Heat Transfer and Film Cooling on a Gas Turbine Blade and Shroud

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HEAT TRANSFER AND FILM COOLING ON A GAS TURBINE BLADE AND SHROUD

A Dissertation

Submitted to the Graduate Faculty of the
Louisiana State University and
Agricultural and Mechanical College
in partial fulfillment of the
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in

The Department of Mechanical Engineering

by

Onieluan Tamunobere
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ABSTRACT

At a fixed pressure ratio, the thermodynamic efficiency and net power output of a gas turbine engine increases with an increase in the turbine inlet temperature. Cooling is necessary because of the thermal limits of the engine materials. This research studies through experimentation, the heat transfer and cooling of critical gas turbine components including the shroud and blade tip. In the first phase, the shroud heat transfer behavior and the effectiveness of shroud cooling under the conditions of rotation is undertaken in a single stage turbine at a design rotation speed of 550 rpm, at off-design speeds and for varying cooling configurations and injection methods. At the design speed, as the blowing ratio is increased, the normalized Nusselt number in the shroud hole region decreases. At the off-design speeds, the results show that the high Nu/Nu₀ region migrates on the shroud surface. This migration affects the coolant coverage in the shroud hole region. The slot cooling study shows increasing cooling effectiveness up to a blowing ratio of 1.25 followed by a drop off in the cooling effectiveness with blowing ratio due to jet lift off. In the second phase of this investigation, the heat transfer and film cooling of a gas turbine blade tip with a blade rotation speed of 1200 rpm have been studied. The heat transfer results for the no coolant injection show a region of high heat transfer on the blade tip near the blade leading edge region. This region of high heat transfer extends and stretches on the tip as more coolant is introduced through the tip holes at higher blowing ratios. The tip film cooling profile is such that the tip coolant is pushed towards the blade suction side due to the effects of blade relative motion. Cooling the blade tip using coolant injection from the shroud holes and slots in combination with tip injection results in better overall cooling coverage of the blade tip with the shroud hole and blade tip cooling combination being the most effective. The level of coolant protection is strongly dependent on the blowing ratio and combination of blowing ratios.
CHAPTER 1: INTRODUCTION

1.1 Motivation

Gas turbine engines are widely used in various industries including aircraft propulsion and other power generation systems. In an ideal gas turbine engine, gases entering the engine undergo isentropic compression in a compressor. Afterwards, the working gas is heated in a constant pressure environment in a combustor. The working gas then undergoes isentropic expansion in a turbine wherein energy is extracted to do work. However in an actual gas turbine engine, the acceleration and compression of the working gas leads to some energy loss due to the effects of friction. Furthermore, the combustion of the working gas results in pressure loss also due to friction. Finally, the expansion of the working gas and the extraction of energy by the turbine are not isentropic as in the ideal case. There is a loss of energy as a result of friction and inefficiencies resulting from the various devices used.

Fig. 1.1: Schematic of Gas Turbine Engines (Illustration by Jeff Dahl via Wikimedia commons)

Therefore, there exists the need to maximize the efficiency of gas turbine engines to extract the maximum possible energy from the working gas. Several methods are available to meet this challenge. One way of achieving this objective is to increase the turbine inlet temperature. Increasing the turbine inlet temperature increases the thermal efficiency of the turbine. The higher temperature of the working gas allows the turbine to extract more energy. However at such high thermal loadings, the heat transfer to the turbine blades also increases. Hot gas flows through the passages between and around the turbine blades.
impacting the shroud and other exposed areas. This high temperature gas runs up against the thermal limits of the turbine blade materials resulting in blade oxidation and potential blade destruction by melting.

To combat these high thermal loadings, various methods have been used to protect the various turbine components exposed to the hot gas. One of the most common and reliable methods is cooling the blade. Coolant air is bypassed from the compressor and bled onto the surface to be cooled in external cooling. Internal cooling is mainly used in blade cooling. In internal cooling, the blade is convectively cooled as coolant is passed through serpentine passages in the blade. Other means of protecting the gas turbine blade include using thermal barrier coatings. Thermal barrier coatings are materials applied to the surface of the blade to protect it from the harshest effects of the high temperature environment. These insulating coatings protect the blade surface from the highest heat loads and improve the oxidation and corrosion resistance of the gas turbine blade since corrosion and oxidation become of particular interest at very high temperatures.

Practically, a combination of the various cooling methods available is used to achieve the best cooling performance for the gas turbine blade. However, a balance has to be maintained when employing the cooling methods because bypassing air from the compressor also adversely affects the overall gas turbine efficiency. Therefore, there remains a pressing need to continually maximize and optimize the effectiveness of the various cooling technologies. To understand the cooling requirements of a gas turbine blade, an understanding of the aerodynamic and heat transfer principles guiding the flow through the gas turbine blade passages is required. It is necessary to study these principles in the various regions of the gas turbine blade to maximize the cooling performance. Furthermore, a proficiency in material characteristics and selection is needed.

To optimize the overall thermal efficiency of a gas turbine engine, there is also a need to properly evaluate the effectiveness of film cooling and the overall performance of any cooling configuration. The adiabatic film cooling effectiveness, defined in Eqn. 1 is a dimensionless temperature used to characterize the cooling performance.
\[ \eta = \frac{T_\infty - T_{aw}}{T_\infty - T_{coolant}} \]  

(1)

The film cooling effectiveness is primarily defined by the adiabatic wall temperature, \( T_{aw} \). This wall temperature is the driving temperature potential for heat transfer to the wall and reducing this temperature is the objective of film cooling.

The heat flux into the wall with film cooling is given in Eqn. 2.

\[ q_c^* = h_c(T_{aw} - T_w) \]  

(2)

Without film cooling, the local heat flux to the wall is given in Eqn. 3

\[ q_{uc}^* = h_{uc}(T_\infty - T_w) \]  

(3)

Comparing the two heat fluxes allows us to compare and evaluate the performance of film cooling in reducing the heat flux to the wall. The representation of the heat transfer coefficients show that there is a difference between the heat transfer coefficients for film cooling and without film cooling. While the heat transfer coefficients can reduce with film cooling in certain cases, experimental evidence has also shown that increased turbulence caused by the interaction of the coolant and mainstream air can also increase the heat transfer coefficient. Therefore, the overall performance of film cooling is evaluated using another dimensionless parameter known as the Net Heat Flux Reduction (NHFR) which is defined as,

\[ NHFR = \frac{q_{uc}^* - q_c^*}{q_{uc}^*} = 1 - \frac{h_c}{h_{uc}} (1 - \eta \theta) \]  

(4)

Typically according to Newton et al (2009), the NHFR is only found in tests at engine representative conditions. \( \theta \) is an unknown and not generally found in laboratory conditions. \( \theta = 1.5 \) is used for experimental cases with engine matching temperatures and is found given an air freestream temperature \( Ta=1900 \ \text{K} \), a blade metal temperature \( Tw=1200 \ \text{K} \) and a coolant total temperature \( Tc=880 \ \text{K} \). A typical set of blade tip results showing the film cooling effectiveness, heat transfer coefficient and Net Heat Flux Reduction is shown in Fig. 1.2.
In this work, the heat transfer on the gas turbine blade tip and shroud regions will be investigated. An understanding of the aerodynamic principles guiding the flow of the working gas through the turbine stage is utilized in explaining the heat transfer in the various components. The effectiveness of cooling the turbine components and the effect of the coolant on the heat transfer of the components is also investigated as a means of maximizing the overall turbine efficiency.

1.2 Blade Aerodynamics

To gain a better understanding of the blade heat transfer, an understanding of the flow dynamics of the hot gas moving through the turbine is necessary. Turbine flow aerodynamics is one of the ongoing areas of study as researchers seek to gain a better understanding of the stator-rotor-stator flow. Due to the interblade passages, this flow is complex, three dimensional, viscous and unsteady. The inter-blade passages also result in other measurable effects on the flow dynamics including secondary flows, leakage flow vortices, horseshoe vortices, pressure side vortices, suction side vortices, counter vortices and flow
separation (S.L. Dixon, 2010). These flow phenomena affect the heat transfer characteristics and cooling effectiveness of the turbine blade as the working gas flows through the turbine.

There is a need to minimize the secondary flow vortices since they account for a large percentage of the losses as shown by Dring and Heiser (1985). The passage flow vortex increases the aerodynamic losses in the turbine blade passages. The resulting secondary flows reduce the overall film cooling effectiveness. This leads to higher thermal loadings on the blade which requires more coolant air to be diverted from the compressors. The diversion of the required coolant air reduces the overall thermal efficiency of the engine. Different strategies resulting from past and ongoing research have been employed in weakening the secondary flow vortex. One of such focus areas in weakening the secondary flow vortex has been the leading edge of the blade.

The leading edge of the blade significantly affects the aerodynamics as well as the heat transfer characteristics of the blade. Changes to the shape of the leading edge have been shown by Sauer et al (2001) to affect the flow as well as the local flow phenomena around the blade. One of the leading edge flow phenomena is the leading edge horseshoe vortex. The leading edge horseshoe vortex is formed at the intersection of the curved leading edge and the endwall. Eckerle and Langston (1987) showed that there is an increase in the static pressure on the leading edge as the free stream air approaches. This increase in static pressure is higher above the endwall boundary layer region than in the boundary layer region because of the higher free stream velocity above the boundary layer region. This causes a pressure gradient resulting in the formation of the horseshoe vortex. The horseshoe vortex splits into a suction side and pressure side wing of the vortex. The vortex is then propagated into the blade passage on the pressure side and suction side by the pressure gradient.
The leading edge of the blade is the origin of most of the thermodynamic phenomena in the blade passage. Therefore, it tremendously influences not only the thermodynamic but the heat transfer characteristics of most of the other blade regions. Numerous strategies have been employed to shape the leading edge of gas turbine blades to optimize the aerodynamic and heat transfer characteristics of some of the other parts of the blades. These strategies include experimenting with different shapes and curvatures of the leading edge (Thole et al, 2005), employing different leading edge cooling strategies (Oke et al, 2001 Pasinato et al, 2002), premixing and influencing the freestream air before its initial contact with the leading edge (Burd et al, 2000), amongst others. Research into various other strategies and their effectiveness is currently active and ongoing. Implementing a combination of some of the above strategies has proven to be most effective in improving the aerodynamic conditions at the blade leading edge region.

The propagation of the leading edge horseshoe vortex into the pressure and suction side leg of the vortex forms the foundation of the secondary flow in the blade passage (Eckerle and Langston, 1987). The separation line separates the adjacent pressure and suction side streamlines. The front section of the blade
passage is defined strongly by the pressure and suction side legs of the leading edge vortex which rotate in opposite directions.

Fig. 1.4: Horse-shoe and corner vortices at the blade leading edge. LE= Blade leading edge, HS=Horse-shoe vortex, Tke=Turbulent Kinetic Energy. (Mahmood et al, 2005)
The pressure side leg of the vortex rotates in a clockwise direction while the suction side leg of the vortex rotates in a counterclockwise direction as seen in Wang et al (1997). The pressure side vortex is driven by the pressure gradient from the blade leading edge to the adjacent blade’s suction side along the separation line. The suction side vortex wraps around the suction side following the separation line. The two vortices rotating in opposite directions merge further downstream in the passage resulting in the passage vortex. The overall passage vortex rotates globally in the direction of the pressure side vortex as shown by Wang et al (1997). This is because the pressure side leg of the vortex is stronger and bigger than the suction
side leg of the vortex when they merge. The newly merged passage vertex is located closer to the suction side of the blade than the pressure side. This is due to the endwall cross flow and the pressure gradient as shown by Mahmood et al (2005). The merged vortex which originally forms close to the endwall grows larger in size and migrates vertically upward towards the midspan and away from the endwall. This vertical movement is the result of a pressure gradient on the blade span on the suction side of the blade.

The formation of the passage vortex by the merging of the pressure and suction side leg of the horseshoe vortex induces another vortex which runs above and with the passage vortex (Wang et al, 1997). This vortex is termed the wall vortex. It is illustrated in the figure below along with the pressure side leading edge corner vortex and the suction side leading edge corner vortex which is also pictured above. These vortices rotate counter to the direction of their respective horseshoe vortices. They are formed at the intersection of the blade leading edge and the endwall. These vortices are small relative to their respective horseshoe vortices. Goldstein et al (1995) illustrates the pressure side and suction side corner vortices which are induced mid passage. These corner vortices both rotate in the counter clockwise direction with the suction side leg of the horseshoe vortex. Corner vortices can cause amongst other things, higher mass transfer rates especially at the suction side. The horseshoe vortex originating at the leading edge and propagated into the blade passage, the pressure and suction side corner vortices and the wall vortex form the primary vortices in the blade passage. These vortices result in secondary flows which are inherently complex and highly three dimensional. The understanding of the structure and propagation of these vortices contribute to the knowledge of the flow field in the blade passages. An understanding of the blade passage flow structure is essential when designing gas turbine blades.
1.3 Blade Tip Heat Transfer

1.3.1 Effect of Geometry

The gas turbine blade tip is one of the most vulnerable regions on a gas turbine blade. This is because a gap or clearance between the blade and the shroud or casing is necessary for blade rotation. With the gap comes an associated over-tip leakage flow. The over-tip leakage flow rolls up into a vortex that combines with the other flow phenomena to affect and define the heat transfer on the blade tip. In a near perfect design of a turbine blade, there would be minimal leakage flows. The over-tip leakage flow is very dependent on the size of the tip clearance as well as on the tip configuration. The larger the tip gap, the larger the over-tip leakage flow and the larger the associated high heat loads on the tip. One of the major objectives in gas turbine blade design is to minimize the aerodynamic losses that result from tip leakage.
flows. There is also the need to minimize the secondary flow vortices. The other major objective of gas turbine blade design is to effectively cool the blade tips using as little coolant air as possible. Different tip design strategies have been developed. The major types of blade tips include: (1) the flat tip (2) the squealer rim with a recessed tip and (3) the tip attached to the shroud. The flat tip is the least used of the tip designs in modern gas turbine engines. It is characterized by high leakage flows, poor tip aerodynamics and an accompanying higher heat load. The recessed tip with a squealer rim is the most commonly used tip design. The squealer rim allows for much smaller tip clearances while mitigating damage to the tip should the blade rub against the shroud. The attached tip is most commonly used in low pressure turbines. While the tip leakage flow is lowest of the three designs, the attachment of the blade to the shroud leads to higher stresses on the blade which makes it unsuitable for use in a high pressure turbine.

The profile of the over-tip leakage flow varies based on the tip configuration and design. The profile of the over-tip leakage flow is generally a good predictor of the heat transfer on the blade tip. With larger over tip leakage flow comes higher tip heat transfer coefficients and higher tip losses. The advantage of the squealer tip over the flat tip is in the presence of the squealer rim which acts as a seal, reducing the tip clearance and thus providing good resistance to the leakage flow. With this resistance comes a decrease in the tip losses and just as important, a decrease in the tip heat transfer coefficients. In the flat tip design, the over-tip leakage flow is characterized by a separation vortex as well as reattachment on the pressure side as shown by Key and Arts (2006). There is also a very strong pressure to suction side component of the tip leakage flow. The squealer tip design is characterized by flow recirculation within the cavity. Both phenomena define the overall tip leakage flow and the subsequent heat transfer profiles on both tip designs. The squealer tip design is further characterized by flow impingement near the leading edge which can and does increase the heat transfer rates in this region. Due to the flow recirculation within the squealer cavity, there is a significant suction side to pressure side component of the leakage flow on the cavity floor as seen in Key and Arts (2006). Despite this, there also remains a strong stream-wise component of the tip leakage flow for the squealer configuration. Other tip leakage flow differences remain between the flat tip and
squealer tip configurations that can explain the heat transfer profiles for both configurations. The flat tip configuration shows more sensitivity to changes in the flow Reynolds number than the squealer tip configuration (Key and Arts, 2006). Higher Reynolds numbers result in higher tip pressure ratios. Generally, the squealer tip design has a weaker leakage vortex and lower aerodynamic losses than the flat tip. This is directly as a result of the tip design with the squealer tip having lower over-tip flow velocities because of the resistance to the over-tip flow provided by the presence of the squealer rim.

Advances in the design of gas turbine blades are made through research. This research is performed both experimentally and using computational methods. Of the three blade tip designs, the flat tip is the easiest to design and to cool. The heat transfer coefficients on a flat tip for different tip gaps and different wall boundary conditions are shown below:

Fig 1.7: Predicted Heat Transfer Coefficients with standard k-ε model using Constant wall heat flux boundary condition (left) and Constant wall temperature boundary condition (right) for tip gaps of 1% span ((a) and (d)), 1.5 % span ((b) and (e)) and 2.5 % span ((c) and (f)) (Yang et al, 2002)
The heat transfer coefficient for the flat tip is highest in the region between the mid chord and pressure side of the blade. This is expected since this is the region with the highest tip leakage flow velocity and thus the highest turbulence levels and pressure ratios. The flat tip design is also characterized by flow separation immediately downstream of the pressure side tip with the highest heat transfer coefficient occurring in the region of flow reattachment and relatively low heat transfer coefficients observed near the leading edge region of the blade tip. Since the flow accelerates as it goes through the blade passage, relatively higher heat transfer coefficients can be expected past the blade mid chord towards the trailing edge in the suction side of the blade tip. The overall heat transfer coefficient on the tip increases with the tip clearance. Therefore, it is important to minimize this clearance when designing the blade tip.

The recessed tip with the squealer rim is one way of minimizing the tip clearance and its associated leakage flow. There are different ways a squealer rim can be arranged on a blade tip. They include pressure side only squealer, mean camber line squealer, pressure and suction side squealer, inner pressure and suction side squealer and suction side only squealer. The five different squealer arrangements and a flat tip as well as profiles of their pressure ratios are shown in Fig. 1.8.

The flat tip is characterized by higher leakage flows rates as seen in the pressure ratio plots than all the squealer tip configurations. This is followed consecutively by the pressure side squealer configuration, the mean camber line squealer, the inner squealer, the pressure and suction side squealer and lastly the suction side squealer configuration. The position of the squealer rim on the tip affects the over-tip leakage flow and the general flow profile near the blade tip region. The pressure ratio for the suction side squealer is low because of the flow separation that occurs upstream of the squealer. With the separated flow, the pressure ratios and the velocities in the region with the flow separation is low. For the pressure side squealer configuration, the over-tip flow immediately encounters the squealer rim on the pressure side and accelerates.
This acceleration increases the over-tip leakage flow velocity as higher pressure ratios are seen as the flow reattaches on the blade tip. The mid camber line squealer configuration is also characterized by flow separation and correspondingly low pressure ratios upstream of the squealer rim. The pressure ratio increases as the flow accelerates past the rim and reattaches on the blade surface. The pressure and suction side squealer rim has the second lowest overall pressure ratios and thus leakage flow velocities. For this configuration, the region of flow reattachment moves towards the leading edge region of the blade tip as the incoming flow separates over the squealer rim and immediately attaches in this region. The rest of the
tip regions for the pressure and suction side squealer configuration experience flow separation between both squealer rims. This results in the low pressure ratios in these regions. Displacing the squealer rims inwards only differs from the pressure and suction side squealer arrangement by the noticeable increase the pressure ratio upstream of the squealer rim. The resulting heat transfer coefficients for the various tip configurations discussed are shown in Fig. 1.9. From the pressure ratios, over-tip leakage flows and the resulting heat transfer coefficient values, the suction side squealer tip provides the best performance as measured by the heat transfer results. This is followed closely by the pressure and suction side squealer tip configuration. The flat tip configuration followed by the pressure side squealer configuration provides poor heat transfer performance relative to the other four configurations.

Fig. 1.9: Heat transfer coefficients for (a) flat tip (b) pressure side squealer tip (c) mid camber line squealer tip (d) pressure and suction side squealer tip (e) inward pressure and suction side squealer tip (f) suction side squealer tip (Acharya et al, 2003)

The highest heat transfer coefficient values can be seen on the squealer rims of both the pressure side squealer configuration and the mean camber line squealer configuration. Nevertheless, the highest area-averaged heat transfer coefficient occurs for the flat tip configuration. After the flat tip configuration,
the next highest area-averaged heat transfer coefficient occurs in the pressure side squealer configuration. This is followed by the mean-camber line configuration, inner squealer configuration, pressure and suction side squealer configuration and finally, the suction side squealer configuration. Although the suction side squealer configuration has lower area-averaged heat transfer coefficients, the pressure and suction side squealer configuration boasts the lowest absolute heat transfer coefficient values. The area-averaged heat transfer coefficient values of the blade tip generally follow the same ranking as the over-tip pressure ratios. However, the position of the squealer rim on the tip does move the region of high heat transfer on the blade tip floor. For the pressure and suction side squealer tip configuration, the region of high heat transfer coefficient on the blade tip floor can be seen near the leading edge region of the blade tip as the incoming flow separates over the squealer rim and reattaches in this region. For the mean camber line squealer rim configuration, the region of high heat transfer coefficient can be seen between the camber line and suction side as the incoming flow separates over the tip and accelerates through the rim and reattaches in this region.

In all the squealer configurations studied, the blade tip is characterized by generally higher heat transfer coefficients on the squealer rim. Thus, the squealer rims are very vulnerable to thermal damage. For smaller tip clearances, the pressure side rim is the most vulnerable part of a squealer tip. However, as the tip clearance increases and the blockage effect decreases, the suction side squealer rim becomes more vulnerable to thermal damage as shown in Kwak et al (2004).

Other derivations of the flat and squealer tip configuration have also been studied for possible use in a gas turbine engine. Two of the most common derivations include the winglet tip and the ribbed tip configurations. The winglet tip brings unique advantages to blade tip design. The blade is thickened further at its thickest point where the pressure difference between the pressure and suction side is largest.
This creates resistance to the leakage flow at a location where the leakage flow velocity is highest. The resistance to the leakage flow results in reduced pressure ratios in the blade tip region compared to what can be expected for a flat tip design. This reduced leakage flow results in reduced heat transfer to the winglet tip compared to the flat tip case.
Fig. 1.11: Schematic and Heat Transfer Coefficients of flat tip, winglet tip, ribbed tip and suction side squealer tip configurations (Acharya et al, 2003)

However, a winglet tip design is still characterized by higher overall area-averaged heat transfer coefficients compared to the suction side squealer design or the pressure and suction side squealer tip design. The decrease in the winglet tip design over the flat tip case is mainly on the suction side region of the tip. In the other parts of the blade tip, the winglet tip and flat tip exhibit similar heat transfer coefficients and profiles.

The ribbed tip is designed with ribs normal to the flow direction. The ribs induce large regions of flow separation on the blade tip with the flow separation resulting in reduced flow impingement and reduced heat transfer in the separation zones. The ribs can be slanted based on the flow dynamics to optimize the profile of the over-tip leakage flow and to induce regions of flow separation on the tip. The ribbed tip design does provide better heat transfer performance on the tip when compared to the flat tip case. The ribbed tip
design also performs better than the winglet tip design. If properly designed, a ribbed tip can exhibit are-
averaged heat transfer coefficient values closer to the suction side squealer design than to the flat tip design.
The ribbed tip design is characterized by regions of very low heat transfer coefficients just downstream of
the ribs where the freestream flow separates over the rib. The heat transfer coefficient values in these regions
can be as low as those seen in the suction side squealer design.

Based on the over-tip leakage flow dynamics and the heat transfer coefficient results, the suction
side squealer configuration provides the best performance for tip design. This is closely followed by the
pressure and suction side squealer configuration. The inner squealer configurations-displaced squealer and
mean camber line squealer, perform better than the pressure side squealer and flat tip configurations. The
winglet tip configurations and the ribbed tip configurations also provide better heat transfer performance
than the flat tip configuration with the ribbed tip configuration being the more effective of the two
configurations.

1.3.2 Effect of Turbulence

The heat transfer coefficient on a blade tip is also affected by such factors as the turbulence levels.
The turbulence in the turbine stage is generated during the mixing of the hot gas and the fuel in the
combustion stage. Knowledge of the turbulence levels as measured by the turbulence intensity is necessary
for a proper estimation of the heat transfer on the blade tip. With increased turbulence, the heat load on the
blade tip is increased which leads to higher cooling requirements or decreased blade lifespan. With
decreased turbulence, the cooling requirements on the blade tip is decreased. The relationship between the
turbulence level and heat transfer profile is also affected by the blade tip configuration. The heat transfer
coefficient increases 15-20% with an increase in turbulence intensity from 6.1-9.7% for the flat tip while
the increase in the heat transfer coefficient for the squealer tip is limited to the pressure side rim and trailing
edge region as shown by Azad et al (2000). The heat transfer coefficient on the cavity floor was not significantly affected by the increase in the turbulence level.

Fig. 1.12: Effect of Turbulence levels on Heat Transfer coefficients on a flat tip (Azad et al, 2000)

In addition to the freestream turbulence from the combustion stage, the heat transfer profile on the blade tip can also be affected by unsteady wakes generated upstream of the turbine blade. These wakes are generated by stator and rotor blades in stages upstream of the current turbine stage and as such impose periodic velocities and turbulence levels on the blade tip. The combination of these unsteady wakes and the freestream turbulence induce further heat transfer increases on the blade tip. This increase in the heat transfer coefficient is more pronounced for the flat tip configuration than for the other tip configurations.
Fig. 1.13: Averaged Heat Transfer coefficients on a squealer tip for (a) $Tu=6.1\%$ (b) $Tu=9.7\%$ (Azad et al, 2000)
1.3.3 Engine Representative Conditions

To accurately predict the heat transfer on a gas turbine blade tip, it is necessary to model the flow inlet and exit conditions such as the Reynolds number, Mach number and temperature ratios amongst others using engine representative conditions. At engine representative conditions, the pressure ratio is large enough such that the flow across the blade tip can become compressible. Compressible flow is characterized by features such as shock waves within the gap. The effect of the shock waves on the heat transfer on the blade tip has not been exhaustively studied in literature due to the difficulty in conducting such tests. Nevertheless, blade tip heat transfer measurements using these conditions have been conducted in various transonic setups as can be seen in the figures below.

![Fig. 1.14: Laterally averaged Nusselt number for three tip gaps (Zhang et al, 2011)](image)

The general relationship between the tip gap and heat transfer seen in subsonic flow is maintained at engine representative conditions. With an increase in the tip gap, the heat transfer on the tip generally
increases as well due to the increased over tip leakage flow. However, this relationship is not maintained for the entire tip as the heat transfer decreases as the tip gap gets smaller in the region less than 50% of the axial chord while the opposite behavior is observed towards the blade trailing edge region. For transonic flow, a lower Mach number is observed for the tip flow as the tip gap decreases. Nevertheless, supersonic flow can still be observed on the blade tip at even smaller tip gaps. All the results suggest that for the most part of a transonic blade tip, the high heat transfer is dominated by the enhanced turbulence thermal diffusion rather than by a direct increase of wall shear stress (Zhang et al, 2011). The structure of the tip leakage flow for transonic flow is different from what is seen in subsonic flow. In a subsonic flow, the flow downstream of the separation bubble typically seen on the tip decelerates due to turbulent mixing. For a tip with transonic flow, the flow downstream of the separation bubble accelerates which can result in supersonic flow conditions on the tip. Shockwaves formed on the tip and in the tip clearance can result in variations in the heat transfer seen on the blade tip.

1.4 Blade Tip Film Cooling

The heat transfer coefficient is not the only parameter utilized in evaluating the overall thermal performance of a tip configuration. In addition to other flow and structural dynamics concerns, the film cooling effectiveness and its effect on the heat transfer coefficient is as important if not more important when evaluating the various tip configurations.

Film cooling is one of the major ways to protect the turbine components from the most detrimental effects of the hot gas. In film cooling, air is bled onto the surface to be cooled using cooling holes on the surface. Air for film cooling is bypassed from the compressor. This reduces the overall efficiency of the engine. Therefore, the surface to be cooled has to be cooled effectively to allow for higher turbine inlet temperatures, higher turbine efficiencies and higher overall engine efficiency. As turbine inlet temperatures in modern gas turbines increase, so too must the various ways to cool the blade tip.
Flow past any blade tip is highly complex. This is due to the presence of the over-tip leakage flow and other flow phenomena in the tip region. With the presence of a secondary fluid in the form of the tip coolant, the flow past the tip becomes even more complex and highly three-dimensional with an associated primary-secondary fluid interaction and the propagation of secondary flow structures in the tip region. Nevertheless, film cooling has been shown to be very effective and highly reliable at cooling the blade tip. Several configurations have been used to cool the blade tip. Among the most common are the use of cylindrical or shaped holes on the tip and/or the pressure side of the blade.

Fig 1.15: Hole configuration on a squealer tipped blade (Tamunobere and Acharya, 2015)

Film cooling using pressure side holes is very effective for a plane tip and not so much for a squealer tip. In some cases, pressure side holes have been shown to have negligible cooling effects on the squealer tip and especially on the cavity floor as shown in Ahn et al (2005). Christophel et al (2005) showed that the effectiveness of cooling the tip of a turbine blade using pressure side holes is more effective for smaller tip clearances than larger tip clearances. With increasing blowing ratio, more of the tip is cooled if the tip gap
is small. The cooling effectiveness of tips with a large clearance is relatively unaffected by an increase in the blowing ratio especially as the tip gap gets larger.

Fig. 1.16: Averaged film cooling effectiveness for plane and squealer tip for different blowing ratios. T=Tip Holes, P=Pressure side holes, TP=Tip and pressure side holes (Ahn et al, 2005)

Film cooling using the pressure side holes also affect the heat transfer coefficient on the blade tip. A film cooled tip with this configuration shows higher heat transfer values over an uncooled tip with increasing heat transfer values with blowing ratio. The increase in the heat transfer coefficient is more pronounced with smaller tip gaps than with larger ones. Cooling a squealer tip is better done using holes on the tip floor. Arranged along the camber line, tip cooling holes show improved effectiveness with blowing ratio.
With enough tip holes, coolant accumulation is observed on the tip floor between the camber line and the pressure side (Ahn et al., 2005). Tamunobere and Acharya (2015) and Rezasoltani et al. (2014) have shown that with blade rotation, the coolant on the tip floor in a squealer configuration is pushed towards the suction side as opposed to the pressure side as in stationary studies. Combining pressure side holes with tip holes is a common cooling configuration for squealer tips as there are additive and carry-over effects due to the inclusion of the pressure side holes as shown by Kwak et al. (2004) and Tamunobere and Acharya (2015). Squealer tip film cooling has also been shown to affect the heat transfer values on a blade tip. While the squealer tip has been shown by Kwak et al. (2004) to reduce the heat transfer coefficient on the blade tip compared to the flat tip case, the presence of film cooling results in higher heat transfer values on the tip.
over the uncooled case (Tamunobere and Acharya, 2015). A typical film cooling profile for a squealer tip configuration with both tip and pressure side holes and blade rotation is shown in Fig. 1.18

![Film Cooling Effectiveness](image)

**Fig 1.18: Film Cooling Effectiveness for a) M=1.0 b) M=1.5 c) M=2.0 d) M=3.0 e) M=4.0 (Tamunobere and Acharya, 2015)**

While film cooling can decrease the heat transfer coefficient on the blade tip, it can also increase the heat transfer coefficient because of the increased turbulence due to the increased mainstream air-coolant interaction. Therefore, the effect of film cooling is important in determining or predicting the heat transfer coefficient on the blade tip.

The overall economy of film cooling and any film cooling configuration needs to be investigated and evaluated before any configuration is chosen by blade designers. As with most other parameters that affect film cooling, the blade tip configuration is an important factor that influences the effect of film cooling on the blade tip heat transfer. In this case, the hole configuration is also an important factor since
the hole exit configuration affects the coolant exit flow profile and thus its interaction with the mainstream flow. The effect of film cooling on the heat transfer coefficient for a squealer tip configuration with two tip holes is shown in the figure below. To evaluate and demonstrate the effect of film cooling on the heat transfer coefficient, the averaged heat transfer coefficient is compared to the baseline averaged heat transfer coefficient without film cooling. For a more comprehensive evaluation, the laterally averaged film cooling effectiveness for this configuration is also provided. It should be noted that the two tip holes are between $x/C_s = 0.2$ and $x/C_s = 0.4$.

For this configuration, the coolant jets do augment and increase the blade tip heat transfer coefficient relative to the no coolant case. Typically for a full squealer configuration, the incoming mainstream flow separates over the squealer rim and reattaches on the blade tip floor near the leading edge region of the blade tip. Thus, high heat transfer coefficient on the blade tip floor can be seen near the leading edge region of the blade tip. With film cooling, there is an enhancement of this region of high heat transfer on the blade tip floor with the region extending axially as more coolant is introduced at higher blowing ratios. The over-tip flow and its complex vortex systems interact with the coolant air and this interaction results in higher turbulence levels and higher resulting heat transfer coefficients on the tip in the affected region. The interaction is most significant in the region where the tip holes are located.

In addition to seeing the effects of film cooling on the heat transfer coefficient using this tip configuration and run conditions, the effect of the blowing ratio on tip cooling can also be seen as the film cooling effectiveness increases with blowing ratio. The coolant coverage also penetrates further axially at higher blowing ratios leading to better coolant coverage downstream of the cooling holes. There is also a lateral spread of the coolant as the coolant reattaches on the tip floor at locations further from the tip holes after separating from the floor at the coolant hole exit. The separation and reattachment of the coolant is observed at higher blowing ratios.

The effect of film cooling is important in evaluating the vulnerabilities of any tip configuration. This particular tip and tip cooling configuration is particularly vulnerable. With only two tip holes, the
coolant accumulation effect on the trailing edge region of the tip which is typically seen in tip configurations with multiple cooling holes on the tip is absent. Furthermore, since the two tip holes are located near the leading edge region of the blade tip, there is a dearth of coolant coverage near the trailing edge of the tip. Combined with the poor cooling effectiveness in the tip trailing edge region, there still remains a not insignificant increase in the heat transfer coefficient in the region and this highlights the vulnerability of this configuration.

Film cooling on a gas turbine blade is achieved using discrete hole cooling or slot cooling. Discrete hole cooling is mainly used for the gas turbine blade tip. There are several factors that affect film cooling. These factors include the blowing ratio, density ratio, surface angle, hole geometry and configuration, amongst others. These factors individually and combined, affect film cooling and are useful to gas turbine designers.

1.4.1 Effect of Hole Geometry

Hole geometry is a broad term for several parameters of cooling holes. These parameters include the hole shape, surface angle, exit condition, hole spacing, hole pitch and general hole configuration.

The shape of the coolant holes on a blade is a primary determining factor in the blade cooling performance. Cylindrically shaped holes form the most basic configuration for coolant hole modeling. Inclining the holes resulting in compound angled holes, reduces the component of the velocity perpendicular to the wall and this allows for increased attachment of the coolant to the surface to be cooled and decreases the mixing out of the coolant with the mainstream air. While decreasing the angle between the hole and the mainstream flow direction decreases the normal velocity component as much as possible, larger angles are used due to the ease of manufacturing and the increased benefit of the larger angles at higher blowing rates (Goldstein and Stone, 1997 and Baldauf et al, 2001). Nevertheless, the smaller angle between the hole axis and the mainstream flow will be more beneficial at lower blowing rates. At higher
blowing rates, a larger angle will be beneficial in addressing the issue of jet lift-off from the surface. At a steeper angle, the coolant jet interacts more with the mainstream air leading to increased spreading of the coolant. With jet lift-off, the coolant reattaches at a smaller distance from the hole at a steeper inclination angle. The hole geometry also affects the heat transfer coefficient as the heat transfer coefficient increases with increasing angle (Baldauf et al, 2002). Typically, a hole inclination angle of 30 degrees is used for most parts of the blade.

The other major cooling hole geometries are extensions and modifications of the base cylindrical hole configuration. In the laidback hole configuration, the hole exit is inclined in the flow direction. With the inclination, the hole exit angle becomes shallower and the potential for jet lift-off is decreased. In the fan-shaped hole configuration, while the majority of the hole remains cylindrical, the hole exit is shaped like the so-called fan such that it expands in the lateral direction. Not only is the hole exit expanded but the expansion in the lateral direction allows for increased diffusion and spreading of the coolant on the surface to be cooled in addition to the benefits of decreased coolant velocity and the potential for jet lift-off. The laidback fan-shaped hole is a combination of the laidback and fan-shaped configurations. In this configuration, the hole exit is expanded in the lateral direction as in the fan-shaped configuration and inclined in the flow direction as in the laid back configuration. A conical hole configuration is obtained by the enlargement of the cylindrical hole exit in every direction such that a cone-like hole is formed. The convergent slot hole (CONSOLE) configuration is a relatively new configuration wherein a hole at the coolant entrance converges to a slot at the coolant exit. The different hole configurations can be seen in the figure below.
Compared to cylindrical holes, fan shaped holes provided a significant increase in film cooling effectiveness and greater surface coverage. Coolant ejected from cylindrical holes lifts off the coolant surface as the blowing ratio increases. On the other hand, shaped holes allow the coolant to remain close to the coolant surface as the coolant exits from the holes thus maximizing the cooling protection further downstream of the holes. The coolant separation from the surface and in effect the hole shape, affects the magnitude of the aerodynamic losses associated with the presence of the cooling holes. The coolant separation from cylindrical holes generates high mixing losses with the freestream. Although fan shaped holes generate significant losses as well, the losses are less than in cylindrical holes. The different kinds of shaped holes have different effects on the film cooling performance and the angles of the laidback and fan shaped hole exits play an important role in not only the film cooling effectiveness but the relationship between the blowing rate and the film cooling effectiveness. Larger exit angles allow for greater cooling effectiveness and this effect is more pronounced at higher blowing ratios. The expansion cooling holes in the lateral...
direction as in fan-shaped holes also helps blade designers in designing a more complete and closed cooling scheme because of the increased lateral spread of the coolant. For each shaped or cylindrical design, the surface and compound angles utilized have to be selected carefully to optimize the film cooling performance.

1.4.2 Effect of Blowing Ratio

The blowing ratio, defined as the ratio of the mass flux of the coolant to the mass flux of the mainstream air is one of the critical parameters that affect the cooling performance of a cooling configuration.

\[ M = \frac{\rho_c U_c}{\rho_m U_m} \]

While very important, the effect of the blowing ratio on the cooling performance is still dependent on some of the other parameters like the hole configuration, density ratio, etc. Unlike some of the other parameters, the effect of the blowing ratio on film cooling has been thoroughly studied. For a simple cylindrical hole configuration on a flat plate, the film cooling effectiveness has been found to increase with blowing ratio up until a peak blowing ratio of about 0.5 or higher at higher density ratios. Increasing the blowing ratio much further results in more coolant separation and lift-off of the coolant jet from the surface to be cooled as illustrated below.

The effect of the blowing ratio and the peak effectiveness for each blowing ratio is highly dependent on the cooling hole configuration and the other parameters to be discussed. With shaped holes, the blowing ratio for the peak effectiveness is higher than for the cylindrical configuration even with compound angle cylindrical holes. While cylindrical holes provide an increase in cooling effectiveness with blowing ratio up to a blowing ratio of 0.5-0.8 followed by a decrease in effectiveness with increasing blowing ratio, shaped holes can sustain the trend of increasing effectiveness with blowing ratios up to blowing ratios of 3.0-4.0. However, the cooling economics has to be considered when deciding to use higher blowing ratios.
as the increase of effectiveness with blowing ratio is not linearly proportional especially at the higher blowing ratios.

![Diagram](image)

Fig. 1.20: Effect of blowing ratio for cylindrical hole configuration at a density ratio of 1.8 (Baldauf et al, 2002)

### 1.4.3 Effect of Density Ratio

The ratio of the density of the coolant to mainstream air is one of the factors that affect the film cooling effectiveness. For modern gas turbine engines, the density ratio is closer to 2.0 than 1.0 as it is for most laboratory experimental tests. Due to this disparity in the density ratio between experimental and engine representative conditions, it is necessary to factor in the effect of the density ratio when evaluating the various cooling configurations. From the definition of the blowing ratio, it can be seen that for a fixed blowing ratio, a lower density ratio implies a higher velocity ratio. With a higher velocity ratio, the effects
of jet lift-off and coolant separation are more readily observed. Therefore, lower density ratio and thus higher velocity ratio jets will separate from the surface to be cooled before higher density ratio jets. Thus, the maximum effectiveness when testing is performed at some laboratory conditions with lower density ratios will be lower than the maximum effectiveness at engine representative conditions (higher density ratios). However, doubling the density ratio from approximately 1 to approximately 2 has been shown to only increase the maximum effectiveness by less than 25% in the region in the vicinity of the cooling hole while the effect of the density ratio is not significant further downstream of the cooling hole (Sinha et al, 1991).

**1.4.4 Effect of Hole Length to Diameter Ratio**

The length of a cooling hole affects the film cooling effectiveness and the cooling performance of the hole. The length of a hole depends on the thickness of the cooling surface and the angle the hole axis makes with respect to the cooling surface. The primary effect of the hole length is on the coolant flow profile especially at the hole exit. At longer hole lengths, the flow is fully developed and significant differences are not observed for the film cooling effectiveness with increasing hole length. However, for shorter hole lengths, \( L/D <5 \), there is an increase in film cooling effectiveness with hole length as shown by Lutum et al (1999) as the flow becomes more fully developed and the velocity field becomes more symmetric.

**1.4.5 Effect of Hole Spacing**

The hole spacing as non-dimensionalized by the hole diameter is an important parameter that affects the cooling effectiveness on the surface to be cooled. Its primary effect is in the interaction of the coolant ejected from a cooling hole with the coolant from neighboring holes. With a small hole spacing, the hole to hole interaction is very high leading to more uniform heat transfer properties in the region around the holes as seen in the figure below. The coolant from a hole does not have to migrate far from the hole to attach to
the coolant from adjacent holes with a small enough hole pitch to diameter ratio. With a larger hole spacing, the cooling effectiveness decreases as one moves further away from the coolant hole until the coolant interacts with the coolant from adjacent holes. This leads to a more sinusoidal and periodic behavior when the coolant hole spacing is sufficiently large.

![Graph](image1.png)

Fig. 1.21: Lateral Film cooling distribution at P/D=4.8 (left) and P/D=10.4 (right) for BR 1.0 at 500 RPM (Tamunobere et al, 2014)

### 1.4.6 Effect of Surface Curvature and Surface Roughness

The geometry of the surface to be cooled also affects the effectiveness of any cooling configuration. The surface to be cooled can be either flat, convex or concave. Convex curvature has been shown to feature increased cooling effectiveness while concave curvature has been shown to feature decreased effectiveness. With surface curvature, the relationship between the tangential component of the coolant velocity and the mainstream velocity changes. A convex surface allows for jet attachment to the wall due to the pressure gradient normal to the cooled surface while the pressure gradient on concave surfaces is such that the coolant is pushed outwards and away from the surface to be cooled (Ito et al, 1978).

Surface roughness in gas turbine engines is mainly as a result of normal wear and tear or the deposition of particulates and other material on the surface to be cooled. The issue of surface roughness is significant in the design of gas turbine engines as most engines experience the detrimental effects of particulate deposition or other wear and tear with the passage of time. Surface roughness leads to increased heat transfer coefficient, increased boundary layer thickness and increased local turbulence on the surface to be cooled.
This in turn can result in decreased effectiveness from film cooling. Practically, particulate deposition can also result in cooling hole blockage that can take some of the cooling holes out of service.

1.4.7 Effect of Slot Cooling

Discrete holes are mostly used for film cooling. For maximum cooling effectiveness, the base configuration of cylindrical holes is modified by inclining the holes to take advantage of the surface angle effect, shaping the holes to take advantage of the positive effects of changing the coolant profile at the hole exit or introducing more holes to take advantage of the hole spacing effect. At the extreme of the hole spacing effect is the use of a slot or a series of holes with such a small hole spacing that they converge to a slot. A slot has an advantage over a row of holes because it maximizes the coolant exit area. The increased coolant flow leads to better coolant coverage and film cooling effectiveness. Slot film cooling effectiveness is greater at higher slot flow rates and allows for more uniform and wide spread coverage especially in regions immediately downstream of the slot than film cooling holes. The use of a slot for film cooling has been continuously investigated as early as the 1940’s by Weighardt (1946). Through the use of experimental data and several modeling techniques, several empirical correlations to model the effectiveness of cooling a surface using a slot have been developed. The most common technique models the slot as a heat sink. In this model, the coolant flow is modelled as a heat sink after injection with the goal of reducing the temperature in the boundary layer downstream and therefore the temperature of the wall. The slot geometry and the flow field at the injection point are two of the most significant factors in predicting the film cooling effectiveness. Goldstein (1971) noted that due to the difficulty in accurately defining the effect of the coolant on the boundary layer, most correlations using the various models predict higher effectiveness values that would usually be found experimentally.

Even with the different models, most of the empirical slot correlations over the years have been of the form

$$\eta_{aw} = CRe_s^{0.2} (\frac{S}{Ms})^{-0.8}$$

(1)
Where \( \eta_{aw} = \frac{T_w(x) - T_\infty}{T_s - T_\infty} \), and \( M = \frac{U_s \rho_s}{U_\infty \rho_\infty} \) \( (2) \)

Slot cooling, as seen by the general form, depends on the slot Reynolds number, \( \text{Re}_s \), distance from the slot and the blowing ratio. The various empirical correlations establish the dependence of the slot on each of these parameters.

In 1952, as part of a larger study on jets discharged parallel to a surface, Tribus and Klein (1952) developed the correlation in Eqn. 3

\[
\frac{T_w(x) - T_\infty}{T_s - T_\infty} = 4.62 \text{Re}_s^{0.2} \left( \frac{x U_\infty \rho_\infty}{s U_s \rho_s} \right)^{-0.8}
\]  \( (3) \)

With \( C = 4.62 \) when compared to the general form of the equation.

Librizzi and Cresci (1964) using a heat sink model and assuming the boundary layer starts at the point of injection arrived at the correlation in Eqn. 4,

\[
\eta = \frac{1}{1 + 0.329 \left( \frac{x}{M_s} \right)^{0.3} \left( \frac{\mu_s \text{Re}_s}{\mu_\infty} \right)^{-0.2}}
\]  \( (4) \)

Stollery and El-Ehwany (1965) assumed that the boundary layer starts at the slot and the total mass flow at the slot is zero arrived at the correlation below.

\[
\eta = 3.03 \left( \frac{x}{M_s} \right)^{-0.8} \left( \frac{\mu_s \text{Re}_s}{\mu_\infty} \right)^{0.2}
\]  \( (5) \)

Kutateladze and Leont’ev (1964) assuming the boundary layer starts upstream of the slot derived the correlation below

\[
\eta = \frac{1}{1 + 0.249 \left( \frac{x}{M_s} \right)^{0.8} \left( \frac{\mu_s \text{Re}_s}{\mu_\infty} \right)^{-0.2}}
\]  \( (6) \)

The empirical correlations derived using the heat sink models contain the same parameters as the general form. Furthermore, the correlations predict the same trend for the film cooling effectiveness. However, the
correlations and the predicted values of the film cooling effectiveness differ due to the wide array of assumptions used in developing the heat sink models.

Using a simpler power law model to fit collected data, Seban et al. (1957) developed the correlation below.

$$\eta = 2.2 \left( \frac{x}{M_s} \right)^{-1/2}$$

(7)

Unlike the earlier models, it does not include the slot Reynolds number. The extent of the correspondence of the various slot cooling correlations for blowing ratios, $M=0.5$ and $M=1.0$ is examined and shown in the figure below:

Fig 1.22: Comparison of Slot Cooling Empirical correlations (Tamunobere et al, 2015)
While the variation in the results decreases at locations further away from the slot, there remains variations in the cooling effectiveness in the region closer to the slot. Since different assumptions were utilized in developing the empirical correlations, the variations are not surprising. Furthermore, since the heat sink model assumes the secondary gas completely mixes with the mainstream air in the mainstream boundary layer, the empirical correlations are only reasonably valid further from the slot (Goldstein, 1971).

1.4.8 Effect of Blade Relative Motion

Although gas turbines are rotating machinery, most gas turbine blade studies are performed experimentally in stationary cascades or computationally assuming stationary blades. This is due to the difficulty in instrumenting and setting up rotating tests in a laboratory and the difficulty in computationally modeling the complex flow fields with blade rotation and validating the model. Even with the inherent difficulties in performing research with blade relative motion, blade relative motion does have a significant effect on the blade over-tip leakage flow, heat transfer coefficient and film cooling effectiveness. Blade relative motion has been shown by Tallman and Lakshminarayana (2001) and Yaras et al (1992) to affect the tip leakage flow and the propagation and interactions of the secondary flow. Blade relative motion decreases the over tip leakage flow as stationary cases have a stronger leakage vortex. The magnitude of the effect of blade relative motion on the blade tip also varies based on the blade tip configuration. Blade relative motion affects the velocity profile on the blade tip as the velocity profile on the blade suction side exit is more uniform for stationary cases while the concentration of the leakage flow at exit locations from the mid chord to the trailing edge results in higher heat transfer in this region. For the squealer tip configuration, this is seen clearly in Yang et al (2010) by the higher heat transfer coefficient on the suction side squealer rim.

Blade relative motion can be modeled with blade rotation or with a moving shroud or casing. Aerodynamic and heat transfer studies have shown more significant differences between the stationary case and the cases
with blade relative motion than between the two blade relative motion cases as can be seen in Yang et al (2010) and Acharya and Moreaux (2012).

Fig. 1.23: Blade Tip Cooling with and without blade relative motion (Acharya and Moreaux, 2012)

Another significant effect of blade relative motion is the apparent push of the coolant to the blade tip suction side as opposed to its orientation to the pressure side in stationary studies. This apparent push becomes more pronounced as the blade rotation speed or shroud velocity increases. This push can be seen in the results in Acharya and Moreaux (2012) and in Rezasoltani et al (2014).
1.5 Shroud Heat Transfer and Cooling

The shroud around gas turbine blades is one of the most susceptible candidates for thermal failure in a gas turbine engine. This is because with the possible exception of the blade itself, the shroud experiences the highest level of thermal loading in the engine. The shroud is exposed to highly three-dimensional and unsteady flow emanating from the interaction of the flow with the earlier turbine stages and the rotor inlets. The shroud is exposed to most of the major flow phenomena the gas turbine blades are exposed to including the over-tip leakage flow. With small tip clearances (less than 2% of the blade span) typically required for blade rotation, the pressure difference between the blade pressure and suction side results in pressure driven flow across the clearance. This over tip leakage flow combined with the secondary flow structures typically associated with the blade tip and clearance region exacts a significant heat transfer penalty on the shroud. Thus, efforts to minimize the secondary flow vortices associated with the blade tip aid in the effort to reduce the thermal loadings the shroud is exposed to. While the blade and especially the blade tip is studied in detail, the shroud is studied to a much lesser extent. Thus, there is a need for detailed studies on the heat transfer behavior of the shroud.

1.5.1 Effect of Tip Design

The magnitude and general profile of the thermal loadings the shroud is exposed to is also highly dependent on the blade tip design and the over tip leakage flow. The flat tip, squealer rim with a recessed tip and the attached tip each pose significant but different thermal loadings on the shroud. The flat tip design imposes the highest thermal loadings on the shroud. This design is characterized by high leakage flows as well as poor tip aerodynamics as discussed in previous sections. Since the flat tip design is characterized by flow separation downstream of the pressure side tip and reattachment in the region between the mid chord and the pressure side, the heat transfer coefficient in this region is highest as can be seen from Fig. 1.26. The squealer tip design which is most commonly used in gas turbine blades allows for smaller tip clearances and thus smaller heat transfer loads on the shroud. The shroud heat transfer load varies bases on
the position of the squealer on the tip. For the pressure side squealer configuration, the overtip leakage flow encounters the squealer rim, detaches from the tip and reattaches on the surface. A similar profile can be seen in the other squealer configurations with the region of high heat transfer occurring as the flow reattaches to the surface. With the flow separation on the rim comes a separation bubble. The size of this bubble is dependent on the tip clearance. In the squealer tip configuration, the squealer rim is especially vulnerable and a prime candidate for failure with high thermal loadings. The squealer rim is generally characterized by higher heat transfer rates than the cavity floor. Some of the tip configurations including the pressure and suction side squealer configuration and the suction side only squealer configurations feature comparatively lower heat transfer coefficients on the shroud than the other configurations. Thus, these configurations are more desirable to gas turbine designers. Nevertheless, it should be noted that the shroud heat transfer profile in Fig. 1.26 is obtained from stationary studies.

![Heat Transfer Coefficients](image)

Fig 1.24: Heat Transfer Coefficients on the shroud for different squealer arrangements (Yang el al, 2002)
1.5.2 Effect of Tip Clearance

The tip clearance affects the magnitude of the heat load on the shroud. The smaller the tip clearances, the smaller the leakage flow. Smaller leakage flows allow for lower heat transfer loads to the tip even in cases with blade relative motion as can be seen in Fig. 1.27. As the clearance height increases, so does the leakage flow velocity and thus the mass transfer rate through the tip. In cases with very small tip gaps as in case 1, the tip leakage flow is too weak to generate a tip leakage vortex (Rahman et al., 2013).

![Predicted Nusselt number on the shroud for three clearance heights](image)

Case 1 (0.6 mm)  Baseline (1.2 mm)  Case 2 (2 mm)

Fig. 1.25: Predicted Nusselt number on the shroud for three clearance heights (Rahman et al., 2013)

The tip leakage vortex becomes stronger and more significant as the clearance height increases. The corresponding increase in the heat transfer in the shroud region above the pressure side of the blade is associated with the significant increase in the strength of the tip leakage vortex.
The tip leakage flow structure in Fig. 3 shows that towards the blade suction side, the tip leakage flow structure detaches from the shroud surface. This is responsible for the relatively lower heat transfer seen on the shroud above the tip suction side for the flat tip design. The detachment of the leakage flow from the shroud surface is as a result of a relative reverse flow very near the shroud. This reverse flow is as a result of the shearing of the tip leakage flow due the relative motion of the shroud. This relative reverse flow acts against the tip leakage flow by shearing it in the opposite direction (Key and Arts, 2006) and its effect on the shroud heat transfer decreases going from the suction side to the pressure side of the blade as the relative reverse flow encounters the tip leakage flow coming from the pressure side of the blade (Rahman et al, 2013).
1.5.3 Engine Representative Conditions

Engine representative conditions are necessary to advance the knowledge of the heat transfer behavior of the shroud. Conditions such as the Reynolds number and Mach number comparable to those in a gas turbine engine can be matched in a laboratory setting to provide more details on the shroud heat transfer behavior. Despite the challenges involved, these conditions have been met in some cases. A contour plot and a graph showing the heat transfer profile on the shroud using a transonic turbine setup is shown in Fig. 1.29.

In this case, the heat transfer rate is measured at 56 discrete positions on the shroud surface and extrapolated to cover the entire surface after its conversion to the Nusselt number using the following formula:

\[
Nu = \frac{Q'}{T_{aw}' - T_w} \frac{C_{ax}}{k}
\]  

(1)

The Nusselt number is normalized with the circumferentially averaged value at – 20 % axial chord. The heat transfer rate on the shroud is highest in the region just upstream of the blade domain position and decreases with the axial chord. This decrease is first gradual and then more sudden further downstream on the shroud surface (Thorpe et al, 2004). The decrease in the heat transfer rate on the shroud surface is mostly as a result of the decrease in the Reynolds number of the working gas as the flow decelerates and expands as work is extracted by the rotor (Thorpe et al, 2004). This decrease in the heat transfer on the shroud with axial position is confirmed by Rahman et al (2013) for different inlet temperature conditions as can be seen in the figure below. Since these are time averaged studies with blade rotation as opposed to earlier studies noting the variation of the heat transfer rate with the tip design or the clearance gap, there is much less circumferential variation in the heat transfer rate.
Fig. 1.27: a) Contour Plot and b) Graph showing Time-Mean Nusselt number on the shroud wall (Thorpe et al, 2004)
While the time-mean heat transfer of the shroud does provide the general information on the shroud heat transfer behavior, the time-resolved results provide context to and explanations of the time-mean results. A more thorough understanding of the physical mechanisms influencing the shroud heat transfer behavior can be obtained using the time-resolved heat transfer data (Thorpe et al, 2004). To that end, the instantaneous shroud heat flux distribution is shown in Fig. 1.30. The heat load on the shroud can be divided
into two major parts. The first part is as a result of the over tip leakage flow and the second part is as a result of the passage flow. With moving blades and a stationary shroud, a fixed point on the shroud is always over a blade or over the passage between blades. The time resolved data allows for a quantification of the contribution of each of these mechanisms to the overall shroud heat load. The work by Thorpe et al (2004) in this regard shows that at least 50 % of the shroud heat load is produced by the over-tip leakage flow despite the tip comprising only about 23 % of the circumferential width. This analysis was performed using the breakdown of the tip vs passage footprint. This result shows the enormous impact the over-tip leakage flow has on the overall shroud heat transfer despite the difference in the relative footprint. Nevertheless, the result shows that using only the over-tip leakage flow contribution as is done in stationary shroud studies with the investigation of the shroud heat transfer behavior above the blade tip, will result in an incomplete study of especially the magnitude of the heat transfer load on the shroud. This is true given the 50 % contribution of the passage flow to the shroud heat load even with a smaller relative contribution compared to its size.

For a proper perspective of the heat load contribution from the passage flow and the over tip flow, the following figures have been provided in Fig. 1.31. The data illustrates the footprint of the passage and over tip leakage flows. The passage is about 3.5 times larger than the blade width in the blade mid-chord region. The rotor passage flow is shown to be extremely high prior to the blade domain given that before the blade, there is no over-tip leakage flow. The heat load upstream of the tip is as a result of the passage flow. While the rotor passage flow decreases with axial distance, the contribution from the over-tip leakage flow first increases from the blade leading edge to a little over 30 % chord and then decreases afterwards.
Fig. 1.29: Instantaneous Shroud Heat Flux Distribution as a Function of Blade-Phase (Thorpe et al., 2004)
While there have been a few studies on shroud heat transfer, studies on shroud film cooling are very sparse in literature. Mostly, shroud cooling designs are adapted from studies performed on a flat plate. From this introduction alone, it is clear that the results of a flat plate study can only partially model the necessary conditions faced by a shroud. Conditions such as the full effect of the over-tip leakage flow, the passage flow or the effects of the blade rotation can only be properly modeled when these elements are
introduced as part of the experimental setup. This gap in the literature on shroud cooling is comprehensively addressed as part of this research work.

1.6 Objectives and Uniqueness of this Study

Gas turbine components like the blade and shroud are exposed to high thermal loadings due to the high inlet temperatures necessary for maximizing the efficiency of the engines. Thus, it is important for gas turbine designers to fully understand the heat transfer behavior of these components in order to design them more efficiently. Providing a greater understanding of the heat transfer behavior of the blade and shroud is one of the primary objectives of this research. Equally important to gas turbine designers is the need to combat the high thermal loadings. Bleeding coolant air onto the surfaces exposed to the hot gases is one of the most common and reliable methods of protecting components such as the blade and shroud. Since the coolant air is bypassed from the compressor stage, the overall engine is subject to a penalty on its efficiency. Thus, balancing the adverse effects on the engine efficiency due to the coolant requirements with the increased efficiency due to the action of film cooling on the components is of paramount importance to gas turbine designers. In other words, to maximize the overall efficiency of the engine, the components exposed to the hot gases have to be cooled efficiently and with as little bypassed air as possible.

The difficulty in instrumenting a rotating rig amongst other obstacles is one of the major reasons for a lack of studies with blade rotating in literature. This applies to both blade heat transfer as well as shroud heat transfer. There is also a lack of stationary shroud heat transfer and cooling studies in literature. Most of the existing knowledge on shroud heat transfer behavior have been developed using flat plate studies that are not conducted with the complexities involved with the presence of the turbine blade.

The primary objectives and uniqueness of this study are as follows:

1) The first major objective of this research is to provide gas turbine designers with a better understanding of the heat transfer behavior of the turbine blade tip and shroud regions including
the effects of film cooling on the heat transfer and especially the effects of blade rotation on the heat transfer behavior of both the shroud and the blade tip. Experimental studies are performed to investigate the heat transfer behavior of the blade. Both types of studies have been ongoing for several years. However, most of the knowledge gathered over the last few decades have been gleaned from experimental studies with non-rotating blades. Furthermore, most of the CFD codes used in advancing the knowledge of blade and shroud heat transfer have been validated using studies conducted in stationary rigs. Blade relative motion has been shown to affect both the heat transfer and film cooling on the turbine blade and shroud. As the blade rotation speed increases, the tip leakage flow strength decreases as a shear layer on the shroud wall forms due to the sweeping action of the blades. The change in flow profile due to the effect of the blade relative motion affects the heat transfer coefficient and especially the profile of the film cooling effectiveness on the blade tip (Tallman J. and Lakshminarayana, B (2001), Yaras and Sjolander (1992)). It is only recently that the effects of blade rotation or blade relative motion have been taken into account in blade tip heat transfer studies. Even at that, it is mostly non-film cooling studies limited to using discrete points on the surface that have been performed (Dunn and Haldeman, 2000, Thorpe et al, 2005, Abhari and Epstein, 1994, Molter et al, 2006). In a study that presented the film cooling results, results of the heat transfer indicated by the heat transfer coefficient, were not found nor presented (Rezasoltani et al, 2014). The lack of tests with blade rotation is due to the inherent difficulty in instrumenting a rotating rig. This study is unique in that it provides both the heat transfer results and the film cooling results on the blade tip and on the shroud under the condition of rotation. Furthermore, these results are not just presented at discrete points but using detailed spatial contour plots on the surfaces to be examined. In addition to the direct advancement of the knowledge of the heat transfer behavior of the shroud and blade tip, the results obtained from this study will be useful in validating future CFD studies involving heat transfer and film cooling of the shroud and blade tip.
2) The second major objective of this research is to investigate the effectiveness of cooling the blade tip and shroud using different cooling configurations. For shroud cooling, these configurations involve discrete hole cooling using a designed cooling hole pattern and lateral injection of the coolant in the direction of blade rotation, using a slot located just upstream of the turbine blade domain to cool the shroud, using the combination of the earlier mentioned discrete hole pattern and the slot to cool the shroud and investigating the effectiveness of forward, backward and lateral coolant injection (both in and opposite the direction of blade rotation) in cooling the shroud. All these tests are performed with blade rotation and for different coolant flow rates. For tip cooling, the configurations studied include the use of tip cooling holes alone, pressure side shaped holes alone and the tip and pressure side holes combined. Other tip cooling configurations studied include the secondary effects of shroud cooling using the discrete hole pattern on the shroud, the slot on the shroud and the combination of the shroud cooling techniques and the tip cooling techniques. Herein lies the other major uniqueness of this study. Very rarely in literature has shroud cooling been studied this in-depth and this is the first study to the author’s knowledge that investigates the secondary effects and effectiveness of shroud cooling on the blade tip.

The results of this study will advance the understanding of the heat transfer behavior of the blade tip and the shroud especially with the effects of blade relative motion as well as provide new information on the effectiveness of different cooling configurations in protecting the tip and the shroud.
CHAPTER 2: EXPERIMENTAL SETUP AND PROCEDURES

2.1 Facility Design and Experimental Setup

2.1.1 Design Goals

The goal of this research project is to experimentally determine the heat transfer coefficients (HTCs) and film cooling effectiveness for a rotor blade tip and shroud under conditions of rotation. In order to test the film cooling strategy employed on the blade tip, pressure side, and shroud, a rotating turbine heat transfer facility has been designed and fabricated. The testing will be done in a heat transfer wind tunnel built to offer scaled flow conditions to test the effectiveness of the film cooling design. In order to obtain heat transfer coefficient and film cooling data, an experiment will be performed using the 1-D semi-infinite solid assumption with a convective boundary condition and a step change in the free-stream air temperature. In order to apply a step change in free-stream air temperature, a bypass system was created, allowing the air to reach a higher temperature in a bypass loop gradually. Once the air in the bypass loop reached the desired temperature, a gate was opened to allow the hot air to flow over the rotor facility. Thus, the heat transfer test is initiated. Liquid crystal thermography is used to find the temperature history of the blade tip and shroud, and to obtain the heat transfer coefficient and film cooling effectiveness from this data.

Multiple blowing ratios would be tested to gain a better understanding of the heat transfer results. Thus, the coolant flow rate must be controlled and measured along with the free-stream air. Tests have been completed on similar blades in a stationary cascade, but the effects of rotation such as the Coriolis forces need to be taken into consideration. To accomplish the goal of this project, a rotating rig which spins the blades at a constant speed has been designed and built. This rotating rig will allow for an investigation of the heat transfer characteristics of the blade and shroud. The facility design is documented in the following subsections.
2.1.2 Wind Tunnel Design and Construction

The wind tunnel is a closed-loop design with two separate flow paths (Fig. 2.1). In bypass mode, air flows through the by-pass wooden duct allowing no airflow through the test section. For testing, the bypass gate will be opened allowing the heated to flow past the bypass duct and into the annular test section. Four duct dampers are used to control and divert the flow. The dampers are opened and closed using switch-activated electronic actuators. These actuators allow one to switch the flow loops in about 1.5 second.

A 20 kW duct heater is used to heat the air in bypass mode prior to the test. It takes about 30 minutes to heat the air from room temperature to a desired temperature of 55°C. Insulation is used on the sheet metal parts to reduce heat loss and speed up this process. Also, turning vanes are used in the corners to reduce pressure drop and straighten the flow.

A large centrifugal blower is used to circulate the air (Fig. 2.2). It has a 30 horsepower motor and is designed to provide up to 5” H₂O (1,244 Pa) differential pressure and 25,000 CFM (11.8 m³/s) flow rate. A blower setting of 20-35 Hz have been use because of the accuracy of the manometer utilized in this measurement. Two flexible bellows have been added to dampen vibrations, allow for any thermal expansion, and to allow for any misalignment. One is at the fan inlet, and the other is just downstream of the test section (Fig. 2.3).

Several validation and benchmark tests on the tunnel have been performed. These tests include test section velocity tests, wind tunnel pressure drop tests, tunnel and test section temperature tests etc. These tests were run to validate and compare the respective measurements in the wind tunnel with the expected performance from the design and calculations. The tests also served to calibrate the instruments utilized in the testing. It should be noted that the calibration tests were conducted in the absence of the rotor and the other components of the test section.
Figure 2.1: Heat Transfer (Wind) Tunnel and Components

Figure 2.2: Picture of the blower
2.1.3 Test Section Velocity

A liquid manometer and pitot-static probe were used to measure velocity of the free-stream air for varying fan speeds. These velocity measurements are performed with an empty test section. The velocities will differ when the added pressure drops of the rotor and guide vanes are introduced. The results are as follows:

<table>
<thead>
<tr>
<th>Fan Setting (Hz)</th>
<th>Velocity (m/s)</th>
<th>Mass Flow (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>19.21</td>
<td>4.17</td>
</tr>
<tr>
<td>25</td>
<td>24.18</td>
<td>5.24</td>
</tr>
<tr>
<td>30</td>
<td>29.08</td>
<td>6.31</td>
</tr>
<tr>
<td>35</td>
<td>34.20</td>
<td>7.42</td>
</tr>
</tbody>
</table>

Fig. 2.4: Table and plot of mainstream velocity and mass flow rate over fan speed setting
2.1.4 Wind Tunnel Pressure Drop

The pressure drop through the test section loop of the wind tunnel has been recorded for varying fan speeds. This, again, is not including the rotor and guide vanes. Pressure was measured at the fan exit and inlet. The difference between the two is the total pressure drop through the tunnel.

<table>
<thead>
<tr>
<th>Fan Setting (Hz)</th>
<th><em>P</em>&lt;sub&gt;exit (&quot;H2O&quot;)&lt;/sub&gt;</th>
<th><em>P</em>&lt;sub&gt;inlet (&quot;H2O&quot;)&lt;/sub&gt;</th>
<th>Δ<em>P</em> (&quot;H2O&quot;)</th>
<th>Δ<em>P</em> (Pa)</th>
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<td>25</td>
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<td>0.00</td>
<td>1.25</td>
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</tbody>
</table>

Figure 2.5: Pressure drop across the length of the tunnel in test mode vs. the fan speed setting

It has been shown that the velocity varies linearly with the fan setting. The pressure drop is proportional to the square of the velocity and this profile is consistent with that.

2.1.5 Tunnel Temperature Profile

Temperature tests in the tunnel have been done to monitor and document the temperature profile in the tunnel. These temperature tests were done by putting thermocouples at different positions in the tunnel.
There is a thermocouple in the bypass loop at a distance of approximately half a foot from the bypass gate. There is a thermocouple in the test section approximately half a foot after the bypass gate. There is also a thermocouple at the same position as the inlet guide vanes. Finally, there is a thermocouple on the outside of the tunnel measuring the room temperature which is used as a reference temperature. The results of the temperature tests are below.

![Temperature Profile in the Tunnel](image)

**Fig. 2.6: Tunnel temperature profile during flat plate heat transfer testing**

The temperature profile in the tunnel is as expected. The slight variation of the temperature with time is accounted for by the Duhamel’s integration technique used in calculating the heat transfer coefficient and film cooling effectiveness.

### 2.1.6 Test Section Design

A rotating test section (Fig. 2.7) has been designed and is currently being fabricated to obtain heat transfer coefficients and film cooling effectiveness contours on the shroud and blade tip. All 64 blades are cooled as well as the entire circumference of the shroud. Pressured coolant air will be fed through the rotary union, which supplies the air to a hollow shaft. The shaft delivers the pressurized air to the hub assembly.
through four 3/8 inch diameter orifices, which connect to the rotor blades by way of 64 coolant holes drilled on the mounting disk of the hub. The mounting disk directly feeds the air to the rotor blades.

Power will be delivered from the other end of the shaft by way of a Baldor (Fort Smith, AZ) motor. The motor will spin the shaft and hub assembly at a constant speed. The motor shaft and rotor shaft are connected by a coupling, securing attachment. Also the shaft is aligned by two mounted ball bearings. The bearings are also connected to bearing supports, which not only align the bearing at the center of the annular section, but also add structural rigidity to the 16 gauge rolled sheet metal annular sections. In addition, there is a motor support that provides similar reinforcement. An exploded view of the assembly can be seen in the figure below.

![Figure 2.7: Drawing of the rotor assembly](image)

Figure 2.7: Drawing of the rotor assembly

![Figure 2.8: Picture of the SLA blades made](image)

Figure 2.8: Picture of the SLA blades made
The blades used for testing have been manufactured using a 3-D printing process called selective laser sintering (SLS) and pictures of the manufactured blades are shown in Fig. 2.8. These blade models have been built to the dimensions of the model given by IHI and a mounting base with coolant air orifice has been added. These blades are directly mounted to the rotating hub by four bolts. This hub receives air from the hollow shaft and stores this air in a large plenum. The air is then distributed to the 64 blades via orifices on the rim (Fig. 2.9).

Figure 2.9: Drawings of the Rotor with blades and their overall assembly

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An Aluminum shroud using a pre-defined IHI hole pattern has been designed and manufactured (Fig. 2.10). The shroud contains two transparent windows for optical access to the blades using a camera. The shroud assembly consists of several components. These include the shroud, the shroud windows, the shroud plenum, the flanges and other supporting structure. The shroud has an inner diameter of 33.725 inches. The shroud was successfully manufactured to within a tolerance of .003 inches. The shroud has 1696 holes each 1mm in diameter. The IHI hole pattern in the figure below (Fig. 2.11) was used in the design. Also shown in Fig. 2.11 is a table of shroud hole dimensions along several rows.

The shroud has two transparent windows for optical access. The windows sit flushed with the shroud and contain the same IHI hole pattern (Fig. 2.12) as the shroud.

Fig. 2.10: Schematic of Shroud
There is a leading edge gap before the leading edge of the shroud. This gap is currently designed to be 1 mm wide. Coolant air would be fed through the gap separately. We have three separate coolant supply lines. The first line feeds the blades through the hub; the second line feeds the shroud holes while the third line feeds the leading edge gap. The plenum for the shroud holes and leading edge are separated using a flange. The assembly is designed to accommodate different leading edge gap dimensions ranging ...
from 0 mm to 9 mm. This was done by using an L-shaped flange that can be switched out for a different one based on the desired leading edge gap. Details of the shroud assembly with the leading edge gap are shown in Fig. 2.13.

![Shroud assembly](image)

Fig. 2.13: A sectional view of shroud 3D drawing (top left), picture of the actual shroud (right and bottom)

The inner test section is housed in an outer shell which attaches to the shroud and completes the wind tunnel circuit. The test section is supported by one inch threaded rods bolted to a steel frame, while the outer shells are bolted to the frame by ¾ inch threaded rod to offer complete adjustability.

Other components that are part of the test section assembly include the coupling, bearing assembly, shaft etc. They are pictured in the figure below.
Additional components not seen in the solid models include: shaft encoder, wireless thermocouple data acquisition system, and battery pack. The wireless DAQ and battery pack (Fig. 2.14) will attach to the shaft using high strength U-bolts and a custom bracket cut from ¼ inch steel.

Fig. 2.14: Additional components in test section assembly
2.1.7 Summary

The main purpose of this project is to experimentally determine the heat transfer coefficient and film cooling data of a shroud and blade under the condition of rotation. To accomplish this task, a rotating heat transfer facility has been designed and fabricated. This facility consists of a wind tunnel and test section.

Several preliminary and actual tests have been performed on this facility to ensure the facility operates as intended. Some of the results from these tests are included in this section. These results show that the facility is working within the design specifications.

When fabricating several test section pieces like the shroud and blades, several changes had to be implemented to ensure these pieces were within the design specification. These changes include adding structural support to the shroud and manufacturing the blades using a different and better method. Test results that are provided using this facility include cooled and uncooled heat transfer data for the shroud and blade. This data is provided for different blowing ratios and flow velocities. Finally, the tests confirm that the overall fabrication of the facility falls within the specifications of the design.
2.2 General Experimental Procedure and Testing Plan

The following is a summary of the experimental procedure generally used in all tests conducted as part of this research. It should be noted that the setup and procedure utilized in each section of experiments slightly differs. Such details and procedures are provided at the beginning of each section.

This study is conducted in a closed loop wind tunnel that houses a one-and-half turbine stage facility. The design of this facility is described in detail in the preceding section in this chapter. For each set of tests, the following procedure is generally followed:

Prior to the start of a test, the mainstream air is heated in a bypass loop to a pre-determined temperature. At time \( t=0 \), the heated mainstream air is introduced into the test section. This results in a step change in the temperature in the test section as shown in Fig. 2.16. Liquid crystal thermography is used to obtain the temperature on the surface to be tested. Along with an image acquisition system, the surface temperature and the time of occurrence of that temperature are recorded.

![Fig. 2.16: Mainstream Temperature, \( T_m \) vs Time, \( t \), immediately upstream of stator blades during testing](image)

With the resulting step change in temperature and the information collected by the image acquisition system, a transient, one dimensional analysis can be conducted to determine the heat transfer coefficient and film cooling effectiveness on the blade tip or shroud.
The following should also be noted for the operation of the rotor: A few minutes before the start of each test, the rotor powered by a motor with a variable frequency drive is set to the target rotor speed. For shroud testing, this speed ranges from 300 RPM to 700 RPM and for blade tip testing, the target speed is 1200 RPM. The 7.5 horsepower motor is equipped with a variable frequency drive which ensures that the rotor is always at the target speed set on the drive. Without air in the test section, the rotor is powered by the motor. When the mainstream air is introduced into the test section, the variable frequency drive adjusts the loading on the motor such that the rotor is now fully spinning as a result of the mainstream air alone. This experimental setup allows for both shroud testing and tip testing using the free-spin speed of the rotor.

Several shroud tests are performed as part of this research. Mainly, the tests are performed using either discrete holes, a slot or the combination of discrete holes and a slot. As part of this research, several features of discrete holes and their effects on the shroud heat transfer and cooling behavior are investigated. As such, different shroud hole patterns are used. The first set of shroud tests involve using discrete holes oriented lateral to the incoming flow direction and angled at 45 degrees to the surface in the direction of rotation. The discrete shroud holes are arranged in a circumferential pattern and consists of five rows with different pitches. Each row of shroud holes, labeled F1 through F5, consists of a unique number of shroud holes corresponding to different pitches. Row 1, labeled F1, has the most number of holes and row 5, labeled F5, has the least. The number of holes in each row decreases from row 1 to row 5. The shroud cooling holes are arranged in a repeating pattern around the shroud circumference. The arrangement is such that 32 iterations of the same pattern is formed. One of the 32 shroud hole patterns is replaced with an acrylic window for visual access. Testing using this shroud hole pattern is performed at the design speed and for two off-design speeds. The testing matrix for the discrete hole cooling using this pattern is shown below:

<table>
<thead>
<tr>
<th>Rotation Speed</th>
<th>Blowing Ratios</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design Speed (400 RPM)</td>
<td>0, 1.0, 1.5, 2.0, 2.5, 3.0</td>
</tr>
<tr>
<td>Below Design Speed (550 RPM)</td>
<td>0, 1.0, 1.5, 2.0</td>
</tr>
<tr>
<td>Above Design Speed (700 RPM)</td>
<td>0, 1.0, 1.5, 2.0, 2.5, 3.0</td>
</tr>
</tbody>
</table>
Cooling for the shroud holes is provided by an independent cooling supply line. Prior to the introduction of the shroud hole coolant to the shroud hole cooling plenum, the coolant air is bypassed. The shroud hole coolant line is equipped with an inline heater that allows the coolant air to be heated to a predetermined temperature. This is necessary since the liquid crystal measurement technique is a two-temperature test technique and the coolant temperatures are different for each of the two tests. The shroud hole coolant air is introduced to the shroud hole plenum which is located above the shroud using 8 air hoses that are attached to a centralized piping system feeding the shroud holes. Since all the discrete shroud cooling holes are supplied by a single plenum that spans the shroud circumference, the blowing ratio reported in this study is the average blowing ratio through the shroud cooling holes.

The slot cooling is done using an uninterrupted slot on the shroud. The slot width matches the shroud-hole diameters and is therefore 1 mm wide, and is located 7 slot-widths upstream of the blade leading edge. Unlike the shroud cooling holes, the slot is not angled. In addition to slot cooling tests, combined discrete hole and slot cooling tests are performed using the cooling hole pattern described earlier. A testing matrix indicating the planned tests using this configuration is shown in the table below.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Blowing Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Slot Cooling</td>
<td>0.5, 0.75, 1.0, 1.25, 1.5, 1.75, 2.0, 3.0</td>
</tr>
<tr>
<td>Slot/Discrete Hole Cooling</td>
<td>0.5/1.0, 0.5/2.0, 0.5/3.0, 1.0/1.0, 1.0/2.0, 1.0/3.0, 1.5/1.0, 1.5/2.0, 1.5/3.0, 2.0/1.0, 2.0/2.0, 2.0/3.0</td>
</tr>
</tbody>
</table>

The slot coolant is supplied by a separate cooling line and can be independently set relative to the discrete holes. The slot cooling line is also equipped with a separate bypass system that allows the slot coolant to be heated prior to its introduction into the test section. The slot coolant is introduced into the slot plenum using 16 air tubing lines located at equal intervals around the slot plenum.

Tip testing is performed on the blade tip. Attached to the rotor are sixty four rotor blades made using a stereolithographic (SLA) machine. Each blade consists of two cylindrical tip holes as well as six
shaped holes on the blade pressure side. The blades feature a cutback squealer rim with a recessed tip and a tip gap set at 1.7% of the blade span. The cooling of the blade tip can be performed using three distinct means: blade tip and pressure side holes, discrete shroud holes and a shroud slot. The blade tip testing is performed using the testing matrix shown in Table 2.3 below.

<table>
<thead>
<tr>
<th>Cooling Configuration</th>
<th>Blowing Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blade Tip Holes</td>
<td>0, 1.0, 1.5, 2.0, 3.0, 4.0</td>
</tr>
<tr>
<td>Shroud Holes</td>
<td>1.0, 2.0, 3.0, 4.0</td>
</tr>
<tr>
<td>Shroud Slot</td>
<td>0.5, 1.0, 2.0, 3.0</td>
</tr>
<tr>
<td>Shroud Holes/Shroud Slot</td>
<td>2.0/0.5, 4.0/1.0</td>
</tr>
<tr>
<td>Shroud Holes/Shroud Slot/Blade Tip Holes</td>
<td>2.0/0.5/2.0, 4.0/1.0/4.0</td>
</tr>
</tbody>
</table>
CHAPTER 3: MEASUREMENT TECHNIQUE

3.1 Heat Transfer Theory

The goal of this study is to experimentally determine the heat transfer coefficient and film cooling effectiveness for the shroud and blade tip under different rotation speeds and blowing ratio. In order to obtain the heat transfer coefficient and film cooling effectiveness, the one dimensional heat conduction equation (shown in Eqn. 1) using the semi-infinite solid assumption with a convective boundary condition at the surface exposed to the coolant is solved numerically for two sets of data points (Ekkad and Han (2000) and Metzger and Larson (1986)).

\[
\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \tag{1}
\]

Applying the convective boundary condition to the uncooled case, Eqn. (1) becomes,

\[
\frac{T_w - T_i}{T_\infty - T_i} = \left[1 - \exp \left(\frac{h^2 \alpha t}{k^2} \right) \text{erfc} \left(\frac{h \sqrt{\alpha t}}{k} \right)\right] \tag{2}
\]

A set of time-temperature data pairs at each pixel location of the image is obtained for each test using liquid crystal thermography.

In this study, a thin coat of narrow band liquid crystal, which has been calibrated for temperature, \(T_w\) is applied to the inside surface of the shroud. The surface is initially at a temperature, \(T_i\). The free-stream air is heated in a bypass loop until it is higher than the free-stream temperature targeted for use in the testing. The coolant air at temperature, \(T_c\), is also run through a separate bypass system. At time \(t = 0\), the test is initiated resulting in a step change in free-stream air temperature as measured by a thermocouple in the test section upstream of the stator. This results in a change in the shroud surface temperature which is governed by the local heat transfer coefficient of the different locations on the shroud surface and thus, causing a pixel by pixel change in color of the liquid crystal coating on the surface (Klein (1968) and Smith et al (2001)). The pixel by pixel change in color on the shroud surface creates a time-temperature pair for each pixel thus tracking the effects of the local heat transfer coefficient along the shroud surface. A color capture
system consisting of a color CCD camera captures the sequence of images during the test and a precise time stamp for those images. The time of appearance of the highest intensity of the color change for every pixel during the transient test is recorded. As such, the time-temperature pair for every pixel on the shroud surface being studied is obtained for each test and is used to obtain the heat transfer coefficient and film cooling effectiveness for each pixel. The heat transfer coefficient is converted to the Nusselt number using the hydraulic diameter for the shroud heat transfer results.

The time-temperature data pair is used to obtain the heat transfer coefficient and film cooling effectiveness by solving Equation 2 where the film temperature defined in 2c is utilized instead of the freestream temperature (Ekkad and Han (2000)).

\[
T_w - T_i = \left[ 1 - \exp \left( \frac{h^2 \alpha t}{k^2} \right) \text{erfc} \left( \frac{h \sqrt{\alpha t}}{k} \right) \right] \times \left[ \eta T_c + (1 - \eta) T_m - T_i \right] \quad (2b)
\]

Where,

\[
T_f = \eta T_c + (1 - \eta) T_m \quad (2c)
\]

Since there are two unknowns to be solved for in Equation 2, a second test is performed with the coolant heated to a different temperature. Thus, two sets of time-temperature pairs are obtained.

The Duhamel’s superposition theorem, utilized in Equation 3 and Equation 4, is used to account for the variation of the temperatures including the heated coolant temperature with time (Klein (1968)).

\[
T_w - T_{i1} = \sum_{j=1}^{N} \left[ 1 - \exp \left( \frac{h^2 \alpha (t_1 - \tau_j)}{k^2} \right) \text{erfc} \left( \frac{h \sqrt{\alpha (t_1 - \tau_j)}}{k} \right) \right] \times \left[ \eta T_{c1} + (1 - \eta)(\Delta T_{m1})_j \right] \quad (3)
\]

\[
T_w - T_{i2} = \sum_{j=1}^{N} \left[ 1 - \exp \left( \frac{h^2 \alpha (t_2 - \tau_j)}{k^2} \right) \text{erfc} \left( \frac{h \sqrt{\alpha (t_2 - \tau_j)}}{k} \right) \right] \times \left[ \eta (\Delta T_{c2})_j + (1 - \eta)(\Delta T_{m2})_j \right] \quad (4)
\]
The two equations are solved numerically for the heat transfer coefficient and film cooling effectiveness for every pixel location on the test piece using the extended Newton-Raphson method for a system of equations.

3.2 Liquid Crystal Thermography

Liquid crystals are an accurate and time-tested method of measuring surface temperatures and heat transfer on gas turbine components. Liquid crystals have been a temperature sensing tool since as early as 1968 (Klein, 1968). In its base form, it is a substance in an intermediate solid-liquid phase. It has several advantages as a sensing tool. One of such advantages is its absolute coverage on the coated surface. In practice, the coverage from liquid crystals is limited to the kind of optical access to the coated surface. Liquid crystals derive their sensing properties from their molecular structure and these structures vary depending on the type of liquid crystal.

3.2.1 Classification of Liquid Crystal by method of formation

Liquid crystals display the useful optical properties of crystals because of their order of orientation. Nevertheless, they lack positional order and thus their liquid-like nature. So named, they can be classified along different lines including method of formation and use. The two major types of liquid crystals according to method of formation are Thermotropic liquid crystals and Lyotropic liquid crystals. Thermotropic liquid crystals reflect light when their temperature falls within a certain range. Their orientation is responsible for their characteristic anisotropic behavior such as their refractive index. (Buffone and Sefiane (2004)). Lyotropic liquid crystals on the other hand are induced when their orientation is altered by the addition of a solvent.

Thermotropic liquid crystals can be subdivided into two mesophases- smectic and nematic (Hay and Hollingsworth (1996)) with the ordering and positioning of the molecules being the primary difference. Smectic liquid crystals are ordered such that they are layered and the longer axis of their molecules are
perpendicular to the plane of the layer. In nematic liquid crystals, the molecules are not layered and the long axes are parallel. A director $n$ is used to describe the general molecular orientation in nematic liquid crystals. This can be seen in fig. 3.2 below.

Fig. 3.1: Classification of Liquid Crystal (Hallcrest Inc Handbook)

Fig. 3.2: Schematic of smectic mesophase (left) and nematic mesophase (right) (Hallcrest Inc Handbook)
Nematic liquid crystals can be further divided into two classes- **chiral** or **achiral**. In a chiral nematic liquid crystal, the molecules of the liquid crystal are asymmetric about the long axis (Jay and Hollingsworth). The forces between each molecular results in a rotation of the molecules compared to neighboring molecules. As such, the director \( n \) shown in Fig. 2 traces a path comparable to that of a helix. Initially, the director \( n \) starts on a parallel path. Eventually, it ends being perpendicular to the molecular plane. The helical path traced by the director \( n \) can be seen in Fig. 3 where the pitch is defined as the distance it takes the director \( n \) to complete one rotation of 360°.

![Diagram](image)

**Fig. 3.3**: Schematic of a chiral nematic liquid crystal (Hallcrest Inc Handbook)

It is the combination of the pitch of the helix and the properties of the incident light including the relative position of the observer that determines the observed color from chiral nematic liquid crystals (Herold and Wiegel, 1981). The primary principle responsible for the reflected light is governed by Bragg’s law and the dominant wavelength reflected by the liquid crystal depends on the pitch of the helix. Since the pitch of a chiral nematic liquid crystal is dependent on the ordering and positioning of the molecules and the ordering and positioning of the molecules is dependent on the temperature, chiral nematic liquid crystals can be used as an effective temperature sensing tool if properly calibrated.
### 3.2.2 Liquid Crystal Calibration

The calibration of a chiral nematic liquid crystal involves an association of the order and position of the molecules to a particular temperature. In practice, the temperature of interest is associated with color since the wavelengths of the reflected light represents a color and this is triggered at certain temperatures. With three primary colors and $C_1$, $C_2$ and $C_3$ representing these colors, the resulting combination of these colors in certain proportions is representative of a particular temperature. This association, known as trichromatic decomposition is further expanded on by Pratt and by Levine. One such association by Akino (1989), is a relatively linear association between the primary colors and temperature seen below

\[ T = aC_1 + bC_2 + dC_3 + e \quad (1) \]

Where $a$, $b$, $d$ and $e$ are constants found in the calibration process using linear regression. Instead of the tri-variables, Levine (1985) has shown that another form can be used for the interpretation of color. This variable-intensity or brightness is directly proportional to the sum of the tri-variables

\[ I \propto (C_1 + C_2 + C_3) \quad (2) \]

As such, the contribution of each primary component to the overall intensity can be quantified. The coordinates of the relative contribution referred to as the chromaticity can be seen in the equation below.

\[ c_1 = \frac{C_1}{I}, \quad c_2 = \frac{C_2}{I}, \quad c_3 = \frac{C_3}{I} \quad (3) \]

Different choices can be made for what system to use for the primary colors. With the advent of CCD cameras, red, green and blue (RGB) is frequently chosen. All colors can be formed from the combination of these three colors and the compatibility to the system used in imaging software makes the RGB system a good choice for any color capture scheme.

While the intensity is the combination of the sum of the tri-variables in a color, the dominant wavelength in a color is defined as the hue. The hue can be obtained from the primary colors using the equation below
\[ H = \arccos \left[ \frac{0.5((R-G)+(R-B))}{((R-G)^2+(R-B)(G-B))^{1/2}} \right] \]  

Using the coordinate system shown in Fig. 3.4, the hue is proportional to the angle between two lines in the coordinate system (Ireland and Jones, 2000). The saturation which is defined as the ratio of OP to OQ is the purity of the color or the distance from the white light point (Hay and Hollingsworth, 1996).

Fig. 3.4: Hue defined as an angle in RGB space. (Ireland and Jones, 2000)

Together with the intensity, the hue and saturation create a color model appropriately named HSI color model schematically shown in fig. 3.5 which allows for in-depth specificity about a color.

Fig. 3.5: Schematic of HSI color representation (www.mdpi.com)
Armed with this information, a HSI calibration technique can be utilized to associate color from liquid crystal with a specific temperature to a high degree of accuracy.

### 3.2.2.1 HSI Calibration Technique

In this technique, a best fit polynomial is used to calibrate the hue-temperature correlation. Since the correlation is non-linear, only data points in a limited portion of the liquid crystal bandwidth are utilized. A sample hue-temperature calibration curve is shown in the figure below. (Smith et al, 2001)

![Calibration Curve](image)

**Fig. 3.6:** An example hue vs temperature calibration curve (Smith et al, 2001)

The steady state HSI technique involves painting the surface to be used for calibration with the liquid crystal. A thermocouple is used to measure the temperature of the surface. To minimize the error involved, a small piece of the surface to be tested is used in the calibration. Heat at a constant heat flux is added to the surface to be tested. After the surface reaches a steady state, the temperature is measured by the thermocouple and the images of the surface are recorded. The heat added to the surface is varied after
each calibration data point. The color on the surface is recorded using a camera. The hue is calculated using equation (4). The heat transfer coefficient can also be calculated from this calibration test using the quotient of the heat flux and the difference between the wall temperature and freestream temperature.

### 3.2.2.2 Transient HSI Testing

In the transient HSI testing, the test section initially at a temperature, \( T_0 \), is suddenly exposed to the freestream air at a higher temperature, \( T_m \). The piece to be tested is made of a low conductivity material and such that in combination with the thickness of the test surface, the 1-D assumption is not violated. Once exposed to the hot freestream air, there is a rapid rise in the temperature in the test section. A camera as part of the image processing software records images of the test section at regular intervals. Each image is time-stamped such that at a particular time, the wall temperature can be obtained from a HSI calibration curve. With the wall temperature, mainstream temperature, time instance of said wall temperature and the properties of the test section piece, the heat transfer coefficient can be calculated using the equation below:

\[
\frac{T_w - T_i}{T_\infty - T_i} = \left[ 1 - \exp\left( \frac{h^2at}{k^2} \right) erfc\left( \frac{h\sqrt{at}}{k} \right) \right]
\]

Using a narrow band liquid crystal, the wall temperature is already pre-defined and associated with a particular color appearance on the surface coated with liquid crystal. With a narrow band liquid crystal technique, the image processing system is set up to record the time of appearance of the color of interest. Since this color of interest is associated with a definite wall temperature, the heat transfer coefficient can be calculated using Eqn. 1. Bunker and Metzger (1990) have shown the use of the green band tracking technique to be more accurate than the wide band technique. Vedula and Metzger (1991) used this technique to obtain both the heat transfer coefficient and film cooling effectiveness. In this case, Rqn 1 is modified by using the film temperature instead of the mainstream temperature. The modified form of the equation is shown below.

\[
T_w - T_i = \left[ 1 - \exp\left( \frac{h^2at}{k^2} \right) erfc\left( \frac{h\sqrt{at}}{k} \right) \right] \times \left[ T_c + (1 - \eta)T_m - T_i \right]
\]
Where,

\[ T_f = \eta T_c + (1 - \eta) T_m \]  

(2c)

### 3.2.3 Liquid Crystal Color Play and Standardization

A thin layer of narrow band liquid crystal (SPNR35C1W from Hallcrest LLC) with a response time on the order of 3 milliseconds (Ireland and Jones, 1987) is applied to any surface to be tested. SPNR35C1W means the liquid crystal has a red start of 35 °C and a bandwidth of 1°C. Thus, green color occurs at 35.2 °C.

![Liquid Crystal Bandwidth](image)

Fig. 3.7: Liquid Crystal Bandwidth (Adapted from hallcrest.com)

For better clarity, the surface is coated with a layer of black paint (SPB-100 from Hallcrest LLC) before the liquid crystal paint is applied. A calibration test for the liquid crystal has been independently conducted and validated to verify the color change temperature. Furthermore, this transient liquid crystal scheme has been validated against published flat plate data shown in an earlier section. The green band tracking technique by Vedula and Metzger (1991) and Metzger et al. (1991) is used for this paper.
3.2.4 Classification of Liquid Crystals by Method of Use

Liquid crystals can be classified by the method of use on a surface. In this, there are three main classifications of liquid crystals. They are unsealed liquids, micro-encapsulated coatings and coating sheets. Each method has its advantages and applications it is more suitable for.

3.2.4.1 Unsealed Liquid

In this form, the liquid crystal applied to a surface is not coated or surrounded by any form of protection. The obvious advantage of unsealed liquid crystals is in the brilliance and sharpness of the display. Nevertheless, when exposed to the elements like most long term applications using liquid crystals are, this form of liquid crystal is unable to serve adequately for more than a short duration of time. Its long term viability is affected by the weather and atmospheric conditions, chemicals and other forms of contaminants. As such, it is used in short term applications where the brilliance of the display is more important than the shelf life of the liquid crystal.

3.2.4.2 Liquid Crystal Coated Sheet

This is the most common form of liquid crystal application because of its many benefits and ease of use. In this form, the liquid crystal is coated between a black background to enhance visual acuity and a transparent substrate for visual access. This is the form of liquid crystal utilized in this project. While still not intended for extreme long term use, it does have the advantage of being easy to apply and use. Furthermore, it is not as exposed to the elements as the unsealed liquid form.

3.2.4.3 Microencapsulated Coatings

The microencapsulated form of liquid crystal is most commonly used in long term applications where maintaining the hue-temperature relationship from the calibration is more important than the sharpness of the images taken of the liquid crystal. In this form, the liquid crystal droplets are encased or encapsulated
in a protective polymer which protects it from the elements and harsh conditions. This form of liquid crystal is typically applied in a spray-able form and thus making it also fairly easy to apply.

### 3.3 Validation of Use of 1-D Semi-Infinite Assumption

Before testing, calculations have been done to determine how long the 1D semi infinite assumption is valid for a particular surface thickness and heat transfer coefficient. The penetration depth in this case has been defined as the time it takes heat to penetrate through the thickness of the surface to be studied. The values on the y-axis of the plot have been multiplied by two. Thus, a safety factor of two has been added to the results and much room has been given to ensure the 1-D assumption is not violated. The results shown in the figure below are for the blade and its material properties.

![Penetration Depth vs. time for different heat transfer coefficient](image)

**Fig. 3.8: Penetration Depth vs. time for different heat transfer coefficient**
From this plot, a test duration of 30 seconds is appropriate for a blade with a penetration depth of 1 cm or a tip thickness of 0.5 cm (due to the added safety factor. The sla material properties are: density = 1250 kg/m$^3$, thermal conductivity = 0.173 W/(m K) and specific heat capacity = 1970 J/Kg K

### 3.4 Measurement Technique Validation

To validate the measurement technique and the wind tunnel design, a flat plate test has been performed in the wind tunnel using the liquid crystal measurement technique. The results of this test is compared to data in literature with the same or similar testing conditions. The validation test is performed for both discrete hole and slot cooling. The results for cooling using a slot is shown in Fig. 3.9. The heat transfer and film cooling results for cooling using discrete holes is shown in Fig. 3.10 and Fig. 3.11.

![Graph showing film cooling validation test results using an upstream slot at M=0.5](image)

- **Present Study, M=0.5, Experimental**
- **Seban et al, M=0.5, Correlation (Best fit line)**
- **Seban et al, M=0.55, Experimental**
- **Librizzi and Cresci, M=0.5, Correlation (Heat Sink Model)**
- **Kutateladze and Leont’ev, M=0.5, Correlation (Heat Sink Model)**
- **F.F. Simon, M=.45, Experimental**

Fig. 3.9: Film cooling validation test results using an upstream slot at M=0.5
Fig. 3.10: Laterally Averaged film cooling effectiveness for discrete hole cooling at M=0.5 for flat plate

Fig. 3.11: Laterally Averaged h/h₀ for discrete hole cooling at M=0.5 for flat plate
3.5 Uncertainty Analysis

Every measurement technique has a level of uncertainty associated with it. The overall uncertainty associated with a technique is a sum of the uncertainties of the individual elements that comprise the results. To calculate the uncertainty associated with this measurement technique, the methodology of Kline and McClintock (1953) is utilized. The methodology is as follows:

Assuming a function \( R \) is composed of several independent variables \( x_1, x_2, x_3, \ldots, x_n \) which can be measured. Therefore, \( R \) can be represented as

\[ R = f(x_1, x_2, x_3, \ldots, x_n) \tag{1} \]

The effect of the uncertainty in a single measurement (i.e. one of the \( x \)'s) on the calculated result, \( R \), if only that one \( x \) where in error is:

\[ \delta R_{x_i} = \frac{\partial R}{\partial X_i} \delta X_i \tag{2} \]

where the partial derivative weights the effect of that uncertainty on the overall uncertainty (sensitivity coefficient). When several independent variables \( (x_1, x_2, x_3, \ldots, x_n) \) are used in calculating the results of the function, \( R \), the individual terms are combined by a root-sum-square method to obtain the uncertainty as shown below

\[ \delta R = \left( \sum_{i=1}^{N} \left( \frac{\partial R}{\partial X_i} \delta X_i \right)^2 \right)^{1/2} \tag{3} \]

and the relative uncertainty is given by \( \frac{\delta R}{R} \).

In the case of the experiments done in this work, the equation to be utilized is

\[ T_w - T_i = \left[ 1 - \exp \left( \frac{h^2 \alpha t}{k^2} \right) \text{erfc} \left( \frac{h \sqrt{\alpha t}}{k} \right) \right] \times [T_m - T_i] \tag{4} \]
Where, \( h = f(T_w, T_i, \alpha, k, t, T_m) \) or \( h = f(f_1, f_2, f_3, f_4, f_5, f_6) \).

The uncertainty is

\[
W_f = \left[ \left( \frac{\partial h}{\partial f_1} W_1 \right)^2 + \left( \frac{\partial h}{\partial f_2} W_2 \right)^2 + \left( \frac{\partial h}{\partial f_3} W_3 \right)^2 + \ldots + \left( \frac{\partial h}{\partial f_n} W_n \right)^2 \right]^{1/2} \tag{5}
\]

For an explicit equation \( y = f(x) \), finding the partial derivative is a straightforward process. However, for an implicit equation like is used here, an approximation is used to numerically solve the equation. In this case, the approximation

\[
\frac{\partial h}{\partial f} = \frac{h(f_b) - h(f_a)}{f_b - f_a} \tag{6}
\]

where \( f_b = f_a + 0.0000001 \) (i.e. \( 1 \times 10^{-7} \)) and \( f_a \) consists of the actual values is used.

An example utilizing the highest uncertainty case for \( h \) in one of the experiments is shown in Table 3.1

<table>
<thead>
<tr>
<th>Factor(f)</th>
<th>Value of the factor</th>
<th>dh/df</th>
<th>Wi (error)</th>
<th>([(dh/df).Wi]^2]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T_w )</td>
<td>35 C</td>
<td>9.916</td>
<td>0.1</td>
<td>0.98327056</td>
</tr>
<tr>
<td>( T_i )</td>
<td>27 C</td>
<td>-7.028</td>
<td>0.2</td>
<td>1.97571136</td>
</tr>
<tr>
<td>( \alpha )</td>
<td>1.0861E-07</td>
<td>-1.68E+08</td>
<td>.03*( \alpha )</td>
<td>0.29832224</td>
</tr>
<tr>
<td>( k )</td>
<td>0.19</td>
<td>316.8634</td>
<td>.03*( \alpha )</td>
<td>3.26207444</td>
</tr>
<tr>
<td>( t )</td>
<td>10 s</td>
<td>-3.0102</td>
<td>0.005</td>
<td>0.00022653</td>
</tr>
<tr>
<td>( T_m )</td>
<td>55 C</td>
<td>-2.8331</td>
<td>1</td>
<td>8.02645561</td>
</tr>
<tr>
<td>Sqrt of Sum</td>
<td></td>
<td></td>
<td></td>
<td>3.81392983</td>
</tr>
<tr>
<td>( h_value )</td>
<td></td>
<td></td>
<td></td>
<td>60.204</td>
</tr>
<tr>
<td>Uncertainty (%)</td>
<td></td>
<td></td>
<td></td>
<td>(sqrt of sum/h_value)*100</td>
</tr>
</tbody>
</table>
For each set of experiments, a Matlab code has been written to solve equation 5 using the approximation in equation 6. A time range of 1 through 60 seconds (so that the calculation includes every possibility within the testing duration) is utilized. A freestream temperature range from 50 to 60 °C is also chosen. Ti, Tw, α and k are constants. The code runs through the possible combinations and the highest uncertainty is picked out of the stack.

Similarly, the film cooling effectiveness is calculated as such with the known heat transfer coefficients and the equation below

\[ T_w - T_i = \left[ 1 - \exp\left(\frac{h^2\alpha t}{k^2}\right) \text{erfc}\left(\frac{h\sqrt{\alpha t}}{k}\right) \right] \left[ \eta T_c + (1 - \eta)T_m - T_i \right] \]  \hspace{1cm} (7)

An example result including the uncertainty for the film cooling effectiveness is shown below

Table 3.2: Example film cooling effectiveness uncertainty results

<table>
<thead>
<tr>
<th>Factor(f)</th>
<th>Value of the factor</th>
<th>deta/df</th>
<th>Wi (error)</th>
<th>[(deta/df).Wi]^2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tw</td>
<td>35 C</td>
<td>-0.1024</td>
<td>0.1</td>
<td>0.000104858</td>
</tr>
<tr>
<td>Ti</td>
<td>27 C</td>
<td>0.0667</td>
<td>0.2</td>
<td>0.000177956</td>
</tr>
<tr>
<td>α</td>
<td>1.0861E-07</td>
<td>2.64E+06</td>
<td>.03*α</td>
<td>7.41107E-05</td>
</tr>
<tr>
<td>k</td>
<td>0.19</td>
<td>-0.8864</td>
<td>.03*α</td>
<td>2.55276E-05</td>
</tr>
<tr>
<td>t</td>
<td>10 s</td>
<td>0.2872</td>
<td>0.005</td>
<td>2.0621E-06</td>
</tr>
<tr>
<td>Tm</td>
<td>55 C</td>
<td>0.0293</td>
<td>1</td>
<td>0.00085849</td>
</tr>
<tr>
<td>Tc</td>
<td>27 C</td>
<td>0.0357</td>
<td>0.1</td>
<td>1.27449E-05</td>
</tr>
<tr>
<td>h</td>
<td>60</td>
<td>0.0023</td>
<td>3.75</td>
<td>7.43906E-05</td>
</tr>
<tr>
<td>Sqrt of Sum</td>
<td></td>
<td></td>
<td></td>
<td>0.036471071</td>
</tr>
<tr>
<td>eta_value</td>
<td></td>
<td></td>
<td></td>
<td>0.4081</td>
</tr>
<tr>
<td>Uncertainty (%)</td>
<td>(sqrt of sum/eta_value)*100</td>
<td></td>
<td></td>
<td>8.936797626</td>
</tr>
</tbody>
</table>
CHAPTER 4: SHROUD HEAT TRANSFER AND COOLING

4.1 Heat Transfer to an Actively Cooled Shroud with Blade Rotation

An experimental study of the shroud heat transfer behavior and the effectiveness of shroud cooling is undertaken in a single stage turbine at low rotation speeds. The shroud consists of a periodic distribution of laterally-oriented cooling holes that are angled at 45 degrees to the shroud surface in a repeating circumferential pattern, and has 5 unique hole pitches in the axial direction. Measurements of the normalized Nusselt number and film cooling effectiveness are done using liquid crystal thermography. These measurements are reported for the no coolant case, and nominal blowing ratios of 1.0, 1.5, 2.0, 2.5 and 3.0. The tests are performed at an inflow Reynolds number of 17,500 corresponding to a scaled down design rotation speed of 550 RPM, and two off-design speeds imposed by a motor: (1) a rotation speed below the design speed (400 RPM) and (2) a rotation speed above the design speed (700 RPM). The results at the design speed show that increasing the blowing ratio increases the area-averaged film cooling effectiveness, while the Nu/Nu₀ in the shroud hole region decreases. As the rotor speed is changed from the design speed, the high Nu/Nu₀ region migrates on the shroud surface. This migration affects the coolant coverage in the shroud hole region resulting in increased coolant coverage at below-design rotation speeds and decreased coolant coverage at above-design rotation speeds.

At all rotation speeds, as the blowing ratio increases the area-averaged film cooling effectiveness in the shroud hole region increases. Decreasing the circumferential shroud coolant hole spacing changes the lateral heat transfer profile from a periodic sinusoidal distribution for a shroud hole spacing of P/D = 10.4 to a more even distribution for a smaller P/D = 4.8.
4.1.1 Test Configuration and Conditions

Before running a heat transfer experiment, the free-stream air is heated by a 20 kW duct heater located just downstream of the 30 HP centrifugal fan and passed through the wind tunnel bypass loop. Use of the bypass loop allows adequate time for the air to reach a pre-set temperature while allowing the test section, to maintain room temperature. The centrifugal fan is controlled using a variable frequency drive that allows the user to operate the fan rotor at different speeds resulting in different mainstream bulk velocities. However, only the mainstream bulk velocity at the design speed is used for the tests. A motor attached to the rotor through the shaft is used to adjust the rotor speed for off-design testing. The stator-rotor-stator schematic and the velocity diagram showing the three testing conditions are shown below.

![Stator-rotor-stator arrangement](image1)

Fig. 4.1: Stator-rotor-stator arrangement

![Rotor inlet velocity triangle](image2)

<table>
<thead>
<tr>
<th>Condition</th>
<th>$U^* = U/U_{design \text{ speed}}$</th>
<th>$\beta_2^* = \beta_2/\beta_{2 \text{ design speed}}$</th>
<th>$\omega_2^* = \omega_2/\omega_{2 \text{ design speed}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Below Design Speed (400 RPM)</td>
<td>0.73</td>
<td>2</td>
<td>1.73</td>
</tr>
<tr>
<td>Design Speed (550 RPM)</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Above Design Speed (700 RPM)</td>
<td>1.27</td>
<td>-1</td>
<td>1</td>
</tr>
</tbody>
</table>

Fig. 4.2: Rotor inlet velocity triangle
Coolant air flows through a supply line and is injected at room temperature or is pre-heated to a predetermined temperature. At test time $t = 0$, the pre-heated free-stream air is introduced into the test section through electrically actuated air dampers, resulting in a step change in temperature in the test section as shown in Fig. 4.3. Simultaneously, the coolant air is introduced into the test section through a pneumatically actuated ball valve. The step-change in temperature in the test section because of the heated free-stream air allows for a transient, 1D analysis of the gas turbine shroud using liquid crystal thermography.

![Freestream Temperature vs Time](chart.png)

Fig. 4.3: Freestream Temperature vs Time immediately upstream of stator blades during testing

Since a two test method is used in the analysis, the coolant air temperature differs on both tests. The coolant air is preheated to a different temperature for the second test. The shroud, which was precision machined using a CNC lathe, is 856.6 mm +/- 0.1 mm in diameter and consists of 32 iterations of a cooling hole pattern. Each pattern consists of laterally oriented holes angled at 45 degrees in the direction of rotation as shown in Fig. 4.4 and was precision machined using a CNC mill.
The shroud holes are arranged in 5 columns labeled F1 through F5 as shown in Fig. 4.4, with column 1 labeled F1 at the leading edge of the shroud. Each column of shroud holes has a unique hole pitch also given in Fig. 4.4, which allows for the study of the effect of the hole pitch on the shroud heat transfer distribution. The hole pitch increases from hole column F1 to hole column F5. Coolant air is supplied to the shroud plenum located above the shroud by 8 air hoses attached independently to a piping system. The coolant air is fed to the holes from the same plenum. Thus, the blowing ratio used in this paper is the average blowing ratio through the holes. Visual access is available on the shroud through optically clear acrylic windows. A network has been setup to obtain the temperature data from the data acquisition system. Thermocouples are installed at multiple locations in the test section and in the facility. A shaft encoder is used to monitor the rotation of the rotor. The coolant air flow is measured and monitored by flow meters.
The free-stream air bypass gate and the coolant flow air bypass valves are actuated and controlled by a switch which allows for the simultaneous opening and closing of all gates and bypass systems. LABVIEW is used for overall data acquisition synchronization.

4.1.2 Results

The tests were conducted to obtain detailed heat transfer coefficient and cooling effectiveness results on the shroud for the no coolant case and the coolant cases with blowing ratios of 1.0, 1.5, 2.0, 2.5 and 3.0 at the design rotation speed of 550 RPM (corresponding to a Re = 17,500 based on the flow properties at the entrance of the test section) and at two off-design conditions corresponding to rotation speeds of 400 RPM and 700 RPM.

The shroud surface has been divided into two regions for further analysis. The shroud hole region is defined as the region containing the shroud holes from x/D = 0 to x/D = 20 while the downstream region represents the region after x/D = 20. The coolant holes have also been labeled as Row 1 through Row 5 such that Row 1 is the row of holes at x/D = 1 and Row 5 is the row of holes at x/D = 18.

The heat transfer coefficient results are converted to a Nusselt number (Nu) and normalized with the Nusselt number (Nu0) as given by the Dittus Boelter equations (1930) for flow in an annulus using the same hydraulic diameter.

\[
Nu_0 = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4}
\]  

(5)

The film cooling effectiveness results \( \eta_f \) as calculated from Equation 2, are also presented. The horizontal and vertical length scales for the shroud have been normalized with the cooling hole diameter, D, and presented as x/D and y/D respectively. The location x/D = 0 is the leading edge of the shroud and the blades rotate counter-clockwise, bottom to top in the contour plots. The 45° angled holes inject coolant from bottom to top with the coolant exit at the top and on the inside surface of the shroud. The results showing
the cooling effectiveness under the different blowing ratios at the design condition (Re=17,500) and the corresponding speed is first presented. The effect of the off-design flow conditions on the results are presented subsequently.

4.1.2.1 Effect of Blowing Ratio:

Testing was done at five different blowing ratios of 1.0, 1.5, 2.0, 2.5 and 3.0 at Re=17,500 and corresponding to the design rotation speed of 550 RPM. A no coolant test with the shroud holes blocked was also done as a reference case. The contour plots for the normalized Nusselt number and film cooling effectiveness results are show in Fig. 4.5 and Fig. 4.6 respectively.

From these contour plots, it is clear that the blowing ratio does have a significant and quantifiable effect on the shroud heat transfer and film cooling effectiveness. Fig. 4.5(a) shows the Nu for the no-coolant case, while Figs. 4.5(b)-4.5(f) show the Nu values for progressively higher values of the blowing ratio. It is clear that the introduction of the coolant reduces the Nu/Nu₀ values in the shroud hole region as well as the region downstream. The coolant air reduces the mixed-mean temperature adjacent to the shroud thus lowering the potential driving temperature difference and the heat flux to the shroud. It is conjectured that the coolant air delivered through the plenum and the coolant delivery tubes have lower turbulence levels than the tip-gap or near-shroud regions and ameliorate the near-shroud heat transfer rates. With increasing blowing ratio, and the introduction of more coolant, the above effect of lowered turbulence is enhanced (since more of the near-shroud flow is originating from the coolant side) leading to decreasing Nu/Nu₀ values as seen in the contour plots in Fig. 4.5. At the higher blowing ratios, the jet footprint on the shroud merge and continuous cooling in the circumferential direction (see also Fig. 4.6) is observed for the holes with the lower circumferential spacings (the first three row of holes). It should be noted that the lower heat transfer coefficients associated with the coolant jets persist downstream of the cooling holes as clearly seen in Fig. 4.6f.
Fig. 4.5: $\text{Nu}/\text{Nu}_0$ at the Design Speed for (a) No Coolant Case (b) BR = 1.0 (c) BR = 1.5 (d) BR = 2.0 (e) BR = 2.5 (f) BR = 3.0
Figure 4.6 shows the film cooling effectiveness contours at different blowing ratios, and shows an increase in the film cooling effectiveness values as the blowing ratio increases. This trend is consistent even at higher blowing ratios as can be seen in a comparison of Fig. 4.6b at BR=1.5 and Fig. 4.6c at BR=2.0. Note that even for BR as high as 3.0, the cooling effectiveness shows an increasing trend which is unlike
the widely reported cylindrical flat plate data, where at the higher BR, the cooling jet tends to blow off. Increasing the blowing ratio also causes the exiting coolant jets to migrate further from the exit holes as seen in Fig. 4.6. For smaller shroud hole pitches, the exiting coolant jet from one hole can migrate far enough at higher blowing ratio to attach to the jet from a neighboring hole and thus provide better coolant coverage in that hole area as can be seen from the first row of holes in Fig. 4.6. Thus, the hole spacing effect combines with the blowing ratio to influence the coolant coverage area. Fig. 4.6 also shows that while the blowing ratio has a significant effect on the shroud hole region, it has a negligible effect downstream of the shroud hole region indicating the loss of coolant coverage.

To summarize the effects of the blowing ratios on the shroud heat transfer behavior, the area averaged Nu/Nu₀ and area averaged film cooling effectiveness, ηᵢ, in the shroud hole region (defined as the region between x/D=0 and x/D=20) are shown in Fig. 4.7 and Fig. 4.8

![Fig. 4.7: Area Averaged Nu/Nu₀ in the shroud hole region as a function of blowing ratio for different rotation speeds.](image1)

![Fig. 4.8: Area-Averaged ηᵢ in the Shroud Hole Region as a Function of Blowing Ratio for different rotation speeds.](image2)
Figure 4.7 shows a monotonic decrease in the area-averaged Nu/Nu₀ values in the shroud hole region for the design (free spin at 550RPM) case as the blowing ratio increases from 0 (the no-cooling case) to 3. For the 550RPM case, the level of decrease is nearly 50% between the no-cooling case and BR of 3.0. At the other off-design RPM’s, comparable reductions are noted.

For the area-averaged film cooling effectiveness shown in Fig. 4.8, a three-fold increase in the area-averaged film cooling effectiveness is observed as the blowing ratio increases from 1.0 to 3.0 for the design speed of 550RPM. This is unlike the results that would be expected with cylindrical holes and streamwise coolant injection as presented, for example, in Colban et al (2011) and Bauldauf et al. (2002). Their results show that cylindrical holes result in an increase in film cooling effectiveness up to BR = 0.8 and above which the film cooling effectiveness decreases with increasing blowing ratio due to coolant separation. In the current tests, the cylindrical holes are not in the streamwise direction as in Kwak and Han (2003), but are oriented with a surface angle of 45° in the direction of rotation which is lateral and at 90° to the incoming axial flow direction. Studies such as Schmidt and Bogard (1996) and Sathyamurthy and Pantakar (1990) have shown that lateral coolant injection even without rotation provide higher film cooling effectiveness at higher blowing ratios than streamwise coolant injection, but even with lateral injection the flat plate data shows a reduction in cooling effectiveness past a critical BR. However in the present study, lateral coolant injection combined with the blade rotation and the corresponding rotor-shroud surface interactions, the coolant jet exiting into the tip gap in the direction of rotation results in increased area averaged film cooling effectiveness with increasing blowing ratio.

Testing for these results were done at a density ratio closer to unity (DR=1.1). While the density ratio has been shown by Goldstein et al (1974) and Pietrzyk et al (1990) to have an effect on film cooling, this effect has not been studied in this chapter.
4.1.2.2 Effect of Off-Design Flow Conditions:

In addition to the design speed, the effectiveness of shroud cooling at off-design speeds was also investigated. Two off-design blade speeds were chosen with one speed (400 RPM) below the design speed and the other (700 RPM) above the design speed. The shroud heat transfer and film cooling results at the off-design speeds are shown in Fig. 4.9 through Fig. 4.12 while the area-averaged results are shown in Figs. 3 and 4.

Fig. 4.9: $\frac{Nu}{Nu_0}$ at Below Design Speed for (a) No Coolant Case (b) BR = 1.0 (c) BR = 1.5 (d) BR = 2.0
Fig. 4.10: $\eta_f$ at Below Design Speed for (a) BR = 1.0 (b) BR = 1.5 (c) BR = 2.0

Fig. 4.11: $\frac{Nu}{Nu_0}$ at Above Design Speed for (a) No Coolant Case (b) BR = 1.0 (c) BR = 1.5 (d) BR = 2.0 (e) BR = 2.5 (f) BR = 3.0
At off-design speeds, the relative inflow angle and velocity changes, and can influence the shroud heat transfer and the cooling effectiveness. When the rotation speed is decreased below the design speed, the relative inflow angle and the relative inflow velocity increase. It should be noted that for the 400 RPM case, the relative inflow angle doubles from 30° in the design case to 60°. Furthermore, the relative inflow velocity is about 1.7 times the design relative inflow velocity. This increase especially of the inflow angle is likely to result in flow separation around the leading edge region of the shroud and flow impingement.
downstream of this region. Thus, higher heat transfer rates are to be expected and are observed away from the shroud leading edge as can be seen in Fig. 4.9. For the higher rotation speed of 700 RPM, the relative inflow velocity is similar to the design case but the inflow angle changes from a positive angle to a negative angle and the approaching flow near the leading edge has an orientation opposite to the pressure gradient in the tip gap. This change in the flow pattern at the above design speed where the flow from the pressure-to-suction side tip-gap region near the leading edge encounters the incidence flow at a negative angle potentially leads to streamline impingement, higher levels of turbulence and higher heat transfer rates near the shroud leading edge region. The changes in the inflow angle and velocity as the rotor speed transitions from the below design speed through the design speed to the above design speed and the resulting effects of these changes on the flow profile and subsequently, the heat transfer rate are responsible for the apparent migration of the critical region of high heat transfer from the shroud trailing surface at below design speeds to the shroud leading surface at above design speeds as observed in Fig. 4.9 and Fig. 4.11.

The change in the flow profile in the shroud hole region is particularly important to understanding the coolant behavior in that region. Going from the design speed at 550RPM to the off design speeds has a significant impact on shroud cooling effectiveness because the freestream flow profile (velocity and flow angle) affects the freestream-coolant air interaction and thus, the capacity of the coolant air to effectively protect the shroud surface. At below design speeds, the region of high freestream impingement and high heat transfer rate on the shroud occurs downstream of the shroud hole region. Thus, the coolant jets exiting the shroud holes can and do attach to the shroud surface forming a protective blanket and a more uniform coolant coverage (Fig. 4.10) in this region than for the design case of 550RPM. At higher than the design rotation speed, there is an increase in flow impingement in the shroud leading edge region where the shroud holes are located. This increased flow impingement in the shroud hole region and the higher turbulence at the above design rotation speeds (700RPM) disrupt the capacity of the coolant jet to attach to the shroud surface by promoting the mixing of the free stream air and the exiting coolant jets and this leads to a highly localized film cooling coverage around the shroud holes (Fig. 4.12).
To clearly demonstrate and summarize the effect of the off-design conditions on the shroud heat transfer behavior, the area-averaged Nusselt ratio values, \( \frac{Nu}{Nu_0} \) and area averaged film cooling effectiveness values, \( \eta_f \), in the shroud hole region for the design speed and different off-design speeds are shown in Fig. 4.13 and Fig. 4.14. The area averaged \( \frac{Nu}{Nu_0} \) downstream of the shroud hole region is shown in Fig. 4.15 while the corresponding effectiveness values downstream of the shroud hole region are not shown because they are very low.

![Graph showing Nu/Nu0](image1)

**Fig. 4.13:** Area Averaged \( \frac{Nu}{Nu_0} \) in the Hole Region as a Function of Rotation Speed for different blowing ratios

![Graph showing \( \eta_f \)](image2)

**Fig. 4.14:** Area-Averaged \( \eta_f \) in the Hole Region as a Function of Rotation Speed for different blowing ratios
Fig. 4.15: Area Averaged $\text{Nu}/\text{Nu}_0$ in the shroud region downstream of the coolant holes as a Function of Rotation Speed for different blowing ratios

The results in Fig. 4.13 and Fig. 4.14 show a monotonic increase in the area-averaged $\text{Nu}/\text{Nu}_0$ values and a monotonic decrease in the area-averaged film cooling effectiveness in the shroud hole region as the transition is made from below design speeds to the above design speeds for all blowing ratios studied. The heat transfer levels increase as much as 70% as the rotation speed increases from 300RPM to 700RPM for the lower BR cases. The cooling effectiveness shows greater changes with rotation speed that is even greater at the higher BR, with a drop in the area-averaged value at BR=1.5 from around 0.25 to 0.11 as the RPM increases from 300 to 700, and at BR of 3.0 the effectiveness drops from 0.4 to below 0.2 with a change in RPM form 550 to 700. The results in Fig. 4.15 show a decrease in the area averaged $\text{Nu}/\text{Nu}_0$ values downstream of the shroud hole region as the blade rotation speed goes from the below design speeds to the above design speeds, but they are relatively insensitive to the blowing ratio since this region is deficient in the film cooling.

4.1.2.3 Effect of Hole Spacing on Shroud Heat Transfer:

The heat transfer profile in the circumferential direction of the shroud is presented via line plots of the Nusselt ratio values and film cooling effectiveness for two locations on the shroud surface. Row 2 at $x/D = 5$, with coolant hole pitch to coolant hole diameter ratio, $P/D = 4.8$, was chosen as one of the locations and Row 4 at $x/D = 13$ with $P/D = 10.4$ was chosen as the other location. For both locations, 15 points in
the circumferential direction, y/D, were chosen around the coolant holes. The line plots were done for different blowing ratios at the design rotation speed and for both the design and off-design speeds at a blowing ratio of 1.5. Fig. 12 shows the lateral Nusselt ratio distribution and lateral film cooling effectiveness distribution, respectively, along Row 2 in the shroud circumferential direction for BR = 1.5 at the design and off-design speeds while Fig. 13 shows the lateral Nusselt ratio distribution and lateral film cooling effectiveness distribution, respectively, along Row 4 in the shroud circumferential direction for BR = 1.5 and at the design and off-design speeds.

Fig. 4.16: Lateral Nu/Nu₀ (top) and Lateral Film cooling (bottom) distribution at Column 2 and BR = 1.5 at different rotation speeds

Fig. 4.17: Lateral Nu/Nu₀ (top) and Lateral Film cooling
The results from Fig. 4.16 and Fig. 4.17 summarize the global trends in the shroud hole region shown in previous figures. Of more interest in these plots, is the heat transfer distribution in the circumferential direction showing the effect of the hole spacing in the lateral heat transfer profile. The five rows of holes all have different hole spacing characterized by the hole pitch which is normalized with the hole diameter. The profiles in Row 2 shown in Fig. 12 and Row 4 shown in Fig. 4.17, show the change in profile of the heat transfer distribution in the circumferential direction as the hole pitch increases. Row 4 with the larger hole pitch, P/D = 10.4, results in a sinusoidal and periodic distribution of the Nusselt number ratios and the film cooling effectiveness results in the circumferential direction. Row 2 with half the hole pitch of Row 4 shows a more even distribution (Fig. 4.16). Thus, the hole spacing clearly has an effect on the heat transfer distribution on the shroud. When the spacing between the holes is small enough, the local heat transfer properties like the local film cooling effectiveness and local heat transfer coefficient around the holes combine to form the global profile in the circumferential direction. With a smaller hole pitch, at higher blowing ratios, the coolant jet migrates enough to attach to the jets from neighboring holes and thus providing better coolant coverage than what can be expected at larger hole pitches. This is particularly useful in regions where the coolant coverage would be disrupted.

To confirm similar trends for different blowing ratios, the lateral Nusselt ratio distributions for the design speed at different blowing ratios including the no coolant case have been plotted for Row 2 and Row 4 in Fig. 4.18. The results from Fig. 4.18 confirm the sinusoidal and periodic behavior for Row 4 and more even distribution for Row 2. The results also confirm the global trend shown in earlier figures of decreased area-averaged Nusselt ratio values with increasing blowing ratios. The no coolant distribution is even for both Row 2 and Row 4 in the absence of the shroud holes whereas every other blowing ratio distribution is sinusoidal for Row 4. This confirms and isolates the effect of the coolant hole spacing on the circumferential distribution. These results show that like a turbine blade or a flat plate, as in other studies done by Gritsch et al (2005) and Dyson et al (2013), the hole spacing P/D does have an effect on the heat transfer behavior.
4.1.3 Summary

The detailed surface heat transfer results on an actively cooled shroud with blade rotation have been presented. An experimental study was performed on a gas turbine shroud with 45° degree laterally-angled, discrete cooling holes at a design rotation speed of 550 RPM for the no coolant case and for coolant blowing ratios of 1.0, 1.5, 2.0, 2.5 and 3.0. The study was then expanded to include two off-design conditions at different rotation numbers for the same blowing ratios studied. Detailed heat transfer results in the form of the normalized Nusselt number, $\frac{Nu}{Nu_0}$, and film cooling effectiveness, $\eta_f$, are provided in contour and line plots. The major conclusions obtained from this study are as follows:

1) The results for the 550 RPM design case shows that the area-averaged film cooling effectiveness in the shroud hole region increases as the blowing ratio increases even at the higher blowing ratios. Conversely, the area averaged Nusselt ratio decreases in the shroud hole region as the blowing ratio increases.
2) Comparing the Nusselt ratio values, \( \frac{\text{Nu}}{\text{Nu}_0} \), for the off-design cases to the design case, it was observed that the area-averaged \( \frac{\text{Nu}}{\text{Nu}_0} \) in the shroud hole region increases at above design rotation speeds and decreases at below design rotation speeds. This is due to the change in flow profile as the flow relative inlet angle and relative velocity change at the off-design speeds. There is a shift of the high heat transfer region from the shroud trailing surface to the shroud leading surface as one goes from below design speeds to above design speeds.

3) The results for the off-design cases show that there is an increase in coolant coverage at below design rotation speeds in the shroud hole region and a decrease in the coolant coverage at above design speeds. The freestream air impinging on the shroud surface at a region downstream of the coolant holes for the below design case allows for the attachment of the coolant to the shroud surface in the shroud hole region while the turbulent mixing of the coolant and free-stream air in the shroud hole region for the above design case disrupts the attachment of the coolant to the shroud surface. Nevertheless, an increase in the area-average film cooling effectiveness in the shroud hole region is observed for even the off-design cases as the blowing ratio increases.

4) The heat transfer profile in the shroud circumferential direction is affected by the coolant hole spacing. The heat transfer distribution is sinusoidal and periodic for Row 4 at \( x/D = 13 \) with a hole pitch to diameter ratio of 10.4 and more even for Row 2 at \( x/D = 5 \) with a hole pitch to diameter ratio of 4.8. This is due to the relative distance between the coolant holes. The closer the coolant holes to each other, the more even the heat transfer and cooling effectiveness distribution.
4.2 An Experimental Investigation of Shroud Cooling Using an Upstream Slot and Angled Discrete Holes: Film Cooling Effectiveness

In this section, the effectiveness of active shroud cooling in a fully shrouded low speed rotating turbine cascade using a 1 mm upstream slot in combination with 1696, 1 mm angled discrete holes is investigated. Measurements of the film cooling effectiveness for the slot alone is performed and compared to the film cooling effectiveness from the shroud holes alone. Both cooling methods are utilized concurrently for a combination of hole blowing ratios from 1.0 to 3.0 and slot blowing ratios from 0.5 to 2.0. The tests are performed at a scaled down design rotation speed of 550 RPM using liquid crystal thermography. The results for the slot cooling alone shows increasing cooling effectiveness up to a blowing ratio of 1.25 followed by a drop off in the cooling effectiveness with blowing ratio due to jet lift off. A comparison between slot cooling alone and shroud hole cooling alone for this geometry shows the slot cooling to provide a higher area averaged film cooling effectiveness up to a blowing ratio of 2.4 after which the shroud hole cooling provides a higher area averaged effectiveness. The results for the combined cooling techniques show improvement in the area-averaged film cooling effectiveness for all blowing ratios studied over the individual results at the same blowing ratios. Furthermore, the combination of the cooling techniques results in better penetration of the coolant coverage further downstream of the slot and shroud holes.

4.2.1 Test Configuration and Conditions

In these tests, film cooling effectiveness measurements are taken on a shroud surface in a test section attached to a closed loop wind tunnel.

For each test, the freestream air is passed through a bypass loop using a centrifugal fan. The fan is controlled with a variable frequency drive allowing the user to choose and set a fan speed corresponding to the design rotor velocity. The air in the bypass loop is heated to a target temperature using a 20kW heater allowing the test section to remain at room temperature. At time $t = 0$, the freestream air is introduced into the test section using an electrically actuated gate valve. This results in a step change in the freestream
temperature in the test section. With this step change in the freestream temperature, there is a change in the shroud surface temperature allowing for data collection.

While the freestream air is being bypassed and heated, the coolant air is also bypassed through a coolant bypass system. Since slot tests and discrete hole tests are being run both individually and concurrently, separate coolant systems for supplying the slot and discrete hole plenums have been designed. The coolant bypass system for both the slot and discrete hole supply systems are equipped with separate inline heaters to allow for a choice of coolant temperatures when running the tests. Pneumatically actuated ball valves wired to a common switch allow for the simultaneous introduction of the coolant air for both the slot and discrete holes for the combined tests and for the introduction of the coolant air for each individual system.

The test section consists of a 1 ½ stage rotating turbine cascade. The rotor is pre-spinned to the target rotor speed with a 2.5 HP motor. The rotor has a tip gap set to 1.7 % of the blade span. The shroud is 856.6 mm +/- 0.1 mm in diameter with 1696 1mm diameter holes and consists of 32 iterations of the discrete hole pattern shown in Fig. 4.19.

Each column of holes has a distinct hole pitch to diameter ratio, P/D. The slot which is 1mm in diameter is designed such that it is uninterrupted around the shroud circumference. It is located 5 slot widths upstream of the first column of holes and about 7 slot widths upstream of the blade leading edge. For analysis, the shroud region is divided into two parts such that the two parts are the same for both the discrete hole and slot cooling analysis. Given that the slot is 1 mm wide and the discrete holes are 1 mm in diameter, shroud region 1 is designated as the region 25 slot widths from the end of the slot or 20 hole diameters from the first column of holes F1. Shroud region 2 is the shroud region downstream of shroud region 1. This designation is such that the discrete holes are located exclusively in region 1.
Visual access to the shroud is achieved using acrylic window inserts in place of two of the 32 iterations of the shroud. Thermocouples at different locations in the wind tunnel and the test section allows for the acquisition and monitoring of temperature data including the acquisition of the freestream temperature immediately upstream of the stator blades. LABVIEW is used for the overall data acquisition and synchronization and MATLAB is used for the data analysis.

A more detailed discussion of the experimental setup and testing conditions including a detailed overview of the test section with schematics, figures showing the step change in the freestream temperature in the test section, and the stator-rotor-stator arrangement can be seen in Tamunobere et al. (2014).
4.2.2 Results

Testing was done to obtain the film cooling effectiveness results for active shroud cooling using a 1 mm upstream slot shown below and a combination of the upstream slot and discrete coolant holes for varying blowing ratios at a design rotation speed of 550 RPM. The shroud hole region has been divided into region 1 (x/s < 25) containing the discrete holes and region 2 (x/s >25) for analysis. The results for the slot and the effect of blowing ratio on slot cooling for 8 different blowing ratios is discussed first. A comparison of the slot cooling results to the discrete hole cooling results for the same setup is then made. Finally, the slot data is compared to available data and correlations in literature. In the next section, the results for the concurrent slot and discrete hole tests for a combination of slot and discrete hole blowing ratios is provided. This is followed by a discussion of the effect of the blowing ratio on the combined slot and discrete hole cooling.

4.2.2.1 Slot Cooling Background:

Film cooling studies have advanced steadily over the last few decades. Most of these advancements have been built on the foundation laid in the early work in the research area. Slot film cooling was investigated as early as 1946 by Wiegardt (1946). Several empirical correlations to model slot cooling have been developed. Most of these correlations utilize a heat sink model wherein the coolant flow is modelled as a heat sink immediately after injection with the goal of reducing the temperature in the boundary layer downstream and therefore the temperature of the wall. Nevertheless, some of the correlations predict higher effectiveness values that would usually be found experimentally. This is primarily due to assumptions made about the magnitude of the effect of the coolant on the velocity boundary layer.

In 1952, Tribus and Klein (1952) as part of their work on boundary layers produced by jets of air discharged parallel to the surface, continued the investigation by Wiegardt (1946) and developed the correlation in Eqn. 3 with air as the mainstream gas.
\[
\frac{T_w(x) - T_\infty}{T_s - T_\infty} = 4.62 \text{Re}_s^{0.2} \left( \frac{x U_\infty \rho_\infty}{s U_s \rho_s} \right)^{-0.8}
\]  \quad (3)

This correlation is of the form

\[
\eta_{aw} = C \text{Re}_s^{0.2} \left( \frac{x}{M_s} \right)^{-0.8}
\]  \quad (4)

Where \( \eta_{aw} = \frac{T_w(x) - T_\infty}{T_s - T_\infty} \), and \( M = \frac{U_s \rho_s}{U_\infty \rho_\infty} \)  \quad (5)

The parameters in Eqn. 4 forms the basis for most subsequent correlations for the film cooling effectiveness from a slot. Thus, slot cooling depends primarily on the blowing ratio, \( M \), distance from the slot, \( x \), and the slot Reynolds number, \( \text{Re}_s \). The extent of this dependence and the influence of other factors are very important when studying slot cooling. Papell and Trout (1959) conducted an investigation to study the importance of each of the parameters to the cooling effectiveness by varying each parameter individually and holding the other constant. Their investigation showed that the correlation in Eqn. 4 is very much dependent on changes in blowing ratio but not by changes in the freestream Mach number for the free stream Mach numbers tested (\( \text{Ma} = 0.15 - 0.70 \)) nor changes in the slot height for the slot heights tested (1/2 in, 1/4 in, 1/8 in, 1/16 in). Goldstein (1971) in his summary of the different correlations available in literature also noted the importance of the geometry and flow field at the point of injection as significant variables in the results. It is noted that for two dimensional slot cooling, including axisymmetric cooling, the external flow is highly two dimensional and the coolant flow is introduced uniformly across the span. This investigation would focus on the effect of the blowing ratio on shroud cooling effectiveness.

### 4.2.2.2 Effect of Blowing Ratio on Slot Cooling:

In these tests, the effect of blowing ratio on shroud cooling effectiveness from a 2-dimensional slot located upstream of the shroud and blade leading edge is investigated. The slot is located 5 slot widths upstream of the shroud leading edge and 7 slot widths upstream of the blade leading edge. The tests are
performed at a scaled down design rotor speed of 550 RPM and for blowing ratios of 0.5, 0.75, 1.0, 1.25, 1.5, 1.75, 2.0 and 3.0. The distance from the slot, \( x \), has been normalized with the slot width, \( s \). The results for the laterally averaged cooling effectiveness vs \( x/s \) for the different blowing ratios is shown in Fig. 4.20.

Fig. 4.20: Laterally averaged film cooling effectiveness vs \( x/s \) at the design speed and different \( M \)

The results in Fig. 4.20 show the significant impact of the blowing ratio on the film cooling effectiveness. Like the correlation in Eqn. 4, the film cooling effectiveness is shown in Fig. 4.20 to increase as the blowing ratio increases. Unlike the correlation, the experimental results show that the increase in film cooling effectiveness with blowing ratio peaks at \( M=1.25 \) and after which, the film cooling effectiveness decreases with increasing blowing ratio. This reduction in film cooling effectiveness is as a result of jet lift-off as the coolant detaches from the surface as the blowing ratio continues to increase. In addition to the increase in the film cooling effectiveness with blowing ratio for \( M<1.25 \), there is an increase in penetration of the coolant coverage with increasing blowing ratio. This is evidenced by the increased lateral averaged effectiveness at locations further downstream of the slot, 10<\( x/s \)<20, as the blowing ratio increases. Nevertheless, for all blowing ratios studied, there is a steep and very noticeable decline in effectiveness as the coolant which exits from a 2-dimensional slot upstream of the rotor enters the highly 3-dimensional rotor domain. This decline in the effectiveness happens immediately upon entry into the rotor domain and leads to the eventual destruction of the coolant coverage for all the blowing ratios studied. For the lower blowing ratios, this destruction in the coverage is swift upon entry of the coolant into the rotor domain. As
the blowing ratio increases up to the blowing ratio with the peak effectiveness, the penetration of the coolant and the subsequent protection of the shroud surface increases relative to the lower blowing ratios. However, this is accompanied by a global decline in the cooling effectiveness and the eventual destruction of the coverage at locations further into the rotor domain.

A plot of the area-averaged effectiveness with the blowing ratio for the slot region defined by \(x/s<25\), has been made. This plot summarizes the relationship between the blowing ratio and the film cooling effectiveness for the slot. For comparison, a similar plot showing the area averaged effectiveness vs blowing ratio for shroud hole cooling has been made in the same figure. The plots for the area-averaged effectiveness vs blowing ratio for slot cooling and shroud hole cooling is shown in Fig. 4.21.

![Fig. 4.21: Area Averaged Film Cooling Effectiveness for the Shroud Hole and Shroud Slot for Region 1 at the design speed](image)

Figure 4.21 summarizes the slot cooling results shown in Fig. 4.20. From this plot, the increase in film cooling effectiveness with blowing ratio up till a blowing ratio of 1.25 is seen. The subsequent decrease in film cooling effectiveness with blowing ratio after \(M=1.25\) due to jet lift off is also seen in Fig. 4.21. The relationship between the slot cooling effectiveness and the effectiveness of cooling using this hole configuration studied is also seen. Since the plots in Fig. 4.21 are area-averaged, the effectiveness of the coolant method in protecting the region in question can be inferred. Thus, while cooling using this pattern of shroud holes might have higher local film cooling effectiveness values near the holes, it can be seen from
Fig. 4.21 that slot cooling has better area-averaged performance in this region than discrete hole cooling for blowing ratios less than about 2.4. At blowing ratios greater than 2.4, the area-averaged effectiveness of discrete hole cooling is greater than the area-averaged effectiveness of slot cooling. While the factors affecting the slot cooling distribution have been discussed, it should be noted that there are also factors that affect the discrete hole cooling. These factors include the coolant hole spacing and the hole orientation and configuration. The area-averaged cooling effectiveness is not only affected by the individual hole cooling performance but the pattern utilized. Specifically, the hole pitch and hole spacing which varies from $P/D = 4$ to $P/D = 24$ play a pivotal role in the coolant coverage area. The hole configuration is such that the holes are oriented laterally in the direction of rotation and at $90^\circ$ to the incoming free stream flow. Furthermore, the holes are angled at $45^\circ$ to the surface. These factors in addition to the rotor-shroud surface interaction results in increased cooling effectiveness with blowing ratio for the discrete holes. This, combined with the jet lift off seen for slot cooling at higher blowing ratios results in the trend seen in Fig. 4.21 with the discrete hole cooling having a higher area-averaged effectiveness at higher blowing ratios whereas the slot cooling has a higher area-averaged effectiveness at the lower blowing ratios.

4.2.2.3 Comparison of Slot Cooling Results with Data from literature:

Several slot cooling investigations were conducted based on the earlier work by Weighardt (1946) and Tribus and Klein (1952). This led to the development of an array of slot cooling correlations based on the heat sink model. Unsurprisingly, most of these correlations contain the same parameters as in Eqn. 4. Surprisingly, there is a substantial variation in the cooling effectiveness predictions from some of the different models especially near the slot region. This is primarily as a result of some of the assumptions used in the different models.
Using a similar heat sink model, Librizzi and Cresci (1964) in the development of their model, assumed that the boundary layer starts at the point of injection and at injection, \( x = 0 \). As such, the film cooling effectiveness is assumed to be equal to 1 at \( x = 0 \). They arrived at the correlation in Eqn. 6,

\[
\eta = \frac{1}{1 + 0.329 \left( \frac{x}{M_s} \right)^{0.8} \left( \frac{\mu_s Re_s}{\mu_\infty} \right)^{-0.2}}
\]  

(6)

Kutateladze and Leont’ev (1964) assumed that the boundary layer starts at a location upstream of the slot such that at the slot, the mass flow rate would be equal to the coolant mass flow rate. As such, the film cooling effectiveness is also assumed to be unity at \( x = 0 \) and the correlation derived is as follows;

\[
\eta = \frac{1}{1 + 0.249 \left( \frac{x}{M_s} \right)^{0.8} \left( \frac{\mu_s Re_s}{\mu_\infty} \right)^{-0.2}}
\]  

(7)

Stollery and El-Ehwany (1965) assumed that the boundary layer starts at the slot and the total mass flow at the slot is zero. This results in a value of infinity for the film cooling effectiveness at \( x = 0 \) and a correlation as follows;

\[
\eta = 3.03 \left( \frac{x}{M_s} \right)^{-0.8} \left( \frac{\mu_s Re_s}{\mu_\infty} \right)^{0.2}
\]  

(8)

Seban et al. (1957) developed a correlation for the film cooling effectiveness. This correlation shown in Eqn. 9 follows a power law on the order of \( \frac{1}{2} \) without the inclusion of the slot Reynolds number.

\[
\eta = 2.2 \left( \frac{x}{M_s} \right)^{-1/2}
\]  

(9)

Seban et al. (1957) indicated that the inclusion of the slot Reynolds number will improve the correspondence of the correlation to experimental data but not enough to warrant its inclusion for normal injection.

To validate the methodology and to add further experimental data to the slot correlations discussed above, a test for the slot film cooling effectiveness at a blowing ratio \( M=0.5 \), without the rotor and other major test section components was conducted and compared to the results from literature discussed above.
The results shown in Fig. 4.22 are in good agreement with the literature results especially away from the near slot region. Since the validity of the correlations are expected to suffer in the near slot regions due to some of the assumptions used in developing the correlations, the disparity between the correlations and experimental data in that region is expected. After the validation test, the test section was fully reassembled as discussed in the experimental section and the slot cooling tests with blade rotation were performed.

To compare the slot correlations developed for the flat surface to data for the slot geometry used in this shroud slot cooling, the lateral film cooling effectiveness has been plotted vs x/Ms for blowing ratios of 0.5, 0.75 and 1.0 in Fig. 4.23.
These blowing ratios were chosen because of the range of validity of the correlations in Eqns. (6, 7, 8, 9). The results show lower overall film cooling effectiveness for the slot than predicted by the correlations and experimental data for a flat plate and the validation studies without the rotor and other major test section components. For analysis, the results would be broken down into two regions: the near slot region and the region further downstream in the rotor domain. As the coolant enters the rotor domain from the two-dimensional slot, it is affected by several factors characteristic of the highly three-dimensional rotor domain such as the tip leakage flow. These factors result in the degradation of the coolant coverage resulting in the rapid decline of the film cooling effectiveness as the distance into the rotor domain increases. This leads to
the eventual destruction of the coolant protection and globally lower values of film cooling effectiveness
than the correlations would predict for this region of the analysis. In the near slot region, in addition to some
of the assumptions used at the injection point noted above, Goldstein (1971) noted an assumption inherent
in the heat sink model itself. The heat sink model assumes complete mixing of the coolant fluid in the
mainstream boundary layer. Thus, the heat sink model is reasonably valid only further from the slot.

4.2.2.4 Combined Slot and Discrete Hole Cooling:

In these tests, the combined effectiveness of discrete hole cooling and slot cooling is investigated.
This investigation is done for the same combination of blowing ratios as the individual discrete hole and
slot hole cooling studies. The film cooling effectiveness results for a slot blowing ratio of 0.5 and discrete
hole blowing ratio of 1.0, 2.0 and 3.0 are presented. Similar combined tests for slot blowing ratios of 1.0,
1.5 and 2.0 and discrete hole blowing ratios of 1.0, 2.0 and 3.0 are also conducted and presented. The results
for this investigation are shown in Figs. 4.24 through 4.27.

Fig. 4.24: η for 0.5 slot and hole blowing ratio of a) BR=1.0 b) BR=2.0 c) BR=3.0 at the design speed
Fig. 4.25: $\eta$ for 1.0 slot and hole blowing ratio of a) BR=1.0 b) BR=2.0 c) BR=3.0 at the design speed

Fig. 4.26: $\eta$ for 1.5 slot and hole blowing ratio of a) BR=1.0 b) BR=2.0 c) BR=3.0 at the design speed

Fig. 4.27: $\eta$ for 2.0 slot and hole blowing ratio of a) BR=1.0 b) BR=2.0 c) BR=3.0 at the design speed
4.2.2.5 Effect of Blowing Ratio on Combined Slot and Discrete Hole Cooling:

Shroud cooling testing was done for a combination of four slot blowing ratios of 0.5, 1.0, 1.5 and 2.0 and for three discrete hole blowing ratios of 1.0, 2.0 and 3.0. The results show that the blowing ratio has a significant impact on the shroud cooling effectiveness as is the case for discrete hole cooling alone and slot cooling alone. The results in Fig. 4.24 show increasing cooling effectiveness in the shroud hole region for slot blowing ratio of M = 0.5 as the discrete hole cooling blowing ratio increases from M=1.0 in Fig. 4.24a to M=3.0 in Fig. 4.24c. A similar set of results is also seen in Fig. 4.25, Fig. 4.26 and Fig. 4.27 for slot blowing ratios of 1.0, 1.5 and 2.0 and discrete hole blowing ratios of 1.0, 2.0 and 3.0 in parts a, b and c of these figures respectively. The combined cooling tests share this characteristic of increased film cooling effectiveness with increasing hole blowing ratio with the discrete hole cooling tests alone. This is mainly attributed to the coolant hole configuration and orientation and the rotor shroud surface interactions.

The results in Figs. 4.24a, 4.25a and 4.26a show increasing area-averaged film cooling effectiveness with slot blowing ratio. This trend of increasing area-averaged effectiveness with blowing ratio is also seen in a comparison of Figs. 4.24b, 4.25b and 4.26b and in Figs. 4.24c, 4.25c and 4.26c which correspond to slot blowing ratios of 0.5, 1.0 and 1.5 respectively. The results in Figs. 4.27a, 4.27b and 4.27c show lower area averaged effectiveness than the corresponding contour plots in Figs. 4.26a, 4.26b and 4.26c. This decrease in area-averaged effectiveness with blowing ratio at higher slot blowing ratio after a preceding increase in effectiveness with blowing ratio at lower slot blowing ratio is also seen in the individual slot cooling tests. It is as a result of the jet lift off from the surface as the slot blowing ratio increases. Like the slot cooling tests alone, with increased blowing ratio up till the peak blowing ratio, the combined coolant axial penetration into the shroud surface increases. This results in a larger coolant coverage area with increasing blowing ratio up till the peak blowing ratio. It should be noted that even with the increase in penetration of the coolant coverage, there is still the global decline in the cooling effectiveness due to the effects of the highly three dimensional rotor domain.
To summarize the effects of the blowing ratio on the combined slot and discrete hole cooling, a plot of the area-averaged effectiveness vs the discrete hole blowing ratio for the different slot blowing ratios including the no slot cooling case has been made in Fig. 4.28.

![Graph](image)

**Fig. 4.28: Area Averaged Film Cooling Effectiveness for Shroud Hole and Slot combination for Region 1 at the design speed**

Figure 4.28 clearly shows both the effect of increasing the discrete hole blowing ratio on the combined slot and discrete hole cooling effectiveness and the effect of increasing the slot blowing ratio on the combined slot and discrete hole cooling effectiveness. Figure 4.28 confirms that with increasing discrete hole blowing ratio, the combined area average effectiveness increases as well. Figure 4.28 also confirms that for the lower blowing ratios, with increasing slot blowing ratio, the area averaged effectiveness increases as well and that this is followed by a decrease in the area-averaged effectiveness with increasing blowing ratio at higher blowing ratios. Furthermore, Fig. 4.28 shows that the combined slot and discrete hole tests provided better protection as measured by the area-averaged effectiveness than the discrete hole cooling alone (slot M = 0) and the slot cooling alone (Discrete hole blowing ratio = 0).

To study the effects of the blowing ratio on the cooling effectiveness further downstream of the discrete holes and slot, an area averaged effectiveness vs discrete hole blowing ratio plot was made for the different slot blowing ratios in Fig. 4.29.
The results show that the combined cooling using the slot and the discrete holes has an effect on the cooling effectiveness downstream given that the effectiveness in this region was relatively small for the discrete hole cooling alone and slot cooling alone.

The results show that for the combined tests, as the slot blowing ratio increases up to the peak blowing ratio, the area averaged effectiveness further downstream of the slot and hole region increases as well. This, in addition to the increase in area averaged effectiveness in the region near the slot and containing the holes, is one of the most significant and quantifiable effects of combining the cooling methods. While there is still a rapid degradation of the coolant coverage and effectiveness as it moves further axially down the shroud, the increase in overall coolant injection into the shroud region allows for a bleed off of the coolant into the shroud regions previously impenetrable to the coolant with the individual cooling methods due to the highly three-dimensional nature of the flow field in this region. It is also noteworthy that for the combined tests, as the slot blowing ratio increases up to the peak blowing ratio for maximum effectiveness, the effectiveness increases downstream. However, the effect of increasing the blowing ratio for the discrete holes for a constant slot blowing ratio does not yield such a noticeable increase in the cooling effectiveness downstream. Nevertheless, it does have an impact on the combined cooling
effectiveness even downstream as evidenced by the lack of penetration of the coolant coverage in the slot cooling tests alone.

4.2.3 Summary

In this section, an experimental investigation was performed on a gas turbine shroud with a 1 mm upstream slot and 1696, 45° degree angled, 1 mm cooling holes at a scaled down design rotation speed of 550 RPM. Individual tests for the slot cooling effectiveness were performed at slot blowing ratios of 0.5, 0.75, 1.0, 1.25, 1.5, 1.75, 2.0 and 3.0. Area averaged results showing the relationship between the blowing ratio and the slot cooling effectiveness were also presented. The results of the slot cooling tests were compared to correlations from literature. In the next series of tests, slot cooling was combined with discrete hole cooling using the shroud hole pattern in Fig. AA. The tests were performed for a combination of slot blowing ratios including 0.5, 1.0, 1.5 and 2.0 and for discrete hole blowing ratios of 1.0, 2.0 and 3.0. Detailed cooling effectiveness results in the form of contour plots showing the film cooling effectiveness, $\eta$, have been provided. Line plots summarizing the results from the tests have also been provided. The major conclusions obtained from this study can be summarized as follows:

1) The slot cooling tests show that the film cooling effectiveness increases with blowing ratio up to a blowing ratio of 1.25. After a blowing ratio of 1.25, the film cooling effectiveness decreases with increasing blowing ratio due to jet lift off from the surface.

2) The film cooling effectiveness results from the experimental investigation is generally lower than the results from correlations in literature at similar slot Reynolds number. Near the slot, this is primarily as a result of the different assumptions at the point of injection in the correlations about the effect of the coolant injection on the velocity boundary layer. Downstream of the near slot region, there is a rapid decline in the film cooling effectiveness and an eventual degradation of the coolant coverage as the coolant from the slot enters the highly three dimensional rotor domain. This results in globally lower
cooling effectiveness due to the effects of the highly three-dimensional rotor domain on the two-dimensional slot injection as the coolant penetrates the rotor domain.

3) For lower blowing ratios, slot cooling provides better area averaged cooling effectiveness than discrete hole cooling for this configuration when a comparison is made at the same blowing ratio and the same region. As the blowing ratio increases past M=2.4, discrete hole cooling provides better area-averaged cooling effectiveness for this configuration. This is primary as a result of the decreasing effectiveness of slot cooling with jet lift off at the higher blowing ratios combined with the still increasing effectiveness of discrete hole cooling with blowing ratio due to this hole configuration and orientation.

4) The results for the combined slot and discrete hole cooling shows that the combined cooling has higher film cooling effectiveness and provides better cooling coverage than the slot cooling or discrete hole cooling alone at their respective blowing ratios. Furthermore, the same peak and drop off in cooling effectiveness as seen in the slot cooling alone is also seen in the combined cooling results. Thus, the best cooling coverage is seen at the peak slot blowing ratio and at the highest discrete hole blowing ratio for this configuration.

5) The combined slot and discrete hole tests provide better penetration of the coolant protection downstream than either the slot tests alone or the discrete hole tests alone which show very low film cooling effectiveness. With increasing slot blowing ratio, the cooling effectiveness downstream for the combined tests increases up to the peak blowing ratio of 1.5 in this study. The same increase in downstream cooling effectiveness is not seen with increasing discrete hole blowing ratio for the combined tests.
CHAPTER 5: BLADE TIP HEAT TRANSFER AND COOLING

5.1 Turbine Blade Tip Film Cooling with Blade Rotation Part I: Tip and Pressure Side Coolant Injection

This is the first in a two-part series of an experimental film cooling study conducted on the tip of a turbine blade with a blade rotation speed of 1200 RPM. In this part of the study, the coolant is injected from the blade tip and pressure side (PS) holes, and the effect of the blowing ratio on the heat transfer coefficient and film cooling effectiveness of the blade tip is investigated. The blade has a tip clearance of 1.7% of the blade span and consists of a cut back squealer rim, two cylindrical tip holes and six shaped pressure side holes. The stator-rotor-stator test section is housed in a closed loop wind tunnel that allows for the performance of transient heat transfer tests. Measurements of the heat transfer coefficient and film cooling effectiveness are done on the blade tip using liquid crystal thermography. These measurements are reported for the no coolant case and for blowing ratios of 1.0, 1.5, 2.0, 3.0 and 4.0. The heat transfer results for the no coolant injection show a region of high heat transfer on the blade tip near the blade leading edge region as the incident flow impinges on that region. This region of high heat transfer extends and stretches on the tip as more coolant is introduced through the tip holes at higher blowing ratios. The cooling results show that increasing the blowing ratio increases the film cooling effectiveness. The tip film cooling profile is such that the tip coolant is pushed towards the blade suction side thereby providing better coverage in that region. The shift in coolant flow profile towards the blade suction side as opposed to the pressure side in stationary studies can primarily be attributed to the effects of the blade relative motion.

5.1.1 Test Configuration and Conditions

This study is conducted in a closed loop wind tunnel that houses a one-and-half turbine stage facility. A schematic of the test facility is shown in Fig. 5.1 and the turbine operating conditions and blade data are shown in Table 5.1.
Prior to the start of a test, the mainstream air is heated in a bypass loop to a pre-determined temperature using a 20 kW duct heater. A few minutes before the freestream air is introduced, the rotor is set to 1200 RPM using a programmable variable frequency drive and a 7.5 horsepower motor. At time $t=0$, which has been universally synchronized amongst all the data acquisition devices using LABVIEW, the heated mainstream air is introduced into the test section with the recorded freestream air temperature rising rapidly (in less than about 2 seconds as recorded by a thermocouple) to about 50 °C. Once the air is introduced into

Table 5.1: Turbine Operating and Blade Data

<p>| | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Rotor Velocity</strong></td>
<td>1200 RPM</td>
<td><strong>Number of Blades</strong></td>
<td>64</td>
</tr>
<tr>
<td><strong>Re</strong></td>
<td>140,000</td>
<td><strong>Blade rel. inlet angle</strong></td>
<td>32.5°</td>
</tr>
<tr>
<td><strong>Mach Number</strong></td>
<td>0.15</td>
<td><strong>Flow Coefficient, $\phi$</strong></td>
<td>0.24</td>
</tr>
<tr>
<td><strong>Axial Chord, $C_x$</strong></td>
<td>40.6 mm</td>
<td><strong>Loading Coefficient, $\psi$</strong></td>
<td>1.06</td>
</tr>
<tr>
<td><strong>Blade Span</strong></td>
<td>73.598 mm</td>
<td><strong>Tip Gap</strong></td>
<td>1.7% Span</td>
</tr>
</tbody>
</table>
the test section, the loading on the motor drops to nearly zero, indicating the rotor is in free spin and is driven by the air flow. It takes the heated air approximately 67 milliseconds to travel from the damper to the blades. Introducing the mainstream air into the test section results in a step change in the temperature in the test section and with the step change in temperature, a transient, one dimensional analysis can be conducted to determine the heat transfer coefficient and film cooling effectiveness on the blade tip.

Attached to the rotor are sixty four rotor blades made using an SLA machine. Each blade consists of two cylindrical tip holes as well as six shaped holes on the blade pressure side. The blades feature a cutback squealer rim with a recessed tip and a tip gap set at 1.7 % of the blade span. A model of the blade and the blade profile is shown in Fig.5.2 and Fig. 5.3.

![Fig. 5.2: Turbine Blade Tip Model](image-url)
The circumferential shroud above the rotor has a transparent acrylic window for visual access. The entire rotor shroud including the acrylic windows has a circumferential cooling hole pattern for coolant injection through the shroud. Visual access to the blade tip is interrupted at locations with the shroud cooling holes. The rotor shroud has a plenum to accommodate the shroud coolant. The plenum is also retrofitted with an optically transparent window so that the blade tip can be directly viewed. In this section of the chapter, no shroud coolant injection is considered. However, that is the subject of the next section in this chapter.

A cooling system has been designed and integrated into the test section for all blade cooling tests. The coolant path for the blade tip cooling is represented by the red line and shown in Fig. 5.4.
Since a two-test method with different coolant temperatures is used for the liquid crystal technique, the coolant system is designed with a bypass. This allows the coolant air to be heated to any pre-determined temperature using an in-line heater prior to the start of a test.

The coolant air is introduced from the coolant bypass line through a hollow channel in the shaft using a pneumatically actuated ball valve. After passing through the hollow shaft, the coolant is ejected into the rotor plenum through discrete holes on the shaft. The coolant air is then distributed from the rotor plenum to the individual blade plenums and out through the blade tip. Since the coolant in the individual
blade plenums are fed from the same rotor plenum, the blowing ratio reported in this study is the average blowing ratio through the holes. A thermocouple is mounted inside the blade plenum to monitor the coolant temperature upstream of the delivery tube.

A data acquisition system that consists of a high speed camera and an optical trigger has been set up as schematically shown in Fig. 5.5. An optical encoder is used to trigger the image acquisition by the high speed camera. The encoder position on the shaft has been aligned such that the trigger pulse it sends to the camera arrives as the target rotor blade enters the field of view of the window. The image acquisition system is setup to isolate and save data from the rotor blade to be studied. The mainstream air bypass gates and coolant air bypass valves are wired to a switch which allows for the simultaneous control of all gates and bypass systems. Several thermocouples are installed at various locations in the test section and at different points throughout the facility to monitor and record all relevant temperatures. The temperature data from the blades and all rotating components is transferred to a data acquisition device using a slip ring. Flowmeters are used to monitor and measure the coolant air flow. Overall data acquisition synchronization is done using LABVIEW. All testing was done at a density ratio (DR) of about 1.1. The density ratio has been shown by Goldstein et al (1974) to have an effect on film cooling. However, this effect has not been studied in this section.

Fig. 5.5: Data Acquisition System
5.1.2 Results

In this section, detailed experimental results showing the heat transfer coefficient and film cooling effectiveness on the blade tip of a gas turbine blade with a blade rotation speed of 1200 RPM are presented. This investigation was performed for the no coolant case and for blowing ratios of 1.0, 1.5, 2.0, 3.0 and 4.0. Coolant is ejected through two tip holes and six pressure side shaped holes. The test geometry is shown in Fig. 5.2. In the first part of this study, the coolant is ejected from the tip holes represented as “Tip” in the figures presented, and from the pressure side holes represented by “PS”. In the latter part of this study, coolant is ejected through the tip holes alone by sealing the pressure side holes and the effect of the tip holes is investigated. Subsequently, coolant is ejected through the pressure side holes alone by sealing the tip holes and the effect of the pressure side holes is investigated.

The contour plots showing the effect of the blowing ratio on the heat transfer coefficient and film cooling effectiveness of the blade tip are shown in Fig. 5.6 and Fig. 5.7, respectively.

![Contour plots of heat transfer coefficient](image)

Fig. 5.6: Heat Transfer Coefficient for a) No Coolant Case b) M=1.0 c) M=1.5 d) M=2.0 e) M=3.0 f) M=4.0
The no coolant heat transfer coefficient result in Fig. 6a shows a region of high heat transfer coefficient on the blade tip near the leading edge region. This high heat transfer coefficient is due to the incoming mainstream air that separates as it passes over the squealer rim and reattaches in this region. A similar phenomena has been seen in other studies such as Acharya and Moreaux (2012) and Krishnababu et al (2009). As the blowing ratio increases, Figs. 5.6b, 5.6c, 5.6d and 5.6e show an extension of the region of high heat transfer coefficient on the tip near the blade leading edge region and an overall increase in the heat transfer coefficient on the blade tip. This is due to increased turbulence and a more complex vortex system from the leakage air-coolant jet interaction. A relatively high heat transfer region can also be seen on the suction side squealer rim. This is primarily due to the leakage vortex as the over-tip leakage flow goes from the pressure side to the suction side. A key observation of the present study is that the film cooling effectiveness distributions show a different distribution pattern from the corresponding stationary studies. In the stationary studies such as those by Kwak and Han (2004) and Ahn et al (2005), the coolant is oriented towards the blade pressure side as it is ejected from the coolant holes. The present results in Fig. 5.7 show that the coolant is pushed towards the blade suction side as it is ejected from the coolant holes. A similar film cooling behavior is shown in other studies with blade rotation such as those by Acharya and
Moreaux (2012) and Rezasoltani et al (2014). This difference in the coolant flow profile clearly shows the effect of the blade rotation on blade tip film cooling.

Numerical studies have also shown this and other differences between stationary and rotation cases for blade tip heat transfer. Stationary cases have a stronger leakage vortex as the blade relative motion reduces the leakage flow over the blade tip. Along with reduced leakage flow strength, blade relative motion results in other differences in the leakage flow profile. For the stationary cases, the velocity profile at the blade suction side exit is more uniform. However for cases with blade rotation, the leakage flow is concentrated at exit locations from the mid chord region to the trailing edge (Yang et al, 2010). Therefore, higher heat transfer coefficients between the mid chord and trailing edge region of the suction side squealer rim is expected and seen in Fig. 5.6. Aerodynamic and heat transfer studies by Yang et al (2010) and Acharya and Moreaux (2012) have shown that while the coriolis and centrifugal forces do contribute to the overall heat transfer profile, the dominant contributor to the differences between stationary and rotation cases is the effect of the blade relative motion. The push of the coolant to the blade tip suction side as seen in Fig. 5.7 is specifically as a result of the blade relative motion as a similar effect seen in cases with blade rotation by Acharya and Moreaux (2012) and Rezasoltani et al (2014) is also seen in cases with a moving shroud.

5.1.2.1 Effect of Blowing Ratio on Blade Tip Cooling:

The results in Fig. 5.6 and Fig. 5.7 show the effect of the blowing ratio on blade tip cooling using two tip holes and six pressure side shaped holes. From the results, it can be seen that as the blowing ratio increases, the film cooling effectiveness and the film cooling coverage on the blade tip increase. The major effect of the increase in blowing ratio is the greater penetration and a corresponding increase in cooling coverage to locations further downstream of the cooling holes. To further this discussion, the lateral averaged heat transfer coefficient ratio ($h/h_0$) and film cooling effectiveness results are shown in Fig. 5.8
and Fig. 5.9 respectively. Note that $h_0$ is the baseline no-coolant value, and therefore the enhancement ratio $h/h_0$ is a measure of how the coolant jets augment or decrease the heat transfer coefficient relative to the baseline.

Fig. 5.8: Laterally Averaged $h/h_0$ vs axial distance for the blade tip at different blowing ratios

Fig. 5.9: Laterally Averaged $\eta$ vs axial distance for the blade tip at different blowing ratios

From Fig. 5.8, it can be seen that introducing the coolant generally increases the heat transfer coefficient over the baseline values, with the highest enhancement ratio values falling between 20% and 40% of the axial chord where the two tip holes are located. The heat transfer coefficient ratio peaks at about 30% of the blade axial chord. The enhancement ratio increases at higher blowing ratios, reaching values in excess of 2.5 at $M=4.0$. This enhancement in heat transfer is expected since the mainstream air-coolant jet interaction is most significant at the point of ejection and increases as more coolant is ejected into this region by increasing the blowing ratio. By increasing the blowing ratio, the region of increased heat transfer
coefficient over the no coolant case also stretches axially (also seen in Fig. 1) as the coolant air penetrates further downstream and the complex vortex systems from the interaction between the coolant and mainstream air also continues downstream.

The results in Fig. 5.9 show that the lateral spread of the film cooling effectiveness increases with M, and is observed to be between 30% and 70% of the axial chord for blowing ratios greater than M=1.0 and between 20% to 50% for M=1.0. At higher blowing ratios, the coolant separates from the tip floor and reattaches further downstream of the ejection hole resulting in better lateral coolant coverage downstream of the holes. At lower blowing ratios, such as M=1.0, only the tip hole region and the region immediately downstream of it show the higher film cooling effectiveness.

From Fig. 5.9, the effect of the pressure side holes on the film cooling effectiveness can be seen. The local increase in effectiveness due to the pressure side holes can clearly be seen for M=1.0, at downstream locations where the effect of the tip hole coolant has diminished. The results show local bumps in laterally averaged film cooling effectiveness around 68% and 84% of the axial chord primarily due to the effects of the pressure side hole cooling. The effect of the pressure side holes on the cooling effectiveness and at higher blowing ratios, the heat transfer coefficient, can also be seen in the contour plots in Fig. 5.6 and Fig. 5.7. Figure 5.6 shows that the pressure side holes result in an increase in the local heat transfer coefficient values around the hole region while Fig. 5.7 and Fig. 5.9 show an increase in both the local and laterally averaged film cooling effectiveness as a result of the pressure side holes.

5.1.2.2 Effect of Tip Only and Pressure Side Only Cooling:

In this part of the study, the effect of tip hole cooling alone and pressure side hole cooling alone is investigated. This investigation is performed for blowing ratios of 1.5 and 3.0. For the tip hole cooling only tests, the pressure side holes are sealed and for the pressure side only tests, the tip holes are sealed. The heat transfer coefficient and film cooling effectiveness results for the blade tip holes only test are shown in Fig.
5.10 and Fig. 5.11, respectively and the results for the pressure side only tests are shown in Fig. 5.12 and Fig. 5.13.

Fig. 5.10: Heat Transfer Coefficient at (a) M=1.5 and (b) M=3.0 for the Blade Tip Holes Only

Fig. 5.11: Film Cooling Effectiveness at (a) M=1.5 and (b) M=3.0 for the Blade Tip Holes Only

Fig. 5.12: Heat Transfer Coefficient for (a) M=1.5 and (b) M=3.0 for the Pressure Side Holes Only

Fig. 5.13: Film Cooling Effectiveness for (a) M=1.5 and (b) M=3.0 for the Pressure Side Holes Only
The individual cooling profiles in Fig. 5.11 and Fig. 5.13 for the tip holes only and pressure side holes only tests match what is seen for the combined tip and pressure side holes results in Fig. 5.7. The results in Fig. 5.11 show that the tip holes contribute more to the overall cooling of the blade tip than the pressure side holes. However, further downstream of the tip holes, the pressure side holes provide additional cooling for the blade tip floor. The results in Fig. 5.10 and Fig. 5.12 show that the tip holes are mostly responsible for the extension of the region of high heat transfer coefficient on the blade tip seen in the combined tip and pressure side hole studies.

To continue the discussion of the individual effects and contributions of the tip holes and pressure side holes, the laterally averaged heat transfer coefficient ratio, \( h/h_0 \), and the laterally averaged film cooling effectiveness have been plotted against the normalized axial distance, \( x/C_x \) in Fig. 5.14 and Fig. 5.15 respectively.

Fig. 5.14: Laterally Averaged \( h/h_0 \) vs axial distance for the blade tip, blade pressure side and blade tip and pressure side holes at blowing ratios M=1.5 and M=3.0
The cooling results in Fig. 5.15 shows that in the region before 68% of the axial chord, the tip holes provide most of the cooling for the blade tip. After 68% of the axial chord, the pressure side holes provide most of the cooling observed in that region. However, the results also indicate that at the lower blowing ratio of M=1.5, the added benefit of the pressure side holes is evident in the region after 36% of the blade chord and for the higher blowing ratio of M=3.0, the added benefit of the pressure side holes is evident in the region after 44% of the axial chord. The cooling effectiveness results show that the pressure side holes provide an added benefit to the tip cooling especially further downstream towards the blade trailing edge. The contributions by both the tip holes and the pressure side holes to the cooling effectiveness are mostly unique and both cooling configurations can be modeled individually and the results combined together by superposition.

The result in Fig. 5.14 shows that the tip holes are mostly responsible for the peak in the laterally averaged heat transfer coefficient ratio seen in the combined tip and pressure side hole tests at about 30% of the
blade axial chord. Thus, it is the tip hole coolant-mainstream air interaction that results in the extension of the region of high heat transfer coefficient on the blade tip.

5.1.3 Summary

In this section, the effect of the blowing ratio on blade tip cooling was investigated for a gas turbine blade with a blade rotation speed of 1200 RPM. While other blade rotation studies have provided only film cooling effectiveness results (Rezasoltani et al (2014)) or only heat transfer results using discrete points on the tip (Dunn and Haldeman (2000)), this study provides detailed contour plots for both the heat transfer coefficient and the film cooling effectiveness. The blade with a cutback squealer rim and a tip gap of 1.7% of the blade span consists of two cylindrical tip holes and six pressure side shaped holes. This investigation was performed for the no coolant case and for blowing ratios of 1.0, 1.5, 2.0, 3.0 and 4.0 with both the tip holes and the pressure side holes open. Further testing was performed at blowing ratios of 1.5 and 3.0 for the tip holes only and for the pressure side holes only. All tests were performed at a Reynolds number of 140,000. The major conclusions of this part of the two part series are as follows:

1) A comparison with stationary studies have shown that the tip coolant is generally oriented towards the tip pressure side for stationary studies while the coolant from the tip holes with blade rotation is pushed towards the suction side of the blade tip. This results in higher coverage on the suction side of the tip and less coverage on the pressure side of the tip. This is specifically due to the effects of the blade relative motion.

2) An increase in the blowing ratio results in a greater penetration and a corresponding increase in coolant coverage to locations further downstream of the coolant holes. At lower blowing ratios, the highest lateral average cooling effectiveness is seen immediately downstream of the tip holes. However, at higher blowing ratios the coolant reattaches on the tip wall further downstream of the ejection holes resulting in better lateral coolant coverage downstream of the holes. The pressure side holes result in higher local and
laterally averaged cooling effectiveness in the regions around the holes particularly towards the blade trailing edge.

3) The heat transfer coefficient results show a region of high heat transfer coefficient near the leading edge region of the blade tip. This forms as the incoming mainstream flow separates as it flows over the squealer rim and reattaches in this region. Increasing the blowing ratio results in an extension of the region of high heat transfer on the blade tip as well as resulting in generally higher heat transfer coefficients on the tip. Unlike the cooling effectiveness, the highest laterally averaged heat transfer coefficient over the no coolant case is seen around the tip holes. This is expected since the most significant mainstream air-coolant air interaction occurs in this region.

4) The combination of the cooling effectiveness results from the tip hole cooling alone and pressure side hole cooling alone gives the same general result as those from the combined tip hole and pressure side cooling tests. In each individual case, as the blowing ratio is increased, the cooling effectiveness increases. The results also show that the tip holes contribute more to the overall cooling of the blade tip than the pressure side holes. Similarly, the tip holes are responsible for the extension of the region of high heat transfer coefficient in the near leading edge region of the blade tip. The contribution of the pressure side holes to the overall heat transfer coefficient becomes more significant towards the blade trailing edge region.

5.2 Turbine Blade Tip Film Cooling with Blade Rotation Part II: Shroud Coolant Injection

In this section, blade-tip cooling is investigated with coolant injection from the shroud alone and a combination of shroud coolant injection and tip cooling. The blade rotates at a nominal speed of 1200 RPM, and consists of a cut back squealer tip with a tip clearance of 1.7% of the blade span. The blade consists of tip holes and pressure side shaped holes, while the shroud has an array of angled holes and a circumferential slot upstream of the rotor section. Different combinations of the three cooling configurations are utilized to
study the effectiveness of shroud cooling as a complementary method of cooling the blade tip. The measurements are done using liquid crystal thermography. Blowing ratios of 0.5, 1.0, 2.0, 3.0 and 4.0 are studied for shroud slot cooling and blowing ratios of 1.0, 2.0, 3.0, 4.0 and 5.0 are studied for shroud hole cooling. For cases with coolant injection from the tip, the blowing ratios used are 1.0, 2.0, 3.0 and 4.0. The results show an increase in film cooling effectiveness with increasing blowing ratio for shroud hole cooling. The increased effectiveness from shroud hole cooling is concentrated mainly in the tip-region below the shroud holes and towards the blade suction side and the suction side squealer rim. Slot cooling injection results in increased effectiveness on the blade tip near the blade leading edge up to a maximum blowing ratio, after which the cooling effectiveness decreases with increasing blowing ratio. The combination of the different cooling methods results in better overall cooling coverage of the blade tip with the shroud hole and blade tip cooling combination being the most effective. The level of coolant protection is strongly dependent on the blowing ratio and combination of blowing ratios.

5.2.1 Test Configuration and Conditions

In this section, the effect of shroud cooling on the heat transfer measurements on the blade tip is studied in a stator-rotor-stator rotating turbine facility with a blade rotation speed of 1200 RPM. Since this is the second in a two-part series, only information relevant to this paper, and not included in the first, is described here. For more information on the facility and its general operation and operating conditions, the reader is referred to the previous chapters.

In the present configuration, the cooling of the blade tip can be performed using three distinct means: blade tip and pressure side holes, discrete shroud holes and a shroud slot. Each configuration has an independent coolant supply line with a dedicated in-line heater. The slot and shroud hole configuration is shown in Fig. 5.16.
The discrete shroud holes are arranged in a circumferential pattern and consists of five rows with different pitches. The shroud holes each 1 mm in diameter, are oriented lateral to the incoming flow direction, and angled at 45 degrees to the surface in the direction of rotation. Each row of shroud holes, labeled F1 through F5, consists of a unique number of shroud holes corresponding to different pitches. Row 1, labeled F1, has the most number of holes and row 5, labeled F5, has the least. The number of holes in each row decreases from row 1 to row 5. The shroud cooling holes are arranged in a repeating pattern around the shroud circumference. The arrangement is such that 32 iterations of the same pattern is formed. One of the 32 shroud hole patterns is replaced with an acrylic window for visual access. The shroud hole information is summarized in Table 5.2.
Table 5.2: Shroud Hole Data

<table>
<thead>
<tr>
<th>Surface Angle</th>
<th>45°</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length/Diameter</td>
<td>7.071</td>
</tr>
<tr>
<td>Hole Shape</td>
<td>Cylindrical</td>
</tr>
<tr>
<td>Distance Between Hole Rows</td>
<td>4.5 D</td>
</tr>
</tbody>
</table>

Cooling for the shroud holes is provided by an independent cooling supply line shown in green in Fig. 5.17.

Prior to the introduction of the shroud hole coolant to the shroud hole cooling plenum, the coolant air is bypassed. The shroud hole coolant line is equipped with an inline heater that allows the coolant air to
be heated to a pre-determined temperature. This is necessary since the liquid crystal measurement technique is a two-temperature test technique and the coolant temperatures are different for each of the two tests. The shroud hole coolant air is introduced to the shroud hole plenum which is located above the shroud using 8 air hoses that are attached to a centralized piping system feeding the shroud holes. Since all the discrete shroud cooling holes are supplied by a single plenum that spans the shroud circumference, the blowing ratio reported in this study is the average blowing ratio through the shroud cooling holes.

The slot cooling is done using an uninterrupted slot on the shroud. The slot is 1 mm wide which matches the shroud-hole diameter, 5 mm deep and is located 7 slot-widths upstream of the blade leading edge. Unlike the shroud cooling holes, the slot is not angled. The blue line in Fig. 5.17 represents the coolant path for the slot cooling. The slot coolant is supplied by a separate cooling line and can be independently set relative to the discrete holes. The slot cooling line is also equipped with a separate bypass system that allows the slot coolant to be heated prior to its introduction into the test section. The slot coolant is introduced into the slot plenum using 16 air tubing lines located at equal intervals around the slot plenum.

A portion of the shroud containing one of the 32 shroud hole cooling patterns has been replaced with a transparent acrylic window to allow for visual access to the blade tip. The acrylic windows have the same discrete hole distribution pattern as the rest of the shroud. Since the plenum outer casing for the shroud lies above the shroud, a portion of the plenum surface in line with the shroud window has also been replaced with an optically transparent acrylic window. A high speed camera is used for image acquisition through both optically clear windows. The high speed camera is triggered by an optical encoder that sends a signal each time the test blade enters the optically clear window. Data (temperatures in the tip coolant plenum) is transferred from the rotor to a data acquisition device using a slip ring. Thermocouples are located at various points in the test section and around the facility to constantly monitor and record all temperature changes. Flow meters are used to monitor the coolant air flow. The shroud cooling and the slot cooling systems have separate bypass systems with pneumatically actuated ball valves. The ball valves are controlled by a single switch which allows for the coolant air to be introduced into the test section simultaneously. Overall data acquisition and synchronization is done using LABVIEW.
TEST CONDITIONS
Various blowing ratios for the tip and pressure side hole cooling, shroud hole cooling and shroud slot cooling are used in this study. Table 5.3 summarizes the combination of tests performed. This combination was chosen to comprehensively investigate the effect of shroud cooling using both the shroud cooling holes and the slot, on the heat transfer coefficient and film cooling effectiveness of the blade tip. All tests in this study are conducted at a blade rotation speed of 1200 RPM, a Reynolds number of 140,000, and a corrected speed of 0.15.

Table 5.3: Test Matrix for Shroud Cooling Tests

<table>
<thead>
<tr>
<th>Cooling Configuration</th>
<th>Blowing Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shroud Holes</td>
<td>1.0, 2.0, 3.0, 4.0</td>
</tr>
<tr>
<td>Shroud Slot</td>
<td>0.5, 1.0, 2.0, 3.0</td>
</tr>
<tr>
<td>Shroud Holes/Shroud Slot</td>
<td>2.0/0.5, 4.0/1.0</td>
</tr>
<tr>
<td>Shroud Holes/Shroud Slot/Blade Tip Holes</td>
<td>2.0/0.5/2.0, 4.0/1.0/4.0</td>
</tr>
</tbody>
</table>

5.2.2 Results
The effects of shroud cooling and its usefulness as a complementary method of cooling the blade tip is investigated. The second through the fifth row of the shroud holes are located directly above the blades, while the first row of shroud holes and the shroud slot are both upstream of the rotating blades.

As part of the broader investigation of the effects of the two shroud cooling methods on the tip cooling, tests using the shroud holes alone, the shroud slot alone, and in combination have been performed. Based on these studies, the appropriate shroud cooling cases were combined with blade tip cooling at different blowing ratios to analyze the effects of these combinations on the blade tip heat transfer and cooling.
5.2.2.1 Effect of Shroud Hole Cooling on Blade Tip Heat Transfer:

In this section, the effect of coolant injection from the shroud holes on the blade tip is presented. This investigation is performed for shroud hole blowing ratios of 1.0, 2.0, 3.0, 4.0 and 5.0. The heat transfer coefficient and film cooling results are shown in Fig. 5.18 and Fig. 5.19 respectively.

![Fig. 5.18: Heat Transfer Coefficient for (a) M=1.0 (b) M=2.0 (c) M=3.0 and (d) M=4.0 (e) M=5.0 for Shroud Hole Cooling](image)

![Fig. 5.19: Film Cooling Effectiveness for (a) M=1.0 (b) M=2.0 (c) M=3.0 and (d) M=4.0 (e) M=5.0 for Shroud Hole Cooling](image)
In interpreting the results in Fig. 5.18 and Fig. 5.19, it should be noted that the shroud cooling holes are laterally oriented with a surface angle of 45°. Thus the ejected coolant from the shroud holes have a very strong lateral component and a correspondingly weak streamwise component. As the coolant exits from the shroud holes, it is pushed further along in the lateral direction towards the blade suction side providing better coverage in that region (Fig 5.19). Combined with the effects of the blade relative motion, the lateral injection of the coolant results in the coolant coverage primarily between the leading edge and mid-chord and concentrated near the SS. A similar push of the coolant toward the suction side due to blade relative motion was observed in the blade tip coolant injection study, and in other tip-cooling studies with rotation (Acharya and Moreaux (2012)). It should be noted that the heat transfer coefficient on the blade tip is relatively unaffected by the shroud coolant (Fig. 5.18. The heat transfer coefficients are high near the blade LE where the separated flow reattaches on the tip floor, and along the trailing half of the blade along the SS due to the leakage vortex in that region.

To further discuss the effect of the shroud cooling on the tip, the laterally averaged heat transfer coefficient ratio, \( h/h_0 \), is presented as a function of the normalized axial distance, \( x/C_x \) in Fig. 5.20, where \( h_0 \) is the heat transfer coefficient for the no-coolant case. It can be seen that the shroud coolant has virtually no impact on the tip heat transfer coefficient implying that there is no direct impingement of the coolant on the tip.

![Laterally Averaged Heat Transfer Coefficient vs Axial Distance for Shroud Hole Cooling](image-url)
Fig. 5.21: Laterally Averaged Film Cooling Effectiveness vs Axial Distance for (a) M=1.0 (b) M=2.0 (c) M=3.0 and (d) M=4.0 (e) M=5.0 for Shroud Hole Cooling

Fig. 5.21 shows the effect of the shroud hole blowing ratio on the blade tip cooling effectiveness. At the lowest blowing ratio studied, the cooling effectiveness from shroud cooling is marginal. The cooling effectiveness and its coverage are mostly limited to the blade regions below the shroud cooling holes, and the effectiveness and its coverage increases with the blowing ratio. However, even with the increased coverage, the coolant protection is still mostly on the blade suction side and is limited to the region upstream of the mid-chord. After 40% of the axial chord, the cooling effectiveness for all but the highest blowing ratio studied is marginal. Since the shroud hole density is highest near the leading edge of the blade, the tip cooling effectiveness is highest in this region and decays rapidly downstream.

5.2.2.2 Effect of Slot Cooling on Blade Tip Heat Transfer:

The effect of slot cooling on the blade tip heat transfer coefficient and film cooling effectiveness using an uninterrupted slot is investigated in this part of the study. Unlike the shroud holes which are primarily in the rotor blade domain, the shroud slot is located upstream of the rotor blades. The investigation is performed at blowing ratios of 0.5, 1.0, 2.0, 3.0 and 4.0. Figure 5.22 and Fig. 5.23 show the effect of shroud slot cooling on the blade tip.
Fig. 5.22: Heat Transfer Coefficient for (a) M=0.5 (b) M=1.0 (c) M=2.0 and (d) M=3.0 (e) M=4.0 for Shroud Slot Cooling

Fig. 5.23: Film Cooling Effectiveness for (a) M=0.5 (b) M=1.0 (c) M=2.0 and (d) M=3.0 (e) M=4.0 for Shroud Slot Cooling
The heat transfer coefficient contours in Fig. 5.22 are essentially identical to those in Fig. 5.18, and $h/h_0$ is nearly 1, as seen in Fig. 5.18, confirming that the shroud coolant injection and modality does not play a major role in the tip heat transfer coefficient distribution. The $h/h_0$ distributions are mostly controlled by the leakage flow aerodynamics. From Fig. 5.23, it can be seen that the effectiveness of shroud slot cooling using this configuration is limited to regions of the blade tip very near the leading edge. Previous tests on the effects of shroud-slot cooling on the shroud also showed a steep decline in area-averaged cooling effectiveness as the coolant which exits from a 2-dimensional slot located 7-slot widths upstream of the blade leading edge, enters the highly 3-dimensional rotor domain. Despite the slot cooling effectiveness on the shroud surface rapidly declining, the slot coolant did show improved cooling effectiveness on the shroud surface in the rotor domain than it does on the blade tip as the results in Fig. 5.23 indicate. This leads to the conclusion that most of the coolant from the slot that enters the rotor domain primarily cools the shroud surface. When detached from the shroud surface, the coolant is rapidly mixed with the highly three dimensional flow regime in the tip gap providing minimal to no cooling on most regions of the blade tip.

![Laterally Averaged Heat Transfer Coefficient vs Axial Distance](image)

**Fig. 5.24:** Laterally Averaged Heat Transfer Coefficient vs Axial Distance for (a) M=0.5 (b) M=1.0 (c) M=2.0 and (d) M=3.0 (e) M=4.0 for Slot Cooling
The results in Fig. 5.25 confirm the conclusions in the preceding paragraph, and show the rapid decline in film cooling effectiveness with axial distance and the limited penetration of the coolant to regions of the tip further downstream of the near-leading edge region of the blade tip. The results in Fig. 5.25 also shows that the film cooling effectiveness increases from M=0.5 to M=1.0 and only marginally to M=2.0 before declining with increasing blowing ratio. With the vertical slot injection, the increased velocity of the coolant air at higher blowing ratios results in the direct penetration of the coolant into the mainstream air. The coolant gets mixed out before entering the rotor domain. This, leads to the decrease in cooling effectiveness at the higher blowing ratios.

### 5.2.2.3 Effect of Combined Shroud Hole Injection and Tip Injection on Blade Tip Heat Transfer:

The individual studies of the effect of shroud cooling alone and slot cooling alone on the blade tip was important in determining which, if any, of the shroud cooling methods was effective in cooling the blade tip. From the earlier results, it can be seen that the shroud holes are more effective in cooling the blade tip than the shroud slot. Therefore, in this part of the study, shroud hole cooling will be investigated as a means of cooling the blade tip in addition to cooling from the tip and pressure side holes. As described in the first paper in this series, the blade tip consists of 2 tip holes and six pressure side shaped holes. In all figures provided, it should be noted that the tip holes are represented as “tip” and the pressure side holes...
are represented as “PS”. A single shroud hole blowing ratio of M=4.0, along with four different tip blowing ratios of M=1.0, M=2.0, M=3.0 and M=4.0 will be used in this study. The effect of the combined tip and pressure side cooling as well as shroud hole cooling is investigated. The heat transfer coefficient and film cooling effectiveness results from the investigation are shown in Fig. 5.26 and Fig. 5.27, respectively.

Fig. 5.26: Heat Transfer Coefficient at Shroud Hole Blowing Ratio of 4.0 and Tip and Pressure side Blowing ratios of (a) M=1.0 (b) M=2.0 (c) M=3.0 and (d) M=4.0

Fig. 5.27: Film Cooling Effectiveness at Shroud Hole Blowing Ratio of 4.0 and Tip and Pressure side Blowing ratios of (a) M=1.0 (b) M=2.0 (c) M=3.0 and (d) M=4.0
The heat transfer coefficient result, as seen in Fig. 5.26, is similar to that of Figs. 5.18 and 5.22, and is dominated by the leakage flow aerodynamics. However, as the tip and PS hole blowing ratios increase, there is a streamwise extension of the region of high heat transfer coefficient. There also does appear to be an increase in the overall heat transfer coefficient values along the entire tip with increasing tip and pressure side hole blowing ratios.

From the results in Fig. 5.27, it can be seen that significant portion of the blade suction side is protected by the combined cooling especially at the higher tip blowing ratios. As in the individual studies, the blade pressure side is less protected by the coolant. The blade relative motion is one of the determining factors in pushing the coolant towards the suction side. Compared to the results shown earlier in the preceding section with PS and tip holes only, the lateral ejection of the shroud hole coolant augments the tip cooling particularly in the leading regions of the blade where the shroud coolant appears to be directed towards. The shroud coolant and the tip coolant effects appear to complement each other in spatially non-overlapping regions, and together provide a more complete coverage of the blade tip. However, downstream of the mid-chord regions, the coolant coverage is still poor in view of the absence of coolant holes in this region. To continue the discussion of the effects of adding shroud hole cooling to the tip and pressure side hole cooling, the laterally averaged heat transfer coefficient ratio, $h/h_0$ and the laterally averaged film cooling effectiveness have been plotted against the normalized axial distance, $x/C_x$ in Fig. 5.28 and Fig. 5.29 respectively. To facilitate this discussion, line plots with the relevant results from the individual studies that make up the combinations used in this part of the study have been added to Fig. 5.28 and Fig. 5.29.

The results in Fig. 5.28 show the dominance of the tip and pressure side hole cooling on the blade tip heat transfer profile. Like the tip and pressure side only heat transfer coefficient profile, the laterally averaged heat transfer coefficient results in Fig. 5.28 peaks at about 30% of the axial chord. The heat transfer coefficient ratio is also generally significantly above 1.0 between 20% and 40% of the axial chord. This is the region with the tip holes.
Fig. 5.28: Laterally Averaged Heat Transfer Coefficient vs Axial Distance for Shroud Hole and Tip Cooling

Fig. 5.29: Laterally Averaged Film Cooling Effectiveness vs Axial Distance for Shroud Hole and Tip Cooling
While maintaining the same general characteristics as the tip and pressure side hole only profile, the combined shroud hole and tip and pressure side hole heat transfer coefficient values are only marginally higher than the individual heat transfer coefficient values for the tip and pressure side profile alone. The increase in the laterally averaged heat transfer ratio values is due to the increase in the amount of coolant air in the mainstream-coolant air mixture and the associated increase in the turbulence in the flow.

The results in Fig. 5.29 show that the effect of combining shroud hole cooling and tip and pressure side hole cooling is the addition of the two individual cooling profiles. In the region less than 30% of the blade axial chord, the cooling profile from the shroud holes is dominant. In the region greater than 30% of the axial chord, the effect of the tip and pressure side holes is dominant. For each part of the overall cooling profile, the results are similar to the individual cooling profiles. In the regions where both coolant coverages overlap in the combined tests, instead of an increase in effectiveness values due to coolant accumulation, there is a decrease in effectiveness. This is especially true at the higher blowing ratios and can be seen between 30% and 50% of the blade axial chord in Fig. 5.29. The results show that when the two coolant profiles overlap, they do interfere with each other leading to a lower overall cooling effectiveness. However, after 60% of the blade axial chord, the combined shroud hole and tip and pressure side cooling does provide an overall better cooling effectiveness than the individual cooling methods alone. The results in Fig. 5.29 like the individual studies, also show an increase in laterally averaged cooling effectiveness with an increase in blowing ratio. Importantly, higher blowing ratios lead to better coolant spread on the blade tip as the increased momentum of the coolant jet takes the coolant further downstream of the coolant holes.

Although the impact of cooling from the slot on the blade tip is limited to only regions of the tip very close to the blade leading edge, four blowing ratio combinations involving the slot were chosen to determine, the additional effects, if any, of adding the slot as a means of cooling the blade tip. The combinations include: 1) The shroud holes and the slot alone with blowing ratios of M=2.0 and M=0.5 respectively. 2) The shroud holes, shroud slot and the tip and pressure side holes with blowing ratios of M=2.0, M=0.5 and M=2.0 respectively. 3) The shroud holes and the slot with higher blowing ratios of M=4.0 and M=1.0 respectively. 4) The shroud holes, shroud slot and the tip and pressure side holes with
higher blowing ratios of M=4.0, M=1.0 and M=4.0 respectively. Only the line plots showing the laterally averaged heat transfer coefficient and laterally averaged film cooling results as well as the associated line plots from the relevant individual and combined cooling methods are presented. The heat transfer coefficient results and film cooling results are shown in Fig. 5.30 and Fig. 5.31 respectively.

![Graph showing heat transfer coefficient and film cooling results](image)

**Fig. 5.30:** Laterally Averaged Heat Transfer Coefficient vs Axial Distance for Shroud Hole, Shroud Slot and Tip Cooling

In Fig. 5.30, it can be clearly seen that the major heat transfer coefficient augmentation over the baseline no-cooling case \( (h_0) \) comes from the tip and PS holes. The shroud holes and slot have virtually no additional impact on the heat transfer coefficient distributions.
From Fig. 5.31, it can be seen that the cooling effectiveness results are spatially-additive. The major impact of adding slot cooling is in the region less than 20% of the blade axial chord. The increase in the cooling effectiveness of the blade tip by the addition of the slot in the region less than 20% of the blade axial chord is as high as 45% for slot blowing ratio, $M=1.0$ over the cases with no slot cooling. There is also a small increase in the peak heat transfer coefficient ratio in the cases with the slot. Figure 5.30 as well as the results in Fig. 5.31 show that the best overall configuration in this study for cooling the blade tip is the combination of the blade tip and pressure side holes and the shroud cooling holes (if one considers the total mass of coolant injected as a criteria).
5.2.3 Summary

In this section, the effect of shroud cooling on the blade tip was investigated for a gas turbine blade with a rotation speed of 1200 RPM. Two shroud cooling methods were investigated as complementary methods of cooling the blade tip in addition to the tip holes. The shroud cooling methods involved using circumferentially distributed holes oriented laterally and with a surface angle of 45° on the shroud above the blade tip and an uninterrupted slot located 7 slot-widths upstream of the blade. For the shroud hole cooling, blowing ratios of 1.0, 2.0, 3.0, 4.0 and 5.0 were investigated. For the shroud slot cooling, blowing ratios of 0.5, 1.0, 2.0, 3.0 and 4.0 were investigated. A combination of blowing ratios for both shroud hole and/or shroud slot cooling were utilized in conjunction with the blade tip hole and pressure side cooling to investigate the complementary effects of shroud cooling on the blade tip. The major conclusions of this part of the two-part series are as follows:

1) The shroud hole cooling is mostly limited to the region upstream of mid-chord and towards the suction side of the blade. The ejected coolant from the shroud holes have a strong lateral component, and with the effects of the blade relative motion, the coolant is pushed towards the blade suction side. As the blowing ratio increases, so does the cooling effectiveness. Furthermore, the associated increase in momentum with increasing blowing ratio results in a better coolant coverage area for the blade tip.

2) The effectiveness of shroud slot cooling on the blade tip using the slot configuration utilized in this study is limited to regions of the blade tip very near the leading edge. There is a steep decline in effectiveness downstream of the blade leading edge as the coolant air exits the 2 dimensional slot upstream of the rotor blade and enters the highly 3-dimensional rotor domain. In the region with any noticeable cooling effectiveness, the blowing ratio increases, reaching a peak at M=2.0 and decrease subsequently.

3) For the shroud coolant injection cases, the heat transfer coefficient distribution is mostly controlled by the leakage flow aerodynamics with the highest heat transfer coefficients in the leading edge
regions where the flow separating near the leading edge squealer reattaches on the tip-floor. High heat transfer coefficients are also seen on the SS squealer rim downstream of mid chord where the leakage vortex rolls up and moves downstream leading to high heat transfer coefficient values.

4) For this configuration, combining discrete shroud hole cooling and blade tip and pressure side cooling results in better overall cooling coverage on the blade tip. At regions less than 30% of the blade axial chord, the cooling from the shroud holes is dominant and in regions greater than 30% of the axial chord, the tip hole and pressure side hole cooling is dominant. For a fixed shroud hole blowing ratio of 4.0, increasing the tip hole and pressure side hole blowing ratio results in increasing laterally averaged cooling effectiveness as well as an increase in the streamwise effectiveness due to the larger coolant spread. The heat transfer coefficient profile for this configuration is similar to the heat transfer coefficient profile for the tip hole and pressure side hole only cooling.

4) The combination of the blade tip holes, pressure side holes and the shroud cooling holes provide the best cooling configuration for the blade tip since the contribution from the shroud slot which is located upstream of the rotor domain, is marginal.
CHAPTER 6: CONCLUSIONS

6.1 Conclusions

The following conclusions have been reached as part of the comprehensive study on Gas Turbine Heat Transfer and Film Cooling:

For Shroud Hole Cooling,

3) The results shows that the area-averaged film cooling effectiveness in the shroud hole region increases as the blowing ratio increases even at the higher blowing ratios. Conversely, the area averaged Nusselt ratio decreases in the shroud hole region as the blowing ratio increases.

4) Comparing the Nusselt ratio values, \( \frac{Nu}{Nu_0} \), for the off-design cases to the design case, it was observed that the area-averaged \( \frac{Nu}{Nu_0} \) in the shroud hole region increases at above design rotation speeds and decreases at below design rotation speeds. This is due to the change in flow profile as the flow relative inlet angle and relative velocity change at the off-design speeds. There is a shift of the high heat transfer region from the shroud trailing surface to the shroud leading surface as one goes from below design speeds to above design speeds.

3) The results for the off-design cases show that there is an increase in coolant coverage at below design rotation speeds in the shroud hole region and a decrease in the coolant coverage at above design speeds. The freestream air impinging on the shroud surface at a region downstream of the coolant holes for the below design case allows for the attachment of the coolant to the shroud surface in the shroud hole region while the turbulent mixing of the coolant and free-stream air in the shroud hole region for the above design case disrupts the attachment of the coolant to the shroud surface. Nevertheless, an increase in the area-average film cooling effectiveness in the shroud hole region is observed for even the off-design cases as the blowing ratio increases.

4) The heat transfer profile in the shroud circumferential direction is affected by the coolant hole spacing. The heat transfer distribution is sinusoidal and periodic for Row 4 at \( x/D = 13 \) with a hole
pitch to diameter ratio of 10.4 and more even for Row 2 at $x/D = 5$ with a hole pitch to diameter ratio of 4.8. This is due to the relative distance between the coolant holes. The closer the coolant holes to each other, the more even the heat transfer and cooling effectiveness distribution.

For Shroud Cooling using an upstream slot and angled discrete holes, the following conclusions have been made:

1) The slot cooling tests show that the film cooling effectiveness increases with blowing ratio up to a blowing ratio of 1.25. After a blowing ratio of 1.25, the film cooling effectiveness decreases with increasing blowing ratio due to jet lift off from the surface.

2) The film cooling effectiveness results from the experimental investigation is generally lower than the results from correlations in literature at similar slot Reynolds number. Near the slot, this is primarily as a result of the different assumptions at the point of injection in the correlations about the effect of the coolant injection on the velocity boundary layer. Downstream of the near slot region, there is a rapid decline in the film cooling effectiveness and an eventual degradation of the coolant coverage as the coolant from the slot enters the highly three dimensional rotor domain. This results in globally lower cooling effectiveness due to the effects of the highly three-dimensional rotor domain on the two-dimensional slot injection as the coolant penetrates the rotor domain.

3) For lower blowing ratios, slot cooling provides better area averaged cooling effectiveness than discrete hole cooling for this configuration when a comparison is made at the same blowing ratio and the same region. As the blowing ratio increases past $M=2.4$, discrete hole cooling provides better area-averaged cooling effectiveness for this configuration. This is primary as a result of the decreasing effectiveness of slot cooling with jet lift off at the higher blowing ratios combined with the still increasing effectiveness of discrete hole cooling with blowing ratio due to this hole configuration and orientation.
4) The results for the combined slot and discrete hole cooling shows that the combined cooling has higher film cooling effectiveness and provides better cooling coverage than the slot cooling or discrete hole cooling alone at their respective blowing ratios. Furthermore, the same peak and drop off in cooling effectiveness as seen in the slot cooling alone is also seen in the combined cooling results. Thus, the best cooling coverage is seen at the peak slot blowing ratio and at the highest discrete hole blowing ratio for this configuration.

5) The combined slot and discrete hole tests provide better penetration of the coolant protection downstream than either the slot tests alone or the discrete hole tests alone which show very low film cooling effectiveness. With increasing slot blowing ratio, the cooling effectiveness downstream for the combined tests increases up to the peak blowing ratio of 1.5 in this study. The same increase in downstream cooling effectiveness is not seen with increasing discrete hole blowing ratio for the combined tests.

For Blade Tip Cooling and the Effects of Shroud Cooling Injection on the blade tip, the following conclusions have been reached:

1) A comparison with stationary studies have shown that the tip coolant is generally oriented towards the tip pressure side for stationary studies while the coolant from the tip holes with blade rotation is pushed towards the suction side of the blade tip. This results in higher coverage on the suction side of the tip and less coverage on the pressure side of the tip. This is specifically due to the effects of the blade relative motion.

2) An increase in the blowing ratio results in a greater penetration and a corresponding increase in coolant coverage to locations further downstream of the coolant holes. At lower blowing ratios, the highest lateral average cooling effectiveness is seen immediately downstream of the tip holes. However, at higher blowing ratios the coolant reattaches on the tip wall further downstream of the ejection holes resulting in better lateral coolant coverage downstream of the holes. The pressure
side holes result in higher local and laterally averaged cooling effectiveness in the regions around the holes particularly towards the blade trailing edge.

3) The heat transfer coefficient results show a region of high heat transfer coefficient near the leading edge region of the blade tip. This forms as the incoming mainstream flow separates as it flows over the squealer rim and reattaches in this region. Increasing the blowing ratio results in an extension of the region of high heat transfer on the blade tip as well as resulting in generally higher heat transfer coefficients on the tip. Unlike the cooling effectiveness, the highest laterally averaged heat transfer coefficient over the no coolant case is seen around the tip holes. This is expected since the most significant mainstream air-coolant air interaction occurs in this region.

4) The combination of the cooling effectiveness results from the tip hole cooling alone and pressure side hole cooling alone gives the same general result as those from the combined tip hole and pressure side cooling tests. In each individual case, as the blowing ratio is increased, the cooling effectiveness increases. The results also show that the tip holes contribute more to the overall cooling of the blade tip than the pressure side holes. Similarly, the tip holes are responsible for the extension of the region of high heat transfer coefficient in the near leading edge region of the blade tip. The contribution of the pressure side holes to the overall heat transfer coefficient becomes more significant towards the blade trailing edge region.

5) The shroud hole cooling is mostly limited to the region upstream of mid-chord and towards the suction side of the blade. The ejected coolant from the shroud holes have a strong lateral component, and with the effects of the blade relative motion, the coolant is pushed towards the blade suction side. As the blowing ratio increases, so does the cooling effectiveness. Furthermore, the associated increase in momentum with increasing blowing ratio results in a better coolant coverage area for the blade tip.
6) The effectiveness of shroud slot cooling on the blade tip using the slot configuration utilized in this study is limited to regions of the blade tip very near the leading edge. There is a steep decline in effectiveness downstream of the blade leading edge as the coolant air exits the 2-dimensional slot upstream of the rotor blade and enters the highly 3-dimensional rotor domain. In the region with any noticeable cooling effectiveness, the blowing ratio increases, reaching a peak at M=2.0 and decrease subsequently. In general, it is concluded that slot coolant injection upstream of the rotor section is not an effective strategy for blade tip cooling.

7) For the shroud coolant injection cases, the heat transfer coefficient distribution is mostly controlled by the leakage flow aerodynamics with the highest heat transfer coefficients in the leading edge regions where the flow separating near the leading edge squealer reattaches on the tip-floor. High heat transfer coefficients are also seen on the SS squealer rim downstream of mid chord where the leakage vortex rolls up and moves downstream leading to high heat transfer coefficient values.

8) Combining discrete shroud hole cooling and blade tip and pressure side cooling results in better overall cooling coverage on the blade tip. For this configuration, at regions less than 30% of the blade axial chord, the cooling from the shroud holes is dominant and in regions greater than 30% of the axial chord, the tip hole and pressure side hole cooling is dominant. For a fixed shroud hole blowing ratio of 4.0, increasing the tip hole and pressure side hole blowing ratio results in increasing laterally averaged cooling effectiveness as well as an increase in the streamwise effectiveness due to the larger coolant spread. The heat transfer coefficient profile for this configuration is similar to the heat transfer coefficient profile for the tip hole and pressure side hole only cooling.

9) The combination of the blade tip holes, pressure side holes and the shroud cooling holes provide the best cooling configuration for the blade tip since the contribution from the shroud slot is marginal.
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