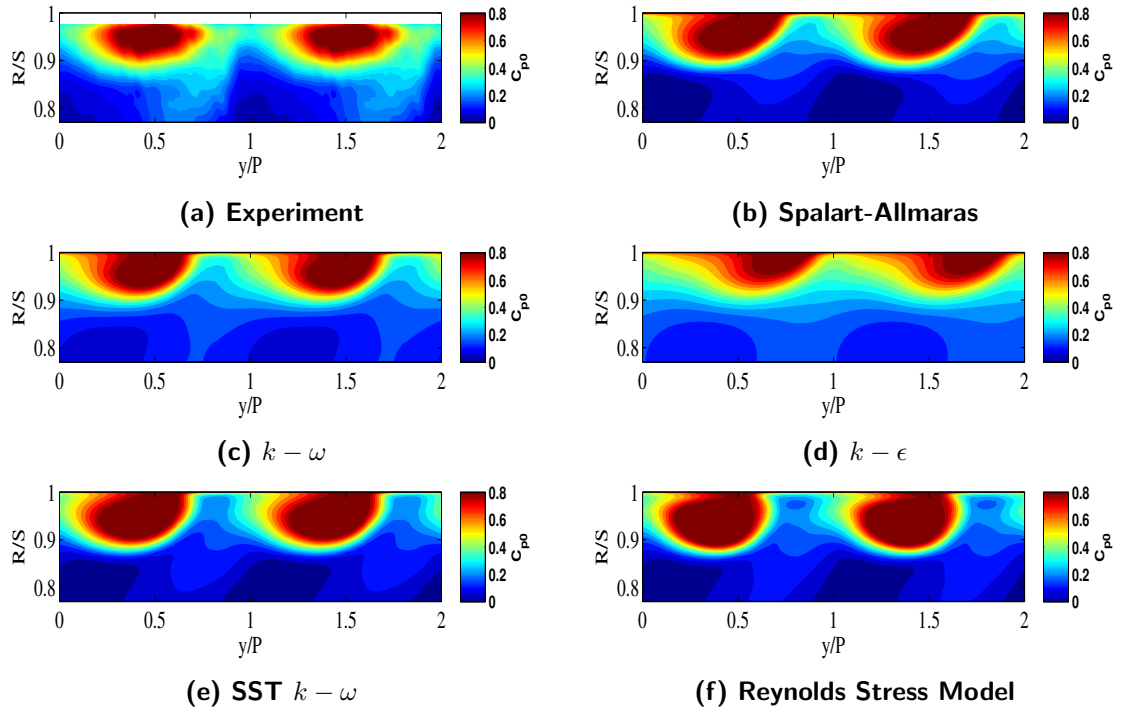


Figure 5.27: Comparison of total pressure loss, C_{p0} , with different turbulence models for 1% tip gap (Ansys CFX)

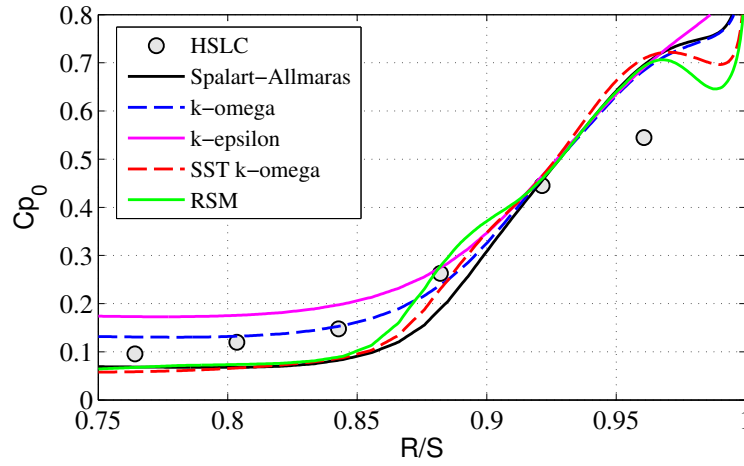


Figure 5.28: Pitchwise-averaged total pressure loss variations with turbulence model for 1% tip gap (Ansys CFX)

winglet tip or squealer tip is being examined. Interestingly, the turbulence modelling predictive capabilities for the aerodynamics and heat transfer are separated. It has been found that the best and worst turbulence models for heat transfer are different to those for the aerodynamic loss prediction.

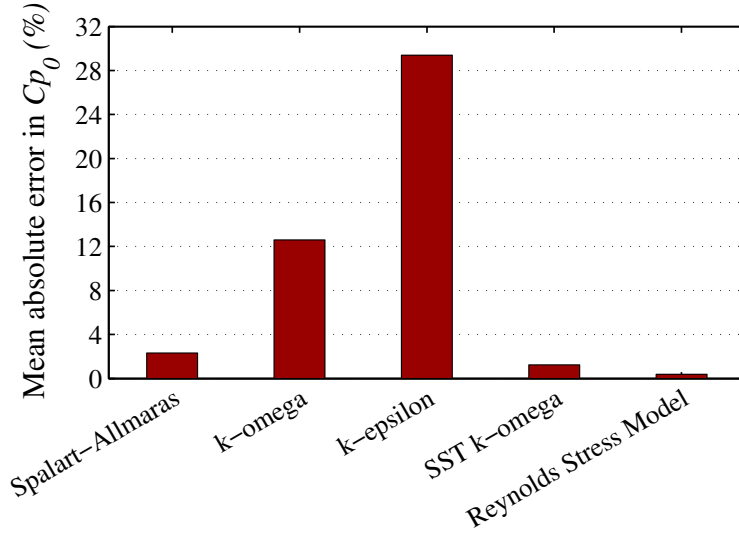


Figure 5.29: Mean absolute error in mass-averaged total pressure loss coefficient for 1% tip gap (Ansys CFX)

5.6 Summary

The thermal and aerodynamic performance of a squealer tip has been experimentally and numerically determined at engine-representative conditions. The results obtained are the first of the kind in respect to a realistic configuration at realistic aerodynamic flow conditions at different tip gaps. Thermal calculations of tip heat transfer coefficient have been determined for three different tip gaps, including detailed maps of the thermal characteristics of a squealer tip. Aerodynamic loss has also been determined at two exit planes downstream of the blade row.

Consolidating the experimental results, numerical CFD results have been obtained. The results show good agreement and captures the majority of flow features found using the measured experimental data. These consistent trends captured for both the tip heat transfer and aerodynamic loss validate the CFD modelling and the predictive capabilities for turbine blade tip aero-thermal performance. The choice of turbulence model has also been assessed.

The following conclusions can be drawn for tip heat transfer:

1. Heat transfer data reveals a large cavity hot spot near the leading edge that is

dependent on tip gap height. Both the cavity hot spot and leading edge area (up to approximately 50-60% axial chord) both show increases in heat transfer coefficient with increased tip gaps.

2. The trend is reversed towards the trailing edge where the heat transfer patterns cross-over showing a reduction in heat transfer with increased tip gaps. At this region, the tip gap flow becomes supersonic, for all tip gaps, indicating the necessity of high-speed flow conditions.
3. The results obtained underline the need for spatially resolved heat transfer measurements and predictions since hot spots and high local gradients over the tip region are well observed.
4. The turbulence model testing shows a heat transfer dependence with the Spalart-Allmaras providing the closest agreement (also in agreement with previous transonic tip research). The results show that high turbulence is generated over the pressure side rim with the Reynolds Stress Model, $k - \omega$, $k - \epsilon$ models, correlating to high rim heat transfer. The Reynolds Stress Model and $k - \omega$ SST models show stronger impingement jets on the frontal cavity region and lead to high heat transfer. The $k - \omega$ and $k - \epsilon$ models perform poorly.

The following conclusions can be drawn for tip aerodynamic loss:

1. Total pressure loss measurements show a trend in aerodynamic loss where the tip leakage loss and vortex core increase with increased tip gaps. The result is consistently observed at the two examined exit planes.
2. As the tip gap increases, the tip leakage vortex is displaced further away from the suction side blade surface. This result is also observed at the two exit planes examined.

3. The OTL mass flow rate over the suction side increases with increased tip gap.
There is no trend for the flow leaving the cavity at the trailing edge, however the distribution of flow correlate to the heat transfer patterns found on the cavity floor.
4. As a result of increased tip gap, the exit yaw angle also increases showing higher flow turning within the tip leakage vortex as expected.
5. The turbulence modelling testing shows less dependence of aerodynamic loss prediction on turbulence models with exceptions of the $k - \omega$ and $k - \epsilon$ models.

Chapter 6

Relative Casing Movement Effects

6.1 Introduction

This chapter covers an extensive CFD analysis regarding the role of casing movement. One of the limitations of the HSLC or any linear cascade environment is the lack of casing motion effects or blade rotation. Although efforts have been made to conduct relative casing movement effects using linear cascades, at high-speed flow conditions this becomes a challenging task. For this reason, the effect of casing movement should not be overlooked, and is therefore evaluated using the validated CFD results. It is of particular interest since the previous data and analyses have not addressed this for a transonic squealer. The basis of this study is to understand the dependence of relative casing movement on tip geometry, specifically, a squealer and a flat tip.

To simulate the effects of blade motion, a tangential casing velocity in a direction from the suction side to the pressure side is imposed and the simulations continue using the HYDRA solver. Firstly, a comparative study is done to compare the stationary cascade results to those with a moving casing. These results are then analysed in depth to show any common trends or differences and how they impact the tip's aero-thermal performance. Secondly, an investigation is carried out to understand the

effect tip geometry has when subject to casing movement. The study also aims to address whether or not the ranking of the aero-thermal performance remains consistent and if this holds for both tip gap variation, as well as geometrical changes.

6.2 Squealer Tip with Relative Casing Movement

Having validated the usage of different CFD solvers capability for heat transfer and aerodynamics loss predictions for a stationary cascade condition, the moving casing effect is now examined. The chosen solver for the study is HYDRA and in the calculations, a relative casing motion is simulated by specifying a tangential velocity of the casing wall boundary in a direction from the suction side to pressure side of the blade. This velocity of the casing wall is calculated using velocity triangles for an axial flow turbine and scaled using supplied engine data provided by Rolls-Royce. The associated flow coefficient, V_x/U , is 0.31. The remainder of the boundary conditions are kept exactly as those prescribed to the stationary setup.

6.2.1 Tip Heat Transfer

As conducted for the stationary setup (Chapter 5), the tip heat transfer coefficient is calculated for the three tip gaps, as studied previously. Figure 6.1 shows heat transfer coefficient maps with relative casing movement imposed. The basic patterns as observed with the stationary wall (Figure 5.2) are also observed here; the cavity region nearest the leading edge, where flow impinges onto the cavity floor can be likened to the stationary case. Furthermore, the pressure side rim also shows relatively little variation when compared to a fixed wall.

As might be expected, there are some particularly interesting changes identified. A region of high heat transfer is formed within the cavity at approximately 40-70% axial chord. This area follows a trend similar to the leading edge cavity region,

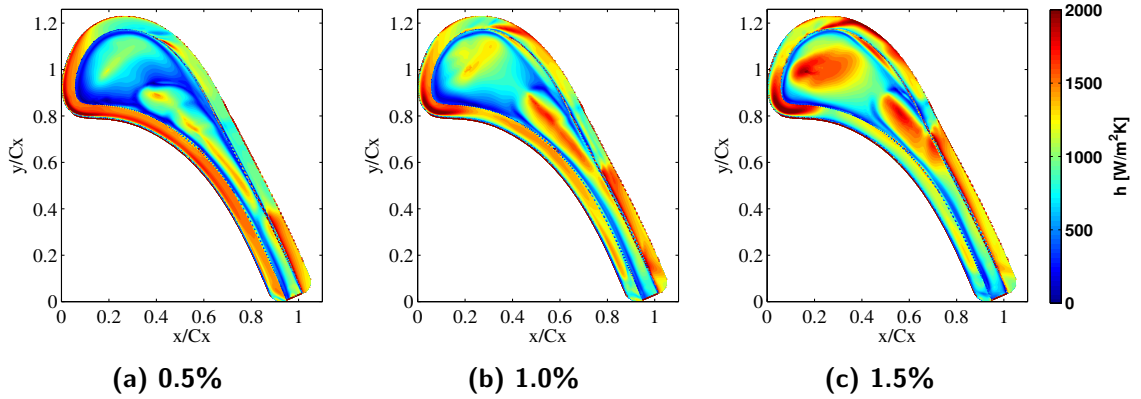


Figure 6.1: Predicted squealer tip surface heat transfer coefficients ($\text{W/m}^2\text{K}$) for tip gaps 0.5%, 1% and 1.5% tip gaps with moving casing condition (HYDRA)

whereby a decrease in tip gap reduces the heat transfer. This is qualitatively in-line with the commonly held view based on low-speed tests that a reduction of leakage flow at a smaller gap leads to a lower heat transfer.

To illustrate this phenomenon more clearly, streamlines are drawn for both the stationary and moving casing setups for a 1% tip gap and are presented in Figures 6.2a and 6.2b. The streamlines provide further insight into the newly formed hot spot located at the mid-cavity region. The casing that is moving in an opposing direction to the fluid, pulls it in the tangential direction. The orientation of the cavity slot at this position combined with casing friction helps develop the in-cavity vortex in two ways: firstly, by promoting reattachment inside the cavity and secondly, by enhancing the vortex core fluid movement along the cavity, which in turn strengthens the vortex itself. As a result, the over-tip fluid in this region is wrapped into a strong and compact vortex inside the cavity rather than continually flowing over the cavity and the suction side rim as in the stationary cascade case. The hot spot region in the mid-cavity between 40-70% axial chord (Figure 5.2) is thus caused by the heat transfer augmentation due to the impinging flow of the mid-cavity vortex.

On the other hand, it should be emphasised that a similar vortex impingement zone with a equivalent heat transfer impact occurs in the main frontal cavity area. Comparing the tip heat transfer between the stationary casing (Figure 5.2) and the

moving casing (Figure 6.1) suggests that the moving casing has a much smaller impact on the main frontal vortex impingement.

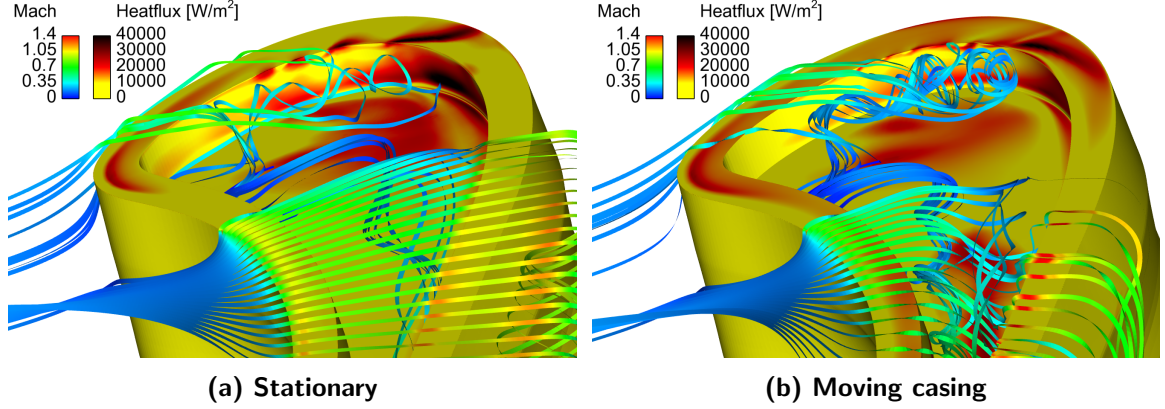


Figure 6.2: Streamlines showing over-tip leakage flow where blade and tip surface contours show heat flux and streamlines coloured by Mach number for 1% tip gap cases (HYDRA)

Circumferentially-averaged heat transfer coefficients for the three moving endwall cases (Figure 6.3) broadly suggest that the local patterns of heat transfer are similar to those shown in the stationary case (Figure 7.13). As the tip gap is reduced, the reduction in h over the subsonic cavity region (10-40% axial chord) is clear. The transonic part (70-90% axial chord) shows a wiggling behaviour that is similar to its stationary counterpart (Figure 5.3b). It is also clear that in both cases the heat transfer trend has contrasting features. The frontal region shows an increase of h with larger tip gaps (10-40% axial chord) and the rear part shows the reverse (90-100% axial chord).

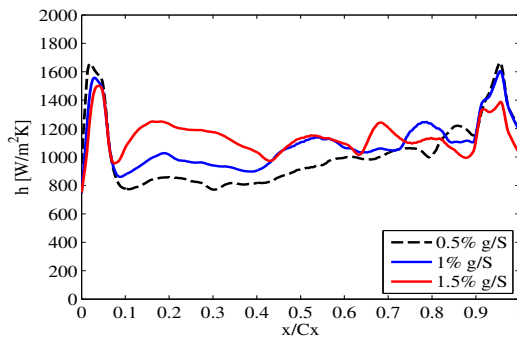


Figure 6.3: Squealer tip circumferentially-averaged heat transfer coefficient (W/m²K) along axial chord with moving casing (HYDRA)

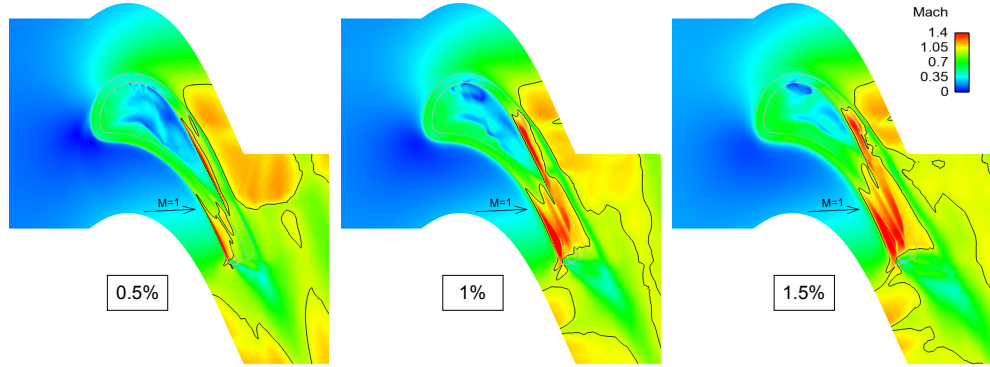


Figure 6.4: Local mid-gap Mach number distributions with moving casing condition (solid line indicates Mach=1) for squealer tip (HYDRA)

The local Mach number distributions calculated for the squealer tip through the mid-tip gap regions, given in Figure 6.4, detail that the flow remains transonic for all tip gaps. The characteristics of the flow over the tip itself are also similar to those of the stationary endwall case (Figure 5.4). The major difference with the applied casing movement is the retarded flow above the frontal cavity region. Here the flow is being developed into the in-cavity vortex as discovered by the tip flow streamlines.

6.2.2 Aerodynamic Loss

The aerodynamic loss expressed in terms of total pressure loss is now considered, in a similar fashion to the stationary analysis. Predictions of the loss are taken at the exit plane, one axial chord downstream of the blade trailing edge for the squealer tip. Contour maps of total pressure loss coefficient, Cp_0 with the moving casing condition for the three tip gaps are shown in Figure 6.5.

Striking differences can be seen by comparing the pressure loss contours for the moving casing case (Figure 6.5) with those for the stationary casing (Figure 5.7). The casing movement is seen to significantly promote the passage vortex, which now has a much clearer loss core identifiable beneath that of the OTL loss core. These are positioned between $0.6 < R/S < 0.85$. Clearly the OTL vortex reduces in size and strength when the tip gap is reduced, as expected, while the passage vortex is much less sensi-

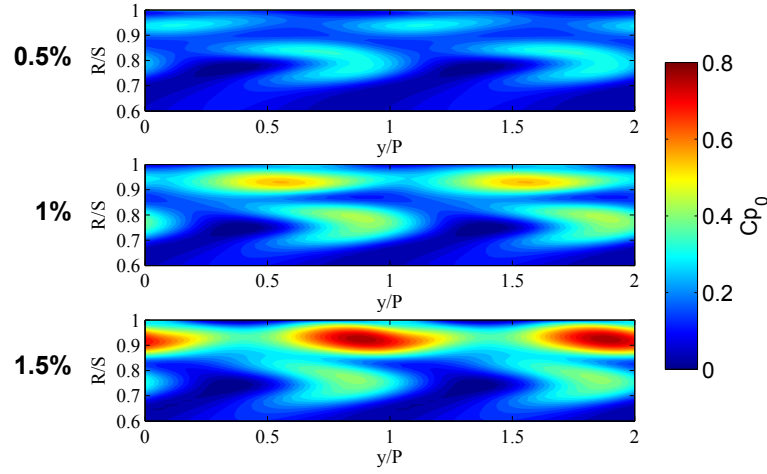


Figure 6.5: Predicted total pressure loss coefficient, Cp_0 , contours for open cavity squealer tip for 0.5%, 1% and 1.5% tip gaps with moving casing at 100% C_x (HYDRA)

tive to the tip gap size. The loss signatures of the passage vortex for the medium and large tip are almost unchanged. The generation of the passage vortex with relative casing movement was also identified and described by Yaras *et al.* (1992b), Tallman & Lakshminarayana (2001a) and Tallman & Lakshminarayana (2001b). They explained that with casing movement, the vortices were dragged towards the suction side and as a consequence the tip gap appeared to be partially blocked. The shifting of the vortices is another feature identified with the inclusion of casing movement, as also found by Palafox *et al.* (2008) where tip leakage and passage vortices were shifted towards the blade suction side and were distorted in shape.

Pitchwise, mass-averaged Cp_0 comparisons for the three tip gaps, stationary and moving casing cases are shown in Figure 6.6. For the stationary cascade configuration, the loss is dominated by the OTL vortex. The loss peaks with this vortex for all tip gaps. On the other hand, for the moving casing condition, there are two clearly identifiable peaks. These peaks are associated with the passage vortex and the OTL vortex respectively. Interestingly, the net contributions of the two interacting vortical flow structures for the moving casing are to some extent, balanced out in comparison with that from the dominant OTL vortex in the stationary case. Although, this trend

becomes less apparent with an increase in tip gap. Further discussions on the overall losses is presented in Section 6.5 which includes a comparison with the performance of a flat tip.

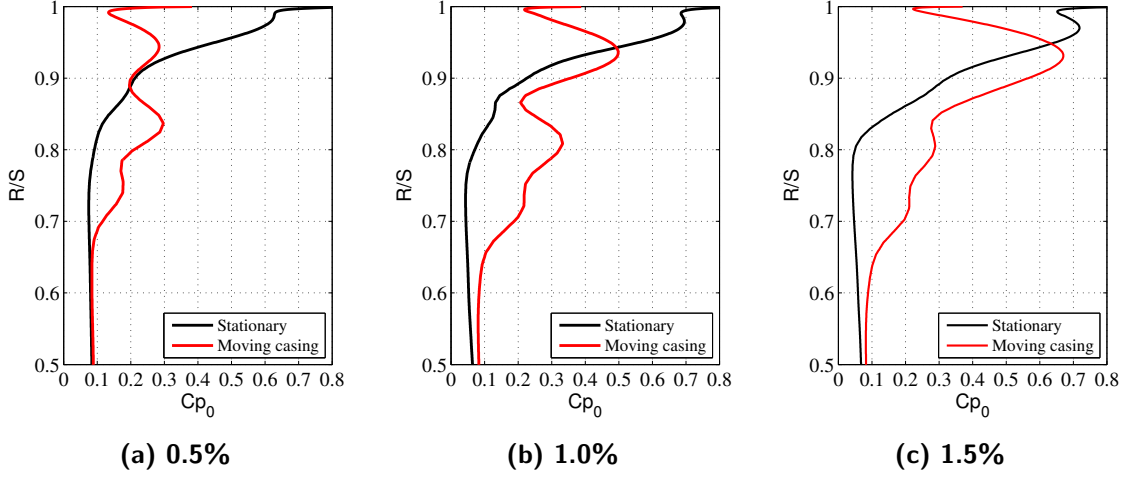


Figure 6.6: Stationary and moving casing pitchwise-averaged total pressure loss coefficients C_{p0} for open cavity squealer tip at tip gaps 0.5%, 1% and 1.5% tip gaps at 100% C_x (HYDRA)

6.3 Over-Tip Leakage Analysis

In Section 5.4, the trailing edge and the cavity gutter of the squealer tip was studied to understand their influences on the near-tip aerodynamic performance. As shown, the relative casing movement has a moderately strong influence on the cavity tip flow and generates additional features that are not present under a stationary setting. However, many of the features still exist with a moving casing and it has been found that mid-gap flow remains supersonic. In order to further understand the formation of the cavity flow with a moving casing, the cavity outflow is examined.

Presented in Figure 6.7 are mass fluxes leaving the gutter, normal to the examined plane for the three tip gaps with a moving casing condition. The blade surface shows contours of the static pressure distribution around the near-tip trailing edge region. For all tip gaps, the moving casing entraps fluid within the cavity and prevents fluid

leaving over the suction side surface. Similarly to the stationary case (Figure 5.19, the gutter outflow is in a vortex-like manner. However in this instance, it occurs for all tip gaps and is greatly intensified by the moving casing. The broken down fluid mass flow rates over the suction side and through the cavity gutter are presented in Figure 6.8. Interestingly, with a moving casing, the mass flow rate leaving the cavity increases with a reduction of the tip gap, previously there was no trend. However, the mass flow rate over the suction side reduces with smaller tip gaps. These results combined indicate that the total OTL mass flow (over the suction side and through the gutter) reduces with tip gap, as expected. This holds for both stationary and moving casing conditions.

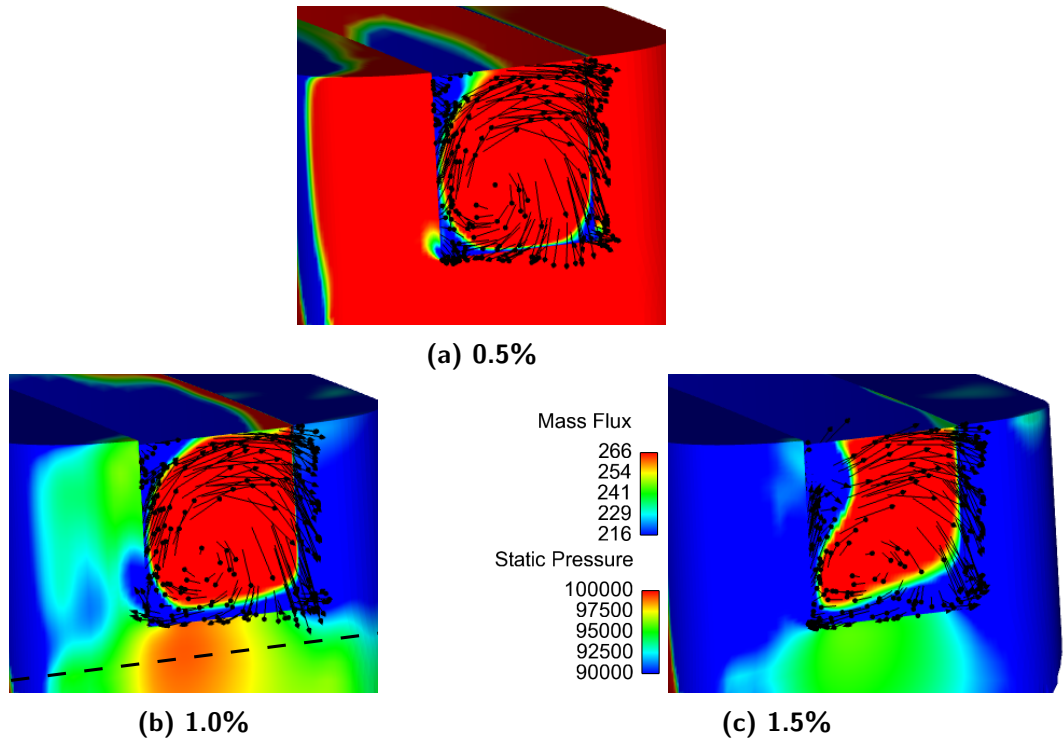


Figure 6.7: Mass flux, ρv_n ($\text{kg}/\text{m}^2\text{s}$), contours on a cut plane through the cavity exit with velocity vector arrows superimposed. Static pressure (Pa) contours also shown on the trailing edge blade surface with a moving casing (HYDRA)

The trailing edge base pressures at 95% span (as also indicated by the dashed line in Figure 6.7b) are presented in Figure 6.9. Unlike the stationary case, the results are reversed and more in-line with expectations. The results with a moving casing show

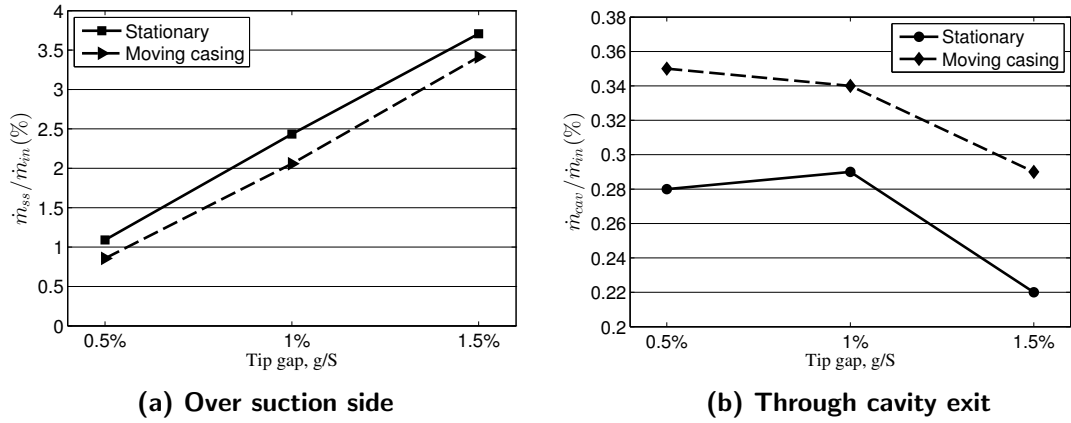


Figure 6.8: Stationary and moving casing total tip leakage mass flow rates normalised by the inlet mass flow rate for tip gaps 0.5%, 1% and 1.5% (HYDRA)

that a lower base pressure coefficient is found when the tip gap reduces. Based on the definition, (Equation 5.1), high base pressure coefficient means higher loss. Although the flow leakage through the gutter is higher with a small tip gap, the trailing edge design shows better performance in comparison to the stationary case when the gap is sufficiently small. Average base pressure coefficients for each tip gap are given in Table 6.1.

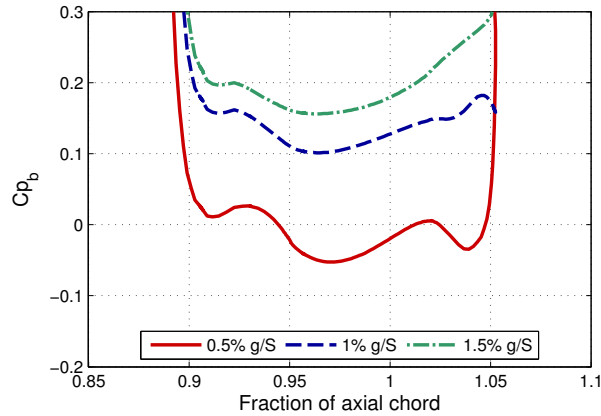


Figure 6.9: Near-tip base pressure coefficient at trailing edge at 95% span with a moving casing (HYDRA)

Tip gap (g/S)	Stationary C_{p_b}	Moving Casing C_{p_b}
0.5%	0.10	-0.01
1%	0.05	0.13
1.5%	-0.06	0.19

Table 6.1: Stationary and moving casing average base pressure coefficient at near-tip trailing edge at 95% span (HYDRA)

6.4 Flat Tip with Relative Casing Movement

In order to identify the sensitivity of tip gap design to relative casing movement, a flat tip is analysed with a stationary and moving casing respectively. Aimed at a consistent comparison, the same blading profile and flow conditions are adopted as those of the squealer case. The flat tip blading profile is a 2D prismatic taken from the 90% span of the original squealer blading, so the only change is the near-tip region. It is recognised that the loading near the tip will have to be tip-configuration dependent. For the present cases, the hub aerodynamic loading has been checked to be mostly the same for both tip configurations (Figure 6.10). In this instance, the hub equates to the hub of the HSLC test section.

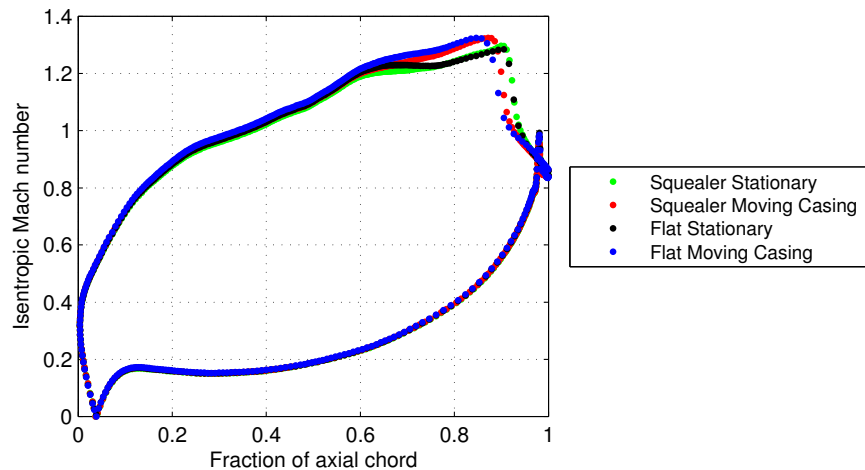


Figure 6.10: Blade loading at hub for stationary and moving casing simulations

6.4.1 Tip Heat Transfer

Heat transfer coefficient contour maps for the flat tip are presented in Figure 6.11, showing the stationary and moving casing results respectively. Figures 6.11a, 6.11c and 6.11e show typical results for a flat tip at a transonic flow condition. A flat tip considered by Zhang *et al.* (2010a) showed a similar leading edge hot stripe that reduces with a decreasing tip gap. In addition, they found that trailing edge becomes more thermally vulnerable at smaller tip gaps. Both of these features are also clearly identified here.

The subsonic behaviour for the near leading edge region produces similar trends to the squealer tip where there is a reduction of heat transfer as tip gap reduces. On the other hand, as the tip gap reduces, there are higher heat loads towards the trailing edge. It becomes evident that the viscous force from the casing motion enhances this effect, suggesting that there is a much larger subsonic region when including casing motion. To illustrate this further, local Mach number plots for the mid-gap region are presented for the three tip gaps in Figure 6.12. These results confirm that the rear, transonic region can be reduced in size by either having a smaller tip gap or by the moving casing condition.

Circumferentially-averaged heat transfer coefficient plots are shown in Figures 6.13a and 6.13b. It is clear that the heat transfer patterns are dependent on the flow regimes over the tip. The subsonic flow part shows a trend of decreasing heat transfer with smaller tip gaps (as seen with the squealer tip also). However, a cross-over region is identified, where these trends are reversed for the transonic part, noticed with both the stationary and moving casing conditions. The cause for the opposite trends of the heat transfer-gap dependence for the subsonic and transonic regions is chiefly linked to the marked reduction of heat transfer in a transonic region, as pointed out by Wheeler *et al.* (2011). This is also the basis for the over-tip shaping to accelerate flow in order to reduce heat transfer by Zhang & He (2011) and also reported by

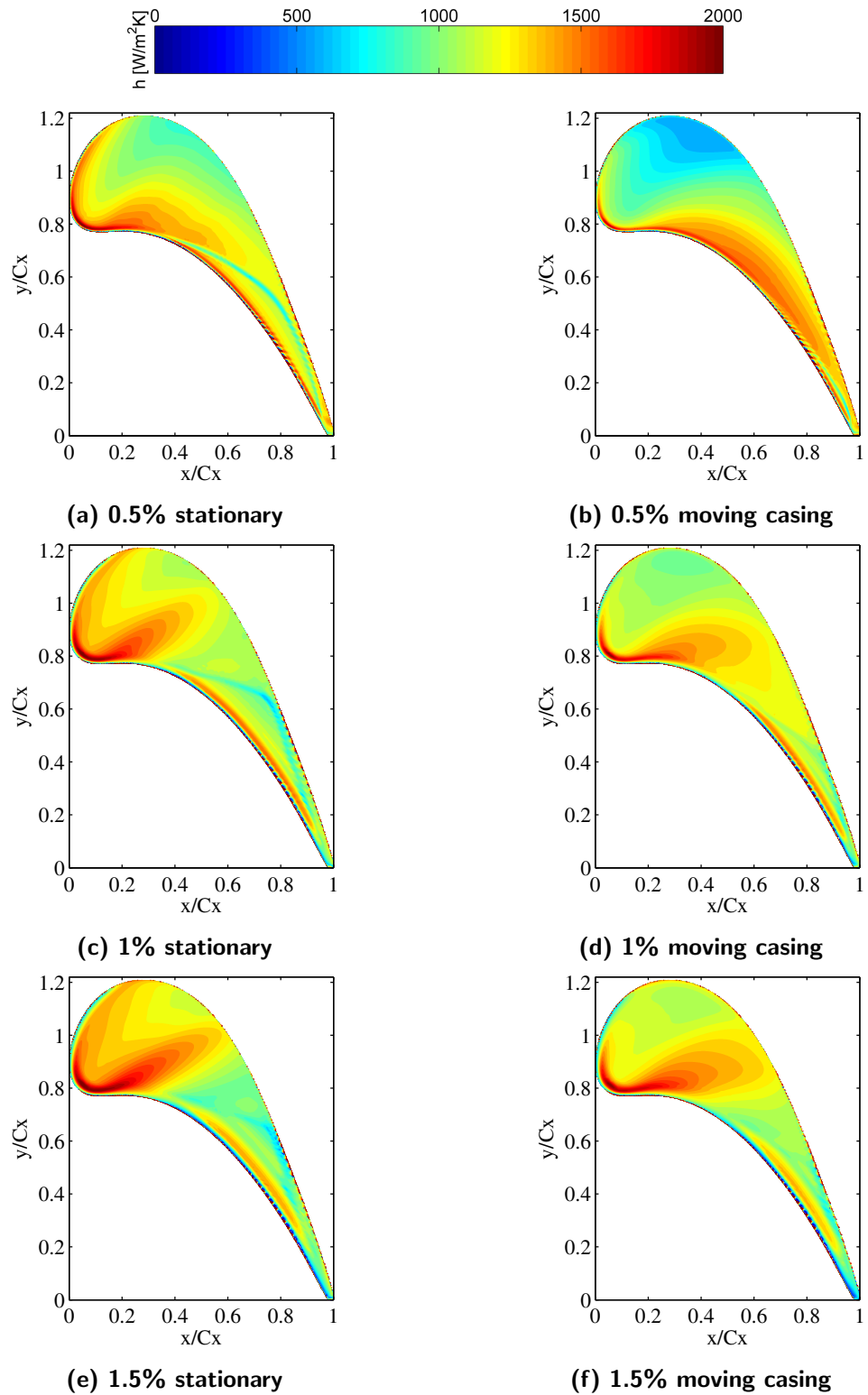


Figure 6.11: Flat tip heat transfer coefficient ($\text{W/m}^2\text{K}$) contours for stationary and moving casing conditions at 0.5%, 1%, and 1.5% g/S tips gaps (HYDRA)

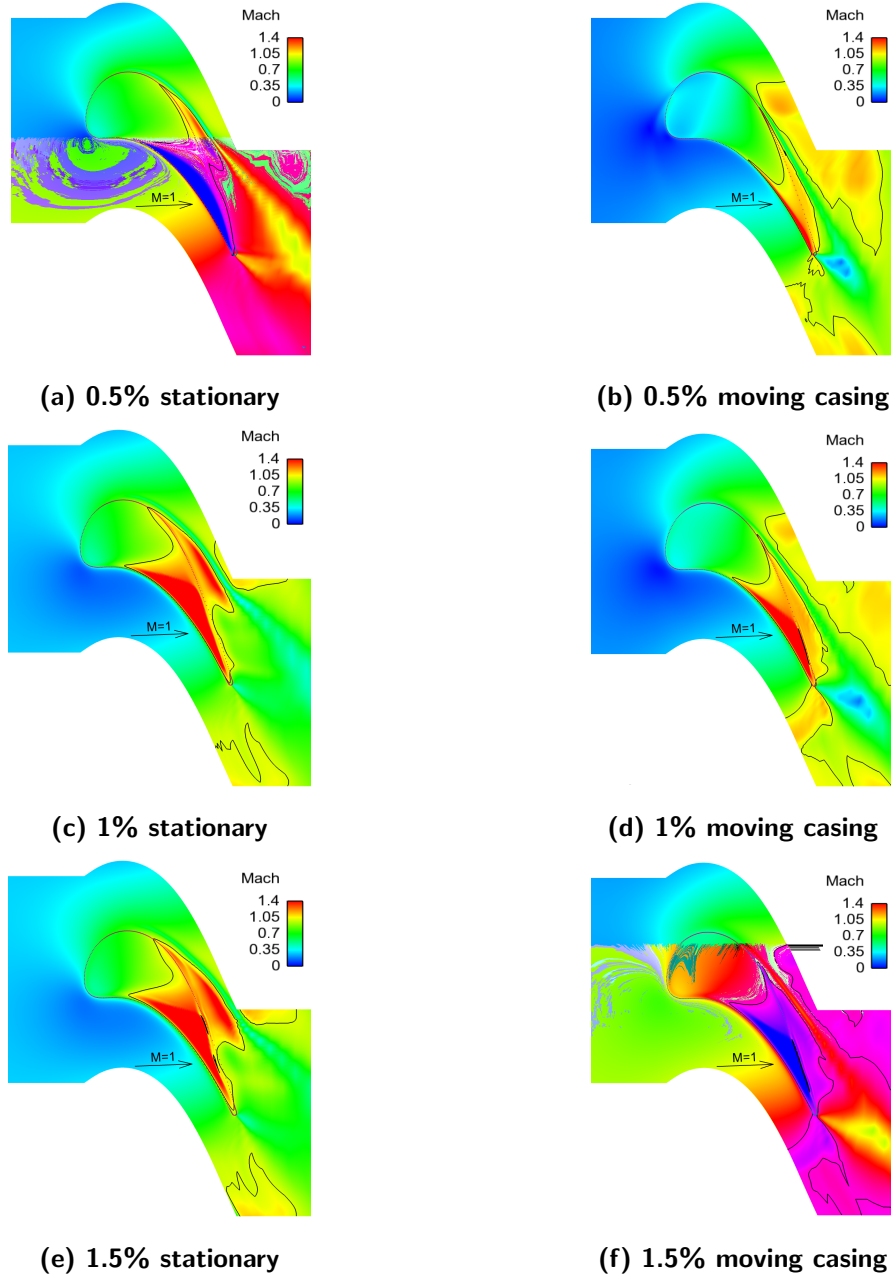


Figure 6.12: Local mid-gap Mach number distributions for stationary and moving casing conditions (solid line indicates Mach=1) for a flat tip (HYDRA)

Shyam *et al.* (2012) and Shyam & Ameri (2011b). A simple possible explanation was hypothesized by He (2012) that the rear transonic region has a lower heat transfer than the subsonic part. Moreover, when the tip gap is decreased, the local transonic flow will be reverted to a subsonic one, at a given point (Figures 6.12a and 6.12b). This change of flow state from transonic to subsonic will have to be associated with

an increase in heat transfer, thus it may be expected that there is a reversed tip gap dependence in the trailing edge region.

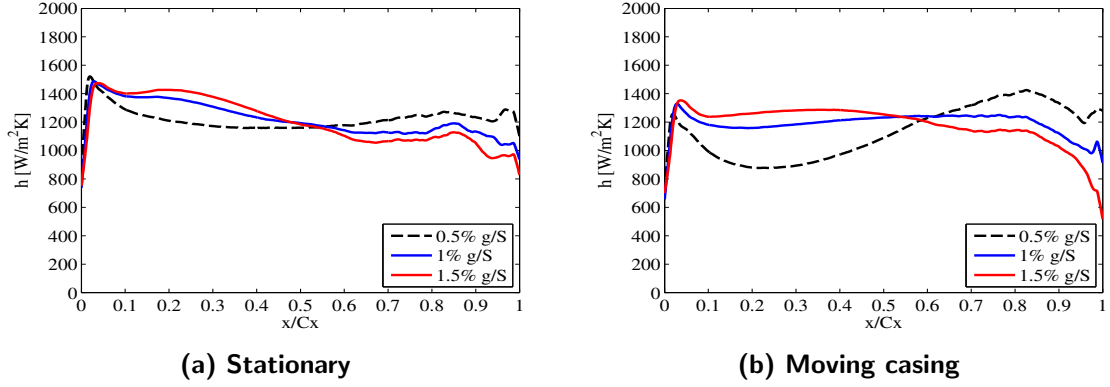


Figure 6.13: Flat tip circumferentially-averaged tip heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$)

Including a moving casing does not qualitatively change the observed opposite trends for the two flow regimes; a reduction of heat transfer when local tip flow is subsonic and an increase when transonic. However, for all tip gaps considered, the over-tip transonic region is reduced in size when the moving casing is employed. This is caused by the casing friction that pulls fluid towards the trailing edge, and hence a slightly more downstream position of the cross-over point from 55% of the axial chord to 60% of the axial chord, as indicated in Figure 6.13b.

Another interesting feature is that the heat transfer is seemingly non-proportionally changed at a small tip gap (0.5% of the span). In other words, there is a much larger difference between the 0.5% tip gap and the 1% tip gap than that between 1% and 1.5% tip gap. This non-proportionality was not observed for the squealer case, with either a moving casing (Figure 5.3a) or a stationary casing (Figure 7.13). This difference in the heat transfer-gap dependence between the two geometry configurations is due to the fact that for a flat tip, the entire over-tip flow is subject to a shear effect from casing that is an equal distance from the blade tip. On the other hand, for a squealer tip, the bulk of flow in the cavity is much further from the casing, and hence less affected by its motion.

In addition to the tip heat transfer, the total pressure loss coefficients, Cp_0 , were also calculated and compared for the moving casing and stationary cases. As possessing very similar characteristics as the squealer tip, contour maps are not included. Nevertheless, pitchwise, mass-averaged Cp_0 are shown in Figure 6.14. Firstly, the results show that for a flat tip when a moving casing is imposed, the over-tip leakage vortex is reduced in strength and the passage vortex is strengthened. Moreover, it has been found that reducing the tip gap consistently reduces the loss. This trend is also consistent for all tip gaps, for both stationary and moving casing conditions. Also shown is a comparison to the squealer tip itself which generally show the flat tip creating higher loss. This comparison is discussed in greater detail in Section 6.5.

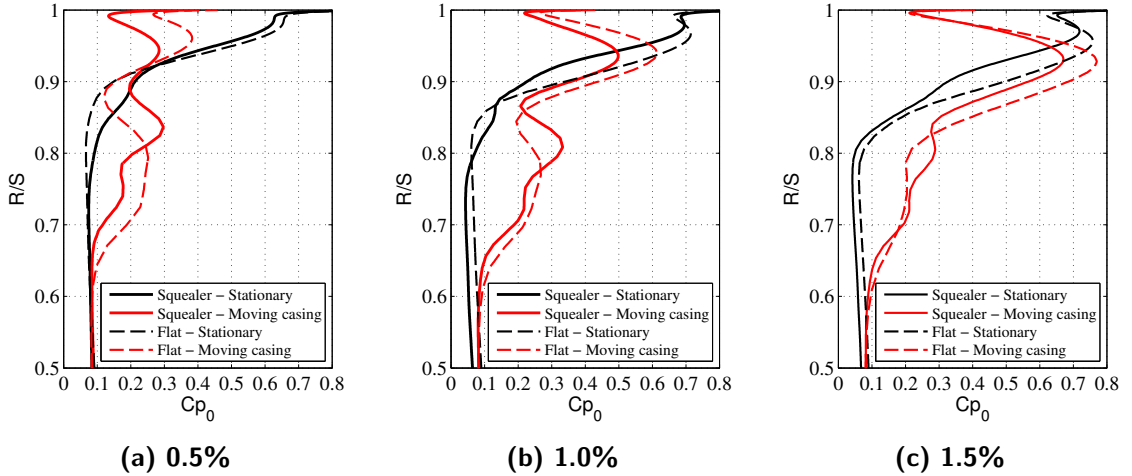


Figure 6.14: Stationary and moving casing pitchwise, mass-averaged total pressure loss coefficients Cp_0 for open cavity squealer and flat tip at tip gaps 0.5%, 1% and 1.5% tip gaps at 100% C_x (HYDRA)

6.5 Overall Aero-thermal Performance

Direct comparisons of the overall aero-thermal performance between the two tip configurations are presented in this section. The focus for heat transfer is the overall tip surface averaged values of heat transfer coefficient and for aerodynamic losses, the mass-averaged entropy rise at the exit plane one axial chord downstream. The

main question of interest is: can the results from a stationary cascade setup rank the performance differences (between different tip gaps and between different tip configurations) consistently?

6.5.1 Overall Heat Transfer Coefficient

The overall area-averaged tip heat transfer coefficient values for the squealer and flat tips are calculated and shown in Figure 6.15.

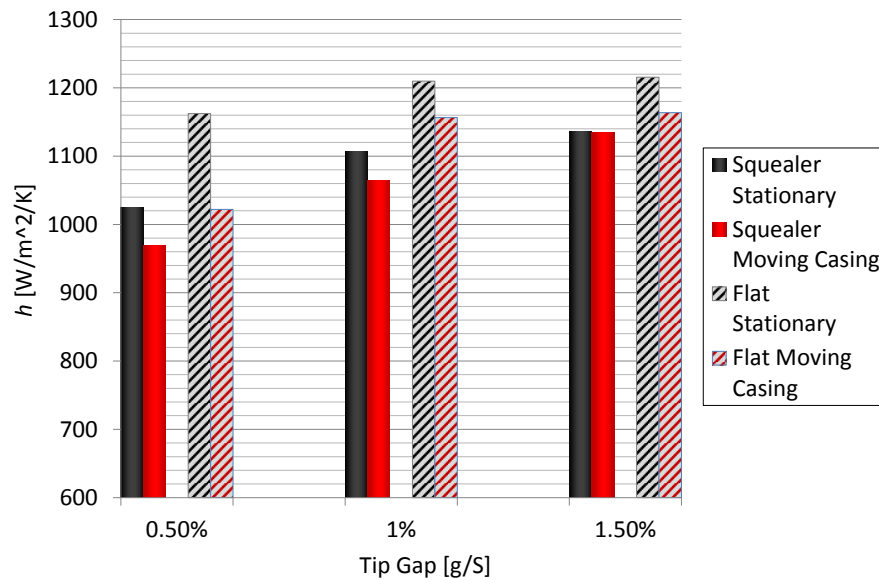


Figure 6.15: Area-averaged tip heat transfer coefficient (W/m²K) for squealer and flat tips, with and without casing movement (HYDRA)

Firstly, for the squealer tip, it is evident that as the tip gap increases, the overall heat transfer increases. This trend is the same for both stationary and moving casing conditions. Coincidentally, for the largest tip gap, the overall values with and without casing movement are almost the same. Although the values are similar, the detailed spatial heat transfer coefficient distributions for the two cases are largely different (Figure 5.2, Figure 6.1).

The flat tip results show similar trends for both stationary and moving casing conditions. Unlike the squealer however, the flat tip casing movement results show

clear reductions in the overall heat transfer coefficient in comparison its stationary case, for all tip gaps. Following the previous discussion of the flat tip, the results at the 0.5% gap are regarded as being dominantly influenced by the moving casing effect as the reduction of heat transfer is substantial.

Comparing geometries, it is apparent that the squealer tip produces lower overall heat transfer coefficients when compared to the flat tip, per tip gap. This observation also holds when the moving casing is applied. This suggests that with respect to heat transfer, the squealer is thermally advantageous over the flat tip, and based on previous research (O'Dowd *et al.*, 2011a), the winglet. It can be seen that the casing movement also lowers the overall heat transfer coefficient. Again this is consistently observed for all tip gaps.

Finally, a cautionary note should be added here when comparing the overall average parameters. It is observed that the local changes for different geometry configurations and casing conditions appear to be much more contrasting than those in terms of the averaged heat transfer coefficient. The averaged heat transfer parameter is adopted here simply as a compact and convenient indicator, which should not be over-relied on without further examination, as presented and discussed earlier. It should be noted that in practice, it is the local burning of the tip which attributes mostly to their failure.

6.5.2 Overall Aerodynamic Loss/Entropy Rise

With the addition of applied work on the fluid by the moving casing, the overall entropy rise (rather than total pressure loss), is used as a measure of flow losses to understand the performance of the two tip configurations. The overall entropy rise is examined at the computational exit plane that is specifically one axial chord downstream. Results are presented in Figure 6.16.

The fundamental gap dependence behaviour of the loss reduction with smaller

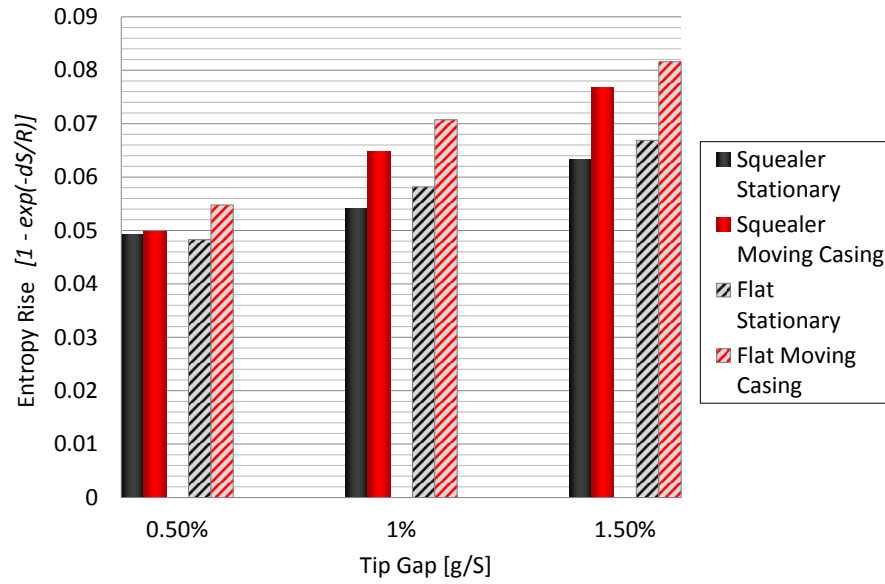


Figure 6.16: Overall entropy rise $[1 - \exp(-dS/R)]$ for squealer and flat tips, with and without casing movement (HYDRA)

tip gaps is clearly shown for both tip configurations consistently with or without the moving casing. The comparison between the two configurations is more complex. At the small tip gap (0.5% g/S), the results with the moving casing still show a performance gain by the squealer over the flat tip, but those with a stationary casing do not. This abnormality is again attributable to the dominant influence of the casing movement at the small gap, which is much larger for the flat tip than for the squealer tip. Table 6.2 shows the percentage change of entropy rise when a moving casing is applied. Focusing exclusively on the two larger tip gaps (1% and 1.5%), it is observed that adding moving casing provides an additional 17-18% rise in terms of the entropy function $[1 - \exp(-dS/R)]$, consistent for both cases of the flat tip and squealer.

Furthermore, Table 6.3 shows the effect of geometry. This shows the percentage change of the entropy function when replacing a flat tip with a squealer tip, both with and without casing movement. For the two larger tip gap cases, the improvement of having a squealer tip is between 6-9% better than having a flat tip.

Tip gap (g/S)	Squealer	Flat
0.5%	1.4%	11.9%
1%	16.5%	17.8%
1.5%	17.6%	18.1%

Table 6.2: Percentage change of entropy rise function when casing movement is applied to stationary case (Reference case is the moving casing)

Tip gap (g/S)	Stationary	Moving Casing
0.5%	1.9%	-9.8%
1%	-7.5%	-9.2%
1.5%	-5.8%	-6.4%

Table 6.3: Improvement of squealer over flat tip (percentage reduction of entropy rise function) (Reference case is the squealer tip)

6.5.3 Further Discussion

As stated in the introduction part, the main motivation of the present work is to identify, if and how, a stationary cascade configuration can be used for rotor blade tip design and analysis. The present results suggest firmly that the moving casing effect depends on the geometry.

A flat tip seems to be more strongly dependent on the moving casing (Figure 6.13). A simple geometrical argument relates the influence on the over-tip flow to the uniformly short distance to the casing wall, when the gap is small. This should, to some extent, explain the non-proportionally large changes at the small tip gap when the moving casing is introduced.

There is another distinctive geometry dependent feature to be noted. In relation to the heat transfer coefficient-gap variation, there is a clear cross-over point in the mid-chord region (50 -60% axial chord) for the flat tip, with or without the moving casing (Figure 6.13). For a squealer tip however, this cross-over point is located much further downstream (Figure 7.13, Figure 6.3). The cross-over point effectively separates the two regions with the opposite trends in heat transfer variation as the tip gap changes. Consequently, for a squealer tip, the overall trend of heat transfer

variation with the tip gaps is clearer and less influenced by other factors including the strong shearing effect due the casing motion as found with a flat tip. The reason for the contrasting behaviour with different tip geometries can be attributed to the differences in flow regime of the bulk over-tip flows between the two tip configurations. For a flat tip, the rear, transonic flow at a large tip gap reverts back to a subsonic state at a very small tip gap. For a squealer tip however, the bulk of the over-tip fluid is trapped inside its cavity and remains subsonic regardless of what the flow state over the pressure side or suction side rims may be in. This bulk subsonic flow within the main squealer cavity dictates the overall gap dependence behaviour of the squealer tip. It therefore results in a more proportional change of heat transfer at a small tip gap as we have observed (Figure 7.13 and Figure 6.3) than its flat tip counterpart (Figure 6.13).

The benefit of the linear cascade is that it is able to provide high resolution spatial tip heat transfer data and a detailed map of the aerodynamic loss. Linking this environment to a rotating one still remains complex and thus, basing rotational results on a linear cascade is challenging. Having obtained moving casing results for two common tip geometries with the addition of a tip gap survey shows the aero-thermal performance has a clear dependence on tip geometry. It has been found that the tip flow under both stationary and moving casing conditions remains transonic and knowledge of this consolidates the requirement for high-speed testing, especially in terms of design work under a rotating frame of reference.

6.6 Summary

To summarise, based on the CFD validation extensive computations have been done to examine the effects of relative casing motion. The results show that the relative casing movement affects the detailed near-tip aero-thermal field considerably.

The following conclusions are made:

1. The present results demonstrate that a significant portion of the over-tip flow remains transonic; with or without the moving casing. The gap dependence of the over-tip heat transfer clearly shows opposite trends for the transonic and subsonic regions, respectively.
2. The moving casing effects are shown to be markedly dependent on both the tip gap size and the tip-geometry configurations. For the three tip gaps considered, the overall heat transfer and aerodynamic losses for all gaps are consistently ranked for both the squealer and the flat tip, with or without the moving casing.
3. At a small tip gap, the over-tip flow for the flat tip is strongly affected by the moving casing due to the uniformly close proximity to the casing wall. In addition, there is a significant difference in the flow state, thought to affect the tip gap dependence of heat transfer coefficient substantially.
4. The bulk fluid in the squealer cavity is always subsonic, regardless of the tip gap size. Hence, the squealer tip exhibits a more proportional variation of heat transfer with tip gap variation than a flat tip.
5. Both squealer and flat tips show that the aero-thermal performance is ranked with tip gap height. Similarly, with a moving casing there is a consistent trend in the aero-thermal performance ranking. Caution should be exercised for a small tip gap as the flat tip shows the local flow behaviour is highly sensitive to the casing motion.

Chapter 7

Cooled Transonic Squealer Tip

7.1 Introduction

This chapter investigates a cooled squealer tip both experimentally and computationally. The same experimental and computational setups are used as done previously, however in both cases, coolant is used. The necessity of film cooling within a real engine stimulates this study. The experimental cooling setup is discussed in Chapter 4 and the computational film cooling procedure via source term modelling is given in Chapter 3. The aero-thermal performance of the cooled squealer tip is examined by the tip film cooling effectiveness and the heat transfer coefficient (thermal), and the kinetic energy loss coefficient and flow exit yaw angle at two exit planes (aerodynamics). The aerodynamic loss is assessed at two exit planes to help provide a better understanding of the loss development and the interaction of the coolant flow with the mainstream flow. The computational analysis uses a simplified cooling model with distributed source terms to identify the capability of predicting the effects of coolant on the tip heat transfer and loss mechanisms.

7.2 High-Speed Linear Cascade Results

7.2.1 Tip Heat Transfer and Cooling Effectiveness

The cooled squealer tip variant is designed and developed by Rolls-Royce plc. The blade used in the HSLC is an adapted version of the uncooled squealer tip, shown in Section 3.5. The actual tested blade differs to the blade, as previously shown in Figure 3.2b, by having the weld beads filed down so that they are level with the rest of the cavity floor. The tested blade is shown in Figure 7.1a. The impact of the weld beads on a real gas turbine blade will have an effect on the aero-thermal performance of the tip geometry. The local cavity flow field will differ and the subsequent heat transfer is likely to be affected also. However, the impact of these manufacturing requirements is not addressed and is recommended for future research that entails a detailed cooling analysis.

The HSLC cooled model is also simplified internally as can be seen in Figures 7.1b and 7.1c. A single coolant plenum is inserted rather than having a multipass cooling configuration. Within the plenum, a slow response thermocouple and a pressure tapping were instrumented to provide measurements of the total pressure and total temperature.

The air supply feeding the coolant was provided through a 650 kPa (100 PSI) air line that was directly connected to an inlet coolant plenum within the HSLC outlet plenum, located below the test section.

To ensure that air supplied to the blade was at a lower temperature than the mainstream flow, it has to be cooled before being supplied to the blade. The application of three vortex tubes with a manifold helps to achieve this. A supply of compressed air is fed to the vortex tubes that spin the airstream towards one end of the tube. Warmer air is forced towards the outer wall of the vortex tube and is expelled. Due to a partial blockage of the exit, cooler air within the centre of the

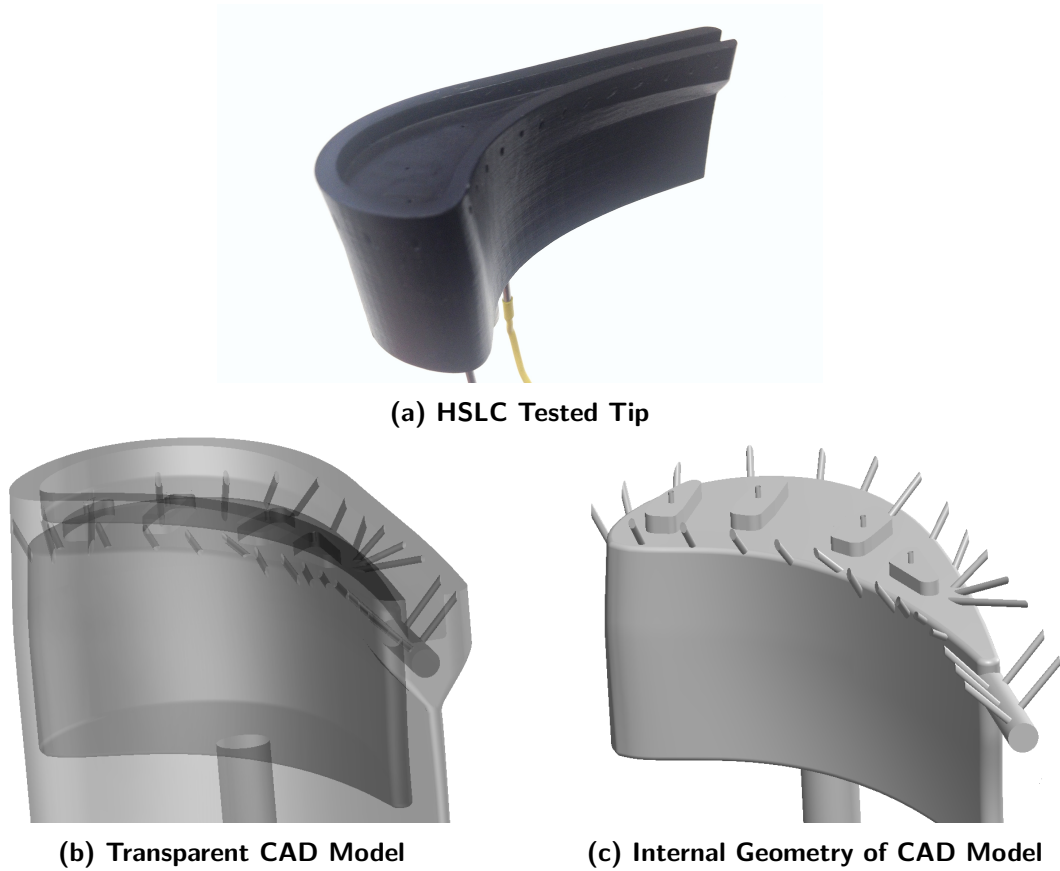


Figure 7.1: Cooled Squealer Tip

tube is rebounded and exits at the other end. The temperature drop of the air can then be used as coolant. The mass flow, in this case, is increased by the employing multiple vortex tubes. Further details of the coolant feed system and setup have been given in Section 4.5.

One of the drawbacks to the cooling setup is temperature ratio achieved. Typically, a gas turbine will operate with a mainstream-to-coolant total temperature ratio of approximately 2, however the HSLC is only capable of reaching a ratio of 1.1. Lastly, similarly to the uncooled blade, a short epoxy resin blade is used and bolted onto a steal base to improve the rigidity of the blade, as well as a small pin near the trailing edge. The neighbouring blades are fitted with the cooled tip variant rather than the uncooled tip to ensure periodicity is met. The two outer blades are fed with uncooled air due to a pressure constraint with the existing system. The HSLC

coolant conditions used for the study are given in Table 7.1

Number of cooling holes	31
Coolant-to-mainstream pressure ratio	1.1
Mainstream-to-coolant temperature ratio	1.1
Coolant-to-mainstream mass flow rate	0.012

Table 7.1: HSLC coolant conditions

7.2.1.1 Cooled Tip Heat Transfer

The cooled squealer tip heat transfer is assessed at the three previously described tip clearances. Figure 7.2 presents heat transfer coefficient contours for the three tip clearances. The cooled squealer tip shows a slight reduction in local heat transfer coefficients for the two larger tips, whilst the smallest tip gap shows lower values on the suction rim and higher values on the pressure side rim. Generally, where cooling holes are present, values are lower but these are also associated with high heat transfer coefficient values closer to the hole locations. It is observed that the heat transfer distributions are largely unchanged, but does have a stronger impact on the pressure side rim. The cavity region (away from the main hotspot) does show higher levels of heat transfer, a trait that has been observed previously with both cooled winglet and winglet-squealer tips. Moreover, as found with the uncooled squealer tip, heat transfer remains to increase when the tip clearance is increased.

Circumferentially-averaged heat transfer coefficients for both the cooled squealer tip at the three tip gaps are shown in Figure 7.3. As described, the results clearly show an increase of heat transfer with increased tip gaps. The characteristics found with the uncooled study are also seen here. However, the cross-over point is less apparent. Direct comparisons to the uncooled results are presented in Figure 7.4. The majority of the squealer tip benefits from tip coolant injection at all tip gaps, however reversed trends are noticed mostly for the two smaller tip gaps. These regions are linked to higher heat transfer coefficients within the cavity. It should be noted

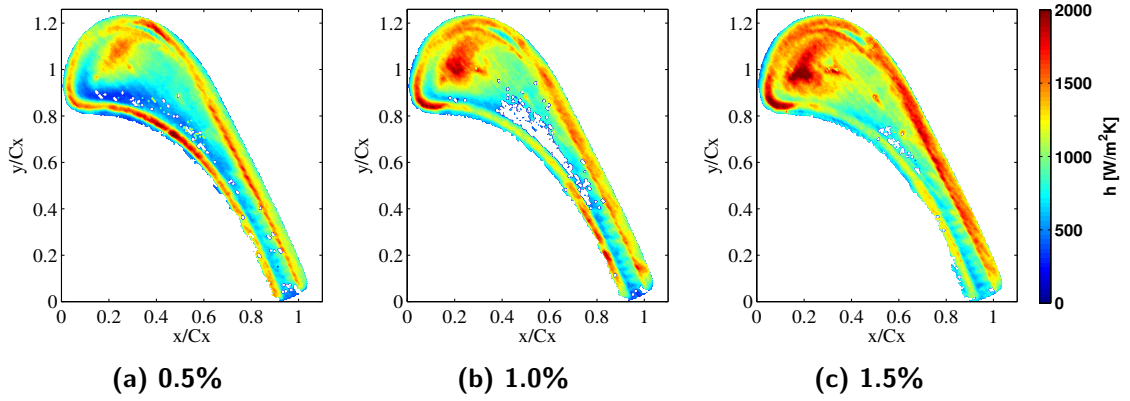


Figure 7.2: Measured cooled squealer tip surface heat transfer coefficients ($\text{W}/\text{m}^2\text{K}$) for tip gaps 0.5%, 1% and 1.5% tip gaps (HSLC)

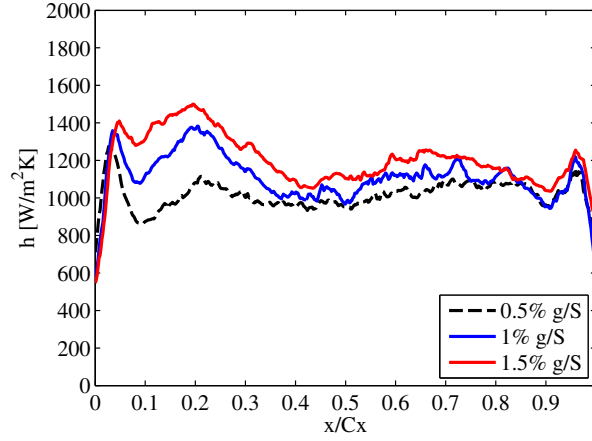


Figure 7.3: Circumferentially-averaged tip heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$) for cooled squealer (HSLC)

that the results are different to those obtained for the winglet (O'Dowd *et al.*, 2013) where a significant rise in heat transfer coefficient was observed due to enhanced thermal mixing and reduction of T_{ad} .

The area-averaged heat transfer coefficient values are computed and shown in Table 7.2. These values are compared to the uncooled tip results and show that the overall coolant impact on tip heat transfer is minimal. It is noticed that at the 0.5% tip gap, the heat transfer coefficient is marginally increased, whilst for the other two gaps, it is marginally decreased. A final point to note is that even with the addition of film cooling, the overall heat transfer remains ranked with tip gap.

The primary goal of the present film cooling design is to protect the highly vul-

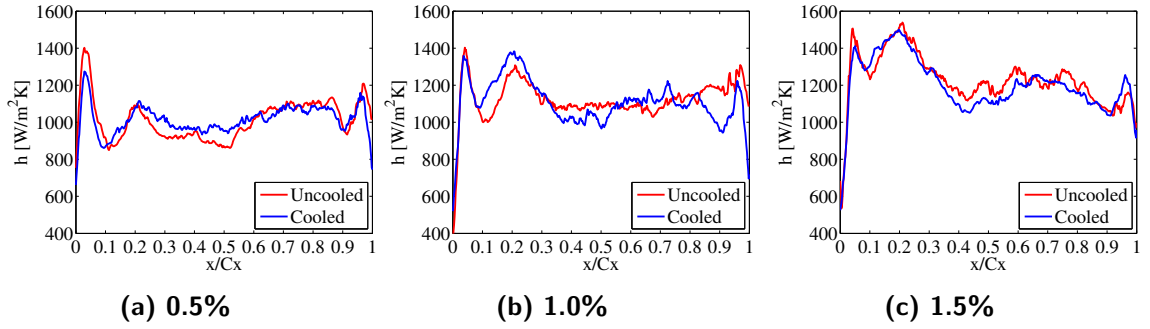


Figure 7.4: Circumferentially-averaged tip heat transfer coefficient comparisons ($\text{W/m}^2\text{K}$) for the uncooled and cooled squealer (HSLC)

nerable squealer rims. The uncooled heat transfer measurements (Figure 5.1) reveal that the rims are susceptible to the oncoming tip leakage flow and care must be taken to reduce the levels of heat transfer the surfaces are exposed to. Drawing attention to the rim surfaces only, the heat transfer coefficient contour maps show larger differences as shown in Figures 7.5 and 7.6. The general observation is that the heat transfer coefficient lowers with film cooling for most cases, on both the pressure side rims and suction side rims. However, the tightest tip gap of 0.5% only shows a reduction towards the leading edge.

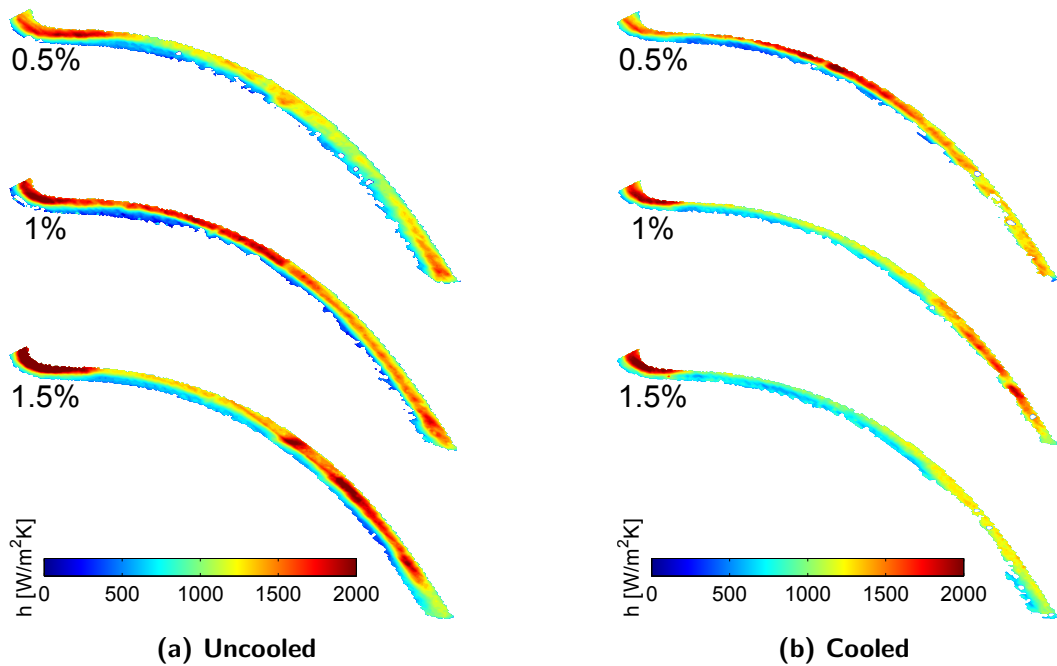


Figure 7.5: Heat transfer coefficient ($\text{W/m}^2\text{K}$) contours on the pressure side rim (HSLC)

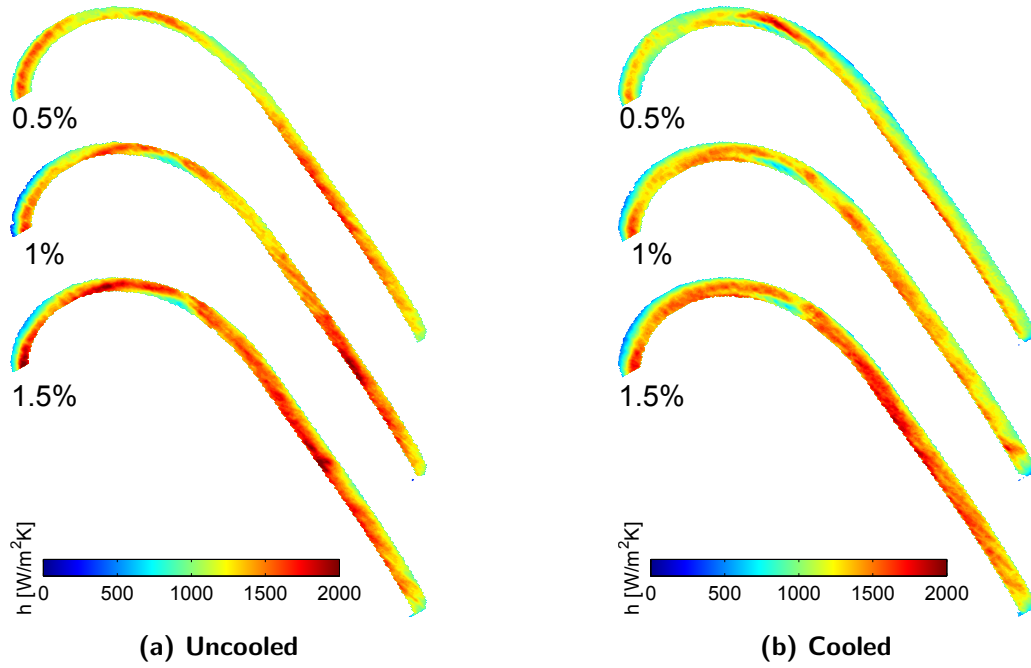


Figure 7.6: Heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$) contours on the suction side rim (HSLC)

The local distributions on the two rims can be seen in Figures 7.7 and 7.8 showing the circumferentially-averaged heat transfer coefficient. The results show significant local reductions in heat transfer with exceptions to the pressure side rim at the two smaller tip clearances. The suction side rim shows lower heat transfer coefficients with film cooling regardless of tip gap height, whereas the pressure side rim has no obvious trend. This is partly caused by a loss of data on the pressure side rim corner where the film cooling effectiveness is high and the difference between the coolant temperature and the initial tip temperature being insufficient to provide a value. Due to this, the averaged results may be misleading, but unlikely to affect the trends based on the spatial maps. The circumferentially-averaged plots describe strong local effects due to film cooling, that are largely tip gap dependent. For example, the aft suction side rim region of the largest tip gap shows only minor differences comparatively to the smaller tip gaps.

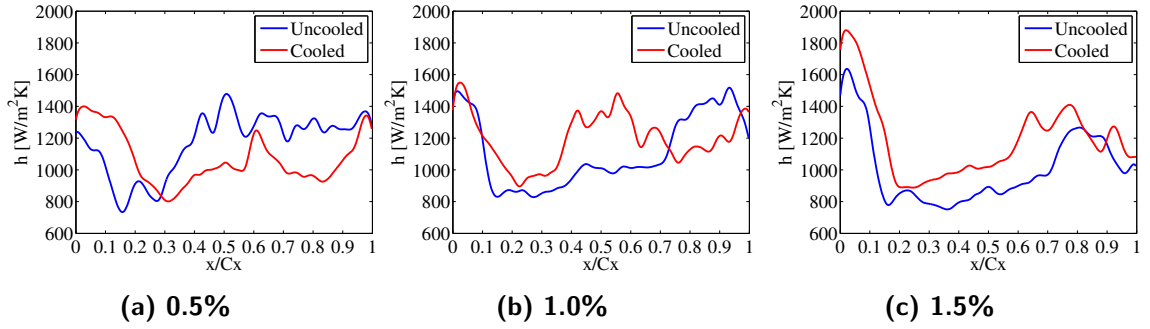


Figure 7.7: Pressure side rim circumferentially-averaged heat transfer coefficient comparisons ($\text{W/m}^2\text{K}$) for the uncooled and cooled squealer (HSLC)

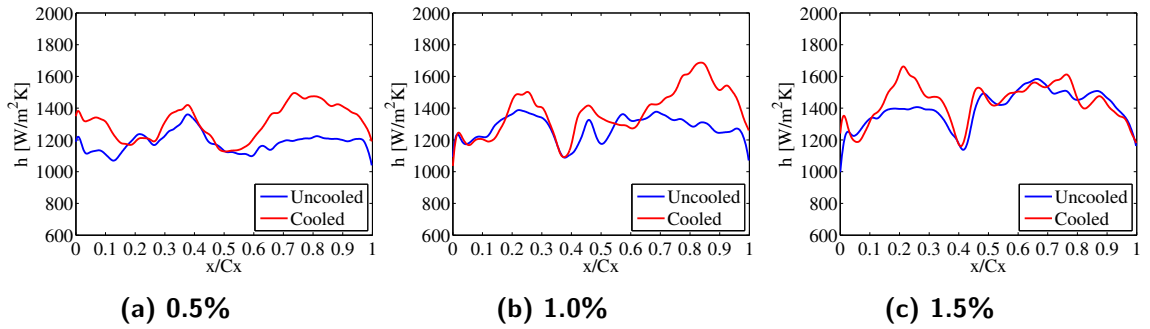


Figure 7.8: Suction side rim circumferentially-averaged heat transfer coefficient comparisons ($\text{W/m}^2\text{K}$) for the uncooled and cooled squealer (HSLC)

Tip gap (g/S)	Uncooled ALL	Cooled ALL	Uncooled PS Rim	Cooled PS Rim	Uncooled SS Rim	Cooled SS Rim
0.5%	1012.6	1014.9	1067.5	1163.3	1298.9	1173.7
1%	1122.0	1109.8	1183.2	1110.9	1355.1	1256.0
1.5%	1228.0	1201.1	1192.9	1007.6	1417.6	1374.2

Table 7.2: Area-averaged tip heat transfer coefficients ($\text{W/m}^2\text{K}$) for the uncooled and cooled squealer tip (HSLC)

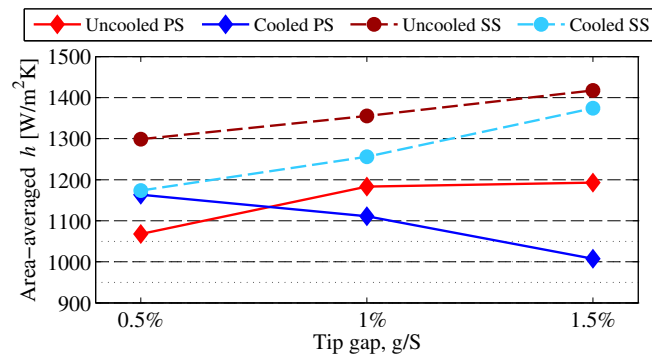


Figure 7.9: Area-averaged heat transfer coefficient ($\text{W/m}^2\text{K}$) for uncooled and cooled squealer rims only (HSLC)

7.2.1.2 Tip Cooling Effectiveness

Figure 7.10 shows film cooling effectiveness distributions for the cooled squealer tip calculated using Equation 4.13. The first most noticeable feature is the extent of the coolant cross flow behaviour with increased tip gaps, downstream of the films. This behaviour might be due to the interaction with the mainstream flow (tip leakage flow). As the gap is increased, the mass flow rate entering the over-tip region is increased. As possessing higher momentum, the tip leakage flow draws the coolant flow with it. In contrast to this, the smallest tip shows less interaction and the coolant appears to be more detached from the surface, however this would in turn mean a lower cooling effectiveness. These findings also indicate the level of mixing occurring within the cavity when coolant is introduced and explain why the trailing edge is more effective at smaller tip clearances. It is likely that at a smaller gap, coolant inside the cavity is driven through the trailing edge gutter exit, as shown in Figure 5.19. Thus, the trailing edge cavity floor might be forced to be cooled better at the smallest gap.

Secondly, the pressure side squealer rim shows high effectiveness and this is solely linked to the array of cooling holes on the pressure side near-tip region. This coolant is carried with the mainstream flow as it enters the tip gap. On the whole, it is observed that the level of cooling effectiveness is low and with tip durability a major concern, tip cooling warrants further, more detailed investigations.

The circumferentially-averaged film cooling effectiveness is shown in Figure 7.11. Generally, the smallest tip gap offers the highest cooling effectiveness, mostly at the rear of the tip (80-100% axial chord). This may be linked to less mixing with the hotter mainstream tip leakage flow, whilst there is a sharp drop for the two larger gaps due to mixing. Apart from this, the effectiveness levels remain largely similar for all cases and there is no clear trend with tip gap variation.

The area-averaged film cooling effectiveness results are given in Table 7.3. The results show a decrease in effectiveness with increased tip gap. This relation suggests

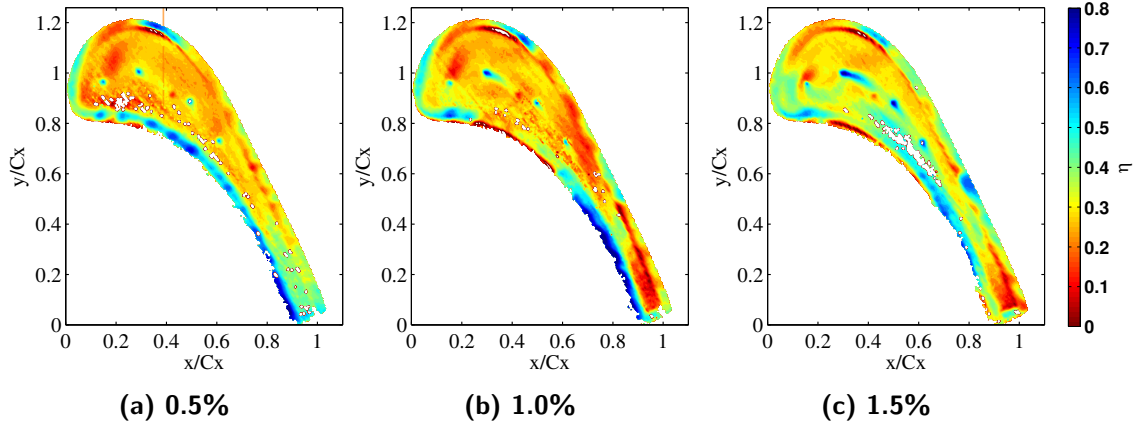


Figure 7.10: Measured cooled squealer tip surface film cooling effectiveness for tip gaps 0.5%, 1% and 1.5% tip gaps (HSLC)

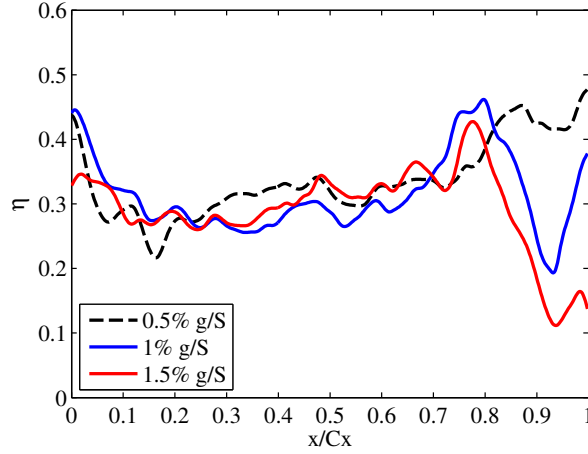


Figure 7.11: Circumferentially-averaged film cooling effectiveness for cooled squealer tip

that at larger tip gaps, the increase in tip leakage flow means more dilution and weakening of the cooling films. However, this trend is less well observed with low-speed squealer tips (Kwak & Han, 2003) where no clear trends were found.

Focusing on the tip rims, it is observed that the highest cooling effectiveness can be found on the pressure side rim, whilst the suction side rim only shows very localised high cooling effectiveness as shown in Figures 7.12a and 7.12b. Generally, as the tip clearance increases, the pressure side rim effectiveness increases. The two smallest tip gaps (0.5% and 1%) have high effectiveness towards the trailing edge, but there are no distinctive features that show a relationship with tip gap height.

Tip gap (g/S)	0.5%	1%	1.5%
η (All)	0.33	0.31	0.30
η (PS Rim)	0.52	0.43	0.39
η (SS Rim)	0.32	0.29	0.34

Table 7.3: Area-averaged effectiveness for squealer tip (HSLC)

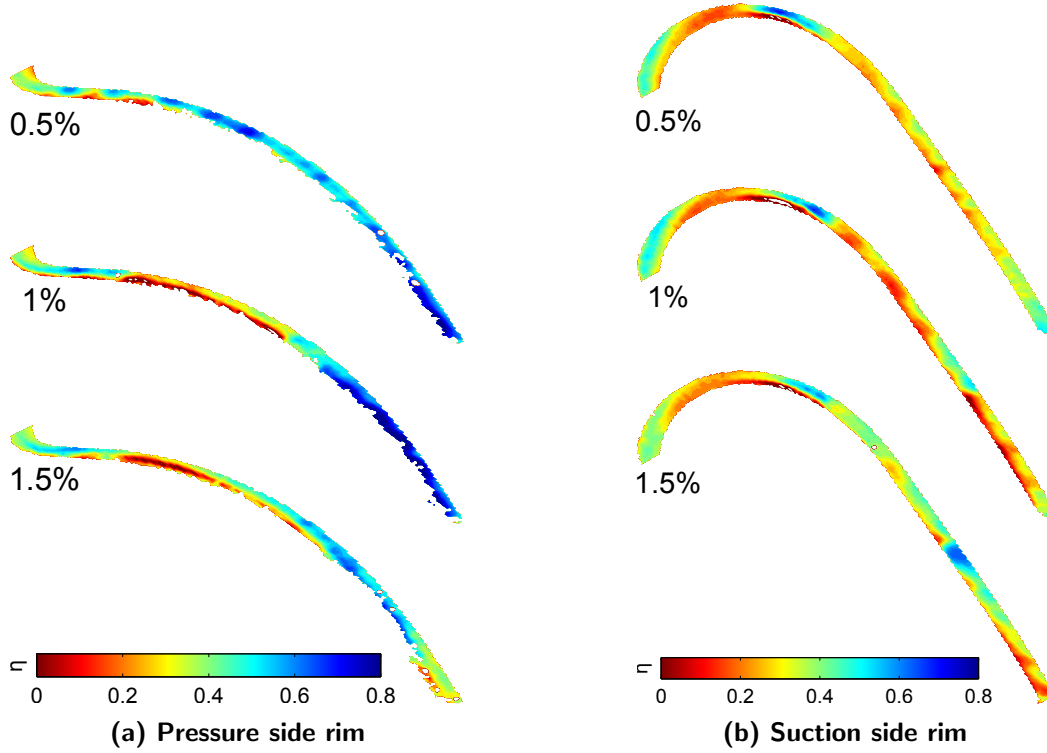


Figure 7.12: Film cooling effectiveness contours on the rims of the squealer tip (HSLC)

The circumferentially-averaged film cooling effectiveness results are presented in Figures 7.13a and 7.13b. The results show that with tip gap variation there is no clear trend on either of the rim surfaces. However, a cooler trailing edge on both rims is generally observed at smaller tip clearances. The pressure side rim shows a disproportional increase in cooling effectiveness around the mid-axial chord region when the tip gap is reduced (30-55% axial chord), whilst the remainder of the rim surfaces show no obvious relationships.

The rims perhaps show higher levels of effectiveness towards the trailing edge due to the interaction with the mainstream flow directing coolant towards this region. The suction side rim has lower levels of cooling effectiveness and is likely due to dilution

of the films, but also because there are limited cooling holes in close proximity to assist in lowering the surface temperature. Also, given in Table 7.3 are area-averaged effectiveness results for the separate rim surfaces. The overall values show slight contrasting behaviour with tip gap: the pressure side rim is ranked with tip gap, whilst there is no relationship on the suction side rim. Furthermore, the low effectiveness levels on the suction side rim reiterate the difficulty of protecting the squealer rims, as well as the remainder of the tip.

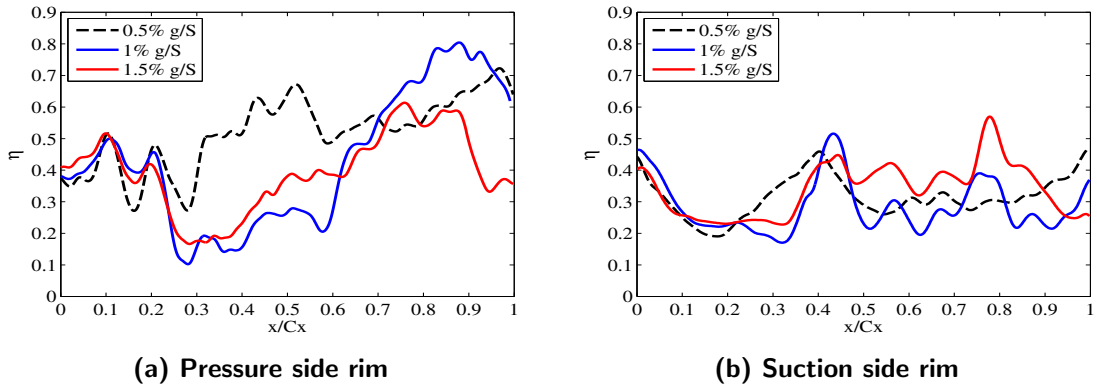


Figure 7.13: Circumferentially-averaged film cooling effectiveness on squealer rims (HSLC)

7.2.2 Cooled Aerodynamic Loss

Another important aspect of cooling design is to understand the impact it has on the loss and how it develops. To understand the development of loss downstream of the rotor, measurements of pressures were undertaken at two exit planes. These planes are both consistent with the uncooled study, namely half an axial chord and one axial chord downstream of the blade row. Moreover, three different tip gaps are examined and results are compared to the uncooled tip.

The use of kinetic energy loss coefficient is preferred to the total pressure loss as it accounts for the addition of coolant within the flow. The equation is described in more detail in Section 3.2.1. Spatial maps of yaw angle are also used to provide further details about the tip flow features with coolant injection.

7.2.2.1 Kinetic Energy Loss Coefficient

Initially, attention is drawn to the closest exit plane to the blade row, located at half an axial chord downstream. The measured kinetic energy loss coefficient distributions with coolant injection are presented in Figure 7.14a and without coolant injection in Figure 7.14b.

The behaviour of the coolant on the loss production near the rotor tip is complex. The spatial maps show that the vortices are moderately stronger than those for the uncooled blade (observed by the deepest shades of red), however the loss cores are larger in size, especially for the two largest tip gaps. The result therefore suggests that the coolant creates more compact and concentrated over-tip leakage vortices. Note that for both cases, a passage vortex is visible, as well as, a contribution of loss in close proximity to the left of the tip leakage vortex (as found with the total pressure coefficients in Section 5.3.2). This feature is also more pronounced without coolant injection.

The pitchwise, mass-averaged distributions of loss are presented in Figure 7.15. The plots show marginal differences, but for the majority of the spanwise positions it is apparent that the coolant reduces the loss. Even though the loss concentration is seemingly higher, the disturbed wake area is less which has more of an impact in this scenario.

The overall, mass-averaged contributions of kinetic energy loss coefficient for the three tip gaps are plotted in Figure 7.16. Similarly to the pitchwise-averaged results, the net effect of the coolant injection is that it reduces the tip leakage loss when directly compared to the uncooled tip when no coolant is present.

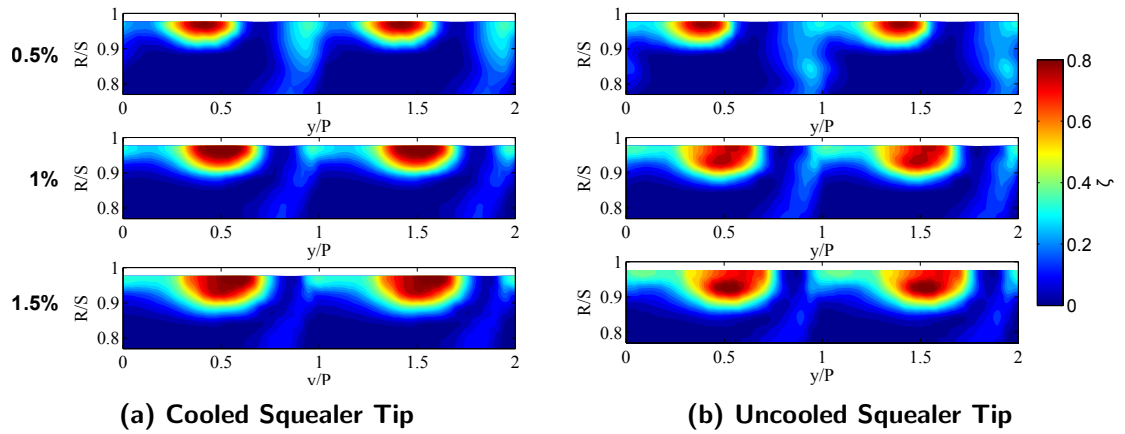


Figure 7.14: Kinetic Energy Loss Coefficient contours at 50% C_{ax} for cooled and uncooled squealer tips (HSLC)

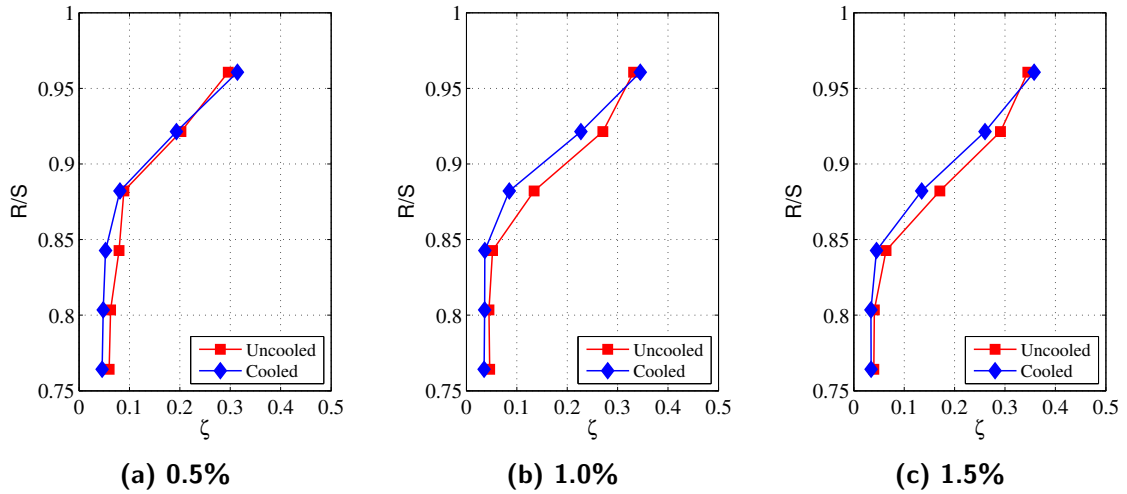


Figure 7.15: Comparison of pitchwise, mass-averaged kinetic energy loss coefficient for tip gaps 0.5%, 1% and 1.5% tip gaps at 50% C_x for cooled and uncooled tips (HSLC)

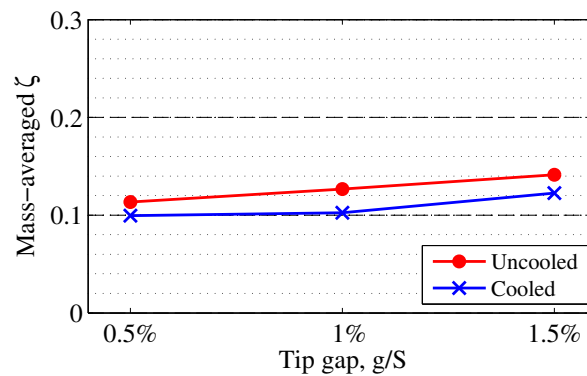


Figure 7.16: Overall mass-averaged kinetic energy loss variations for tip gaps 0.5%, 1% and 1.5% tip gaps at 50% C_x (HSLC)

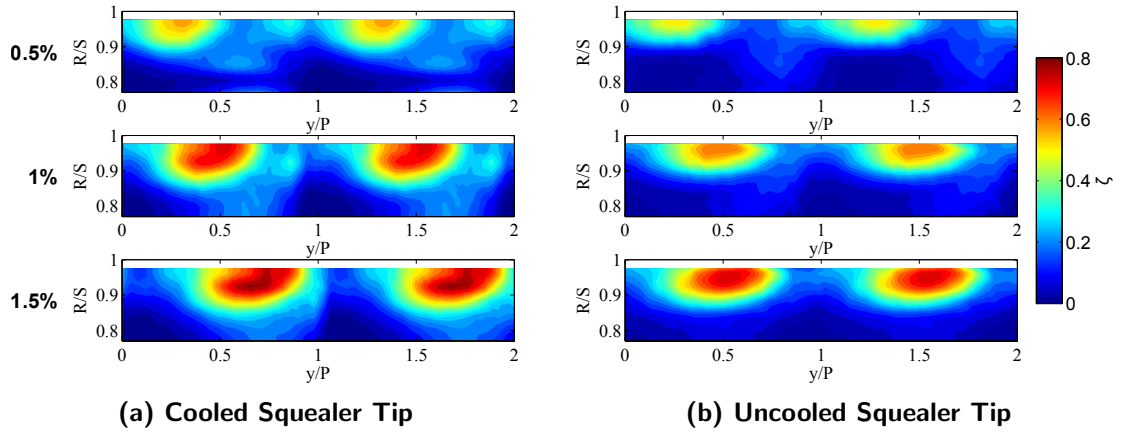


Figure 7.17: Kinetic Energy Loss Coefficient contours at 100% Cax for cooled and uncooled squealer tips (HSLC)

The last set of aerodynamic loss measurements are conducted at the exit plane located one axial chord downstream of the blade row. In a similar fashion to above, a cooling analysis is undertaken to compare the cooled tip to the uncooled tip. Presented in Figures 7.17a and 7.17b are the measured kinetic energy loss coefficient distributions with coolant injection and without coolant injection.

The major observation found with the cooled version of the tip, is the significant impact the coolant has on the loss production. The contribution of the coolant addition is more apparent and shows it has developed considerably from the previous measured location. The tip leakage vortices are both larger in strength and size. Furthermore, there are slightly noticeable smaller vortices spotted to the right hand side of the tip leakage vortices for the two smaller tip gaps (0.5% and 1%).

These are believed to be linked to features observed from the previous measurements, half an axial chord downstream. This region of flow is lossy due to the blunt trailing edge at the tip, acting as a bluff body disturbing the flow and creating a wake downstream. This disturbed flow continues to interact with the mainstream flow and prevail when coolant is present. It is likely that these features are more pronounced when coolant is used and when blowing through the near-tip suction side film cooling holes, it helps to propel the wakes and transfers momentum. It is also identified that

at the largest tip gap (1.5%) these features are less visible. However, it is apparent that the tip leakage vortex has experienced a substantial amount of shifting in comparison to the uncooled geometry. This displacement is likely to be due to a gain in momentum provided by the coolant encapsulated within the vortex or an interaction between the passage vortex and the tip leakage vortex.

The increase in loss observed in this case was also found by Hofer & Arts (2009) with a double squealer at high-speed test conditions. Their results showed an increase in loss with only pressure side and cavity cooling holes and it is understood by the results obtained in this case that by introducing suction side cooling holes, the losses can be significantly high. Also, through simulations by Wang *et al.* (2015) demonstrated that the coolant blockage effect is relatively weak for squealer tips under a transonic condition and may improved by introducing a partial flat tip. Following these results, it is understood that schemes to improve the aero-thermal performance of cooled squealers under a transonic flow regime are not well established and can be greatly improved.

Pitchwise, mass-averaged distributions of kinetic energy loss coefficient are presented in Figure 7.18. Unlike the previous exit plane, the coolant contributions show reversed trends. The results reiterate that the tip cooling now increases the loss in comparison to the uncooled case and moreover, has a higher penalty. The area-averaged contributions are also quantified in Figure 7.19. The results show a clear increase the present cooling configuration has on the overall loss, but also a non-linear behaviour as tip gap is increased. This non-linearity is also observed at the previous measurement plane (Figure 7.16) and is expected as the near-tip pressure loading will vary with tip gap height. In regards to the film cooling, this will lead to variation in the coolant mass flow exiting the suction side cooling holes and thus, produces a non-linear variation in the overall loss examination. Furthermore, at low-speed conditions, Zhou & Hodson (2011) found that a cooled cavity tip at small tip gaps do

not perform well in comparison to a cooled flat tip, however this trend reversed at larger tip gaps. It is possible that this non-linearity is due to the flow physics within the tip gap and how the coolant creates more blockage at larger tip gaps.

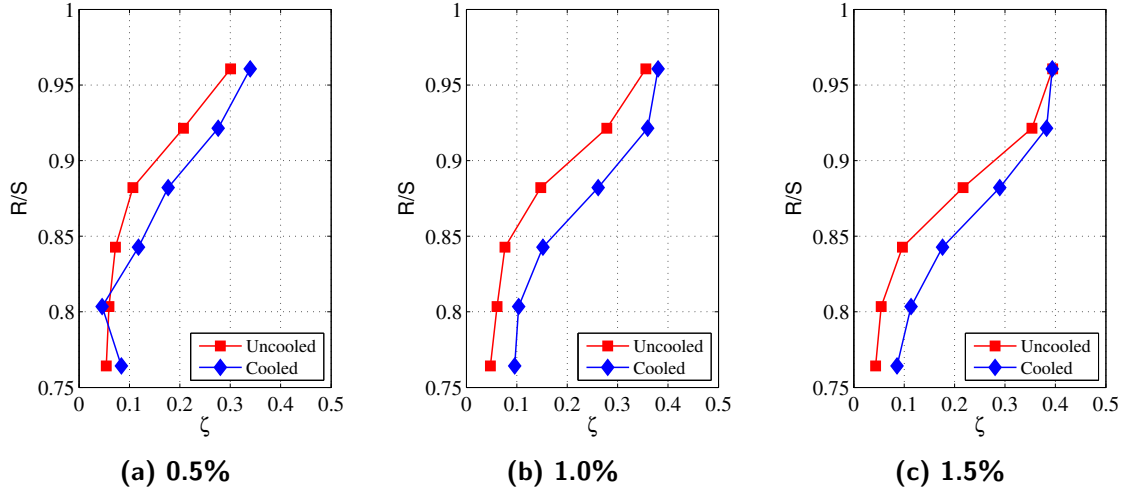


Figure 7.18: Comparison of pitchwise, mass-averaged kinetic energy loss coefficient for tip gaps 0.5%, 1% and 1.5% tip gaps at 100% C_x for cooled and uncooled tips (HSLC)

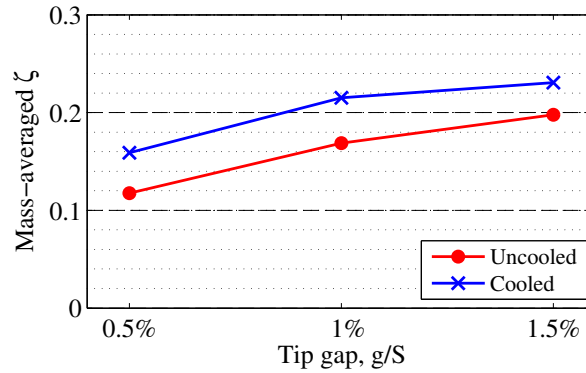


Figure 7.19: Overall mass-averaged kinetic energy loss variations for tip gaps 0.5%, 1% and 1.5% tip gaps at 100% C_x (HSLC)

7.2.2.2 Exit Flow Angle

The final aerodynamic analysis of the tip cooling configuration is an examination of the exit flow angle through measurements of the yaw angle. These results are compared to the uncooled configuration and also analysed at both exit planes.

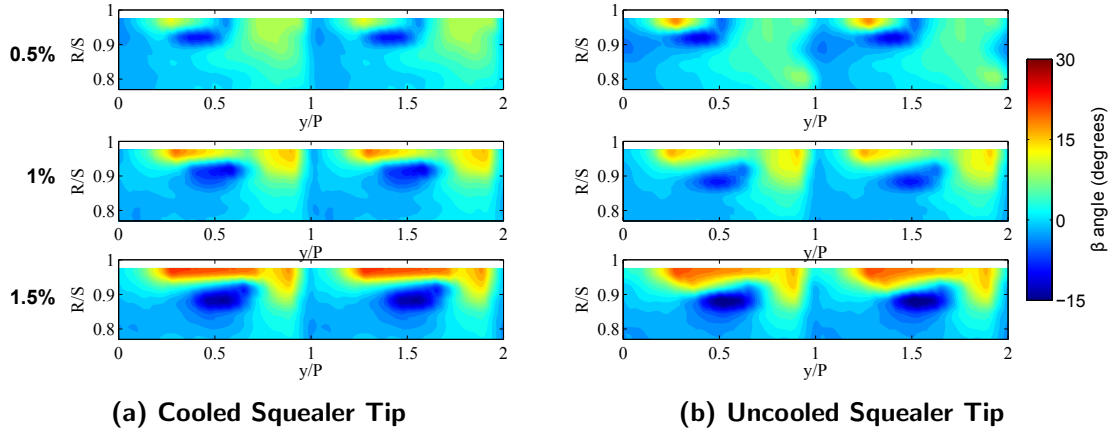


Figure 7.20: Yaw angle contours at 50% C_{ax} for cooled and uncooled squealer tips (HSLC)

Firstly, the exit yaw angle contour maps for the cooled and uncooled tips at half an axial chord downstream are presented in Figures 7.20a and 7.20b. The two cases are fairly comparable and the angles show marginal differences. These are more apparent near the casing where the uncooled tip has larger flow angles. The pitchwise, mass-averaged comparisons for each tip gap are shown in Figure 7.21. The results show a reduction in yaw angle when coolant is introduced into the mainstream flow. These results consolidate and agree with the kinetic energy loss findings such that the cooled tip produces lower loss at the exit plane closer to the blade row.

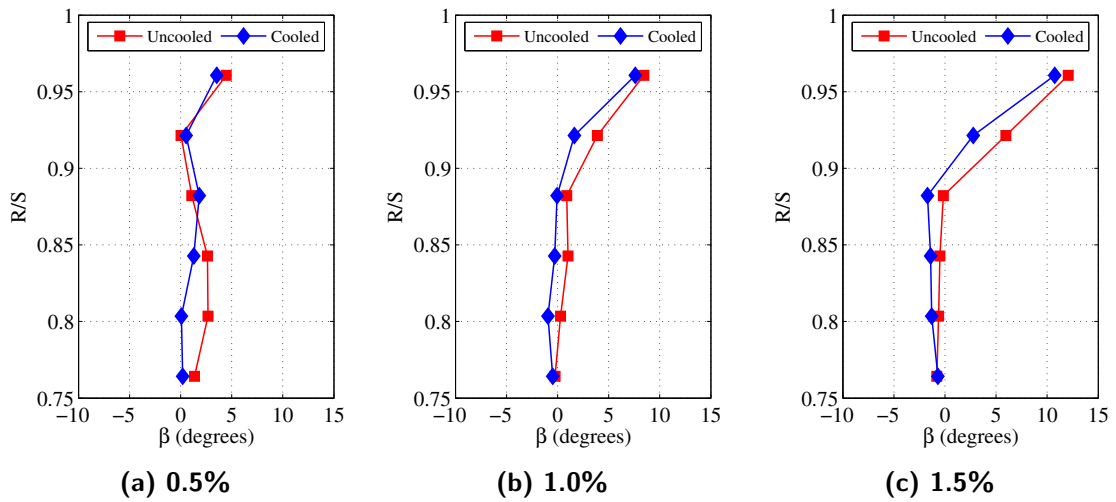


Figure 7.21: Comparison of pitchwise, mass-averaged yaw angle for tip gaps 0.5%, 1% and 1.5% tip gaps at 50% C_x for cooled and uncooled tips (HSLC)

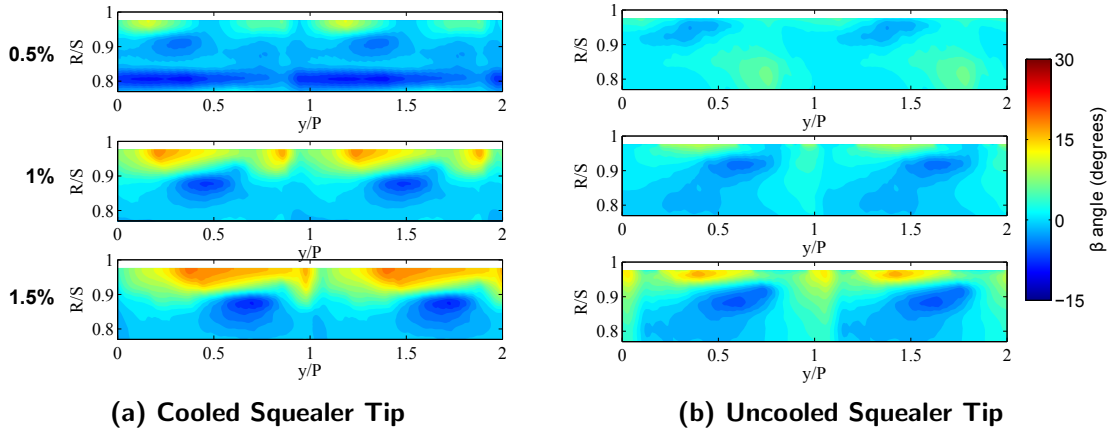


Figure 7.22: Yaw angle contours at 100% Cax for cooled and uncooled squealer tips (HSLC)

Secondly, the yaw angle contour maps at one axial chord downstream are presented in Figures 7.22a and 7.22b. In contrast to the previous study, the results show remarkable differences. Focusing on the two larger tip gaps, 1% and 1.5%, there is a significant amount of under-turning occurring near the casing region when cooling is used. These results justify the large increase in kinetic energy losses as previously found. The results at the smallest tip gap, 0.5%, show an unexpected amount of overturning (negative angles). These are a band of low angles at $R/S = 0.8$ between $0 < y/P < 0.4$ and are deemed anomalous. The remaining area follows a similar trend in comparison to the other tip gaps where larger, positive angles are visible near the casing and larger, negative angles are found within the mid-portion of the tip region ($R/S = 0.9$). The amount of flow turning for all tip gaps with cooling is much greater than those for without cooling and due to the larger deviations away from the blade metal exit angle, the cooling configuration produces higher loss.

Pitchwise, mass-averaged yaw angles are plotted in Figure 7.23. By considering these results, the extent of the flow disruption caused by the tip cooling is clear. For the two larger tip gaps in particular, flow angles nearer the casing are much larger than the uncooled setup. These results indicate that it is mostly the tip leakage flow that is intensified. With exception to the anomalous result found at the smallest tip gap ($R/S = 0.8$), the remaining results show a similar behaviour to the two larger

tip gaps but do show greater overturning in comparison. On the whole, the findings show that the coolant initially helps to reduce the losses with the film cooling acting as a sealant. However, further downstream the coolant shows strong interaction with tip leakage vortex where the turning is greatly increased and is associated with higher overall loss.

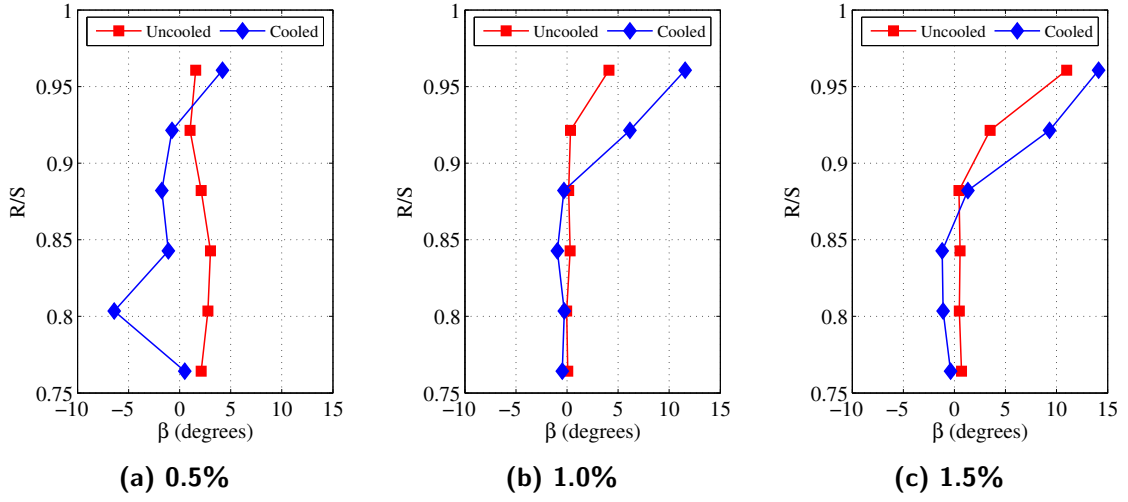


Figure 7.23: Comparison of pitchwise-averaged yaw angle for tip gaps 0.5%, 1% and 1.5% tip gaps at 100% C_x for cooled and uncooled tips (HSLC)

7.3 Computational Results

The following section provides computational results of the cooled HSLC study. The results use the HYDRA solver with the inclusion of source terms as introduced in Section 3.4. Source terms are often used, for example, to model chemical reactions or to generate heat. In this instance, source terms are used to model the film cooling. The basis of source (or if negative, a sink) terms is that they are scalar properties added to the conservation laws of mass, momentum and energy (Equations 3.10, 3.11 and 3.12). For HYDRA, source terms are specified usually through rows of film cooling holes using the total mass flow rate for the row, injection angle, total area of holes in the row, temperature and density of the coolant. These rows are mapped

onto the grid using known coordinates of the cooling holes.

The results presented in this section give an indication of the solver's capability in predicting the tip heat transfer and aerodynamic losses with film cooling deployed. It should be noted that the computational study is a relatively simple approach and the results may be vastly improved with a full modelling approach. Since the latter requires a highly complex grid and is computationally expensive, the source term model is used to assess its capabilities based on a number of assumptions.

7.3.1 Computational Cooled Tip Heat Transfer Results

As conducted for the HSLC study, the following addresses tip heat transfer and cooling effectiveness. Numerically calculated results of tip heat transfer coefficient for the three tip gaps are presented in Figure 7.24. In comparison to the HSLC (Figure 7.2), the results are generally over-predicted. Large differences are found on the squealer rims where it is clear that the coolant flow is having a larger influence, especially from the pressure side cooling array. Although, the reduction of heat transfer coefficient with increased tip gaps on the pressure side is captured. There is also less indication of reduced levels near the main cavity hot spot, but the surrounding cavity trends shows better agreement.

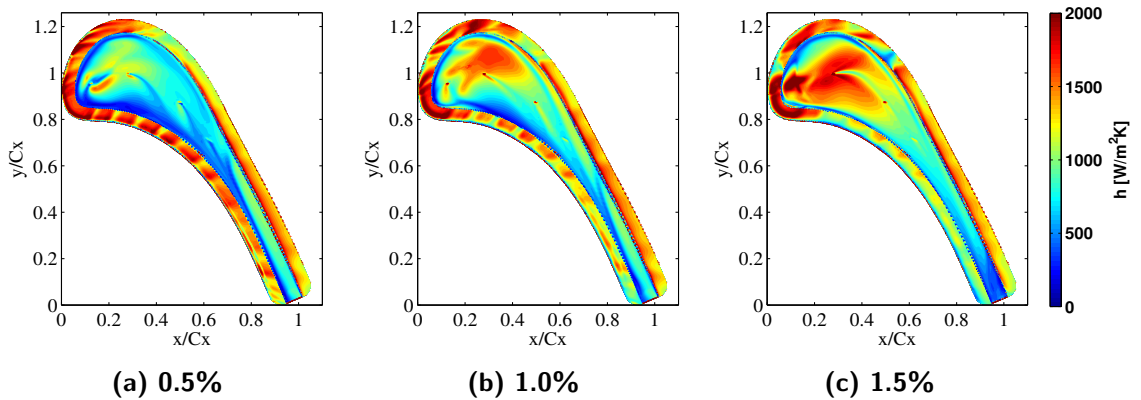


Figure 7.24: Predicted cooled squealer tip surface heat transfer coefficients ($\text{W/m}^2\text{K}$) for tip gaps 0.5%, 1% and 1.5% tip gaps (HYDRA)

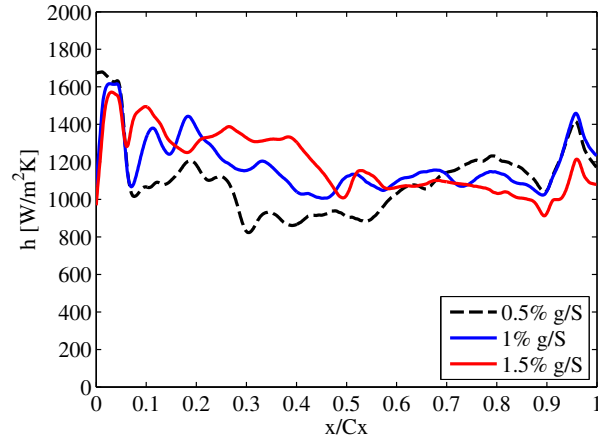


Figure 7.25: Predicted circumferentially-averaged tip heat transfer coefficient ($\text{W/m}^2\text{K}$) for cooled squealer (HYDRA)

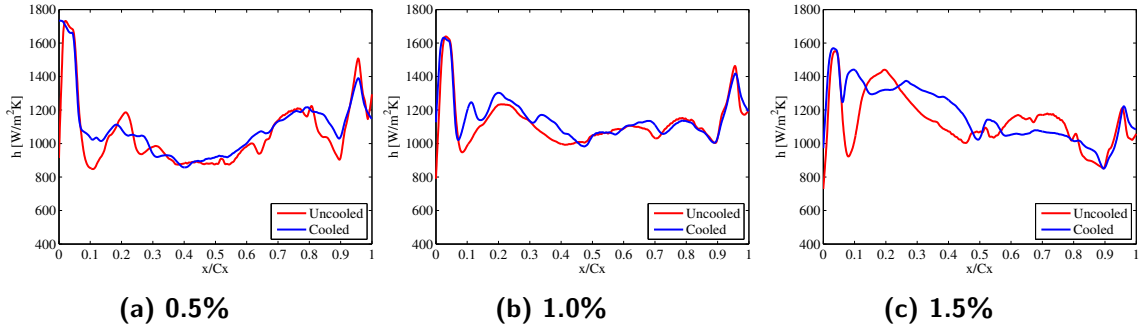


Figure 7.26: Predicted circumferentially-averaged tip heat transfer coefficient ($\text{W/m}^2\text{K}$) comparisons for the uncooled and cooled squealer (HYDRA)

Tip gap (g/S)	Uncooled h [$\text{W/m}^2\text{K}$]	Cooled h [$\text{W/m}^2\text{K}$]
0.5%	1025.4	1054.2
1%	1107.2	1137.1
1.5%	1136.9	1176.1

Table 7.4: Predicted area-averaged tip heat transfer coefficients for the uncooled and cooled squealer tip (HYDRA)

Comparing the cooled tip to the uncooled tip predictions, it is noticed that the heat transfer coefficient predictions are comparable in magnitude. However, with coolant injection there is slightly more area with higher heat transfer levels. Whilst the measured results did not show a clear reduction or increase when locally averaging the tip, the overall results suggested lower values. With the source term modelling technique, circumferentially-averaged heat transfer coefficients are shown in Figures

7.25 and 7.26. The area-averaged values are also given in Table 7.4. Both results show mostly increases in h , albeit only marginal. The affected regions coincide mostly with the frontal tip region and in particular, the leading edge rim.

Predicted cooling effectiveness maps are presented in Figure 7.27. Generally, in comparison to the measured data, the source term method has a tendency to under-predict the cooling effectiveness for all tip gaps. However, the cooling films themselves tend to show higher effectiveness and have a tendency to be more penetrative. This creates very localised high effectiveness regions, whilst the experimental setup shows that the films spread more over the tip surface. Linking this result to the tip heat transfer coefficients explains why the overall heat transfer coefficients trends differ. Experimentally, more of the tip surface has higher effectiveness, thus reducing the driving temperature of the convective heat transfer. Whilst computationally, the coolant has less interaction with the mainstream flow. The final observation is that as the tip gap is reduced, the effectiveness increases, as also found with the experimental results. It was observed that the highest effectiveness was towards the trailing edge of the smallest tip gap and the predicted results capture this trend well.

The circumferentially-averaged cooling effectiveness predictions are shown in Figure 7.28. The results show a greater difference to those measured. Firstly, by showing lower levels of effectiveness and secondly, the leading edge (0-5% C_{ax}) has very little to zero effectiveness. Towards the rear, it can be seen that the effectiveness is ranked accordingly with tip gap and the smallest tip gap shows the highest levels of cooling effectiveness. This result does compare well with those measured. Furthermore, the non-proportionality is less pronounced, but it is noticeable.

The area-averaged effectiveness values are given in Table 7.5. The results confirm that the source term modelling under-predicts the effectiveness, and whilst the predictions show effectiveness is ranked with tip gap, it is less transparent as those measured.

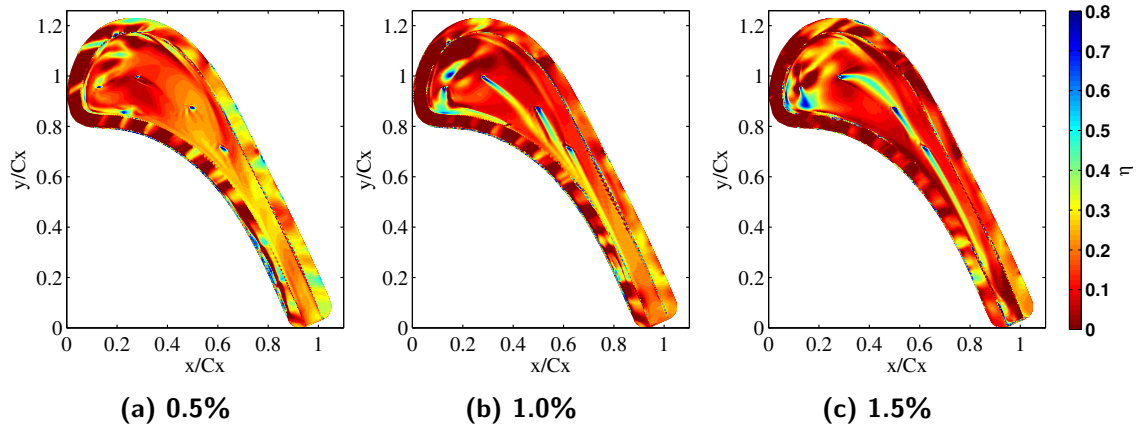


Figure 7.27: Predicted cooled squealer tip surface film cooling effectiveness for tip gaps 0.5%, 1% and 1.5% tip gaps (HYDRA)

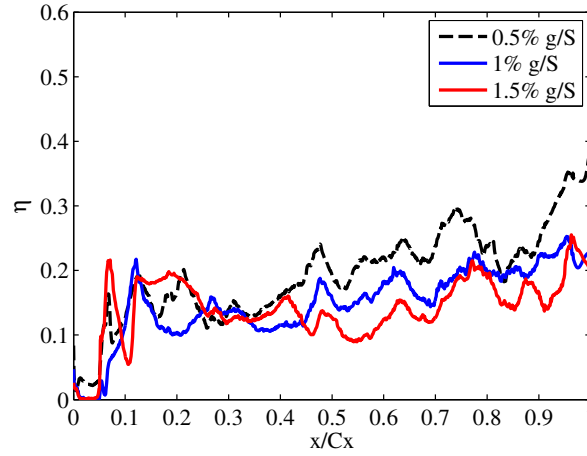


Figure 7.28: Predicted circumferentially-averaged film cooling effectiveness for cooled squealer tip (HYDRA)

Tip gap (g/S)	0.5%	1%	1.5%
η	0.19	0.15	0.14

Table 7.5: Predicted area-averaged effectiveness for squealer tip (HYDRA)

On the whole, the results found suggest that source term modelling for tip heat transfer with coolant injection is reasonable considering the level of sophistication involved. The overall differences in heat transfer coefficient compared to the uncooled case are considered small and therefore, suggest that the present cooling configuration has a small impact on the top surfaces of the tip. This is consolidated by the low overall levels of cooling effectiveness. However, as these results only consider the top,

flat surfaces, the intended cooling purposes may be missed. The pressure side rim shows a degree of lower heat transfer rates and as a result, the internal and external walls of the rim surfaces (near-tip pressure side, near-tip suction side and internal rim surfaces) are unaccounted for and may be the main areas targeted to protect. It is thus recommended that future studies consider these surfaces also, especially with film cooling.

7.3.2 Computational Cooled Aerodynamic Loss Results

The following section addresses predicted aerodynamic loss results in the form of kinetic energy loss coefficient at the exit planes previously described. The exit yaw angles are excluded since the source terms do not seem to interact with the external flow velocities. This may be due to inaccuracy in the assumed and specified coolant injection angles from the surface. These have been checked and show no greater than 0.01% change in the overall mass-averaged exit flow angle between the uncooled and cooled setups.

7.3.2.1 Kinetic Energy Loss Coefficient

Firstly, at the half axial chord exit domain downstream, kinetic energy loss coefficient contours are plotted in Figure 7.29. Figure 7.29a and Figure 7.29b show the cooled and uncooled squealer tip results, respectively. The results show that with film cooling there is little impact on the tip leakage loss. The predictions also show qualitative similarities to the measured results (Figure 7.14) including the branch of loss feeding off the main tip leakage vortex.

The low impact of the film cooling can also be seen in the pitchwise, mass-averaged kinetic energy loss coefficient curves shown in Figure 7.30. The results indicate a feature found experimentally where towards the casing, losses are seemingly higher with the cooled tip in comparison to the uncooled tip. This is with an exception to

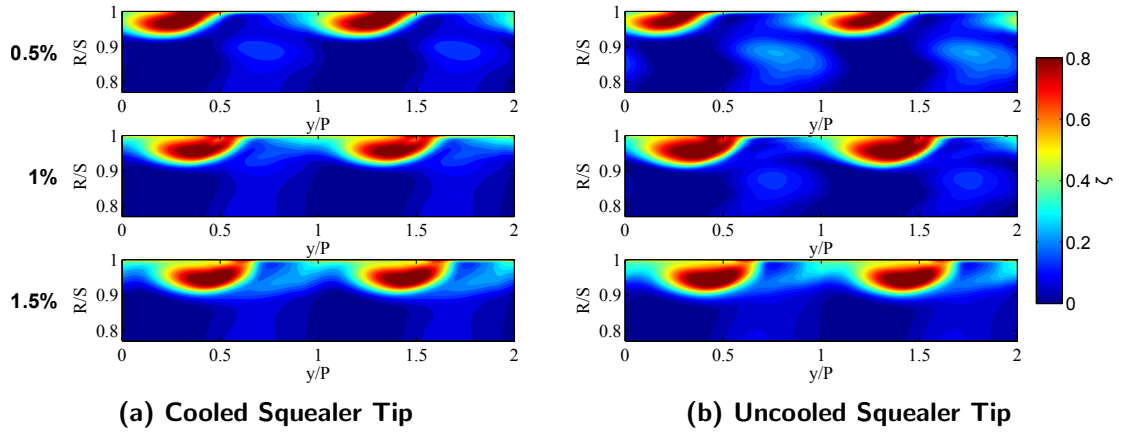


Figure 7.29: Predicted kinetic energy loss coefficient contours at 50% C_{ax} for cooled and uncooled squealer tips (HYDRA)

the predicted results for the largest tip gap. This finding shows an initial increase of loss with film cooling injection which in this case does not create a loss penalty. Figure 7.31 gives the overall kinetic energy loss coefficients that show the little effect coolant has on the loss development.

As found with the validation case, the values are again largely under-predicted. This discrepancy suggests that HYDRA with source terms struggles to predict the early stages of loss development or there was a misplacement of the total pressure probe at the desired exit plane. Even though there is a discrepancy between the measured and predicted results, the measured result showed that losses are reduced with coolant injection. This is also found here with an exception to the middle tip gap. As the differences are small, it may be concluded that there is little contribution from coolant at the nearest exit domain.

Travelling further downstream to the exit plane one axial chord away, the predicted kinetic energy loss coefficients are presented in Figures 7.32a and 7.32b. They show that with film cooling the losses are higher, as indicated by the sizes of the tip leakage vortices. There is also a shift in the tip leakage vortices, however this is not observed for the largest tip gap.

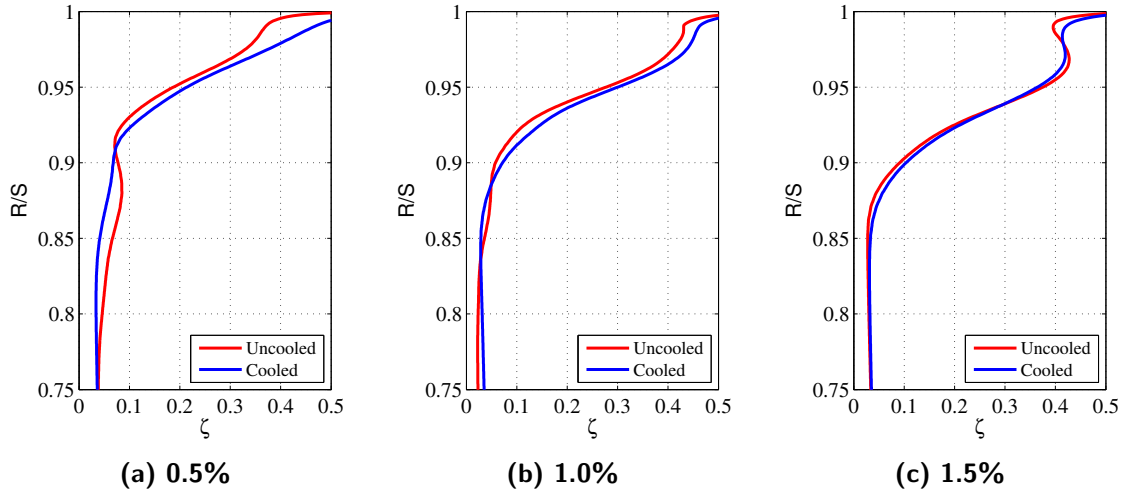


Figure 7.30: Comparison of pitchwise, mass-averaged kinetic energy loss coefficient for tip gaps 0.5%, 1% and 1.5% tip gaps at 50% C_x for cooled and uncooled tips (HYDRA)

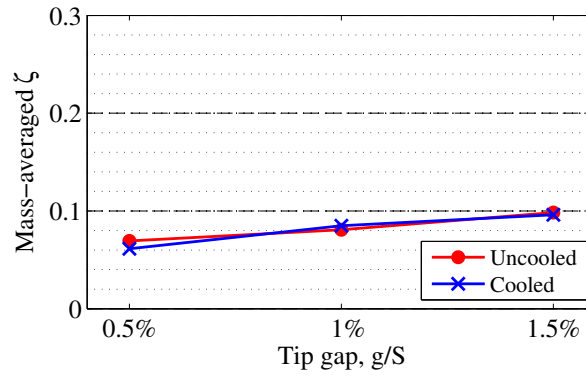


Figure 7.31: Overall mass-averaged kinetic energy loss variations for tip gaps 0.5%, 1% and 1.5% tip gaps at 50% C_x (HYDRA)

The pitchwise, mass-averaged graphs are shown in Figure 7.33. Interestingly, the near casing loss for the two larger tip gaps (1% and 1.5%) is lower than the uncooled case. This result is credible as the experimental data shows a similar trait that can be seen in Figure 7.18. The measurement locations at 0.93 and 0.96 R/S show the impact of coolant on the loss production reduces as the tip gap increases.

The losses with coolant injection become more significant at this exit plane in comparison to the previous plane and agreeing with those measured (examining the same area as that which has been measured). These contributions are shown in Figure 7.34 describing the overall mass-averaged kinetic energy loss coefficients. The

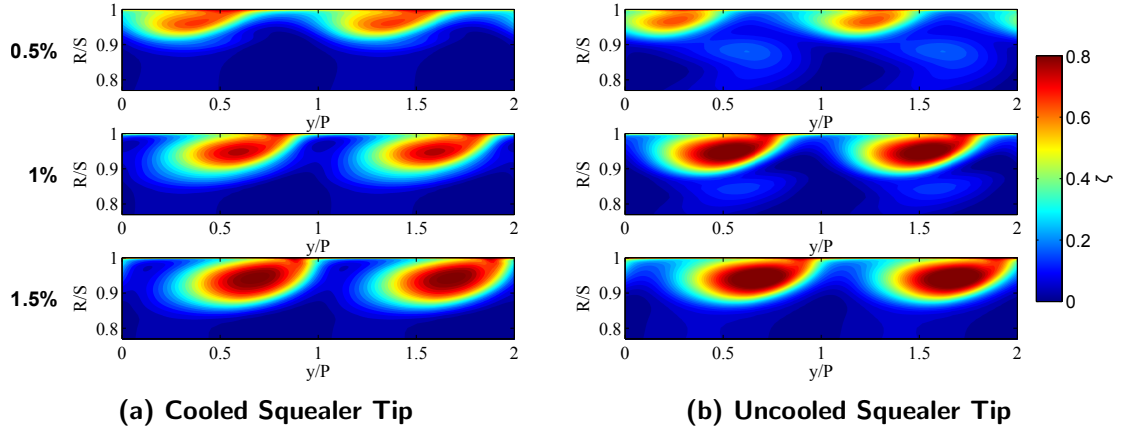


Figure 7.32: Predicted kinetic energy loss coefficient contours at 100% C_{ax} for cooled and uncooled squealer tips (HYDRA)

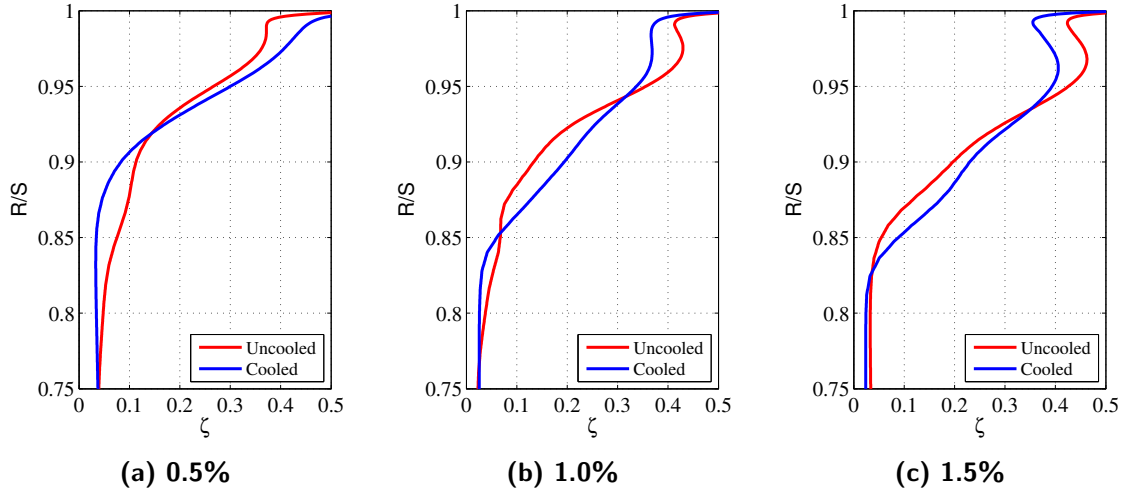


Figure 7.33: Comparison of pitchwise, mass-averaged kinetic energy loss coefficient for tip gaps 0.5%, 1% and 1.5% tip gaps at 100% C_x for cooled and uncooled tips (HYDRA)

uncooled results shows a large discrepancy with the 1% tip gap, but other tip gaps are mostly in good agreement. The source term approach shows that the cooled losses are under-predicted, which can be expected based on the modelling technique used, but the increases in loss with cooling present in comparison to the uncooled tip is a positive finding. These are such that the predictions detect the majority of features and trends that are found experimentally.

Considering the low-fidelity approach of modelling the film cooled squealer tip, it has been shown that the source terms are reasonably capable of predicting the ma-

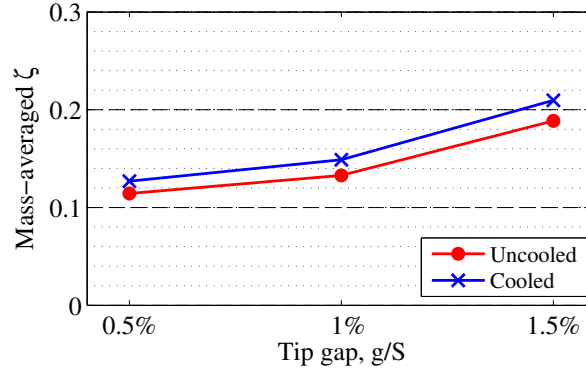


Figure 7.34: Overall mass-averaged kinetic energy loss variations for tip gaps 0.5%, 1% and 1.5% tip gaps at 100% C_x (HYDRA)

jority of trends found through experimental testing. In most instances, the technique has a tendency to under-predict the aerodynamic and thermal performance. Based on the computational cost saving, the coolant modelling is useful in establishing common trends, however the limitation is its ability in predicting absolute values. However, without a direct comparison to a higher fidelity modelling approach such as resolving each cooling hole individually, it cannot be concluded how much this technique is lacking. A study on a winglet using resolved cooling holes, rather than modelled, was conducted under low-speed conditions by Zhou *et al.* (2013). The results showed that the effectiveness was 11% under-predicted giving an indication of the present CFD capability in predicting cooled tip configurations and the continual requirement of computational fluid dynamics development.

7.4 Summary

In summary, heat transfer and aerodynamic loss has been determined for a cooled squealer tip both experimentally and numerically at engine-representative aerodynamic conditions. The results presented are the first of the kind in regards to assessing a cooled squealer tip at different tip gaps at a transonic flow condition. Thermal measurements of tip heat transfer coefficient and cooling effectiveness have been deter-

mined. Aerodynamic loss measurements, expressed in terms of the kinetic energy loss coefficient, have also been determined at two exit planes downstream of the squealer.

Comparatively, a low-fidelity cooled CFD analysis using specified source terms has been done to determine the capability of predicting the cooled aero-thermal performance of the squealer tip.

The conclusions are as follows:

1. The experimentally determined cooling impact on the squealer tip heat transfer coefficients is minimal and shows no clear trend to whether or not it increases or decreases with coolant injection.
2. The biggest effect the coolant has is found on the squealer rims and generally shows a reduction of heat transfer coefficient with coolant. This result is unexpected based on previous experience, for example the winglet. The results are also ranked with tip gap when considering the overall and suction side rim heat transfer. However, the pressure side rim shows a reversal of trends with coolant and also has a higher heat transfer coefficient than the uncooled tip at the small tip gap.
3. Measurements of film cooling effectiveness show coolant behaviour is dependent on the tip clearance. The flow features found at a large tip gap show regions of high effectiveness on the cavity floor downstream of the coolant holes. This effect reduces with smaller tip clearances and indicates more of a jetting effect with little interaction from the mainstream flow. Furthermore, as the tip clearance reduces, the effectiveness is greater, in particular towards the trailing edge. On the other hand, the overall area-averaged effectiveness follows no trend. Collectively, the tip cooling effectiveness is considered to be low.
4. The measured aerodynamic loss results indicate a contrasting behaviour at two downstream traverse planes. At the exit plane closest to the blade row, the

coolant acts to reduce the tip losses generated, whilst further downstream, the coolant largely increases the loss. Thus, indicating vigorous mixing with the mainstream flow occurring towards the more downstream portion of the fluid domain.

5. Similarly to the loss results, the exit yaw angles show little deviation at the near exit plane and much larger deviation at the far exit plane. This occurs within the tip leakage vortex and shows greater turning and fluid mixing when coolant is present.
6. The numerical simulations of the cooled squealer tip show less convincing results compared to the uncooled study. Although the absolute values calculated are not well predicted, the trends are reasonably well captured, in particular for the cooling effectiveness.

Chapter 8

Enhanced Internal Cooling By Squealer Tip Shaping

8.1 Introduction

The choice of a squealer tip is regarded, largely based on the recent findings, as the best compromise for overall aero-thermal performance. However, the aerodynamic benefit may be outweighed by the difficulty of cooling the rims that are subject to a very high thermal load.

Incorrectly determining the heat transfer within a gas turbine and the blade metal temperature can result in a significant reduction of the blade life. Furthermore, reducing the mean temperature by only 15 K can double the life expectancy of the blade (Friedrichs, 1997; Han, 2013). Therefore, it is critical that the local heat transfer coefficients and local blade temperature are accurately predicted.

The design of a durable turbine blade tip relies on the knowledge of local hot spots and high heat transfer regions. One of the main challenges imposed is to strategically maximise the internal cooling before introducing film cooling. With respect to this, the present chapter introduces a shaped squealer tip concept that aims to offer an

improvement thermally without hindering the aerodynamic benefit offered with the choice of a nominal squealer. The shaped squealer concept was first proposed by He *et al.* (2013). The major task of the present work is to examine the configurations both experimentally and computationally to prove the concept. Furthermore, the detailed analyses of a new shaped squealer configuration provides a direct indication of its feasibility as a realistic high-pressure turbine blade geometry under working conditions, and potential benefits for future design applications.

The proposed designs also acknowledge that the effects of external cooling by films can influence the aerodynamic performance both positively and negatively which has been found to be fairly dependent on the choice of tip and cooling strategy used. Moreover, feeding coolant to the tip itself is a difficult task and can have a significant influence on the external flow field, such as the unknown effects of film cooling weld beads. Thus, the proposed designs aim to address these by focusing on the internal cooling design aspect that can also potentially improve the external film cooling strategy.

Tip shaped designs tend to begin with an aerodynamically optimised tip and then consider the thermal aspects and rely on film cooling for protection. Ideally, the design should consider the interaction between both the aerodynamics and thermal management and the following chapters aim to address this.

8.2 Squealer Tip Shaping Hypothesis

Unshrouded tip configurations are largely driven by their ability to reduce the undesirable over-tip leakage flow. Based on the poor aerodynamic performance of the flat tip, unshrouded tips have evolved with a purpose to reduce this aerodynamic penalty. As discussed, such designs include squealers, winglets and winglet-squealer tips. A study focusing on different types of unshrouded tip by Coull *et al.* (2014b) shows

that by including a tip cavity, the over-tip leakage flow can be significantly reduced, whether it be for a plain tip or plain-winglet tip. It was shown that the leakage flow could be reduced by approximately 20% by including a cavity. The major drawback to any type of winglet tip is the increased heat load mostly due to the larger wetted area. Note that this is still less than a full shroud, and its heat load can be balanced against efficiency.

The reputable squealer tip has demonstrated its ability in reducing the over-tip leakage loss. Extensive research on the squealer tip, as covered in Section 2.5.2, has shown this capability, but findings have also indicated the major weakness lies with the squealer rims and their high thermal load. This problem is also exacerbated by the difficulty in cooling them.

In terms of the physical mechanism behind a squealer tip, the squealer can be likened to that of a “thin blade” as discussed by Denton (1993). Focusing on a squealer tip, to reduce the leakage flow, the design relies on the flow separation over the pressure side corner, as shown in Figure 8.1. Fluid is driven over the pressure side through the gap (g) over to the suction side. In this instance, the flow separation occurs over the pressure side corner creating a blockage effect. This blockage then leads to a reduction of the discharge over the tip. Likewise to a “thin blade”, a thinner rim (or smaller width, W) is preferred from an aerodynamic perspective. However, a thin squealer rim has a consequence thermally due to the increased area-to-volume ratio. This results in a higher specific heat load per unit volume. Given that from an aerodynamic standpoint, the squealer is the desired choice. However, the thermal performance is restricted.

Recent efforts to improve the current unshrouded tip designs have been discussed and although many offer thermal improvements, they impose a loss penalty. The present work aims to address this by developing a tip that can produce a similar level of aerodynamic performance as a standard squealer tip, but can offer a significant

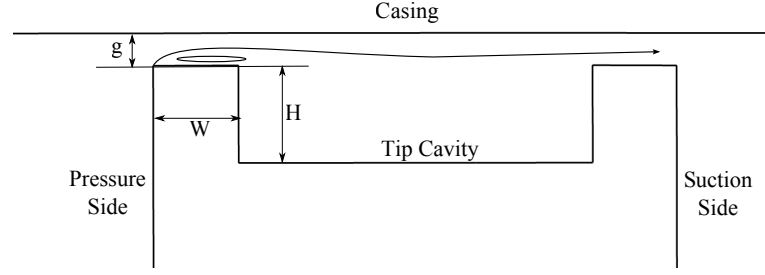


Figure 8.1: Squealer tip leakage flow blockage through separation (Reproduced from He et al. (2013))

improvement thermally by proposing an enhanced internal cooling method. This may also offer the potential of using less coolant and further increase the overall performance.

As discussed, the clearance gap, g , to width, W , is an important aerodynamic parameter for squealer tip design. Another is the rim width, W to cavity depth, H . The width must be small enough to prevent flow from reattaching on the surface (Metzger *et al.*, 1989) and the depth acts to assist in terms of heat transfer. A deeper cavity can generally provide lower heat transfer coefficients (Bunker & Bailey, 2001).

Given the importance of the rim width, W , in controlling OTL flow, it was proposed to shape the squealer rim as such that it still provides much of the desired aerodynamic effects whilst enhancing the cooling ability of the rim (He *et al.*, 2013). Schematics of a nominal and a shaped squealer are given in Figure 8.2.

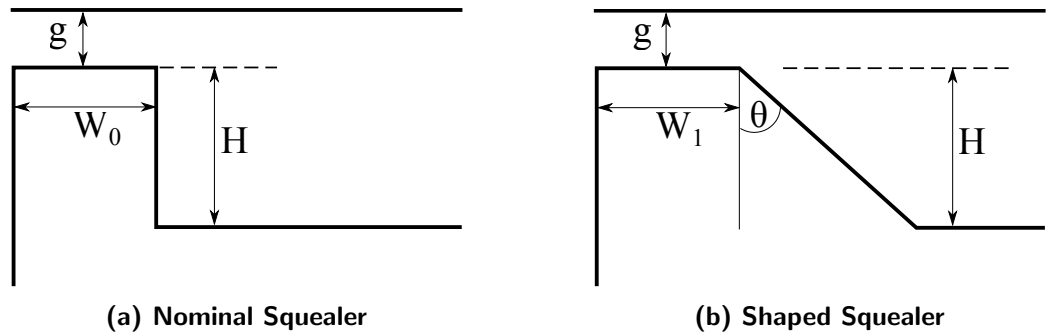


Figure 8.2: Squealer rim configurations (Reproduced from He et al. (2013))

Figure 8.2 shows that in regards to the aerodynamic performance, a small width is preferred. However, with regards to the thermal performance, the external surface

area exposed to convective heat transfer to the corresponding volume of the solid is limited. In relation to the surface area, A , of the rim to the solid volume, V , for a unit depth gives:

$$\left(\frac{A}{V}\right)_{nominal} = \frac{2H+W_0}{HW_0}$$

Thus, considering the extreme case for a small width (as $W_0 \rightarrow 0$):

$$\left(\frac{A}{V}\right)_{nominal} \rightarrow \infty$$

For a shaped squealer with a rim inclination angle (θ) from the vertical rim, the surface area to solid volume (for a unit depth) gives:

$$\left(\frac{A}{V}\right)_{shaped} = \frac{H + \frac{H}{\cos(\theta)} + W_1}{HW_1 + \frac{1}{2}H^2 \tan(\theta)}$$

For a small width (as $W_1 \rightarrow 0$) shows that the width is no longer a limiting factor:

$$\left(\frac{A}{V}\right)_{shaped} \rightarrow \frac{\cos(\theta)+1}{\frac{1}{2}H \sin(\theta)}$$

As a result, the rim width may be made smaller whilst at the same time providing an improved thermal performance with a reduced external heat transfer. However, the biggest advantage a shaped squealer can offer is the enhancement of the internal heat transfer by cooling. Current designs are limited by the internal heat transfer effectiveness as the rims tend to be at an unreachable distance from the internal cooling passages and are usually protected by film cooling. However, with a shaped squealer additional cooling passages may be repositioned (closer to the rims) and can still maintain the minimum wall thickness required. This concept is drawn in Figure 8.3 giving a representation of the improved cooling scheme in relation to type of squealer.

In order to justify the design of a shaped squealer that offers a high potential from enhanced internal cooling methods, the aerodynamic sensitivity to the rim inclination angle must be assessed and is the key focus of the present effort. For the basis of the design concept to work, the dependence on the over-tip leakage flow must be

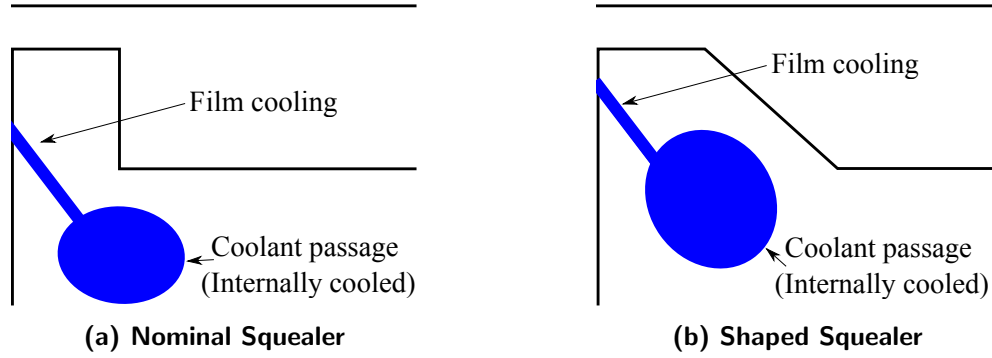


Figure 8.3: Enhancing the internal cooling by squealer tip shaping (Reproduced from He et al. (2013))

minimal. A set of squealers have been designed with different rim inclination angles ($\theta_h = \theta - 90^\circ$). These are more specifically with angles of $\theta_h = 90^\circ, 60^\circ, 45^\circ$ and 30° from the horizontal or the casing wall and can be seen in Figure 8.4. The blade geometry parameters are kept consistent, however a closed trailing edge was chosen.

8.3 Aerodynamic Sensitivity to Shaping

8.3.1 High-Speed Linear Cascade Results

In order to understand the aerodynamic sensitivity to the tip shaping concept, tests are conducted with the HSLC. An aerodynamic loss study is undertaken as conducted previously for the open cavity squealer. The test conditions are maintained as the blade is only adjusted near the tip region and the loss is examined at one axial chord downstream of the blade row. The three hole probe is also used and is traversed at 22 locations in the pitchwise direction, covering one complete blade pitch and 9 spanwise positions, of which 2 are done with a single hole probe close to the casing endwall. It was decided to decrease the pitchwise measurements locations and increase the spanwise measurement locations to provide a better resolution, based on the previous aerodynamic loss results obtained.

The detailed total pressure loss coefficient results for the four tip designs are



(a) 90° Rim



(b) 60° Rim



(c) 45° Rim



(d) 30° Rim

Figure 8.4: Nominal and shaped squealer models for aerodynamic sensitivity

presented in Figure 8.5. With respect to the rim shaping, the contour maps show inconsiderable differences between the four cases. For all cases, the magnitude and size of the loss core is mostly unaffected.

To further clarify the sensitivity to tip shape, pitchwise, mass-averaged total pressure loss coefficients are plotted for each design in Figure 8.7. The results show only

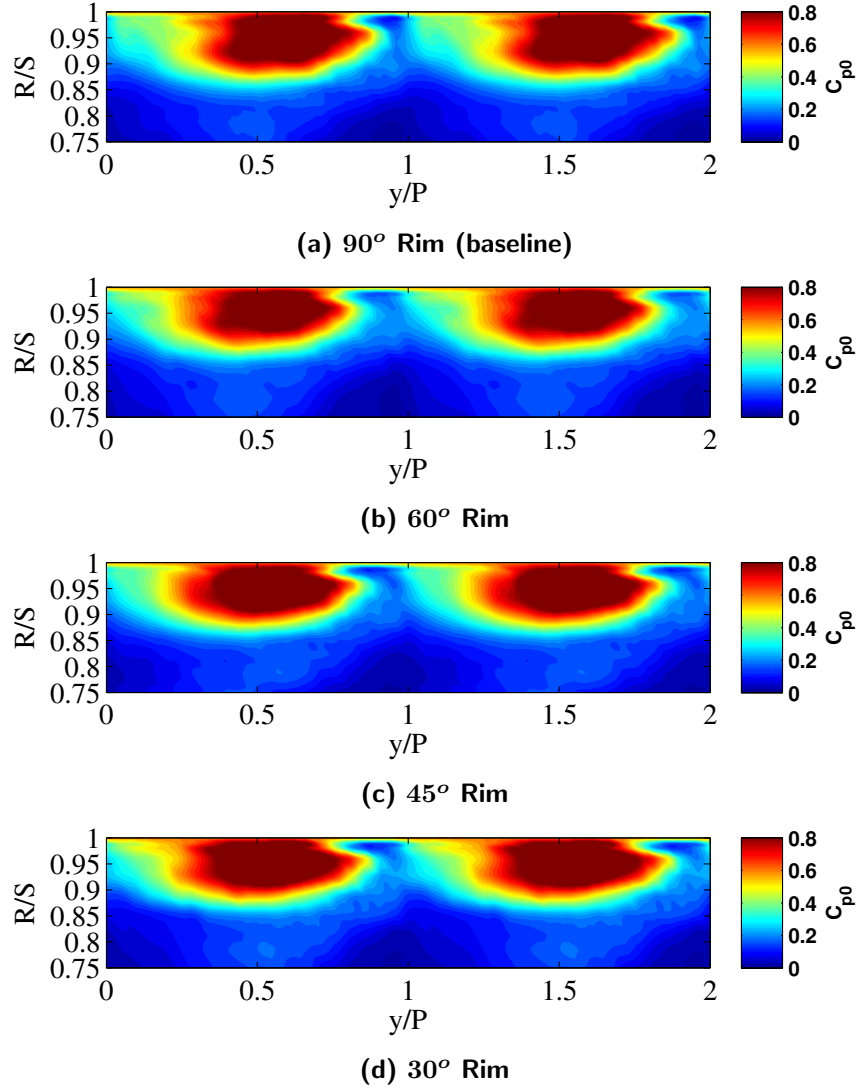


Figure 8.5: Measured total pressure loss coefficient, C_{p0} , for nominal and shaped squealer tips (HSLC)

marginal differences when comparing the nominal squealer (90° rim angle) to the shaped squealer tips. The majority of the points lie close to one-another and the largest differences are found with the shallowest rim case (30° rim angle) and may suggest that this tip forces a limiter on the shaping concept. Even though it still has a cavity, the blockage effects are not as strong as the steeper rim cases.

The final examination of the aerodynamic sensitivity is a study of the measured exit yaw angle and is based on the three-hole probe measurements. Contour maps of the exit yaw angle for the different shaped configurations are given in Figure 8.6. The

detailed yaw angle plots show small and insignificant differences as found with the pressure losses. These can be seen more clearly by examining the pitchwise averaged yaw angle results as shown in Figure 8.8. For the majority of the radial positions, the measured yaw angles are within half a degree of each other and the largest difference found shows a two degree margin.

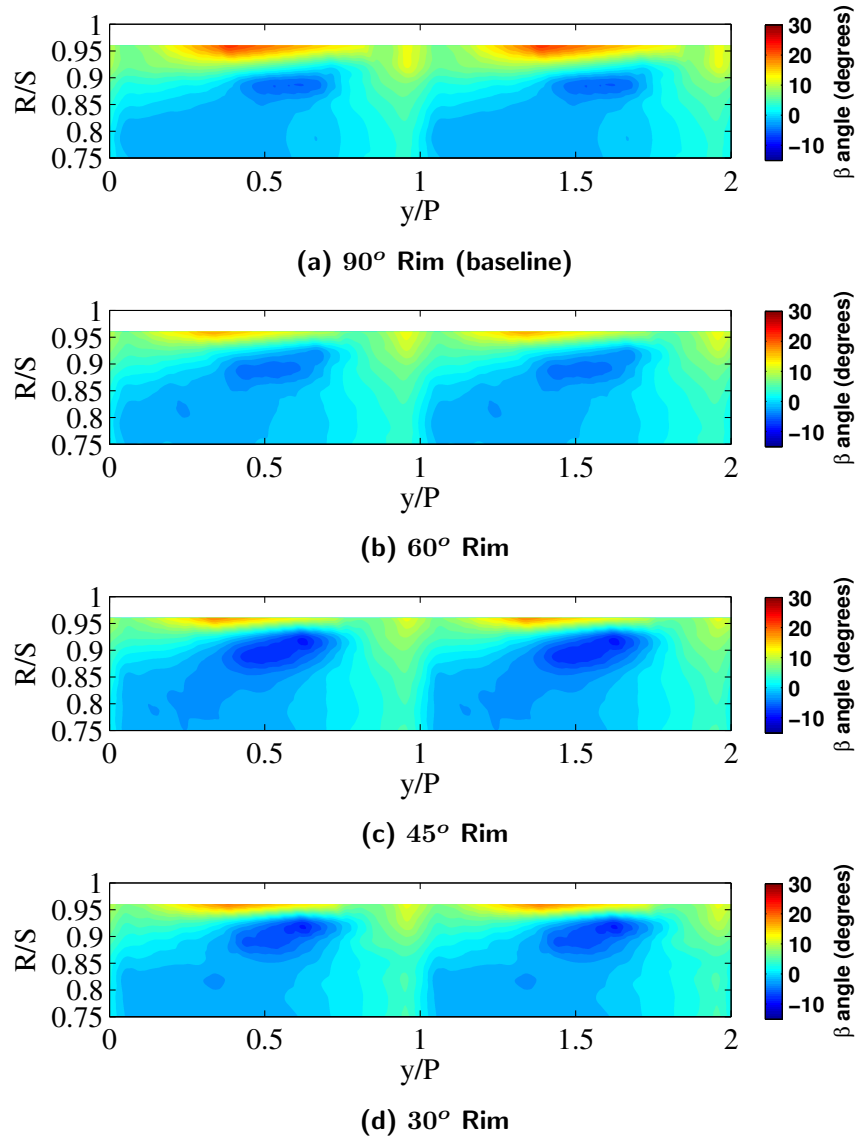


Figure 8.6: Measured exit yaw angle distributions for nominal and shaped squealer tips (HSLC)

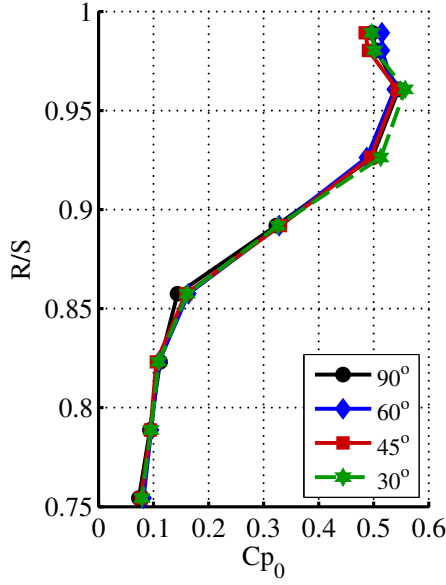


Figure 8.7: Mass-averaged pitchwise total pressure loss coefficients, C_{p0} , for nominal and shaped squealer tips (HSLC)

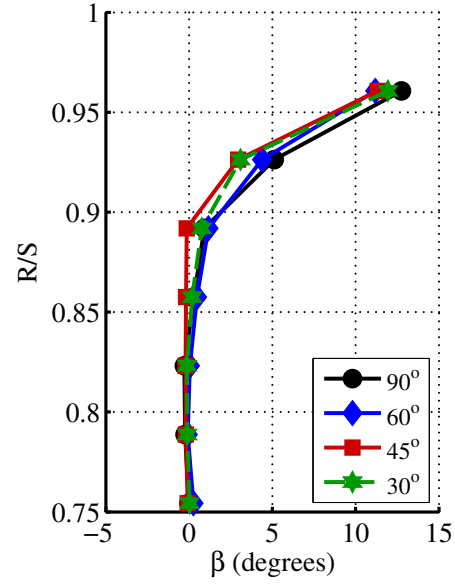


Figure 8.8: Mass-averaged pitchwise exit yaw angles for nominal and shaped squealer tips (HSLC)

8.3.2 Computational Results

The following study addresses the loss sensitivity through CFD using the HYDRA solver. The four different tip configurations are run with the equivalent boundary conditions to match the HSLC setup, as introduced in Chapter 3. The following aerodynamic CFD study serves a purpose to prove the aerodynamic concept of insensitivity to tip shaping and with the addition of the experimental measurements, the CFD can be validated.

As conducted using the HSLC, contour maps of the total pressure loss coefficient and yaw angle at the exit plane one axial chord downstream are calculated. The results are presented in Figures 8.9 and 8.10, respectively. Following the experimental results, HYDRA shows very little sensitivity to the tip shaping configurations suggesting that the over-tip leakage flow and loss develops in a similar manner for all cases, as shown in Figure 8.11

To further clarify the insensitivity to the tip shaping, pitchwise, mass-averaged plots of total pressure loss coefficient and yaw angle are presented in Figures 8.12

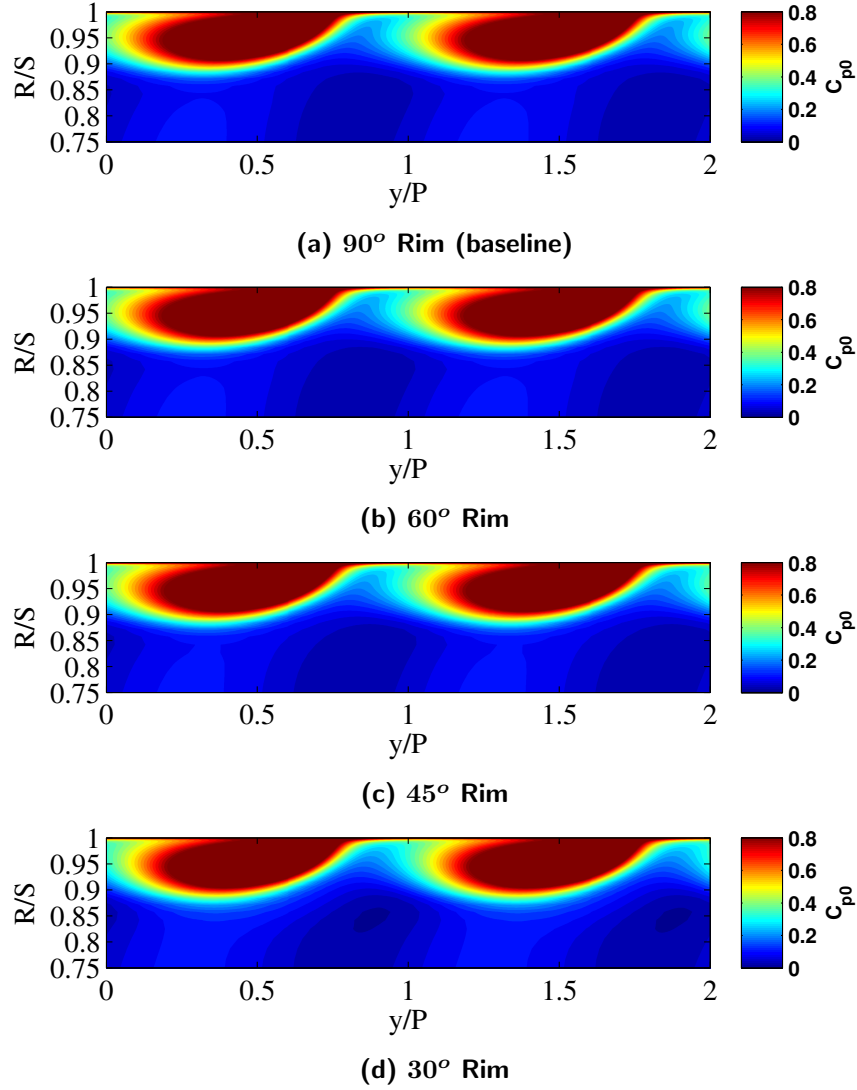


Figure 8.9: Predicated total pressure loss coefficient, C_{p0} , for nominal and shaped squealer tips (HYDRA)

and 8.13, respectively. Both computed sets of results reiterate the insensitivity to the squealer rim shaping. In comparison with the measured results, HYDRA compares well but fails to obtain the correct magnitudes of loss close to the casing wall. The exit yaw angles compare well. HYDRA also shows less sensitivity in comparison to the measured results and to an extent, does not capture the detailed flow physics. As indicated in Section 5.5, there may also be some dependence on the turbulence model choice and inlet boundary layer.

Lastly, overall mass-averaged total pressure loss coefficient results for the nominal

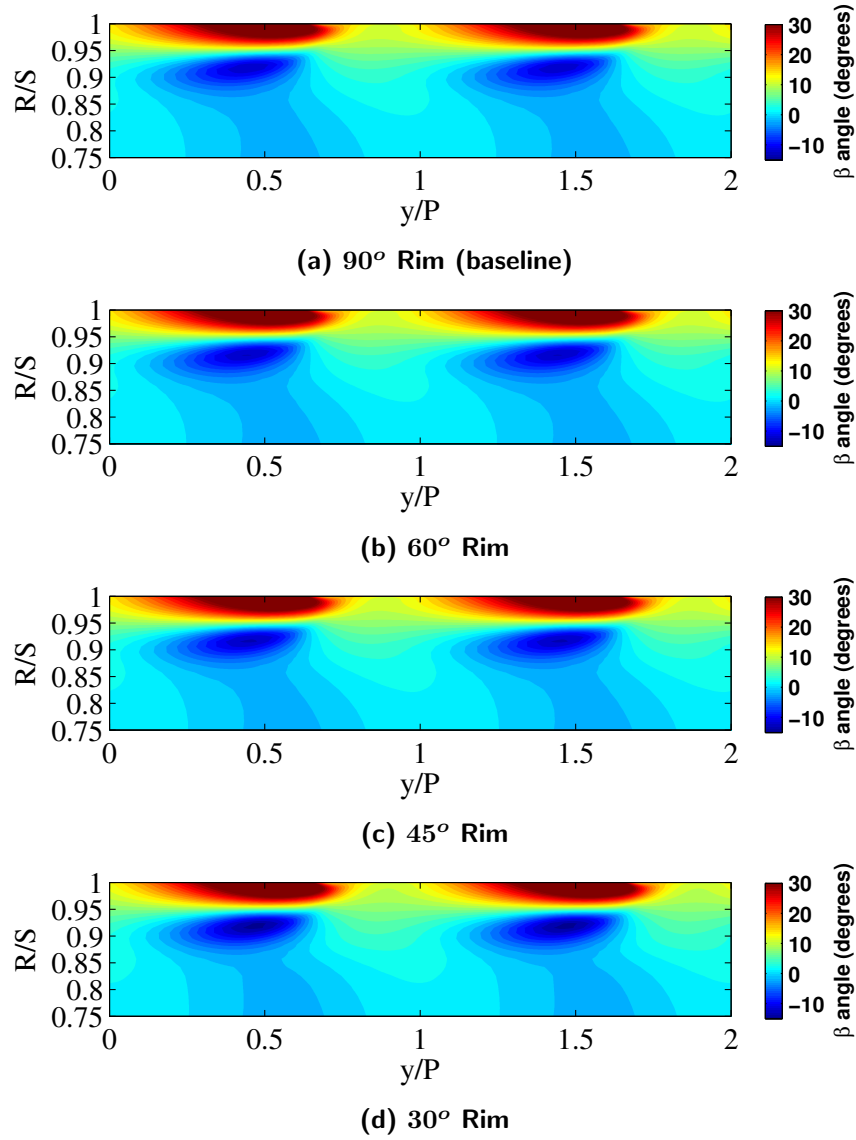


Figure 8.10: Predicated exit yaw angles for nominal and shaped squealer tips (HYDRA)

and shaped squealers for a 1% tip gap are given in Table 8.1. The average values are obtained from the uppermost region of the blade span equivalent to the upper 25% of the blade span. The percentage increase in pressure loss for each of the shaped squealers are 1.5%, 1.1% and 3.8% for the 60°, 45° and 30° rim inclination angles obtained using the HSLC, notably showing no trend. Similarly, HYDRA gives a 0.4%, 1.4% and 1.8% percentage increase for the respective shaped squealers. For all cases analysed the percentage increase in total pressure losses are no greater than

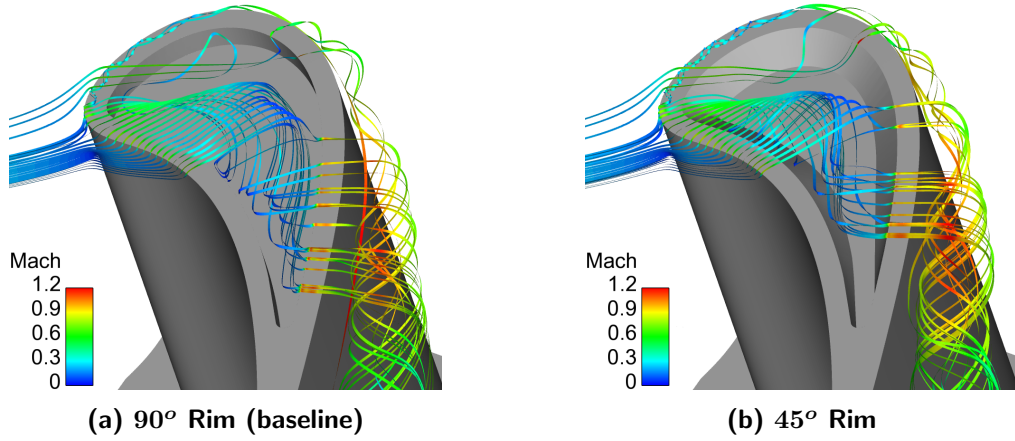


Figure 8.11: Streamlines showing the development of the flow patterns for nominal and shaped squealer (HYDRA)

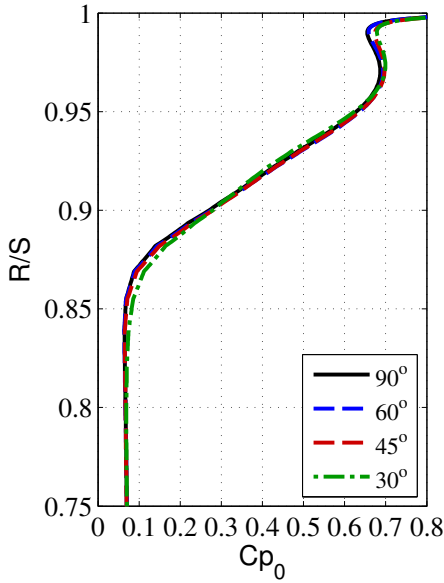


Figure 8.12: Mass-averaged pitchwise total pressure loss coefficients, C_{p0} , for nominal and shaped squealer tips (HYDRA)

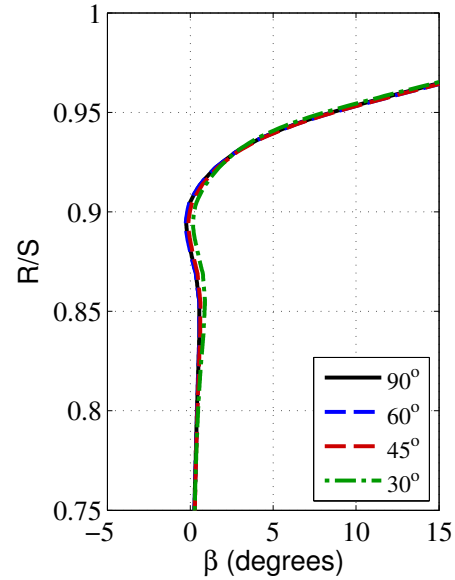


Figure 8.13: Mass-averaged pitchwise exit yaw angles for nominal and shaped squealer tips (HYDRA)

3.8%, however, up until 45°, the measured losses are no greater than 1.5%. Therefore, it suggests that the tip concept should only be applied to angles greater than 45° since the increase in the tip aerodynamic loss, when the angle reduces from 45° to 30°, is noticeably higher. The predicted results do not predict this trend and it is likely that the tip flow behaviour is influenced by the highly skewed elements with a 30° rim. For structured grids, the skewness of an element is the deviation from a quadrilateral

element. At 30° this would equate to 66% skewness and is likely linked to not being able to capture the correct flow physics near the rims.

Shaped Squealer	HSLC	HYDRA
90° Rim	0.262	0.278
60° Rim	0.266	0.279
45° Rim	0.265	0.282
30° Rim	0.272	0.283

Table 8.1: Measured and predicted overall mass-averaged total pressure loss coefficients for nominal and shaped squealer tips (HSLC & HYDRA)

8.4 External Heat Transfer Sensitivity to Shaping

The proposed benefit of the tip shaping is the enhancement of the rim cooling ability. Ideally, both the tip aerodynamic loss and external heat transfer are indifferent, or largely indifferent, with the shaping compared to a nominal squealer with perpendicular rims. Essentially, the specific heat load, that is the product of the heat transfer coefficient and the tip area (hA), is expected to reduce with each inclined rim angle of the shaped squealers since the overall tip area reduces.

In order to test this concept, results are obtained using HYDRA. The choice is primarily based on the previous validation exercise using HYDRA (Chapter 5) and secondly, due to the ability to include the inner squealer rim walls for the nominal squealer that are not accounted for when using the HSLC infrared thermography results.

Presented in Figure 8.14 are tip heat transfer coefficients for the nominal and shaped squealers. The nominal 90° rim squealer shows the same characteristics of heat transfer as found with the open cavity squealer in Section 5.2. The hot spot region at the leading edge is identifiable along with the very high heat transfer at the leading edge rim. The low region of heat transfer coefficient is also visible where flow separates over the pressure side rim. Apart from the change to a closed squealer tip,

the heat transfer patterns are consistent.

The shaped squealer tips also show comparative results. The high heat transfer found on the leading edge rim and the cavity hot spot are present. Also, the pressure side cavity region shows low heat transfer coefficient levels. However, the large scale flow separation effects behind the rim are less pronounced. Nevertheless, the majority of the heat transfer characteristics are similar for all cases.

In terms of the overall heat transfer coefficient (including all tip wall surfaces), the shaped squealers show an increase with shallower rims as shown by Figure 8.15. However, an important parameter in to consider is the tip area, which is significantly decreased by the tip shaping. For the present uncooled configurations, this result is advantageous as the heat load is reduced. Additionally, the potential for an improved internal cooling design can be addressed by having internal coolant in closer proximity to the rim walls. The overall tip area and heat load for the nominal and shaped squealers are presented in Figures 8.16 and 8.17. The results show that the shaped squealer concept is capable of reducing the specific heat load by up to 4.1% and 4.5% for the 60° and the 45° cases, respectively. As well as proving to have minimal impact on the tip aerodynamics, the tip shaping provides slight reductions in the overall external heat load. This makes the shaped squealer more advantageous.

8.5 Experimental Results

As a consequence of a lab incident, heat transfer results for the shaped squealer tips are limited due to multiple power supply and heater mesh failures over the past 12 months. This meant that a complete group of heat transfer experiments for these tips could not be obtained. The results provided in this section are merely for indicative purposes and only represent a partial validation of the numerical heat transfer results. Thermal data has only been obtained for the 60° rim squealer tip, nevertheless, the

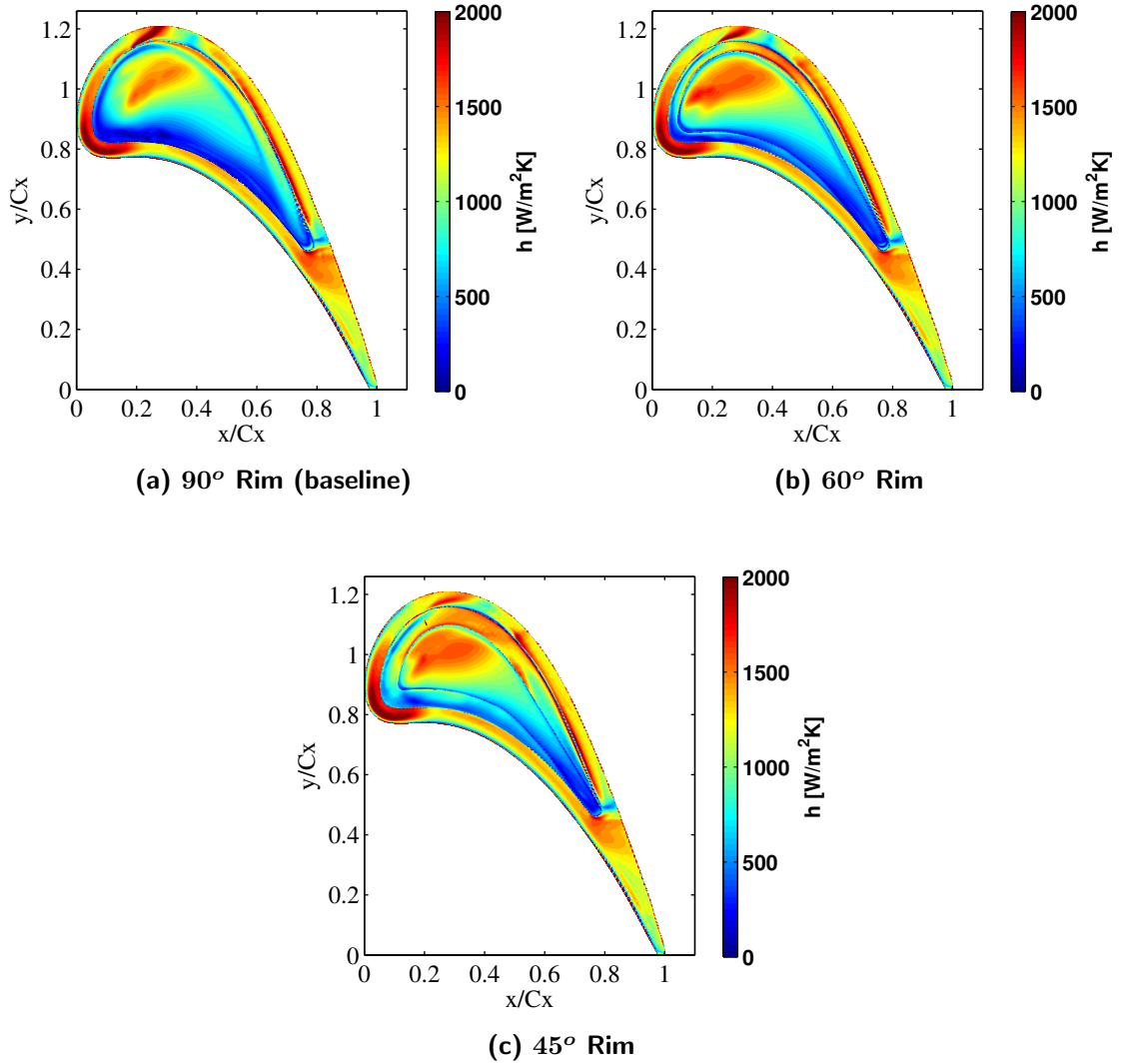


Figure 8.14: Predicted tip heat transfer coefficient contours, (W/m²K), for nominal and shaped squealers (HYDRA)

result is of high value as the data provides an insight into the behaviour of the inner squealer rims which were unable to be captured previously. Although the results have not been obtained for the nominal squealer and the 45° rim shaped squealer tip, the results obtained and presented in this thesis provide adequate information regarding the heat transfer patterns and trends over a transonic squealer, both unshaped and shaped with open and closed trailing edges. Moreover, the experimental results reveal a high level of detail of the tip heat transfer that is provided by a linear cascade facility and through the use of infrared thermography.

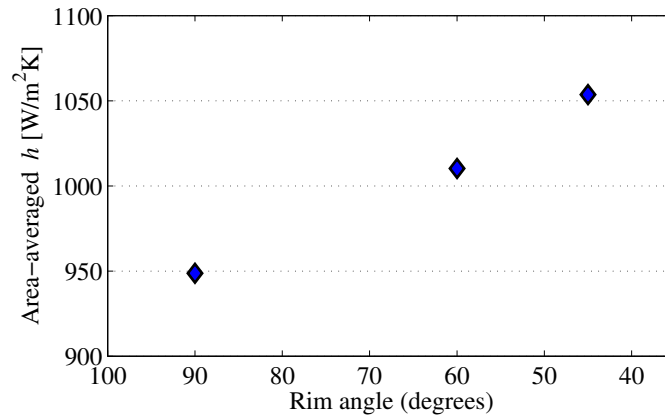


Figure 8.15: Area-averaged heat transfer coefficient, ($\text{W/m}^2\text{K}$), for nominal and shaped squealers (HYDRA)

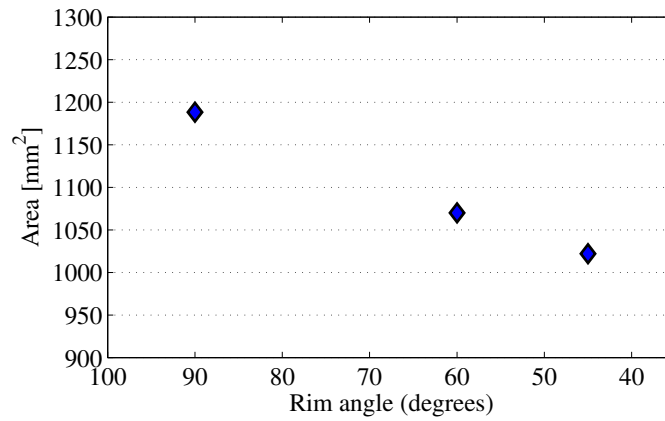


Figure 8.16: Total area of nominal and shaped squealer tips including inner rim walls (mm^2)

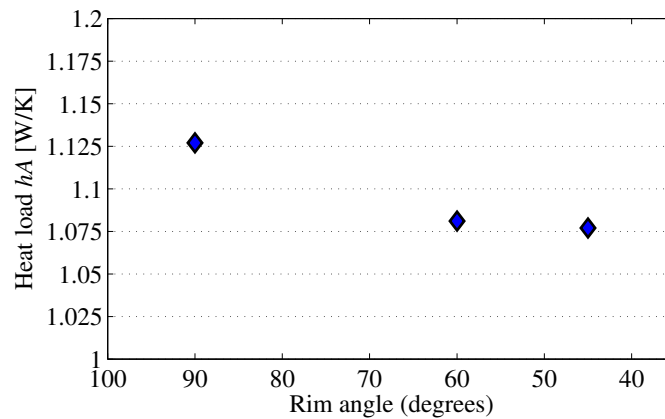


Figure 8.17: Specific heat load, (W/K), for nominal and shaped squealers (HYDRA)

The result presented in Figure 8.18 shows data obtained for the 60° rim shaped squealer. The results shows full coverage of the heat transfer coefficient over the tip,

including the inner shaped rims. The result obtained shows a similar pattern of heat transfer to the previously measured open cavity squealer (Chapter 5).

The leading edge region shows the highest levels of heat transfer coefficient where reattachment of flow on the leading edge rim is apparent and flow impingement is occurring within the cavity. Notably, the rear portion of the tip behaves like a flat tip as expected (Chapter 6). Most significantly, the inner rim heat transfer can now be identified and shows high heat transfer coefficients on the suction side rim. Conversely, on the pressure side rim, low levels of heat transfer coefficient are found which is a consequence of the flow separation over the rim. This result resembles the predicted tip heat transfer coefficient result shown in Figure 8.14b well and once more shows that HYDRA is well capable of predicting the tip heat transfer for turbine blade tips under a transonic flow regime. The area-averaged heat transfer coefficient for the measured squealer tip is $1047.70 \text{ W/m}^2\text{K}$ and the predicted squealer tip is $1010.20 \text{ W/m}^2\text{K}$.

It should be noted that up until 60° away from the IR camera viewing angle, the uncertainty on the angled rims should be no greater than previously measured. For a black painted tip, IR emission remains roughly constant to an angle of approximately 60° , therefore the sensitivity to the angle of observation is low.

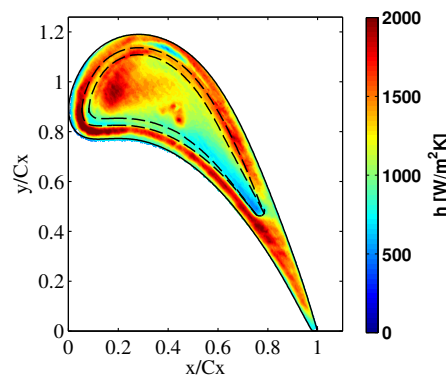


Figure 8.18: Measured heat transfer coefficient maps ($\text{W/m}^2\text{K}$) for the 60° shaped squealer tip (HSLC)

8.6 Open Versus Closed Cavity Squealer

Having opted for a closed cavity squealer to examine the sensitivity of the rim design, it is also useful to compare the aero-thermal performance compared to the open cavity variant of the squealer tip. Note that the closed cavity differs in a few ways compared to the open cavity squealer where the cavity depth is constant for the closed squealer and it also is based on a normal extrusion of the open cavity squealer mid-span blade profile removing the presence of any blade lean.

8.6.1 Overall Aerodynamic Loss

Given in Table 8.2 are aerodynamic total pressure loss coefficients for the open cavity and the 90° closed cavity squealer tips. The results include losses for the tip region only (upper 25% of blade span, including the tip gap) and the overall passage loss. The overall losses are expected to be lower than expected as the hub wall was prescribed an inviscid boundary condition. The reported values are calculated at one axial chord downstream and in terms of the mass flow rate, it is calculated over a plane over the suction side surface and is located from the tip surface to the casing wall.

	Open Cavity	Closed Cavity
Upper 25% blade span Cp_0	0.284	0.278
Overall Cp_0	0.161	0.164
Upper 25% exit yaw angle	4.68°	4.65°
Overall yaw angle	1.67°	1.65°
$\dot{m}_{OTL}/\dot{m}_{in}$ (%)	2.72	3.96
Near-tip Cp_b at 95% span	0.05	-0.12

Table 8.2: Aerodynamic performance parameters for open and closed squealer tips examined at a 1% tip gap (HYDRA)

Interestingly, focusing exclusively on the near-tip aerodynamic total pressure loss, the closed squealer shows better performance. However, the overall loss shows that the open cavity squealer performs marginally better and also has a substantially lower OTL mass flow rate leaving the suction side rim (including the gutter for the open

cavity squealer). The latter would also be true if the cavity exit flow was included for the open cavity squealer. The large difference in the mass flow rates over the suction side surface shows the gain in aerodynamic performance a squealer has, acting as a double seal. The closed squealer tip towards the trailing edge is essentially a plain tip and does not act as an effective seal, whilst for the open cavity squealer, the reversal is true. Examining the mass flux distribution along the curve length of the two tip geometries as presented in Figure 8.19 shows that the frontal portions of the tip behave similarly, but towards the trailing edge higher over-tip leakage is experienced when the rims are reverted to a flat tip. In terms of the trailing edge base pressure, the closed squealer performs better as expected having a lower base pressure coefficient (Equation 5.1) or more physically, a higher base pressure. Ideally, the cavity should extend to the trailing edge as much as possible to maximise the aerodynamic benefits. However, introducing a blunt trailing edge shows an increase in the near-tip losses showing the complexity of the tip leakage flow.

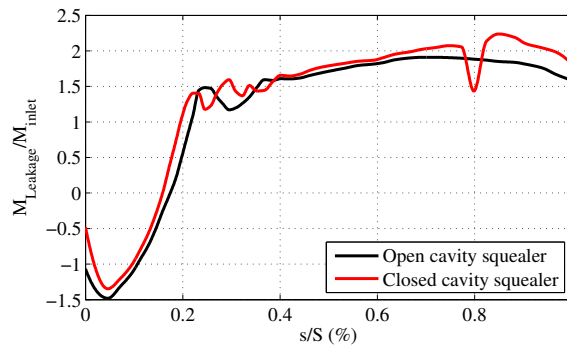


Figure 8.19: Mass flux distributions along the curve length of the suction side rim for open and closed cavity squealers with 1% tip gap (HYDRA)

8.6.2 Overall Heat Transfer

In terms of the thermal performance of the two tips, the results are area-averaged and include all tip surface walls. For the open cavity squealer, this is notably different to those given previously. This is because the previous basis was to validate the

numerical code based on the measured HSLC data, so only the top, flat surfaces were accounted for.

Table 8.3 gives the overall, area-averaged heat transfer coefficient and the heat loads for the open and closed squealer tips. The open cavity squealer shows higher heat transfer levels and an increased heat load. The latter occurs inherently since the overall area of this tip is much greater than the closed squealer. The closed squealer shows larger low heat transfer regions (separation from the pressure side rim) as can be seen in Figures 5.2b and 8.14a and is the main reason for the lower overall heat transfer coefficient.

	Open Cavity	Closed Cavity
Heat transfer coefficient [W/m ² K]	1020.4	948.7
Heat load [W/K]	1.55	1.13

Table 8.3: Overall thermal performance for open and closed squealer tips (HYDRA)

8.7 Summary

In this chapter, a modified squealer has been introduced with the aim to improve its durability whilst maintaining its aerodynamic performance. The biggest challenge of implementing unshrouded blades is the concern of the tip life and means to cool it effectively. Specifically, the squealer offers the best overall compromise between many design aspects and protection against tip-casing rubbing, but its rim durability is most problematic. The purpose of the concept design is to maximise the rim and near-tip internal cooling, prior to applying films. Three variations of the shaped tip concept have been examined and results analysed.

The main conclusions are:

1. Experimental results using the High-Speed Linear Cascade show that the aerodynamic losses are largely insensitive to the tip shaping and the rim inclination

angle. The results show at most a 3.8% increase in the overall, mass-averaged, pressure loss coefficient compared to a typical (nominal) squealer. Less sensitivity is found for rim angles greater than 45° providing an increase no greater than 1.5%. Smaller than a 45° rim angle, the tip shaping concept is deemed unfavourable based on experimentally determined results.

2. Exit yaw flow angles are also found to be insensitive the tip shaping, showing up to half a degree of deviation for most spanwise positions with a maximum increase of two degrees.
3. Numerical predictions using HYDRA show less sensitivity to the shaped squealers studied. Mass-averaged total pressure loss coefficients show no higher than 1.6% increase in comparison to the nominal squealer. Similarly, exit yaw angles show inconsiderable deviations.
4. Numerical calculations of the tip heat transfer coefficient reveal that it does increase with each rim inclination angle. However, since the area of each tip reduces, the tip heat load decreased by 5%. These results are not confirmed by experiments.
5. The combined aerodynamic and heat transfer studies reveal that the shaped squealer tip is capable of reducing the heat load whilst showing inconsiderable variations to the tip aerodynamic loss. These largely indifferent results with inclined rim angles are preferred and forms the basis of the enhanced cooling method, via internal cooling, to be introduced in the next chapter.
6. Comparing the nominal and previously studied open cavity squealer, the aero-thermal performance is dependent on the performance parameter. For the parameters considered, the closed squealer ranks higher. Caution should be exercised as the relative casing movement effects have not been examined.

Chapter 9

Conjugate Heat Transfer Analysis of Shaped Squealer

9.1 Introduction

Considering a squealer as a primary candidate for unshrouded tip development, it is important to address the thermal behaviour and its impact on durability. Therefore, this chapter continues to develop the shaped squealer and builds on its ability to house additional internal cooling passages within the aims to extend the tips lifespan. In particular, the rims appear to be the limiting factor in terms of the tip's lifespan given the very high external heat load they are exposed to, and their much larger area/volume ratios compared to other parts.

The development of a near-tip internal passage is made possible due to the increase in volume gained by the tip shaping. The approach taken for the following study is to enable and maximise the internal cooling as a main priority rather than rely on film cooling that is known to be challenged by complex flow features.

A conjugate heat transfer (CHT) CFD analysis is then conducted in order to identify and understand the thermal performance potential with the additional inter-

nally cooled passage for a tip under transonic flow conditions, comparing a nominal squealer to a chosen shaped squealer.

9.2 Conjugate Heat Transfer

The conventional approach taken to predict the blade metal temperatures is based on the surface convective heat transfer coefficient and adiabatic wall temperature (recovery temperature) calculated from a fluid solution using CFD and then used as boundary conditions for a conduction analysis using Finite Element Analysis and are often described as loosely coupled (or segregated) conjugate methods.

The alternative approach is to use a fully coupled conjugate method which solves both the fluid and solid zones simultaneously using the same CFD solver. This approach is also more efficient as the loosely coupled approach may require many iterations to improve the accuracy of the solution, whilst the fully coupled solution does not require the heat transfer coefficient or the recovery temperature as described by He & Oldfield (2011).

The governing equations to solve must be updated to account for the thermal conduction of the solid, so with no thermal source, the equation to solve is:

$$\frac{\partial \rho C_p T}{\partial t} = \nabla \cdot (k \nabla \cdot T) \quad (9.1)$$

On the interface of the fluid and the solid region, energy conservation must be satisfied so the heat flux must be equivalent across the interface. Also, the temperature must be equivalent on both interfaces as shown in Figure 9.1, or in other words to satisfy temperature continuity:

$$\frac{k_s}{\Delta x_s} (T_w - T_s) = \frac{k_f}{\Delta x_f} (T_f - T_w) \quad (9.2)$$

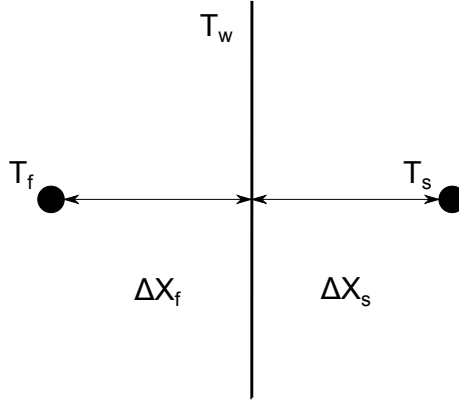


Figure 9.1: Fluid and solid domain interface

Or, alternatively:

$$T_w = \frac{T_s(k_s/\Delta x_s) + T_f(k_f/\Delta x_f)}{k_s/\Delta x_s + k_f/\Delta x_f} \quad (9.3)$$

9.3 Formulation and Design Process

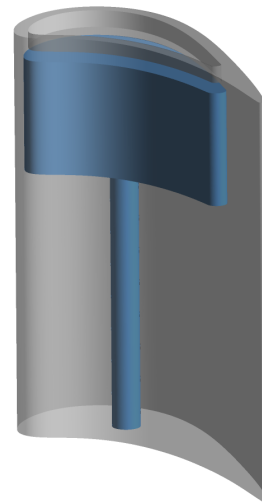
Following the verification that the over-tip aerodynamic loss is relatively insensitive to the rim shaped angle, a lower overall heat load and the high potential of increasing the tip durability through the design of additional internal cooling passages, a conjugate heat transfer study is conducted.

For the conjugate study, the solid must also be modelled. As the priority is to maximise the internal cooling, no film cooling is modelled. A simple, single internal cooling channel is fitted and in the case of the shaped squealer, this channel feeds extra cooling passages within rims. Incorporation of several cooling passages with introduced surface roughness, rib turbulators or dimples are not explored in the present study. Modern gas turbine blades can tolerate a minimum wall thickness of less than 1 mm and up to 0.5 mm for trailing edge slot designs and these two conditions serve as constraints in the cooling design process.

The two initial designs sought are shown in Figures 9.2 and 9.3. The former is a



(a) External geometry

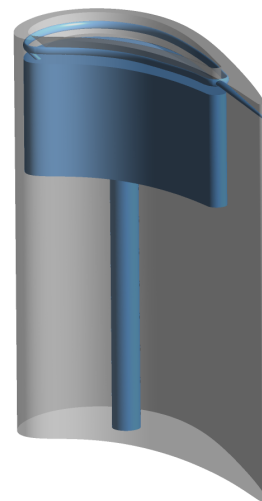


(b) Internal geometry

Figure 9.2: Nominal squealer solid model with a simplified cooling chamber



(a) External geometry



(b) Internal geometry

Figure 9.3: Shaped squealer solid model with cooling chamber and extra cooling passage within rims

nominal squealer with a simplified internal cooling passage and the latter is a shaped squealer (Figure 8.4c) that is also fitted with an extra cooling passage within the rims. Improvements to the passage and trailing edge slot are also examined (Section 9.6.1).

The chosen material property for the solid is a Nickel superalloy (Inconel 718) that has material properties given in Table 9.1. Typically, the alloy is a composition

of Nickel, Chromium, Iron and other elements and is often used in within turbine engines for its resistance to degradation, high temperature strength and toughness, as described by Pollock & Tin (2006).

Property	Value
Density	8193 kg/m ³
Specific Heat Capacity	435 J/KgK
Thermal Conductivity	11.4 W/mK

Table 9.1: Material properties of Inconel 718

9.4 Flow Solver, Boundary Conditions and Numerical Grid

The chosen CFD solver for the CHT analysis is Ansys CFX since a fully coupled analysis was not developed for HYDRA Version 6.1.16. The fluid domain for the CHT study is largely the same as those used for the uncoupled studies with a few forced amendments to the blocking structure to ensure a conformal grid with the solid volume. The calculations are steady state.

The boundary conditions applied to the CHT study are unchanged on the inlet and outlet, although the inlet total temperature is set to 300 K and since the external fluid-solid temperature is now resolved, no thermal boundary conditions on the external solid wall surfaces are required. All walls are supplied with no-slip wall conditions and the hub and casing are set to adiabatic to solely examine the effects of the extra cooling passages within the rims on the tip heat transfer. In order to consistently compare the nominal squealer to the shaped squealer, the internal coolant fluid is not resolved. This is because without film cooling, the flow would remain stagnant within the nominal squealer's internal chamber. Thus, the internal plenum and chamber have imposed wall conditions with an assumed fluid temperature set to 150 K giving

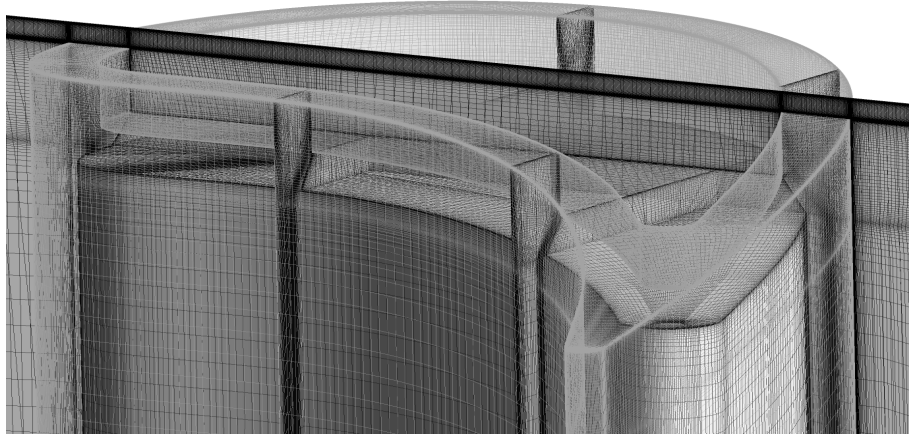


Figure 9.4: Nominal squealer grid used for conjugate study

a typical inlet-to-coolant temperature ratio of 2, and internal heat transfer coefficient, h_i , set to $700 \text{ W/m}^2\text{K}$. Lastly, to close the turbulence stress terms the SA model is used with a later study to examine turbulence sensitivity using $k - \omega$ SST. It must be stressed that the external fluid is solved as studied previously and only additional internal cooling boundary conditions are supplied.

The numerical grids used for the study are generated using Pointwise. The grid for the nominal squealer is fully structured for both fluid and solid domains and shown in Figure 9.4. However, due to the complexity of the geometry part of the solid for the shaped squealer forced an unstructured grid comprising of tetrahedra and pyramids. Since the solid is simpler to resolve in comparison to the fluid, the compromise was inconsequential. The shaped squealer grid is shown in Figure 9.5.

To ensure the numerical solution does not depend on the grid, refinements were made to both the solid and fluid regions, for both the nominal and shaped squealer models. As stated, the meshes of the fluid domains studied previously were maintained as well as possible, however some adjustments were forced due to the blocking structure of the solid domains. Four meshes were tested and the minimum number of cells required was 19.7 million cells. An refinement of this mesh to 22.7 million cells gave a change of 0.1% in the area averaged tip temperature. The mesh dependency

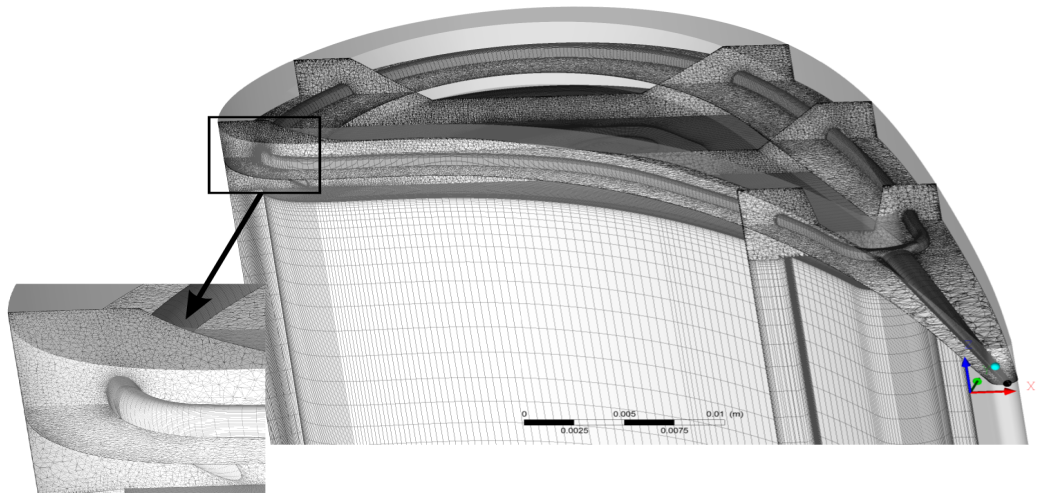


Figure 9.5: Shaped squealer hybrid grid used for conjugate study

study results for the conjugate mesh are shown in Figure 9.6. In contrast, the coarsest mesh (14.5 million cells) gave a change of 2%.

Likewise, the shaped variant of this geometry with an additional internal coolant passage, consisted of 24.4 million cells to establish a similar mesh independent solution. A refined mesh with 27.6 million cells gave a 0.2% overall change in tip surface temperature. The conjugate mesh used for the shaped variant was a hybrid type with mixed elements including hexahedra, tetrahedra and pyramids, displayed in Figure 9.5.

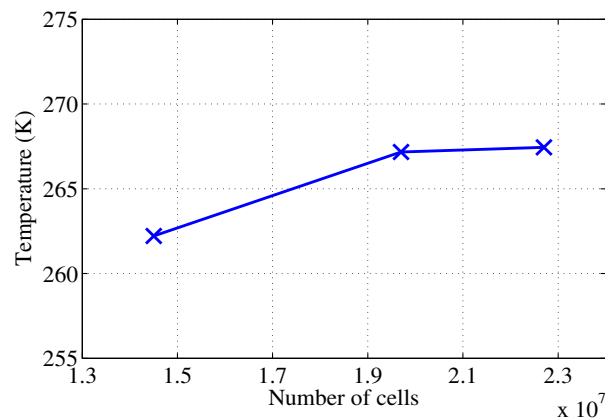


Figure 9.6: Area-averaged tip surface temperature for closed squealer conjugate meshes (Ansys CFX)

9.5 Significance of Biot Number

When quantifying heat transfer, the Biot number is a dimensionless group used to match the conductive resistance to the convective resistance of a solid body and is defined by:

$$Bi = \frac{hL}{k} \quad (9.4)$$

As a parameter used in many gas phase equations including Reynolds number, Nusselt number and Prandtl number and in solid phase equations such as with the Fourier number, the role of the Biot number should not be overlooked. Typically, in a gas turbine, the Biot number may have an expected value of approximately 0.5 based on a high convective heat transfer coefficient of $10,000 \text{ W/m}^2\text{K}$ and a length scale ($2L$) of 1 mm based on the blade wall thickness and a thermal conductivity of 10 W/mK . Whereas in the case of the simulated conjugate heat transfer for the scaled model, the Biot number has been calculated to be approximately 0.05 based on a high convective heat transfer coefficient of $2,000 \text{ W/m}^2\text{K}$ blade. Thus, it should be noted that the Biot number does not match between the tested blade and a real engine configuration blade and that the results should only be used as an indication of the expected solid temperature profiles. It is recommended that future calculations account for the change in Biot number between engine and test scenarios in order to broaden the understanding of turbine tip metal temperatures and their profiles along the tip.

9.6 Tip Heat Transfer Results

The conjugate heat transfer results for the nominal and shaped squealer are presented in Figure 9.7. By equipping the shaped tip with an extra cooling passages within

the rims provides a significant improvement in surface temperature in comparison to a nominal squealer without the passages. The targeted areas are the pressure and suction side rims and having an extra passages offers a global reduction in wall temperature over the rim regions. The conductive heat transfer effect also propagates towards the cavity floor.

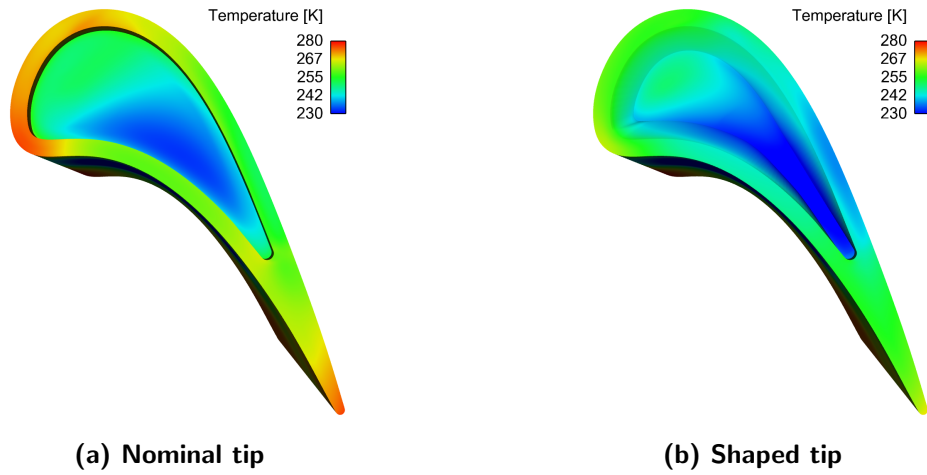


Figure 9.7: Tip surface temperature contours with SA turbulence model

To help understand the effect of the additional coolant passages within the solid, a cut plane is made through the solid to show how the temperature profiles compare for the two tip designs. These are shown in Figures 9.8 and 9.9. Even with an elliptic shaped passage equipped, the rims can show a significant response. Improvements to the passage shape may also reduce the surface temperature further by maximising the internal area available whilst maintaining wall thickness tolerances, such as, a passage with a quadrilateral shape. Figures 9.10 and 9.11 show the effect on the suction and pressure side rims and the results show potential local reductions to the tip temperature by approximately 10%. Importantly, there is a significant impact on the mid-axial chord rims. These regions are prone to over-heating and often noted as areas of fatigue.

The effect of turbulence modelling was also studied to test the sensitivity on the CHT results. The chosen model was $k - \omega$ SST. Unlike the uncoupled simulations,

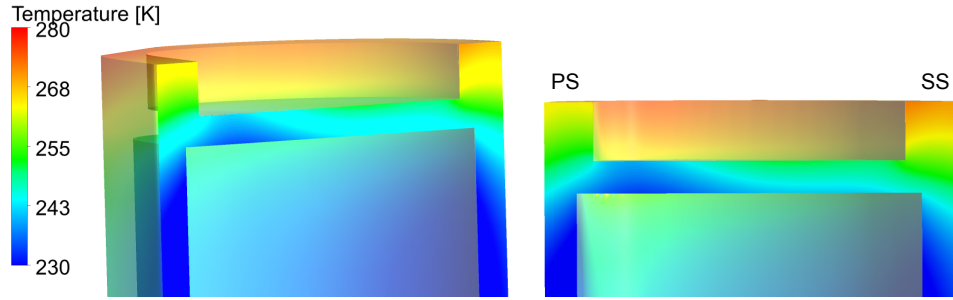


Figure 9.8: Nominal squealer temperature contours through solid (SA turbulence model)

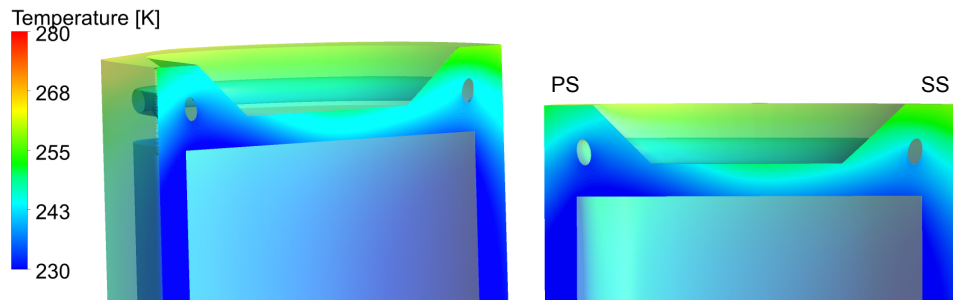


Figure 9.9: Shaped squealer temperature contours through solid (SA turbulence model)

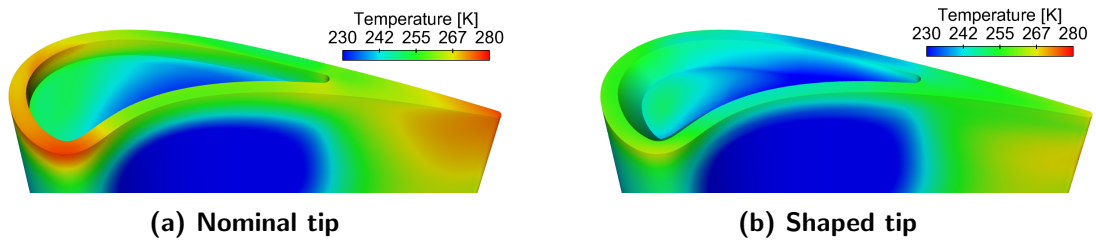


Figure 9.10: Suction side rim surface temperature contours with SA turbulence model

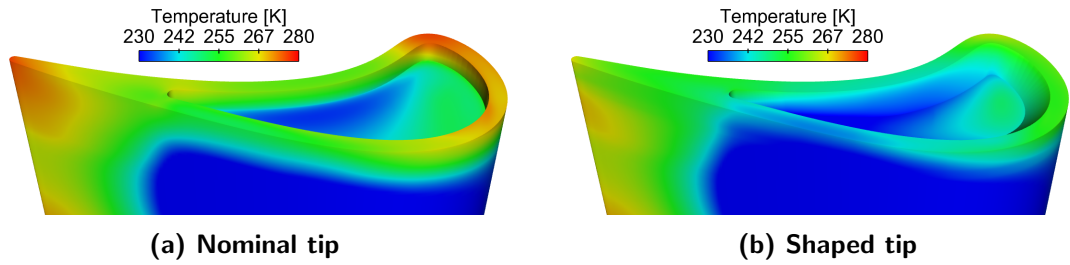


Figure 9.11: Pressure side rim surface temperature contours with SA turbulence model

the CHT results show less dependence. Contour maps of the tip surface temperature are presented in Figure 9.12. The maps resemble those as shown in Figure 9.7 and the difference is mostly a shift in the predicted temperature range. Apart from the temperature shift, which is actually an increase, the effect of the additional internal

cooling passages is similar. The overall percentage change of the temperature drop over the tip with the additional cooling passages using the SST model is 3.1% and 4.1% with the SA model.

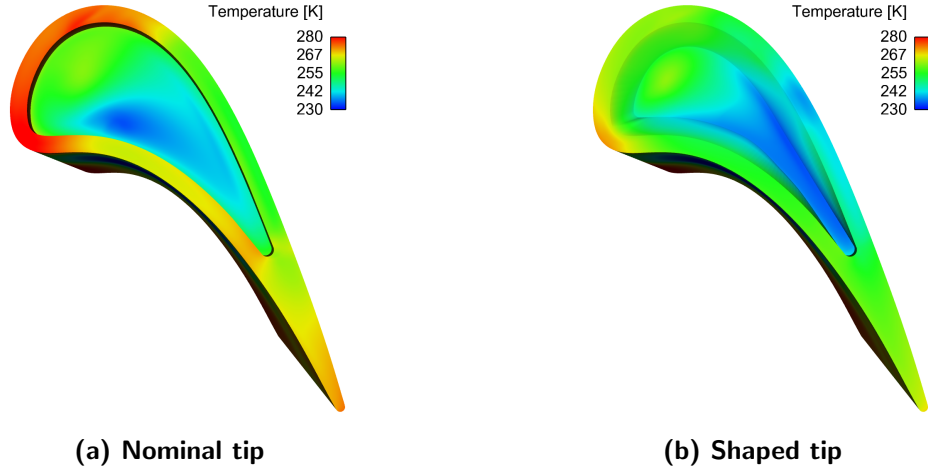


Figure 9.12: Tip surface temperature contours with $k - \omega$ SST turbulence model

The nominal squealer for both cases shows high thermal loads on the rims, typically these are classed as “burnout” regions. The main purpose of the shaped squealer is to house additional coolant passages, so it is of interest to understand the effects on the rim surfaces alone.

Extracting the top rim surface for both the nominal and shaped squealer tips as shown in Figure 9.13a, the calculated temperatures are circumferentially-averaged and plotted in Figures 9.13b and 9.13c for the SA and $k - \omega$ SST models, respectively.

The distribution of rim surface temperature is similar for both turbulence models, although the $k - \omega$ SST model shows noticeably higher predicted temperatures. Regardless of the turbulence model, the reduction in surface temperature is clear when an additional passage is equipped to a squealer tip with shaped rims. The significance of this result provides the desired outcome, where the goal was to better control the high thermal stresses on the rims. From an initial design perspective, being able to house such a passage shows great potential. The overall reduction in rim temperature the current design gives is 5.6% for the SA model and 4.8% for the $k - \omega$ SST model.

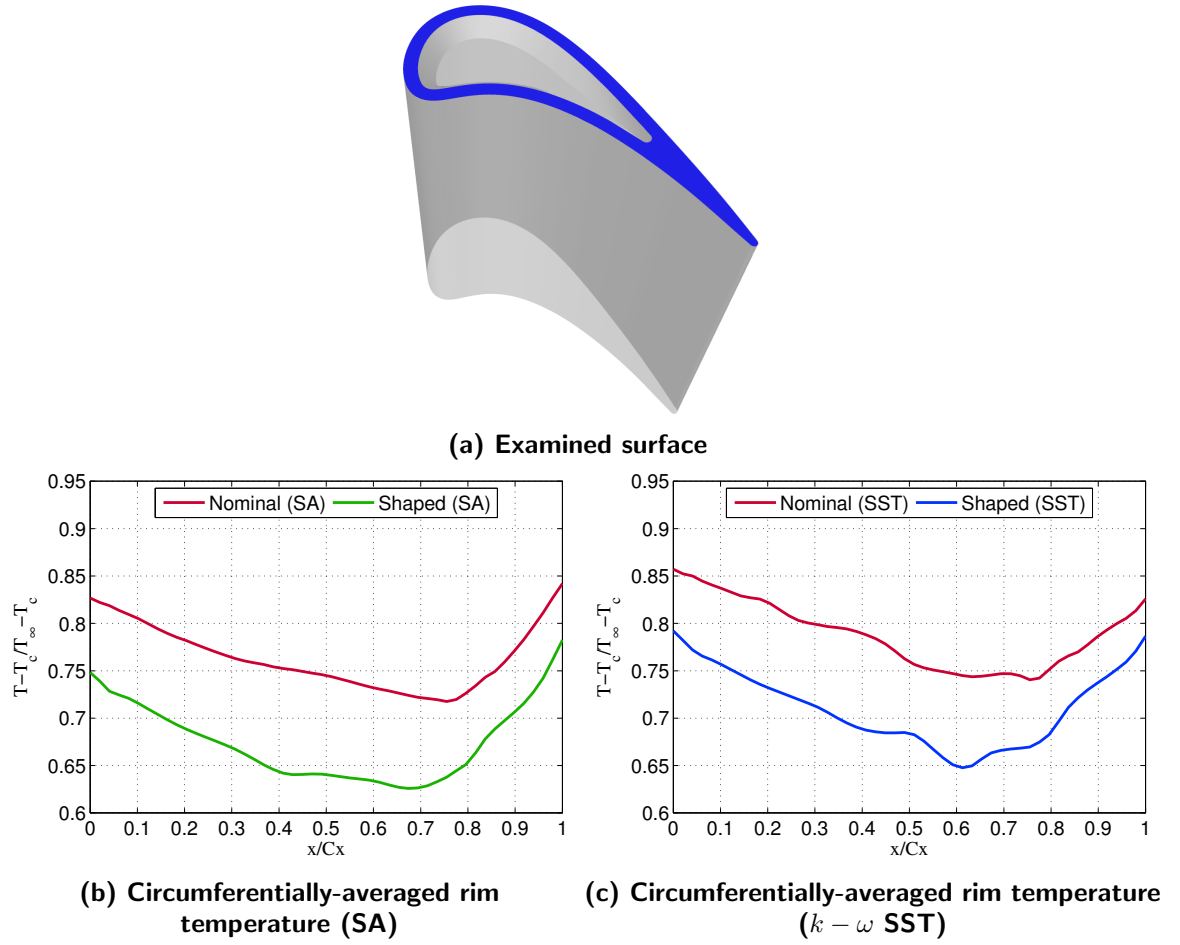


Figure 9.13: Turbulence model sensitivity

The conjugate heat transfer results show an insignificant dependence on turbulence modelling.

For all calculations, assumed thermal boundary conditions are supplied to the internal cooling chamber. One of the conditions as mentioned is a set internal heat transfer coefficient, h_i . In order to understand the dependence of the solution to the internal boundary conditions, additional cases have been run with a lower internal wall heat transfer coefficient of $400 \text{ W/m}^2\text{K}$. Using this condition, whilst maintaining the inlet-to-coolant temperature ratio, the tip surface temperature of the nominal and shaped squealer tip are shown in Figure 9.14.

As expected, the surface temperature reduction is less than that with a higher wall

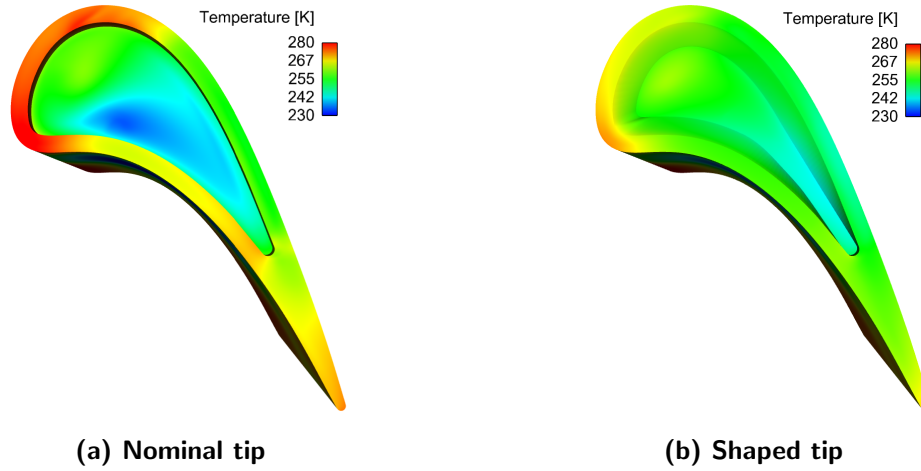


Figure 9.14: Tip surface temperature contours with $h_i = 400 \text{ W/m}^2\text{K}$ (SA)

heat transfer coefficient ($h_i = 700 \text{ W/m}^2\text{K}$), but importantly the additional cooling passage within the rims is still capable of reducing the surface wall temperature, particularly on the targeted rim areas. Overall, the shaped squealer tip surface absolute temperature reduces by 3.2% compared to the nominal squealer. Comparatively, the rim surface reduces by 3.9%. In both instances, the additional passages still prove to be beneficial. For the nominal case, the dependence on h_i is only slight. However, it is more noticeable for the shaped squealer where the set internal thermal conditions has more of an impact on the overall tip and rim temperatures. The distributions of rim surface temperature from the leading edge to the trailing edge for the nominal and shaped squealer with a lower internal heat transfer coefficient are plotted in Figure 9.15.

The present study provides an initial basis to improving the tip durability and improvements to the inlet of the coolant passage can be made, so too can the shape of the passage. Furthermore, film cooling fed by the additional passages may also be explored, an advantage that can also help provide better film coverage over the tip region itself. This is a common weakness for unshrouded tip design along with the understanding of cooled tip heat transfer. Generally speaking, there have been far fewer tip cooling studies conducted in comparison to vane and rotor blade cooling.

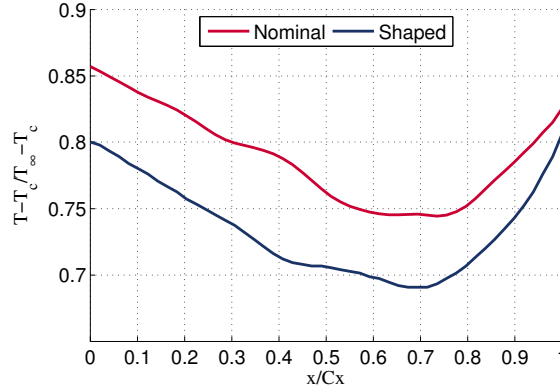


Figure 9.15: Circumferentially-averaged rim temperature with $h_i = 400 \text{ W/m}^2\text{K}$ (SA)

Increasing the understanding in tip cooling and engineering advanced cooling methods may greatly increase the lifespan of unshrouded turbine blade tips and ensure longer periods of maximum efficiency is gained during the in-service cycle.

9.6.1 Improved Cooling Passage Design

As mentioned, the primary goal is to maximise the internal cooling of turbine blade tips and one option is to improve the design of the passage. The previously considered design utilised an elliptical shaped passage. However, by considering a quadrilateral shaped passage, the internally cooled area can be maximised whilst still maintaining the minimum wall thickness required.

The updated design also improves the trailing edge cooling gallery by replacing the geometry with a slot, that is also more representative of a turbine blade, as well as improving the trailing edge cooling. The design of the trailing edge slot requires a landing edge in order to properly satisfy the wall thickness constraint. The updated shaped squealer with improved additional internal cooling passages is shown in Figure 9.16.

The CHT results for the modified coolant passage are shown in Figure 9.17. Note that present simulation utilises the $k-\omega$ SST turbulence model only, as the turbulence effect has been shown to be minimal. The results indicate a further reduction in

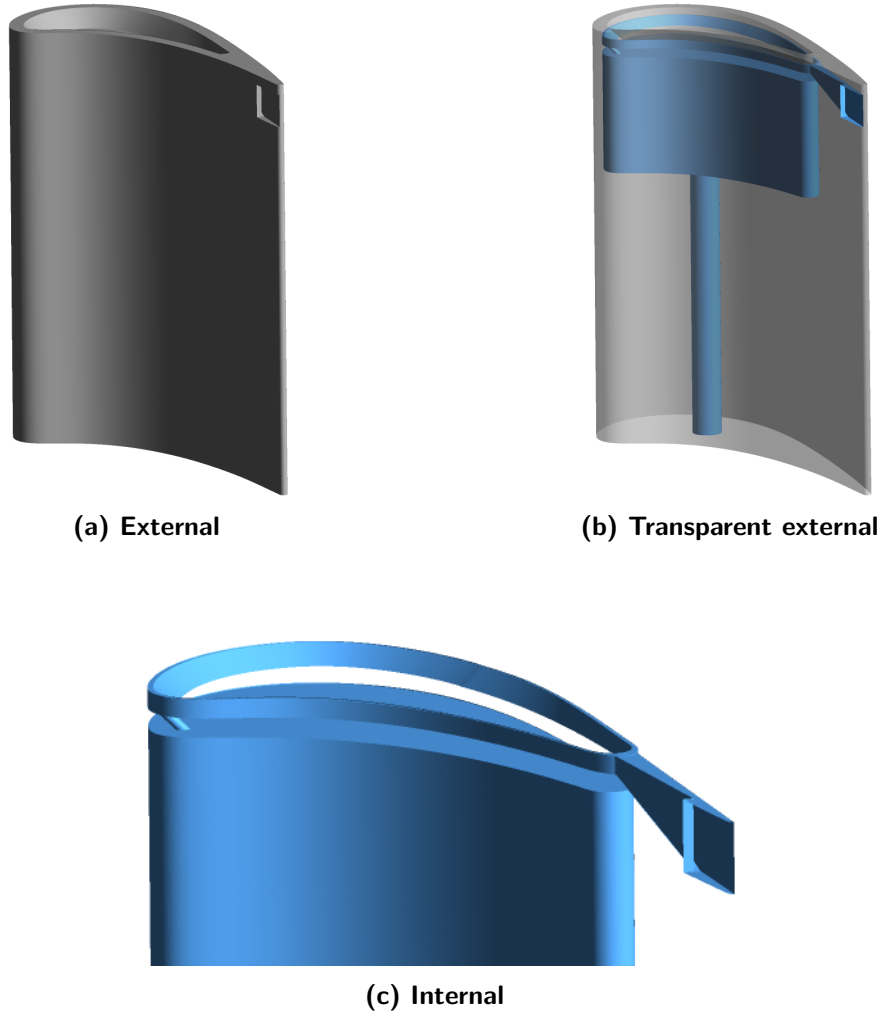


Figure 9.16: Improved internal coolant passage and trailing edge slot design

surface temperature can be achieved by maximising the interface area of the internal coolant passage. The targeted rim regions show a drop in surface temperature and notably, the modified passage reduces the leading edge tip corner heat transfer as well as significant reductions towards the trailing edge due to the improved slot design. The circumferentially-averaged rim surface temperatures are given in Figure 9.18.

The reduction in tip temperature shows a percentage change of 7.0% for the whole tip surface and 6.7% for the top rim surface only. The overall tip surface temperature shows the most significant improvement, giving over a two-fold difference. The rim surface also shows a reasonable improvement. The weight of these results when scaled

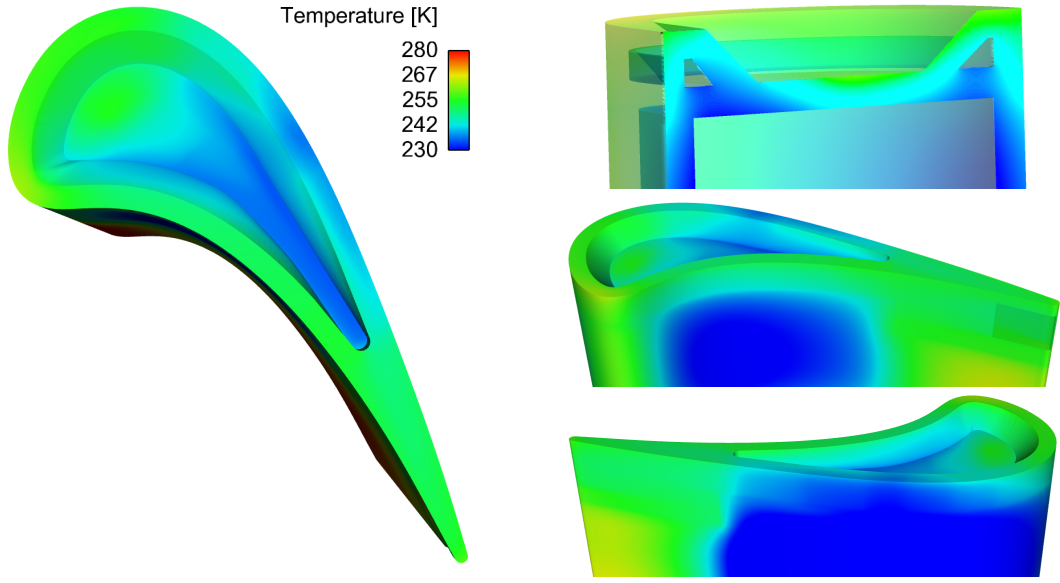


Figure 9.17: Tip surface and solid model temperature contours for modified internal cooling passages ($k - \omega$ SST)

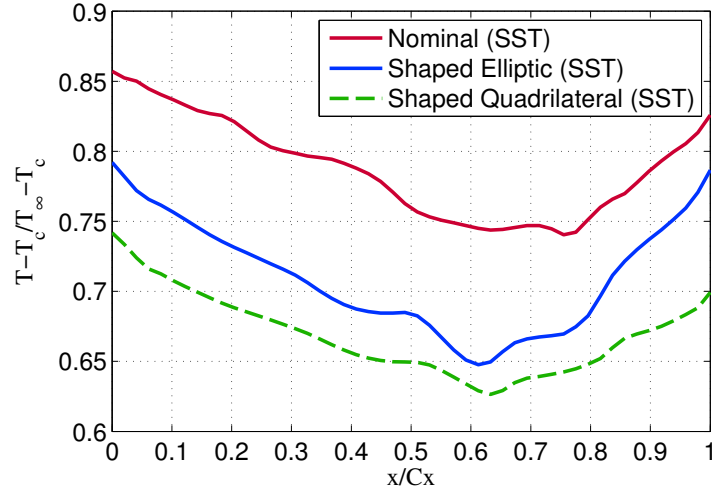


Figure 9.18: Circumferentially-averaged rim temperature comparison for nominal and shaped squealers with elliptic and quadrilateral internal cooling passages ($k - \omega$ SST)

to engine-realistic temperatures offers great potential benefits considering turbine inlet temperature are soaring above 1800°C mark. From an initial design perspective, the results show a possible configuration of managing the tip thermal stresses from a bottom-up approach. Naturally, design constraints and restrictions may hold, however the implementation of shaped squealer tips that can house additional internal cooling passages offer great potential in protecting squealer tips known to be exposed

to overheating and fatigue. Moreover, since the tip shaping effect on the aerodynamics has been shown to be minimal, the sought design can offer the high efficiency levels a typical squealer provides, but also a significant potential to protect the tip and increase its durability and life.

9.7 Summary

To summarise, a conjugate heat transfer CFD analysis has been performed to assess the tip metal temperature of a shaped squealer equipped with a proposed, near-tip internal cooling passage. The solution is solved at engine-representative aerodynamic conditions with an assumed coolant-to-mainstream temperature ratio and coolant wall heat transfer coefficient. The metal material is also chosen to a Nickel superalloy representing the state-of-the-art in gas-turbines.

The major conclusions are:

1. Simple additional internal cooling passages designed in conjunction with a shaped squealer has demonstrated its ability in lowering the surface temperature of the tip and rim surfaces. An approximate wall temperature reduction (T_w), of 5% can be achieved on the rims and 3-4% overall.
2. Improvements to the cooling passages provides greater reductions, a difference of 7% of T_w on both the rims and the overall tip surface is discovered. Noting that the Rolls-Royce Trent 900 operates at 1800°C, an equivalent surface temperature reduction is deemed significant.
3. The aim of designing a tip to help protect it from extreme gas temperatures and to promote durability has, from a preliminary perspective, been achieved.

Chapter 10

Conclusions and Recommendations

10.1 Overall Context

Gathered from the open literature, there has been a shift in the importance of the flow regime used to study unshrouded high-pressure turbine blades. Ideally, all research and data should be obtained from rotating rigs operating at high-speed in order to best replicate a real engine. However, obtaining data is often challenging and considerably limited. Measurements on the tip are often confined to only a few locations.

As a result, many studies have been undertaken within stationary platforms and at low-speed. Their offering of highly resolved and detailed data has meant that they are often preferred. For these reasons, low-speed cascades have become extremely popular and their results have strongly impacted the wisdom and understanding of the flow physics of unshrouded blade tips. However, as recently discovered, without matching engine-realistic conditions the true tip flow and physical features are missed and have a profound impact, particularly on turbine blade tip heat transfer. Taking into account the complexity and costs, ultimately, the next closest setups to high-speed rotating rigs are high-speed cascades since they can assist in capturing well resolved data.

Based on the requirements for highly resolved spatial measurements of the tip flow and blade surface heat transfer, the High-Speed Linear Cascade was developed to understand the behaviour of unshrouded blades at engine-representative aerodynamic conditions. The results obtained for a flat tip showed considerable differences to low-speed flow and to a large extent, showed reversed trends in tip heat transfer and going against the common conventional wisdom. These results were also in agreement with numerical CFD simulations.

Following the flat tip studies, approaches were taken to minimise the tip aerodynamics. At high-speed, this was done with the use of winglets and winglet-squealers. However, both tips were prone to high heat loads. A common approach used to improve efficiency and heat transfer, is by the use of a squealer; a flat tip with a cavity. The literature has reported its good overall performance, but the research of a squealer tip under a high-speed regime has been minimal. Also, to the author's knowledge, there are no detailed studies of a tip clearance study on high-speed squealer tips, the cooling performance at different tip gaps or the role of casing motion at different tip gaps in open literature.

10.2 Conclusions

10.2.1 Heat Transfer

Using infrared thermography and the Impulse Response method, the tip heat transfer of a transonic squealer was determined. The results revealed distinctive heat transfer patterns as the tip gap was varied. Importantly, this finding is new as the tip gap survey has not been undertaken for a squealer tip at high-speed. The results found that the frontal portion of the tip showed an increase in heat transfer coefficient with larger tip gaps. Also discovered was that towards the trailing edge region, there was a reversal in the heat transfer trend. Typically, the larger tip gap accounts for a

higher leakage mass flow rate over the tip and therefore, higher heat transfer. As the flow becomes choked, this becomes a limiter. Using a CFD analysis, the tip gap flow was shown to be supersonic in the rear portion and on the rims only, and also revealed the dependence of heat transfer with flow regime. Similar trends were found including that of a smaller tip gap being prone to higher heat transfer towards the trailing edge. The CFD results showed good agreement with those measured and in both cases, the heat transfer was ranked accordingly with tip gap.

Subsequently, this dependence was also found for a flat tip and showed similar trends with the over-tip leakage flow regime. Studying both the squealer and flat tips with a relative casing movement condition, the tip heat transfer was also ranked similarly. Both the squealer and flat tip with a moving casing condition showed the existence of supersonic tip flow. It was also shown that the supersonic region was mostly affected by the moving casing over the flat tip, whereas for the squealer tip it was more proportional, for both stationary and moving cases. Thus, showing a dependence on the geometry.

The main discovery with the moving casing was the geometrical influence on the heat transfer patterns; the squealer showed generally equally proportional changes of heat transfer, whilst the flat tip showed a large, non-proportional behaviour with tip gap variation. For this reason, the heat transfer was strongly affected at the small tip gap over the flat tip.

10.2.2 Aerodynamic Loss and Exit Flow Angles

An aerodynamic loss study was also conducted using using pressure measurements taken downstream of the blade row at two exit plane locations; half and one axial chord downstream. An expected trend of increased losses with larger tip gaps was observed. The numerical results also showed agreement with the experiments. However, at the half chord exit plane the magnitudes were less well predicted. The exit

flow angles showed the increase of flow turning with increased tip gaps coinciding with the increase in loss. Only fair agreement was found with the simulated results.

With a moving casing, the losses at one axial chord downstream showed a significant difference to the stationary case; the enhancement of the passage vortex was found showed little variation with gap height. However, the tip leakage vortices continued to be dependent on the gap. For the stationary case, the squealer was generally shown to be superior to a flat tip, with exception at the smallest tip gap where they are largely similar. With a moving casing, the performance of the squealer was shown to be superior to a flat tip at all tip gaps.

10.2.3 Film Cooling

Experimentally, applying film cooling to a squealer tip at high-speed showed overall low effectiveness with only the pressure side films having most impact. Within the squealer cavity, the film coverage is limited and seem to be slightly influenced by the tip gap. At larger tip gaps, streaks of high effectiveness were found. These reduce with tip gap and likely to be influenced by the mainstream flow.

Compared to the uncooled case, the aerodynamic loss was affected in two ways. The kinetic energy loss coefficient showed a reduction of loss at half an axial chord downstream of the blade row and an increase of loss at one axial chord downstream. These results also showed similar traits with the exit yaw angles measured. It is apparent that the coolant suppresses the tip leakage vortex initially, but contributes to higher mixing further downstream.

Numerically, a low-fidelity approach using source terms was used to predict the cooling effectiveness. Although it was able to predict the correct trends for both aerodynamic loss and film cooling effectiveness, the magnitudes were less well predicted.

10.2.4 Enhanced Internal Cooling by Tip Shaping

Shaped squealer tips were designed and tested to show their insensitivity to the tip aerodynamics. The results showed that for the near-tip region, the total pressure losses increased by, at most, 1.5% compared to a standard squealer tip when restricted to squealer rim angles greater than 45° from the top (horizontal) rim surface. Exit yaw angles also showed little variation in comparison to the standard squealer tested. CFD simulations also confirmed this result and showed less sensitivity.

As a consequence of the tip shaping, the reduction of tip area resulted in a reduction of the tip heat load, calculated by numerical simulations. These results offer the potential to reduce the tip heat transfer and confirmed the intentions of the shaping design. As far as the aerodynamic OTL flow losses and the external heat transfer are concerned, aero-thermal indifferences were found. Thus, these results support the approach to enhancement of the internal cooling by a shaped squealer rim.

Following the proof of concept in terms of the external aero-thermal indifferences, the 45° rim angle shaped squealer tip was modified by equipping extra cooling passages within the rims, only enabled by the increase in volume gained by a shaped squealer tip. The goal was to provide better protection of the rim surfaces for improved tip durability. Based on an initial conjugate heat transfer CFD study, the additional passages were able to provide a 7% reduction in metal temperature. Although considered small, it is to be noted that a 15 K reduction in wall temperature can double the lifespan of a blade (Friedrichs, 1997). Assuming a wall temperature of 1000 K, if a 5% reduction could be achieved, this would equate to a 50 K reduction and a considerable impact on the tip durability. However, it should be noted that there will be some significance of the Biot number between the modelled case and the engine case and these values should be likely to differ once matched. Nevertheless, the tip cooling concept is further benefited by a proven, aerodynamically efficient tip design: a squealer.

10.3 Recommendations for Future Work

Based on the contributions of the present research, the following suggestions are made for future studies on unshrouded turbine blade tips.

Firstly, on the experimental side it is imperative that the heater mesh setup is improved due to the high failure rate experienced during the later part of the present study. It is advised to consider a double heater mesh setup with them running in series to provide an adequate step-wise increase in mainstream temperature whilst reducing the thermal load on each mesh. Following this, the tip heat transfer of the shaped squealer tips should be measured to understand the predictive capabilities and check whether the obtained results are valid. A correction may be required for the angled rims, where the infrared energy may be less than expected.

Secondly, based on the squealer tip design, it is recommended to further develop the shaped squealer concept and understand possible means to further improve the aero-thermal performance. This may potentially be by incorporating a larger cavity that extends closer to the trailing edge or perhaps by introducing an opening in the form of an open or cut-back squealer.

Thirdly, since only an initial study has been conducted on improving the internal cooling of a squealer tip, it is highly recommended to investigate this further by incorporating a realistic internal passage. There is also potential for increased film cooling coverage on the tip using additional cooling passages and is recommended to consider possible means to do so. A plausible design can then be tested using the High-Speed Linear Cascade.

On the film cooling side, it is advised to thoroughly study the effect of pressure side and suction side cooling holes on the aerodynamic loss and tip heat transfer. This may initially be done through plugging cooling holes and examining the contributions, especially with regards to the loss generated. It is highly recommended to follow this study with a means to optimise the tip cooling arrangement as there is currently a

lack of research on cooled unshrouded tips. Incorporating tip abrasives or tip coatings and a study of tip welds that are present due to the manufacturing process would also be beneficial as it better replicates an engine-scenario and has not had much exposure in terms of research. Numerically, a higher fidelity method of predicting the film cooling is advised so that a better comparison can be made with experimental results and to help provide an in-depth analysis of the film cooling performance and how it can be improved for unshrouded blade tips.

Notably, the results obtained from the High-Speed Linear Cascade have been conducted in a stationary environment. Although extremely difficult, it may be possible to introduce a transparent rotating disc at high-speed (Kakade *et al.*, 2009). This setup would be able to provide an excellent representation of a real-engine whilst being able to provide a map of tip heat transfer. Not only would this data be useful, it would also help for CFD validation and tip development. It is also advised to use a higher-resolution infrared camera for better spatial maps of heat transfer. On the numerical side, a casing speed sensitivity study on the tip aero-thermal performance would be useful as the goal of achieving higher rotational speeds is desired. Furthermore, the effects of unsteadiness on a rotor blade tips in a high-pressure turbine stage would also be beneficial in enhancing the understanding of unshrouded blades.

Lastly, calibration of a four-hole probe would be beneficial for the aerodynamic loss studies as it would enable additional measurements of pitchwise yaw angles. Furthermore, the loss at half an axial chord downstream should be further examined to understand why the discrepancy between the experiments and the CFD exists. In addition, means to improve the PID controller to maintain longer steady inlet pressure would be useful for providing a higher resolution of the downstream aerodynamic loss as it is currently restricted by the run time.

Bibliography

- Acharya, S., & Moreaux, L. 2014. Numerical Study of the Flow Past a Turbine Blade Tip: Effect of Relative Motion Between Blade and Shroud. *Journal of Turbomachinery*, **136**(3), 031015.
- Acharya, S., Yang, H., Ekkad, S. V., Prakash, C., & Bunker, R. S. 2002. Numerical Simulation of Film Cooling on the Tip of a Gas Turbine Blade. *Pages 1051–1062 of: ASME Turbo Expo 2002: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Acharya, S., Yang, H., Prakash, C., & Bunker, R. S. 2003. Numerical Study of Flow and Heat Transfer on a Blade Tip With Different Leakage Reduction Strategies. *Pages 471–480 of: ASME Turbo Expo 2003, collocated with the 2003 International Joint Power Generation Conference*. American Society of Mechanical Engineers.
- Acharya, S., Kramer, G., Moreaux, L., & Nakamata, C. 2010. Squealer Tip Heat Transfer with Film Cooling. *Pages 1869–1877 of: ASME Turbo Expo 2010: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Ahn, J., Mhetras, S., & Han, J. C. 2005. Film-Cooling Effectiveness on a Gas Turbine Blade Tip Using Pressure-Sensitive Paint. *Journal of Heat Transfer*, **127**(5), 521–530.
- Allen, H. W., & Kofskey, M. G. 1955. Visualization Study of Secondary Flows in Turbine Tip Regions. *NASA Technical Note 3519*.

- Ameri, A. A. 2001. Heat Transfer and Flow on the Blade Tip of a Gas Turbine Equipped with a Mean-Camberline Strip. *Journal of Turbomachinery*, **123**(4), 704–708.
- Ameri, A. A., & Bunker, R. S. 2000. Heat Transfer and Flow on the First - Stage Blade Tip of a Power Generation Gas Turbine: Part 2 – Simulation Results. *Journal of Turbomachinery*, **122**(2), 272 – 277.
- Ameri, A. A., & Steinthorsson, E. 1995. Prediction of Unshrouded Rotor Blade Tip Heat Transfer. *ASME Paper No. 95-GT-142*.
- Ameri, A. A., Steinthorsson, E., & Rigby, D. L. 1998a. Effects of Tip Clearance and Casing Recess on Heat Transfer and Stage Efficiency in Axial Turbines. *ASME Paper No. 98-GT-369*.
- Ameri, A. A., Steinthorsson, E., & L., Rigby D. 1998b. Effect of Squealer Tip on Rotor Heat Transfer and Efficiency. *Journal of Turbomachinery*, **120**(4), 753–759.
- Anderson, E. 2015 (Jan). *Dubai Overtakes Heathrow to Become World's Busiest Airport*. [Online] Available from: <http://www.telegraph.co.uk> [Accessed 23 March 2015]. Accessed: Online.
- Anderson, J. D., & Wendt, J. F. 1995. *Computational Fluid Dynamics*. Vol. 206. Springer.
- Ansys, Inc. 2012. Ansys CFX - Solver Theory Guide, Release 14.5.
- Atkins, N. R., Thorpe, S. J., & Ainsworth, R. W. 2012. Unsteady Effects on Transonic Turbine Blade-Tip Heat Transfer. *Journal of Turbomachinery*, **134**(6), 061002.
- Azad, G. S., Han, J. C., Teng, S., & Boyle, R. J. 2000. Heat Transfer and Pressure Distributions on a Gas Turbine Blade Tip. *Journal of Turbomachinery*, **122**(4), 717 – 724.

- Azad, G. S., Han, J. C., & Boyle, R. J. 2000b. Heat Transfer and Flow on the Squealer Tip of a Gas Turbine Blade. *Journal of Turbomachinery*, **122**(4), 725–732.
- Azad, G. S., Han, J. C., Bunker, R. S., & Lee, C. P. 2002. Effect of Squealer Geometry Arrangement on a Gas Turbine Blade Tip Heat Transfer. *Journal of Heat Transfer*, **124**(3), 452–459.
- Behr, T., Kalfas, A. I., & Abhari, R. S. 2008. Control of Rotor Tip Leakage Through Cooling Injection From the Casing in a High-Work Turbine. *Journal of Turbomachinery*, **130**(3), 031014.
- Bindon, J. P. 1980. Exit Plane and Suction Surface Flows in an Annular Turbine Cascade with a Skewed Inlet Boundary Layer. *International Journal of Heat and Fluid Flow*, **2**(2), 57–66.
- Bindon, J. P. 1989. The Measurement and Formation of Tip Clearance Loss. *Journal of Turbomachinery*, **111**(3), 257 – 263.
- Booth, T. C. 1985. Importance of Tip Clearance Flows in Turbine Design. *VKI Lecture Series*, **5**, 1–34.
- Booth, T. C., Dodge, P. R., & K., Hepworth H. 1982. Rotor-Tip Leakage: Part I – Basic Methodology. *Journal of Engineering for Power*, **104**(1), 154 – 161.
- Bunker, R. S. 2001. A Review of Turbine Blade Tip Heat Transfer. *Annals of the New York Academy of Sciences*, **934**(1), 64–79.
- Bunker, R. S. 2013. Blade Tip Cooling Requirements and Design. *IGTI-VKI Lecture Series, Turbine Blade Tip Workshop, Hamburg*.
- Bunker, R. S., & Bailey, J. C. 2001. Effect of Squealer Cavity Depth and Oxidation on Turbine Blade Tip Heat Transfer. *Pages V003T01A038–V003T01A038 of: ASME*

- Turbo Expo 2001: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Bunker, R. S., Bailey, J. C., & Ameri, A. A. 2000. Heat Transfer and Flow on the First - Stage Blade Tip of a Power Generation Gas Turbine: Part 1 – Experimental Results. *Journal of Turbomachinety*, **122**(2), 263 – 271.
- Camci, C., Dey, D., & Kavurmacioglu, L. 2003. Tip Leakage Flows Near Partial Squealer Rims in an Axial Flow Turbine Stage. *ASME Paper No. GT2003-38979*.
- Chana, K. S., & Jones, T. V. 2003. An Investigation on Turbine Tip and Shroud Heat Transfer. *Journal of Turbomachinety*, **125**(3), 513–520.
- Chyu, M. K., Moon, H. K., & Metzger, D. E. 1989. Heat Transfer in the Tip Region of Grooved Turbine Blades. *Journal of Turbomachinety*, **111**(2), 131 – 138.
- Coleman, H. W., & Steele, W. G. 1999. *Experimentation and Uncertainty Analysis for Engineers*. John Wiley & Sons.
- Coull, J. D., & Atkins, N. R. 2015. The Influence of Boundary Conditions on Tip Leakage Flow. *Journal of Turbomachinety*, **137**(6), 061005.
- Coull, J. D., Atkins, N. R., & Hodson, H. P. 2014. Winglets for Improved Aerothermal Performance of High Pressure Turbines. *Journal of Turbomachinety*, **136**(9), 091007.
- Coull, J. D., Atkins, N. R., & Hodson, H. P. 2014b. High Efficiency Cavity Winglets for High Pressure Turbines. *Pages V02CT38A005–V02CT38A005 of: ASME Turbo Expo 2014: Turbine Technical Conference and Exposition*. American Society of Mechanical Engineers.
- da Silva, L. M., Tomita, J. T., Barbosa, J. R., & Bringhenti, C. 2014. Evaluation of Different Squealer Cavity Configurations in a HPT Blade Tip Region and its

- Influence on the Heat Transfer. *Pages V02CT38A034–V02CT38A034 of: ASME Turbo Expo 2014: Turbine Technical Conference and Exposition*. American Society of Mechanical Engineers.
- Day, C. R. B., Oldfield, M. L. G., & Lock, G. D. 1999. The Influence of Film Cooling on the Efficiency of an Annular Nozzle Guide Vane Cascade. *Journal of Turbomachinery*, **121**(1), 145–151.
- De Maesschalck, C., Lavagnoli, S., & Paniagua, G. 2013. Blade Tip Shape Optimization for Enhanced Turbine Aerothermal Performance. *Pages V03CT14A011–V03CT14A011 of: ASME Turbo Expo 2013: Turbine Technical Conference and Exposition*. American Society of Mechanical Engineers.
- De Maesschalck, C., Lavagnoli, S., & Paniagua, G. 2015. Blade Tip Carving Effects on the Aerothermal Performance of a Transonic Turbine. *Journal of Turbomachinery*, **137**(2), 021005.
- Denton, J. D. 1993. Loss Mechanisms in Turbomachines. *Journal of Turbomachinery*, **115**(4), 621 – 656.
- Dey, D., & Camci, C. 2000. Development of Tip Clearance Flow Downstream of a Rotor Blade with Coolant Injection From a Tip Trench. *Pages 572–579 of: Proceedings of the 8th ISROMAC Conference, Honolulu, Hawaii*.
- Dey, D., & Camci, C. 2001. Aerodynamic Tip Desensitization of an Axial Turbine Rotor using Tip Platform Extensions. *Pages V001T03A069–V001T03A069 of: ASME Turbo Expo 2001: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Dishart, P. T., & Moore, J. 1990. Tip Leakage Losses in a Linear Turbine Cascade. *Journal of Turbomachinery*, **112**(4), 599–608.

- Dring, R. P., & Joslyn, H. D. 1981. Measurement of Turbine Rotor Blade Flows. *Journal of Engineering for Gas Turbines and Power*, **103**(2), 400–405.
- Dunn, M. G., & Haldeman, C. W. 2000. Time-Averaged Heat Flux for a Recessed Tip, Lip, and Platform of a Transonic Turbine Blade. *Journal of Turbomachinery*, **122**(4), 692–698.
- Dunn, M. G., Rae, W. J., & Holt, J. L. 1984a. Measurement and Analyses of Heat Flux Data in a Turbine Stage: Part I Description of Experimental Apparatus and Data Analysis. *Journal of Engineering for Gas Turbines and Power*, **106**(1), 229–233.
- Dunn, M. G., Rae, W. J., & Holt, J. L. 1984b. Measurement and Analyses of Heat Flux Data in a Turbine Stage: Part II Discussion of Results and Comparison with Predictions. *Journal of Engineering for Gas Turbines and Power*, **106**(1), 234–240.
- Economic Benefits from Air Transport in the UK. 2011. *Economic Benefits from Air Transport in the UK*. Oxford Economics.
- El-Ghandour, M., Shatat, M. M. E., & Nakamura, Y. 2012. On the Physics of Vortex Formation at the Tip of a Turbine Blade. *Pages 1647–1654 of: ASME Turbo Expo 2012: Turbine Technical Conference and Exposition*. American Society of Mechanical Engineers.
- Fischer, A., König, J., Czarske, J., Rakenius, C., Schmid, G., & Schiffer, H. P. 2013. Investigation of the Tip Leakage Flow at Turbine Rotor Blades with Squealer Cavity. *Experiments in Fluids*, **54**(2), 1–15.
- Friedrichs, S. 1997. *Endwall Film-Cooling in Axial Flow Turbines*. Ph.D. thesis, University of Cambridge, Department of Engineering.

- Gao, Z., Narzary, D., Mhetras, S., & Han, J. C. 2009. Effect of Inlet Flow Angle on Gas Turbine Blade Tip Film Cooling. *Journal of Turbomachinery*, **131**(3), 031005.
- Han, J. C. 2013. Fundamental Gas Turbine Heat Transfer. *Journal of Thermal Science and Engineering Applications*, **5**(2), 021007.
- Han, J. C., Dutta, S., & Ekkad, S. V. 2012. *Gas Turbine Heat Transfer and Cooling Technology*. CRC Press.
- Harvey, N. W., & Ramsden, K. 2001. A Computational Study of a Novel Turbine Rotor Partial Shroud. *Journal of Turbomachinery*, **123**(3), 534–543.
- Harvey, N. W., Newman, D. A., Haselbach, F., & Willer, L. 2006. An Investigation Into a Novel Turbine Rotor Winglet: Part I - Design and Model Rig Test Results. *Pages 585–596 of: ASME Turbo Expo 2006: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- He, L. 2012 (Oct. 29). Tip Gap Dependence of Heat Transfer in Transonic Region. Turbine Tip Forum, Rolls-Royce, Bristol.
- He, L. 2013 (Oct. 3). “After-thought of 2013 ASME Turbine Blade Tip Symposium”, Personal Communication. Rolls-Royce/RAEng Chair of Computational Aerothermal Engineering, University of Oxford.
- He, L., & Oldfield, M. L. G. 2011. Unsteady Conjugate Heat Transfer Modeling. *Journal of Turbomachinery*, **133**(3), 031022.
- He, L., Zhang, Q., Wheeler, A. P. S., & Atkins, N. R. 2010. Over-Tip-Shaping for Heat Load Reduction. *UK Patent No. GB, 1017797*(October).
- He, L., Zhang, Q., & Rawlinson, A. J. 2013 (Dec. 9). *Turbine Blade*. Google Patents. US Patent App. 14/100,784.

- Heyes, F. J. G., Hodson, H. P., & Dailey, G. M. 1992. The Effect of Blade Tip Geometry on the Tip Leakage Flow in Axial Turbine Cascades. *Journal of Turbomachinery*, **114**(3), 643–651.
- Hirsch, C. 2007. *Numerical Computation of Internal and External Flows: The Fundamentals of Computational Fluid Dynamics*. Vol. 1. Butterworth-Heinemann.
- Hofer, T., & Arts, T. 2009. Aerodynamic Investigation of the Tip Leakage Flow for Blades With Different Tip Squealer Geometries at Transonic Conditions. *Pages 1051–1061 of: ASME Turbo Expo 2009: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Jørgensen, F. E. 2001. *How to Measure Turbulence with Hot-Wire Anemometers: A Practical Guide*. Dantec Dynamics.
- Kakade, V. U., Lock, G. D., Wilson, M., Owen, J. M., & Mayhew, J. E. 2009. Accurate Heat Transfer Measurements Using Thermochromic Liquid Crystal. Part 2: Application to a Rotating Disc. *International Journal of Heat and Fluid Flow*, **30**(5), 950 – 959.
- Key, N. L., & Arts, T. 2006. Comparison of Turbine Tip Leakage Flow for Flat Tip and Squealer Tip Geometries at High-Speed Conditions. *Journal of Turbomachinery*, **128**(2), 213–220.
- Kline, S. J., & McClintock, F. A. 1953. Describing Uncertainties in Single-Sample Experiments. *Mechanical Engineering*, **75**(1), 3–8.
- Krishnababu, S. K., Newton, P. J., Dawes, W. N., Lock, G. D., Hodson, H. P., Hannis, J., & Whitney, C. 2007a. Aero-Thermal Investigation of Tip Leakage Flows in Axial Flow Turbines. Part I - Effect of Tip Geometry and Tip Clearance Gap. *ASME Paper No. GT2007-27954*.

- Krishnababu, S. K., Dawes, W. N., Hodson, H. P., Lock, G. D., Hannis, J., & Whitney, C. 2007b. Aero-Thermal Investigation of Tip Leakage Flow in Axial Flow Turbines: Part II Effect of Relative Casing Motion. *ASME Paper No. GT2007-27957*.
- Krishnababu, S. K., Hodson, H. P., Dawes, W. N., Newton, S. J., & Lock, G. D. 2009. Numerical and Experimental Investigation of Tip Leakage Flow and Heat Transfer Using Idealised Rotor-Tip Models at Transonic Conditions. *Aeronautical Journal*, **113** (1141), 165–175.
- Kwak, J. S., & Han, J. C. 2003. Heat Transfer Coefficients and Film Cooling Effectiveness on the Squealer Tip of a Gas Turbine Blade. *Journal of Turbomachinery*, **125**(4), 648–657.
- Kwak, J. S., Ahn, J., Han, J. C., Lee, C. P., Boyle, R., Gaugler, R., & Bunker, R. S. 2003. Heat Transfer Coefficients on the Squealer Tip and Near Tip Regions of a Gas Turbine Blade with Single or Double Squealer. *Pages 681–690 of: ASME Turbo Expo 2003, collocated with the 2003 International Joint Power Generation Conference*. American Society of Mechanical Engineers.
- Lavagnoli, S., Paniagua, G., De Maesschalck, C., & Yasa, T. 2013. Analysis of the Unsteady OverTip Casing Heat Transfer in a High Speed Turbine. *Journal of Turbomachinery*, **135**(3), 031027.
- Ledezma, G. A., Allen, J., & Bunker, R. S. 2013. An Experimental and Numerical Investigation Into the Effects of Squealer Blade Tip Modifications on Aerodynamic Performance. *Pages V001T03A002–V001T03A002 of: ASME 2013 Turbine Blade Tip Symposium*. American Society of Mechanical Engineers.
- Li, W., Jiang, H., Zhang, Q., & Lee, S. W. 2014. Squealer Tip Leakage Flow Characteristics in Transonic Condition. *Journal of Engineering for Gas Turbines and Power*, **136**(4), 042601.

- Lomakin, N., Granovskiy, A., Belkanov, V., & Szwedowicz, J. 2013. Effect of Common Blade Tip Squealer Designs in Terms of Tip Clearance Loss Control. *Pages V001T03A005–V001T03A005 of: ASME 2013 Turbine Blade Tip Symposium*. American Society of Mechanical Engineers.
- Lu, K., Schobeiri, M. T., & Han, J. C. 2013. Numerical Simulation of Film Cooling on Rotating Blade Tips Within a High-Pressure Turbine. *Pages V03BT13A032–V03BT13A032 of: ASME Turbo Expo 2013: Turbine Technical Conference and Exposition*. American Society of Mechanical Engineers.
- Main, A. J., Day, C. R. B., Lock, G. D., & Oldfield, M. L. G. 1996. Calibration of a Four-Hole Pyramid Probe and Area Traverse Measurements in a Short-Duration Transonic Turbine Cascade Tunnel. *Experiments in Fluids*, **21**(4), 302–311.
- Matsunuma, T. 2006. Effects of Reynolds Number and Freestream Turbulence on Turbine Tip Clearance Flow. *Journal of Turbomachinery*, **128**(1), 166–177.
- Mayle, R. E., & Metzger, D. E. 1982. Heat Transfer at the Tip of an Unshrouded Turbine Blade. *Proceedings of the Seventh International Heat Transfer Conference*, **3**, 87 – 92.
- McCarter, A. A., Xiao, X., & Lakshminarayana, B. 2001. Tip Clearance Effects in a Turbine Rotor: Part II Velocity Field and Flow Physics. *Journal of Turbomachinery*, **123**(2), 305–313.
- Mee, D. J., Baines, N. C., Oldfield, M. L. G., & Dickens, T. E. 1992. An Examination of the Contributions to Loss on a Transonic Turbine Blade in Cascade. *Journal of Turbomachinery*, **114**(1), 155–162.
- Menter, F. R., Kuntz, M., & Langtry, R. 2003. Ten Years of Industrial Experience with the SST Turbulence Model. *Turbulence, Heat and Mass transfer*, **4**, 625–632.

- Metzger, D. E., Bunker, R. S., & Chyu, M. K. 1989. Cavity Heat Transfer on a Transverse Grooved Wall in a Narrow Flow Channel. *Journal of Heat Transfer*, **111**(1), 73 – 79.
- Metzger, D. E., Dunn, M. G., & Hah, C. 1991. Turbine Tip and Shroud Heat Transfer. *Journal of Turbomachinety*, **113**(3), 502 – 507.
- Mhetras, S., Yang, H., Gao, Z., & Han, J. C. 2005. Film-Cooling Effectiveness on Squealer Rim Walls and Squealer Cavity Floor of a Gas Turbine Blade Tip Using Pressure Sensitive Paint. *Pages 397–408 of: ASME Turbo Expo 2005: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Mhetras, S., Narzary, D., Gao, Z., & Han, J. C. 2008. Effect of a Cutback Squealer and Cavity Depth on Film-Cooling Effectiveness on a Gas Turbine Blade Tip. *Journal of Turbomachinery*, **130**(2), 021002.
- Mischo, B., Burdet, A., B., Thomas, & S., Abhari. R. 2007. Control of Rotor Tip Leakage Through Cooling Injection From the Casing in a High-Work Turbine: Computational Investigation Using a Feature-Based Jet Model. *ASME Paper No. GT2007-27669*.
- Mischo, B., Behr, T., & Abhari, R. S. 2008. Flow Physics and Profiling of Recessed Blade Tips: Impact on Performance and Heat Load. *Journal of Turbomachinery*, **130**(2), 021008.
- Moffat, R. J. 1988. Describing the Uncertainties in Experimental Results. *Experimental Thermal and Fluid Science*, **1**(1), 3–17.
- Moinier, P., & Giles, M. B. 1998. Preconditioned Euler and Navier-Stokes Calculations on Unstructured Grids. *In: 6th ICFD Conference on Numerical Methods for Fluid Dynamics, Oxford, UK, March*.

- Molter, S. M., Dunn, M. G., Haldeman, C. W., Bergholz, R. F., & Vitt, P. 2006. Heat-Flux Measurements and Predictions for the Blade Tip Region of a High-Pressure Turbine. *Pages 49–60 of: ASME Turbo Expo 2006: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Moore, J., & Elward, K. M. 1993. Shock Formation in Overexpanded Tip Leakage Flow. *Journal of Turbomachinery*, **115**, 392 – 399.
- Moore, J., & Tilton, J. S. 1988. Tip Leakage Flow in a Linear Turbine Cascade. *Journal of Turbomachinery*, **110**(1), 18–26.
- Moore, J., Moore, J. G., Henry, G. S., & Chaudhry, U. 1989. Flow and Heat Transfer in Turbine Tip Gaps. *Journal of Turbomachinery*, **111**(3), 301 – 309.
- Morphis, G., & Bindon, J. P. 1988. The Effects of Relative Motion, Blade Edge Radius and Gap Size on the Blade Tip Pressure Distribution in an Annular Turbine Cascade with Clearance. *Page 1988 of: ASME, Gas Turbine and Aeroengine Congress and Exposition, Amsterdam, Netherlands*.
- Narzary, D., Liu, K., Han, J. C., Mhetras, S., & Landis, K. 2014. Turbine Blade Tip Film-Cooling and Heat Transfer Measurements at High Blowing Ratios. *Pages V05BT13A025–V05BT13A025 of: ASME Turbo Expo 2014: Turbine Technical Conference and Exposition*. American Society of Mechanical Engineers.
- Nasir, H., Ekkad, S. V., Bunker, R. S., & Prakash, C. 2004a. Effects of Tip Gap Film Injection from Plain and Squealer Blade Tips. *ASME Paper No. GT2004-53455*.
- Nasir, H., Ekkad, S. V., Bunker, R. S., & Prakash, C. 2004b. Effects of Tip Gap Film Injection From Plain and Squealer Blade Tips. *Pages 419–427 of: ASME Turbo Expo 2004: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.

- Nasir, H., Ekkad, S. V., & Bunker, R. S. 2007. Effect of Tip and Pressure Side Coolant Injection on Heat Transfer Distributions for a Plane and Recessed Tip. *Journal of Turbomachinery*, **129**(1), 151–163.
- Nathan, S. 2011 (Jun). *Composite Class: Developing the Airbus A350-XWB*. [Online] <http://www.theengineer.co.uk/in-depth/the-big-story/composite-class-developing-the-airbus-a350-xwb/1009059.article> [Accessed 6 April 2015]. Accessed: Online.
- Newton, P. J., Lock, G. D., Krishnababu, S. K., Hodson, H. P., Dawes, W. N., Hannis, J., & Whitney, C. 2006. Heat Transfer and Aerodynamics of Turbine Blade Tips in a Linear Cascade. *Journal of Turbomachinery*, **128**(2), 300 – 309.
- Newton, P. J., Lock, G. D., Krishnababu, S. K., Hodson, H. P., Dawes, W. N., Hannis, J., & Whitney, C. 2007. Aero-Thermal Investigation of Tip Leakage Flow in Axial Flow Turbines: Part III Tip Cooling. *Pages 317–327 of: ASME Turbo Expo 2007: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Nho, Y. C, Park, J. S., Lee, Y. J., & Kwak, J. S. 2012. Effects of Turbine Blade Tip Shape on Total Pressure Loss and Secondary Flow of a Linear Turbine Cascade. *International Journal of Heat and Fluid Flow*, **33**(1), 92–100.
- Norris, G. 2013 (Feb). *A350 Trent XWB Fuel Consumption Improves in Tests*. [Online] <http://aviationweek.com/awin/a350-trent-xwb-fuel-consumption-improves-tests> [Accessed 6 April 2015]. Accessed: Online.
- O'Dowd, D. O. 2010. *Aero-Thermal Performance of Transonic High-Pressure Turbine Blade Tips*. Ph.D. thesis, University of Oxford, Department of Engineering.
- O'Dowd, D. O., Zhang, Q., Usandizaga, I., He, L., & Ligrani, P. M. 2010. Transonic Turbine Blade Tip Aero-Thermal Performance With Different Tip Gaps: Part II

- Tip Aerodynamic Loss. *Pages 347–356 of: ASME Turbo Expo 2010: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- O'Dowd, D. O., Zhang, Q., He, L., Oldfield, M. L. G., Ligrani, P. M., Cheong, B. C. Y., & Tibbott, I. 2011a. Aerothermal Performance of a Winglet at Engine Representative Mach and Reynolds Numbers. *Journal of Turbomachinery*, **133**(4), 041026.
- O'Dowd, D. O., Zhang, Q., He, L., Ligrani, P. M., & Friedrichs, S. 2011b. Comparison of Heat Transfer Measurement Techniques on a Transonic Turbine Blade Tip. *Journal of Turbomachinery*, **133**(2), 10 pages.
- O'Dowd, D. O., Zhang, Q., He, L., Cheong, B. C. Y., & Tibbott, I. 2013. Aerothermal Performance of a Cooled Winglet at Engine Representative Mach and Reynolds Numbers. *Journal of Turbomachinery*, **135**(1), 011041.
- Oldfield, M. L. G. 2008. Impulse Response Processing of Transient Heat Transfer Gauge Signals. *Journal of Turbomachinery*, **130**(2), 9 pages.
- Oldfield, M. L. G., Jones, T. V., & Schultz, D. L. 1978. On-line Computer for Transient Turbine Cascade Instrumentation. *Transactions on Aerospace and Electronic Systems*, 738–749.
- Palafox, P., Oldfield, M. L. G., LaGraff, J. E., & Jones, T. V. 2008. PIV Maps of Tip Leakage and Secondary Flow Fields on a Low Speed Turbine Blade Cascade with Moving Endwall. *Journal of Turbomachinery*, **130**(1), 9 pages.
- Palafox, P., Oldfield, M. L. G., Ireland, P.T., Jones, T. V., & LaGraff, J. E. 2012. Blade Tip Heat Transfer and Aerodynamics in a Large Scale Turbine Cascade With Moving Endwall. *Journal of Turbomachinery*, **134**(2), 021020.

- Papa, M., Goldstein, R. J., & Gori, F. 2002. Effects of Tip Geometry and Tip Clearance on the Mass/Heat Transfer from a Large-Scale Gas Turbine Blade. *Pages 285–293 of: ASME Turbo Expo 2002: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Park, J. S., Lee, D. H., Rhee, D. H., Kang, S. H., & Cho, H. H. 2014. Heat Transfer and Film Cooling Effectiveness on the Squealer Tip of a Turbine Blade. *Energy*, **72**, 331–343.
- Patel, K. V. 1980. *Research on a High Work Axial Gas Generator Turbine*. Tech. rept. SAE Technical Paper.
- Payne, S. J., Ainsworth, R. W., Miller, R. J., Moss, R. W., & Harvey, N. W. 2003. Unsteady Loss in a High Pressure Turbine Stage. *International Journal of Heat and Fluid Flow*, **24**(5), 698–708.
- Pollock, T. M., & Tin, S. 2006. Nickel-Based Superalloys for Advanced Turbine Engines: Chemistry, Microstructure and Properties. *Journal of Propulsion and Power*, **22**(2), 361–374.
- Povey, T., Oldfield, M. L. G., & Haselbach, F. 2008. Transonic Turbine Vane Tests in a New Miniature Cascade Facility. *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, **222**(5), 529–539.
- Rains, D. A. 1954. *Tip Clearance Flows in Axial Flow Compressors and Pumps*. Ph.D. thesis, California Institute of Technology.
- Rao, N. M., Gumusel, B., Kavurmacioglu, L., & Camci, C. 2006. Influence of Casing Roughness on the Aerodynamic Structure of Tip Vortices in an Axial Flow Turbine. *Pages 893–903 of: ASME Turbo Expo 2006: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.

- Rolls-Royce. 2005. *The Jet Engine*. 6th edn. Rolls-Royce plc.
- Saha, A. K., Acharya, S., Bunker, R. S., & Prakash, C. 2006. Blade Tip Leakage Flow and Heat Transfer with Pressure-Side Winglet. *International Journal of Rotating Machinery*, **2006**.
- Saxena, V., & Ekkad, S. V. 2004. Effect of Squealer Geometry on Tip flow and Heat Transfer for a Turbine Blade in a Low-Speed Cascade. *Journal of Heat Transfer*, **126**(4), 546–553.
- Schabowski, Z., & Hodson, H. P. 2014. The Reduction of Over Tip Leakage Loss in Unshrouded Axial Turbines using Winglets and Squealers. *Journal of Turbomachinery*, **136**(4), 041001.
- Schabowski, Z., Hodson, H. P., Giacche, D., Power, B., & Stokes, M. R. 2014a. Aeromechanical Optimization of a Winglet-Squealer Tip for an Axial Turbine. *Journal of Turbomachinery*, **136**(7), 071004.
- Schabowski, Z., Hodson, H. P., Giacche, D., Power, B., & Stokes, M. R. 2014b. Aeromechanical Optimization of a Winglet-Squealer Tip for an Axial Turbine. *Journal of Turbomachinery*, **136**(7), 071004.
- Schneider, P. J. 1955. *Conduction Heat Transfer*. Addison-Wesley Pub. Co.
- Shavalikul, A., & Camci, C. 2008. A Comparative Analysis of Pressure Side Extensions for Tip Leakage Control in Axial Turbines. *Pages 1279–1288 of: ASME Turbo Expo 2008: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Shyam, V., & Ameri, A. A. 2011a. Comparison of Various Supersonic Turbine Tip Designs to Minimize Aerodynamic Loss and Tip Heating. *Pages 887–895 of: ASME*

- 2011 Turbo Expo: Turbine Technical Conference and Exposition*. American Society of Mechanical Engineers.
- Shyam, V., & Ameri, A. A. 2011b. Comparison of Various Supersonic Turbine Tip Designs to Minimize Aerodynamic Loss and Tip Heating. *Pages 887–895 of: ASME 2011 Turbo Expo: Turbine Technical Conference and Exposition*. American Society of Mechanical Engineers.
- Shyam, V., Ameri, A. A., & Chen, J. P. 2012. Analysis of Unsteady Tip and Endwall Heat Transfer in a Highly Loaded Transonic Turbine Stage. *Journal of Turbomachinery*, **134**(4), 041022.
- Sjolander, S. A., & Cao, D. 1995. Measurements of the Flow in an Idealized Turbine Tip Gap. *Journal of Turbomachinery*, **117**(4), 578 – 584.
- Srinivasan, V., & Goldstein, R. J. 2003. Effect of Endwall Motion on Blade Tip Heat Transfer. *Journal of Turbomachinery*, **125**(2), 267–273.
- Starke, C., Janke, E., Hofer, T., & Lengani, D. 2008. Comparison of a Conventional Thermal Analysis of a Turbine Cascade to a Full Conjugate Heat Transfer Computation. *Pages 1013–1024 of: ASME Turbo Expo 2008: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Tallman, J. A., & Lakshminarayana, B. 2001a. Numerical Simulation of Tip Leakage Flows in Axial Flow Turbines, With Emphasis on Flow Physics: Part I Effect of Tip Clearance Height. *Journal of Turbomachinery*, **123**(2), 314 – 323.
- Tallman, J. A., & Lakshminarayana, B. 2001b. Numerical Simulation of Tip Leakage Flows in Axial Flow Turbines, with Emphasis on Flow Physics: Part II – Effect of Outer Casing Relative Motion. *Journal of Turbomachinery*, **123**(2), 324–333.

- Tallman, J. A., Haldeman, C. W., Dunn, M. G., Tolpadi, A. K., & Bergholz, R. F. 2009. Heat Transfer Measurements and Predictions for a Modern, High-Pressure, Transonic Turbine, Including Endwalls. *Journal of Turbomachinery*, **131**(2), 021001.
- Tang, B. M. T. 2011. *Unshrouded Turbine Blade Tip Heat Transfer and Film Cooling*. Ph.D. thesis, University of Oxford, Department of Engineering.
- Tang, B. M. T., Palafox, P., Gillespie, D. R. H., Oldfield, M. L. G., & Cheong, B. C. Y. 2010. Computational Modelling of Tip Heat Transfer to a Super-Scale Model of an Unshrouded Gas Turbine Blade. *Journal of Turbomachinery*, **132**(3), 8 pages.
- Teloxa, J., Narzary, D., Sierra, F. Z., Lei, J., Carrillo, F., & Han, J. C. 2013. Measurement and Modelling of Blade Tip Cooling in Gas Turbines. *Pages V03BT13A034–V03BT13A034 of: ASME Turbo Expo 2013: Turbine Technical Conference and Exposition*. American Society of Mechanical Engineers.
- Thorpe, S. J., Yoshino, S., Ainsworth, R. W., & Harvey, N. W. 2004a. An Investigation of the Heat Transfer and Static Pressure on the Over-Tip Casing Wall of an Axial Turbine Operating at Engine Representative Flow Conditions. (I). Time-Mean Results. *International Journal of Heat and Fluid Flow*, **25**(6), 933–944.
- Thorpe, S. J., Yoshino, S., Ainsworth, R. W., & Harvey, N. W. 2004b. An Investigation of the Heat Transfer and Static Pressure on the Over-Tip Casing Wall of an Axial Turbine Operating at Engine Representative Flow Conditions. (II). Time-Resolved Results. *International Journal of Heat and Fluid Flow*, **25**(6), 945–960.
- Thorpe, S. J., Yoshino, S., Thomas, G. A., Ainsworth, R. W., & Harvey, N. W. 2005. Blade-Tip Heat Transfer in a Transonic Turbine. *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, **219**(6), 421–430.

- Virdi, A. S., Zhang, Q., He, L., Li, H. D., & Hunsley, R. 2015. Aerothermal Performance of Shroudless Turbine Blade Tips with Relative Casing Movement Effects. *Journal of Propulsion and Power*, 1–10.
- Vogel, G., Agrawal, A., & Nannapaneni, P. 2009. Turbine Blade Tip Cooling Modeling and Optimization. *Pages 139–147 of: ASME Turbo Expo 2009: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Wadia, A. R., & Booth, T. C. 1982. Rotor-Tip Leakage: Part II - Design Optimization through Viscous Analysis and Experiment. *Journal of Engineering for Power*, **104**(1), 162 – 169.
- Wang, Z., Zhang, Q., Liu, Y., & He, L. 2015. Impact of Cooling Injection on the Transonic Over-Tip Leakage Flow and Squealer Aerothermal Design Optimization. *Journal of Engineering for Gas Turbines and Power*, **137**(6), 062603.
- Wheeler, A. P. S., & Saleh, Z. 2013. Effect of Cooling Injection on Transonic Tip Flows. *Journal of Propulsion and Power*, **29**(6), 1374–1381.
- Wheeler, A. P. S., & Sandberg, R. D. 2013. Direct Numerical Simulations of a Transonic Tip Flow With Free-Stream Disturbances. *Pages V001T01A002–V001T01A002 of: ASME 2013 Turbine Blade Tip Symposium*. American Society of Mechanical Engineers.
- Wheeler, A. P. S., Atkins, N. R., & He, L. 2011. Turbine Blade Tip Heat Transfer in Low Speed and High Speed Flows. *Journal of Turbomachinery*, **133**(4), 041025.
- Wheeler, A. P. S., Korakianitis, T., & Banneheke, S. 2013. Tip-Leakage Losses in Subsonic and Transonic Blade Rows. *Journal of Turbomachinery*, **135**(1), 011029.
- Xiao, X., McCarter, A. A., & Lakshminarayana, B. 2001. Tip Clearance Effects in

- a Turbine Rotor: Part I - Pressure Field and Loss. *Journal of Turbomachinery*, **123**(2), 296–304.
- Yamamoto, A. 1989. Endwall Flow/Loss Mechanisms in a Linear Turbine Cascade with Blade Tip Clearance. *Journal of Turbomachinery*, **111**(3), 264–275.
- Yamamoto, A., Tominaga, J., Matsunuma, T., & Outa, E. 1994. Detailed Measurements of Three-Dimensional Flows and Losses Inside an Axial Turbine Rotor. *ASME Paper No. 94-GT-348*.
- Yang, D., Yu, X., & Feng, Z. 2010. Investigation of Leakage Flow and Heat Transfer in a Gas Turbine Blade Tip with Emphasis on the Effect of Rotation. *Journal of Turbomachinery*, **132**(4), 041010.
- Yang, H., Acharya, S., Ekkad, S. V., Prakash, C., & Bunker, R. S. 2002. Numerical Simulation of Flow and Heat Transfer Past a Turbine Blade with a Squealer-Tip. *Pages 295–307 of: ASME Turbo Expo 2002: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Yang, H., Chen, H. C., & Han, J. C. 2004. Numerical Prediction of Film Cooling and Heat Transfer With Different Film-Hole Arrangements on the Plane and Squealer Tip of a Gas Turbine Blade. *Pages 177–187 of: ASME Turbo Expo 2004: Power for Land, Sea, and Air*. American Society of Mechanical Engineers.
- Yang, H., Chen, H. C., & Han, J. C. 2006. Turbine Rotor with Various Tip Configurations Flow and Heat Transfer Prediction. *Journal of Thermophysics and Heat Transfer*, **20**(1), 80–91.
- Yang, T. T., & Diller, T. E. 1995. Heat Transfer and Flow for a Grooved Turbine Blade Tip in a Transonic Cascade. *ASME Paper No. 95-WA/HT-29*, 8.

- Yaras, M., Yingkang, Z., & Sjolander, S. A. 1989. Flow Field in the Tip Gap of a Planar Cascade of Turbine Blades. *Journal of Turbomachinery*, **111**(3), 276 – 283.
- Yaras, M. I., & Sjolander, S. A. 1991. Measurements of the Effects of Winglets on Tip-Leakage Losses in a Linear Turbine Cascade. *Pages 127–135 of: International Symposium on Air Breathing Engines, 10 th, Nottingham, England.*
- Yaras, M. I., & Sjolander, S. A. 1992a. Effects of Simulated Rotation on Tip Leakage in a Planar Cascade of Turbine Blades: Part I – Tip Gap Flow. *Journal of Turbomachinery*, **114**(3), 652 – 659.
- Yaras, M. I., & Sjolander, S. A. 1992b. Prediction of Tip-Leakage Losses in Axial Turbines. *Journal of Turbomachinery*, **114**(1), 204–210.
- Yaras, M. I., Sjolander, S. A., & Kind, R. J. 1992b. Effects of Simulated Rotation on Tip Leakage in a Planar Cascade of Turbine Blades: Part II – Downstream Flow Field and Blade Loading. *Journal of Turbomachinery*, **114**(3), 660 – 667.
- Zhang, D. H., Zeng, M., & Wang, Q. W. 2009. The Influence of Rotating Speed on Film Cooling Characteristics on GE-E3 Blade Tip With Different Tip Configurations. *Pages 1017–1025 of: ASME Turbo Expo 2009: Power for Land, Sea, and Air.* American Society of Mechanical Engineers.
- Zhang, Q. 2012. Personal Communication. Lecturer, Department of Mechanical Engineering and Aeronautics, City University London.
- Zhang, Q., & He, L. 2011. Over-Tip Choking and Its Implications on Turbine Blade Tip Aerodynamic Performance. *Journal of Propulsion and Power*, **27**(5).
- Zhang, Q., & He, L. 2013. Tip-Shaping for HP Turbine Blade Aerothermal Performance Management. *Journal of Turbomachinery*, **135**(5), 051025.

- Zhang, Q., & He, L. 2014. Impact of Wall Temperature on Turbine Blade Tip Aerothermal Performance. *Journal of Engineering for Gas Turbines and Power*, **136**(5), 052602.
- Zhang, Q., O'Dowd, D. O., He, L. Oldfield, M. L. G., & Ligrani, P. M. 2010a. Transonic Turbine Blade Tip Aero-Thermal Performance with Different Tip Gaps: Part I - Tip Heat Transfer. *Journal of Turbomachinery*, **133**(4), 9 pages.
- Zhang, Q., O'Dowd, D. O., He, L., Wheeler, A. P. S., Ligrani, P. M., & Cheong, B. C. Y. 2011. Overtip Shock Wave Structure and Its Impact on Turbine Blade Tip Heat Transfer. *Journal of Turbomachinery*, **133**(4).
- Zhang, Q., He, L., & Rawlinson, A. 2014. Effects of Inlet Turbulence and End-Wall Boundary Layer on Aerothermal Performance of a Transonic Turbine Blade Tip. *Journal of Engineering for Gas Turbines and Power*, **136**(5), 052603.
- Zhou, C., & Hodson, H. P. 2011. The Tip Leakage Flow of an Unshrouded High Pressure Turbine Blade with Tip Cooling. *Journal of Turbomachinery*, **133**(4), 041028.
- Zhou, C., Hodson, H. P., Tibbott, I., & Stokes, M. 2012. Effects of Endwall Motion on the Aero-thermal Performance of a Winglet Tip in a HP Turbine. *Journal of Turbomachinery*, **134**(6), 061036.
- Zhou, C., Hodson, H. P., Tibbott, I., & Stokes, M. 2013. The Aerothermal Performance of a Cooled Winglet Tip in a High Pressure Turbine Cascade. *Journal of Turbomachinery*, **135**(3), 031005.