ABSTRACT

OKOYE, OBINNA CHIBUZOR. Gas Turbine Vane Film Cooling in a Subsonic Cascade. (Under the direction of Dr. Srinath Ekkad).

In order to increase the efficiency of gas turbine engines, the rotor inlet temperature is increased. The rotor inlet temperature of modern gas turbine engines far exceeds the metallurgical limits of gas turbine components. Therefore, these components have to be adequately cooled to ensure they meet their design life span. Air from the compressor is used to cool the gas turbine hot gas path. Cooling gas turbine components decreases the thermodynamic efficiency of the engine, as it reduces the amount of compressed air available for combustion. Hence, cooling the components of gas turbine engines has to be optimized. Film cooling along with other cooling technologies are used to cool the gas turbine hot gas path, for example components such as airfoils. In film cooling of gas turbine airfoils, cooling air from the internal cooling passages exits via holes in the airfoil to provide a layer protection on the surface of the airfoil from the hot combustion gases. Shaped holes are the industry standard in cooling gas turbine airfoils.

In this work, the effects of compound angle, multiple rows of holes and freestream turbulence on adiabatic effectiveness were experimentally investigated on the pressure side of a first-stage scaled-up GE-E3 vane in a subsonic gas turbine linear vane cascade using infrared thermography technique. The 7-7-7 shaped hole geometry was studied. All experiments were performed at Reynolds number based on mainstream inlet velocity and axial chord of the vane of 175,340 at density ratio of approximately 1. The effect of blowing ratio was investigated as well. The compound angles orientations studied were 0° (axial orientation), 30° and 60°. The cooling performance of the 7-7-7 shaped holes was compared with that of cylindrical holes for the same test conditions. Numerical simulations were performed using the realizable k-ε turbulence model with enhanced wall treatment available in ANSYS Fluent to complement experimental results. It
was found that inclining cylindrical holes at 30° compound angle resulted in greatly reduced adiabatic effectiveness levels at turbulence intensity of 8.9%. Also, the numerical simulations did not capture coolant reattachment to the vane surface downstream of the first and second rows of holes. In addition, freestream turbulence intensity of 18.2% mostly resulted in greatly reduced film cooling performance for the cylindrical and 7-7-7 shaped holes. The baseline results for this research are the results obtained at turbulence intensity of 8.9%. Thus, a strong benchmark at this turbulence intensity level to compare with previous studies is not available. All trends were compared on the basis of the results for turbulence intensity of 8.9%.
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Gas Turbine Vane Film Cooling in a Subsonic Cascade

by

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A dissertation submitted to the Graduate Faculty of North Carolina State University in partial fulfillment of the requirements for the degree of Doctor of Philosophy

Mechanical Engineering

Raleigh, North Carolina 2022

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DEDICATION

To my wife, Ogochukwu. Thank you for your love.
BIOGRAPHY

Obinna Chibuzor Okoye was born in Lagos, Nigeria. He received a bachelor’s degree in mechanical engineering from the University of Nigeria, Nsukka, Nigeria. He earned a master’s degree in Energy Systems and Thermal Processes from Cranfield University, United Kingdom. Thereafter, he was appointed a lecturer in the Department of Mechanical Engineering, Federal University Oye-Ekiti, Nigeria. He commenced his Ph.D. program in the Department of Mechanical and Aerospace Engineering at North Carolina State University in 2018. He joined the Thermal Energy Research and Management Laboratory (ThERMaL) working under the supervision of Dr. Srinath Ekkad. His research involved work on film cooling of a gas turbine vane in a subsonic linear cascade. He has gained extensive experience in designing and fabricating the subsonic gas turbine linear vane cascade, experimental data acquisition and processing, as well as in conducting numerical simulations.
ACKNOWLEDGMENTS

I wish to express my sincerest gratitude to my advisor, Dr. Srinath Ekkad for giving me the opportunity to work in his laboratory, for his guidance and mentorship, as well as for the financial support. I am grateful to the members of my committee, Dr. Richard Gould, Dr. Alexei Saveliev, Dr. Kenneth Granlund and Dr. Jason Patrick, for their helpful comments and suggestions and for reviewing my dissertation. In addition, I thank Drs. Gould, Saveliev and Granlund for their courses which were very helpful in my research.

I thank the Petroleum Technology Development Fund, Nigeria for the Ph.D. scholarship I was offered to study at North Carolina State University.

I am deeply grateful to my dear wife, Ogochukwu, for her constant love, care, and encouragement. I thank my parents for their steadfast and enduring support and encouragement. I thank my siblings and their families and my in-laws for their unwavering support.

I am grateful to all my colleagues and friends at Thermal Energy Research and Management Laboratory (ThERMaL). Thank you all for the helpful discussions, comments and suggestions. I thank all who, in one way or another, have been of help to me over the course of my studies at North Carolina State University. I am very appreciative of your help.

Above all, I am grateful for the grace and mercy of God. To God be all the glory.
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INTRODUCTION

Research Background

Gas turbines engines are used for aircraft propulsion, marine propulsion and power generation. They are used extensively to meet the ever-increasing global energy demand. The efficiency of gas turbine engines increases with increase in the rotor inlet temperature. The rotor inlet temperature of modern gas turbine engines is far above allowable limits of the materials used to produce turbine components [1,2]. Air is routed from the compressor to cool the components of the engines. Advanced military engines have turbine inlet temperatures that exceed 1600°C and the turbine inlet temperatures of land-based turbines exceed 1400°C [3]. The metal softening temperature for current large commercial aircraft and ground-based power generation turbines is 1260°C [1]. Thus, cooling of the gas turbine engine components is inevitable. Cooling of the engine components has to be accomplished with the minimum amount of cooling air, as air used for cooling purposes reduces the amount of air available for combustion which thereby reduces the thermodynamic efficiency of gas turbine engines. Cooling of gas turbine engines is not limited to only the high-pressure turbine, it also includes cooling of casings, turbine disks, exhaust nozzles and the combustor system to mention but a few [1]. The turbine cooling sub-system of the entire gas turbine engine cooling has the greatest economic importance accounting for up to 30% of the total costs of the development and repair of engines, while only the turbine could use 20 to 30% of compressed air for cooling, leakage as well as purge flows [1]. Therefore, the optimum use of cooling air in gas turbine engines is a major factor for the efficient operation of the engines.

Gas turbine engine components are cooled both internally and externally. This research work focuses on film cooling of a gas turbine vane. Gas turbine components, which are cooled actively or passively include, but are not limited to, vanes and blades of the high pressure turbine stages,
shrouds and combustor liners; all of them taken together are known as the hot gas path \([1,4]\). Film cooling technology is an external form of cooling. Film cooling of a turbine vane involves the ejection of cooling air (coolant) from holes on the vane surface, so that the coolant produces a layer of protection for the vane surface from the hot combustion gases. Film cooling differs from other methods of introducing fluids to the boundary layer of surfaces to be protected, like transpiration cooling. In film cooling, the cooling fluid offers protection to the surface not only at the coolant injection locations, but downstream of the coolant injection locations as well \([5]\).

Film cooling has been studied extensively for over four decades both experimentally and numerically. There are reviews of gas turbine film cooling in the literature \([2,3,6–8]\). A lot of factors affect gas turbine film-cooling performance. Bogard and Thole \([3]\) identified six factors that greatly impact film-cooling performance. They are the blowing ratio (or the mass flux ratio), the momentum flux ratio, surface curvature, hole shape, the film cooling hole angle (injection and compound) and freestream turbulence. They noted that film cooling performance is difficult to predict because these individual factors do not act in isolation, as each combination of the factors possibly results in different film-cooling performance, thus the number of operating conditions are very great. Many researchers have investigated the different factors affecting of gas turbine film cooling. Majority of the experimental studies on film cooling have been conducted using flat plates \([2,3]\). While investigations using simplified geometries like flat plates are not typical of the actual engine conditions, they help in understanding film cooling physics and have been instrumental in studying novel configurations in order to improve overall effectiveness over turbine airfoils \([2]\). There is a lot of experimental and numerical investigations of gas turbine airfoil film cooling available in the literature, as well as other aspects of gas turbine film cooling.
Objectives of the Research

In commercial and military applications, the film-cooling holes used are round (cylindrical) or shaped holes, and shaped holes are the industry standard in gas turbine film cooling [6]. The laid-back fan shaped holes are the most common in practice of all the shaped holes [6,9]. The work presented in this dissertation focuses on the 7-7-7 shaped hole introduced by Schroeder and Thole [9], which is intended to serve as a basis for comparison of new hole geometries instead of comparing new geometries to the cylindrical hole. Jet lift-off is known to occur for cylindrical holes at high blowing ratios. The 7-7-7 shaped hole is a laid-back fan shaped hole with 7° expansion angle in both the forward and lateral directions (Figure 1.2) and has been studied mostly using flat plates. Relatively few studies of the 7-7-7 shaped holes have been conducted on gas turbine airfoils. This work extends the database for the 7-7-7 shaped hole by considering the effect of compound angle and multiple rows of holes on the pressure surface of a gas turbine vane. Steady-state experiments were conducted in a subsonic gas turbine linear vane cascade using infrared thermography technique to obtain the adiabatic film cooling effectiveness also known as the adiabatic effectiveness. All experiments were performed at density ratio of 1 ± 0.1 and at Reynolds number based on the mainstream inlet velocity and axial chord of the vane of 175,340. The subsonic gas turbine linear vane cascade was designed and fabricated in the course of this research and is located in the Thermal Energy Research and Management Laboratory (ThERMaL) at North Carolina State University. The vane studied is a scaled-up GE-E3 vane. Numerical simulations were also performed using ANSYS Fluent, a commercial computational fluid dynamics (CFD) package to support experimental results.
Outline of Dissertation

In the first chapter of this dissertation, the experimental investigation of the effect of compound angle and multiple rows of holes of the 7-7-7 shaped hole on the adiabatic effectiveness of the pressure side of the vane is presented. The schematic diagram of the vane studied is presented in Figure 1. There were three rows of holes at different normalized axial locations \(X/C_x = 0.15, 0.3 \) and \(0.65\) on the pressure side of the vane. Each row had pitch-to-hole diameter ratio \((P/D)\) of 3. The compound angles investigated are \(0^\circ\) (axial orientation), \(30^\circ\) and \(60^\circ\). The blowing ratios, defined as the ratios of the mass flux of the coolant to the mass flux of the mainstream, tested are 1, 1.5, 2, 2.5, and 3. The film cooling performance of the 7-7-7 shaped hole was compared to that of cylindrical holes for the same experimental conditions.

![Schematic diagram of the vane studied](image)

Figure 1: Schematic diagram of the vane studied (not to scale)

In the second chapter, steady-state three-dimensional Reynolds-averaged Navier-Stokes simulations were performed using the realizable \(k-\varepsilon\) turbulence model with enhanced wall treatment available in ANSYS Fluent to study the adiabatic film cooling effectiveness of the
pressure side of the vane. Three rows of holes on the pressure side of the vane each with P/D = 3 were used to study the interaction of coolant flowing from the different rows of holes. The 7-7-7 shaped holes and cylindrical holes were studied at 0°, 30° and 60° compound angle inclinations at blowing ratios of 1, 2 and 3. The simulations were performed to complement the experimental results presented in the first chapter, and the numerical results were compared with experimental results. In addition, from numerical simulations, the adiabatic effectiveness contours on the normal plane at the hole exit and five hole diameters (5D) downstream of the hole exit were presented along with local adiabatic effectiveness plots at both locations for the 7-7-7 shaped holes as well as the cylindrical holes. They were used to compare the film cooling performance of both hole geometries for the distance of one film cooling hole pitch.

The experimental study of the effect of compound angle and freestream turbulence on the adiabatic film cooling effectiveness of the pressure side of the vane is presented in the final chapter. The effect of multiple rows of holes on the pressure side film cooling with each row having P/D = 3 is also presented. The compound angle orientations investigated are 0°, 30° and 60° at blowing ratios of 1.5, 2 and 2.5. The turbulence intensity levels studied are 8.9% and 18.2% with turbulence length scales of 0.8D and 3.8D respectively. The film cooling performance of the 7-7-7 shaped holes was compared with that of cylindrical holes for the same test conditions.
REFERENCES


CHAPTER 1

THE EFFECT OF COMPOUND ANGLE OF A SHAPED HOLE ON VANE PRESSURE SIDE FILM COOLING: EXPERIMENTAL STUDY

1.1. ABSTRACT

To increase gas turbine engine efficiency, adequate cooling methods are required for the hot gas path components of the engine. Film cooling is a very important technology for cooling gas turbine components. This is especially true for the first-stage vanes of gas turbine engines which experience the greatest heat load in the turbine section of the engine. In this study, the effect of compound angle of 7-7-7 shaped film cooling hole is experimentally investigated on the pressure side of a scaled-up first-stage GE-E3 nozzle guide vane in a subsonic linear cascade. Steady-state experiments were performed using infrared thermography technique to obtain the adiabatic film cooling effectiveness. Three rows of holes were placed on the pressure side to study the effect of the interaction of fluid emanating from rows of holes on film cooling. Each row of holes has P/D = 3. Two different compound angles of 30° and 60° in addition to the axial orientation (0° compound angle) of the 7-7-7 holes were studied. The range of blowing ratios studied is 1 to 3 at density ratio of approximately 1. The results obtained for the 7-7-7 shaped holes were compared to those for cylindrical holes for the same configurations. From this study, it was determined that axial 7-7-7 shaped holes at blowing ratio of 3 performed better than all the other configurations tested. Also, inclining the holes at 30° compound angle significantly reduced adiabatic film cooling effectiveness for the cylindrical holes.
1.2. INTRODUCTION

Film cooling of gas turbine hot gas path components has continued to be studied extensively because it is a very useful technology for ensuring higher efficiency of gas turbine engines as it allows for higher rotor inlet temperatures (RIT). Han et al. [1.1] presented a review of gas turbine film cooling up to year 2010. A review of factors which affect film cooling performance such as the geometry and configuration of cooling holes and mainstream and surface effects are presented by Bogard and Thole [1.2]. Goldstein et al. [1.3] introduced shaping of the exits of cylindrical holes and showed that the shaped holes performed better than the cylindrical holes. They used fan-shaped hole with expansion angle of 10° on both sides. The shaped hole had significantly greater film cooling effectiveness downstream of either just one hole or of a hole in a row of holes. Also, they noted greater lateral spread of the secondary fluid downstream of the shaped hole as compared to the cylindrical hole. Many researchers have studied different film cooling hole shapes [1.4–1.6]. The potential of each type of hole is dependent on ease of manufacturing, cost effectiveness and functionality [1.7]. It is well known that shaped holes are the industry standard for film cooling of gas turbine airfoils [1.7]. As Bunker [1.8] noted, round film holes are still used in regions which do not permit exit shaping or where exit shaping offers no significant value.

Shaped holes have been studied extensively. Some examples are [1.9–1.12]. Saumweber and Schulz [1.13] studied the effect of geometric and flow parameters on the performance of fan-shaped hole. They noted that the geometric parameters of greatest importance are the diffuser’s expansion angle, the angle of inclination of the cooling hole in addition to the length of the cylindrical part of a diffuser cooling hole. Gritsch et al. [1.14] experimentally studied the effect of different hole geometry parameters such as hole length, L/D, and the compound angle of the hole.
They noted that compound angle addition results in rather lower film cooling performance at higher blowing ratios as compared to an axial hole. They suggested using non-symmetric diffusion to overcome this. Yu et al. [1.15] studied the overall film cooling performance of three different hole shapes using transient liquid crystal technique. The shapes are straight circular hole, laid-back hole with 10° forward diffusion and laid-back fan-shaped hole with both forward and lateral diffusion of 10°. They also carried out flow visualization using a pulsed laser sheet seeded with alumina particles. They reported that the laid-back fan-shaped hole produces the greatest film effectiveness and overall heat transfer reduction when compared to the other hole geometries investigated. The most frequent hole shape in literature is the laidback fan-shaped hole [1.16]. The laidback fan-shaped hole expands in both the forward and lateral directions. Schroeder and Thole [1.16] noted variation of trends in the results of different film cooling hole shapes and that most of the studies in the literature compare novel cooling hole designs with cylindrical holes. They proposed the 7-7-7 cooling hole design as a basis for comparing novel hole designs since jet detachment at high momentum flux ratios occurs with cylindrical holes. This baseline shaped hole they proposed has 7° expansion angle in the forward and lateral directions and inclination angle of 30°. The length of the hole is 6 hole diameters while the hole exit-to-inlet area ratio is 2.5. They noted that for the low- and high-density ratio cases peak adiabatic effectiveness was obtained at blowing ratio of about 1.5. Simon et al. [1.17] carried out numerical simulations to compare the performance of k-ε Realizable model in ANSYS Fluent with a high-fidelity solver Lattice-Boltzmann Method computed using Simulia PowerFLOW for 7-7-7 shaped hole. The density ratio, blowing ratio and turbulence intensity were 1.5, 1.5 and 0.5% respectively. They compared their results with the experimental results of Schroeder and Thole [1.16]. They reported that laterally-averaged adiabatic effectiveness for k-ε Realizable model agrees reasonably well with
experimental results while centerline and the lateral extremities have over predicted adiabatic effectiveness values. They obtained excellent results with Lattice-Boltzmann Method. Schroeder and Thole [1.18] studied the effect of freestream turbulence intensity on both flow field and adiabatic effectiveness of the 7-7-7 shaped holes. They used PIV to obtain flowfield data. Haydt et al. [1.19] noted that errors in the alignment of the metering and diffuser section of film cooling holes can occur during manufacturing. They studied the impact of meter-diffuser offset using 7-7-7 shaped hole. They obtained adiabatic effectiveness measurements for blowing ratios which range from 0.5 to 3 at density ratio of 1.2. They reported that only fore offset, which is one of the five offset directions they tested, performed slightly better than the baseline 7-7-7 hole. Hossain et al. [1.20] compared the performance of a row of sweeping jet film cooling hole to a row of 7-7-7 holes on the suction side of a nozzle guide vane both experimentally and numerically. They obtained measurements of adiabatic effectiveness, thermal field and velocity boundary layer at freestream turbulence intensity of 0.6% and 14.3% in a low-speed linear cascade for blowing ratios ranging from 0.5 to 3.5. They obtained adiabatic film cooling effectiveness using infrared thermography technique. They reported that sweeping jet hole gives better cooling effectiveness at high blowing ratios. Hossain et al. [1.21] compared the performance of a row of sweeping jet holes with the 7-7-7 holes on the suction side of a vane in a transonic turbine cascade. They reported slight increase in effectiveness for the 7-7-7 hole as result of increased lateral spreading of coolant at high blowing ratios and high freestream turbulence level, and slight decrease in effectives for sweeping jet hole at high blowing ratios.

Several parameters affect gas turbine airfoil cooling, one of which is the compound angle which is the angle the cooling hole centerline makes with the streamwise direction. Compound angle is used to enhance the rate at which coolant spreads, thus resulting in more even coolant
coverage [1.22]. McClintic et al. [1.23,1.24] studied the effect of crossflow velocity on axial and compound angle 7-7-7 shaped hole. It was noted that the magnitude of internal crossflow velocity significantly influences film cooling effectiveness of axial shaped holes. For compound angle holes, they reported that the direction of crossflow influences the bias of the jets within the diffuser. While the bias of in-line crossflow is toward the diffuser’s downstream side, that of counter crossflow is toward the diffuser’s upstream side. Haydt and Lynch [1.25] studied the effect of compound angle of 7-7-7 shaped hole for pitchwise spacing of P/D = 3 and 6 for a range of blowing ratios from 1 to 4. They reported that increasing the compound angle resulted in greater laterally averaged effectiveness at high blowing ratios.

Studies of 7-7-7 shaped holes have mostly been conducted on flat plates. There is need for studies of 7-7-7 shaped holes on gas turbine airfoils, especially investigations of the influence of compound angle. To the best of the author’s knowledge, the effect of compound angle of the 7-7-7 shaped holes on the adiabatic film cooling effectiveness of a nozzle guide vane has not been studied. Furthermore, the effect of multiple rows of the 7-7-7 shaped holes needs to be investigated. The aim of this study is to experimentally determine how multiple rows of 7-7-7 holes and compound angle inclinations affect adiabatic film cooling effectiveness on the pressure side of a first-stage gas turbine nozzle guide vane. The results will be compared with those of cylindrical holes for similar test conditions.

1.3. EXPERIMENTAL SETUP

The experiments were performed in the subsonic gas turbine vane cascade located in the Thermal Energy Research and Management Laboratory (ThERMaL) at North Carolina State University. The schematic diagram of the experimental setup is presented in Figure 1.1. The rig is
an open-loop linear cascade wind tunnel. At the tunnel inlet, air is supplied by a pressure blower having a maximum flow rate of 3000 CFM. The blower is connected to a 20-HP motor with a maximum speed of 3450 R.P.M. The desired air flow rate into the wind tunnel is set by PowerFlex 70 Adjustable Frequency AC Drive.

Figure 1.1: Subsonic Gas Turbine Vane Cascade.

The blower pushes air through a diverging section, a straight section then it enters a converging section. Then the air enters a transition duct before it gets to the test section. The test section is a three-vane, two-passage linear cascade. The vanes which are scaled-up GE – E3 first-stage vanes are made of acrylonitrile butadiene styrene (ABS) which has low-thermal conductivity. Two tailboards ensure periodic flow around the central vane. The blower supplies mainstream air at about 25 – 29°C while the secondary fluid is air supplied by the laboratory compressor and passes through two buffer tanks before it is heated to about 45 – 66°C using an inline heater. The secondary fluid is fed into a plenum from which it enters into rotameters that feed the individual
rows of film cooling holes. The amount of secondary fluid to obtain a particular blowing ratio is set using the rotameters.

The middle vane is modular and is changed in order to study different film cooling hole configurations. The parameters of the vane are presented in Table 1.1. The diameter of the metering section of the film cooling holes (D) is 1.5 mm. The film cooling hole geometries used in this study are the 7-7-7 shaped hole [1.16] and the cylindrical hole. The design of the 7-7-7 shaped hole is presented in Figure 1.2 and the geometric parameters of hole are presented in Table 1.2. The film cooling holes are inclined to the vane surface at an angle of 30° and have L/D = 6. The effect of compound angle of the holes was investigated. The compound angles investigated were 0° (axial hole), 30° and 60°. It should be noted that it was not possible to get L/D of 6 for the 7-7-7 shaped holes inclined at 30° and 60° compound angles due to the curvature of the pressure surface. However, the profile of the diverging section of the shaped hole for these compound angle orientations followed that of the 7-7-7 hole and the length of the metering section was the same as for the axial 7-7-7 holes. Three cooling-hole rows were placed on the pressure side of the vane at normalized axial positions, X/Cx of 0.15, 0.3 and 0.65. There were 16 holes in each row of cooling holes and the holes were placed in the central 60% of the span of the vane. The schematic diagrams of the vanes tested in the study are shown in Figure 1.3. The region of interest (ROI) is defined by X/Cx from 0.1 to 0.68 and Z/H from 0.36 to 0.56. A quartz window was placed in the transition duct in order to view the region of interest. The schematic of the ROI is shown in Figure 1.4.
Table 1.1: GE–E3 vane parameters for this study.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Vanes</td>
<td>3</td>
</tr>
<tr>
<td>Hole inlet diameter</td>
<td>1.5 mm</td>
</tr>
<tr>
<td>Hole Injection Angle</td>
<td>30°</td>
</tr>
<tr>
<td>Compound Angle</td>
<td>0°, 30° and 60°</td>
</tr>
<tr>
<td>Hole Spacing (P/D)</td>
<td>3</td>
</tr>
<tr>
<td>Hole length (L/D)</td>
<td>6</td>
</tr>
<tr>
<td>Number of Cooling Hole Rows</td>
<td>3</td>
</tr>
<tr>
<td>Number of Cooling Holes in each Row</td>
<td>16</td>
</tr>
<tr>
<td>Normalized Axial Position of Cooling Hole Rows</td>
<td>0.15, 0.3 and 0.65</td>
</tr>
<tr>
<td>Span (H)</td>
<td>0.1143 m</td>
</tr>
<tr>
<td>Axial Chord (C_x)</td>
<td>0.2216 m</td>
</tr>
<tr>
<td>Vane Pitch (P)</td>
<td>0.2286 m</td>
</tr>
</tbody>
</table>
Figure 1.2: The 7-7-7 shaped hole design [1.16].
Table 1.2: Geometric parameters of the 7-7-7 shaped hole [1.16].

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection Angle, $\alpha$</td>
<td>30°</td>
</tr>
<tr>
<td>$L_m/D$</td>
<td>2.5</td>
</tr>
<tr>
<td>$L_{lat}/D$, $L_{fwd}/D$</td>
<td>3.5</td>
</tr>
<tr>
<td>$L/D$</td>
<td>6</td>
</tr>
<tr>
<td>Laidback Angle, $\beta_{fwd}$</td>
<td>7°</td>
</tr>
<tr>
<td>Lateral Angle, $\beta_{lat}$</td>
<td>7°</td>
</tr>
<tr>
<td>Coverage Ratio, $t/P$</td>
<td>0.35</td>
</tr>
<tr>
<td>Area Ratio, AR</td>
<td>2.5</td>
</tr>
<tr>
<td>Sharpness of Inlet at Breakout Edges</td>
<td>Sharp</td>
</tr>
<tr>
<td>Rounding of Four Edges Inside</td>
<td></td>
</tr>
<tr>
<td>Diffuser, $R/D$</td>
<td>0.5</td>
</tr>
</tbody>
</table>
Figure 1.3: Schematic diagrams of the vanes tested in this study (a) axial cylindrical hole, (b) cylindrical hole with 30° compound angle, (c) cylindrical hole with 60° compound angle, (d) axial 7-7-7 shaped hole, (e) 7-7-7 shaped hole with 30° compound angle, (f) 7-7-7 shaped hole with 60° compound angle.

Figure 1.4: Region of interest (ROI).
1.4, EXPERIMENTAL PROCEDURE AND DATA REDUCTION

The static and total pressures at inlet and exit of the test section were obtained using pressure taps and Kiel probes respectively. All were connected to the Scanivalve ZOC 33 pressure scanner. The static and total pressures at inlet to the test section was obtained at 0.25C\(_x\) upstream of the leading edge of the vane while the static and total pressures downstream of the test section was obtained 0.2C\(_x\) downstream of the trailing edge of the vane. The inlet static pressure was obtained in order to confirm that the flow at the inlet to the test section was uniform. Also, static pressure taps were placed at 50\% of the span on the pressure side to measure the wall static pressure. This subsonic gas turbine vane cascade was benchmarked with another GE – E3 vane cascade by comparing the vane surface Mach number distribution on the pressure surface of the vane in this study with that of Ramesh et al. [1.26]. The comparison of the vane surface Mach number distribution is presented in Figure 1.5. A very good agreement can be seen in the data from the two studies.
The IR camera used to obtain the surface temperature of the vane was the FLIR A6750sc camera. Imaging of the pressure surface of the vane using the IR camera was done through the quartz window located in the transition duct. Therefore, calibration was necessary since the IR camera imaging was done through the quartz window. For calibration, one thermocouple was placed near the exit of one hole in each of the three film cooling hole rows. Thus, three thermocouples were used in all. The temperatures obtained using the thermocouples were matched with the temperature of the particular pixels of the image from the IR camera corresponding to the thermocouple locations. This was done in order to determine the relationship between the actual vane surface temperature and the temperature obtained from the IR camera. Calibration was done for the entire range of temperatures typical of all experiments and during calibration data logging from the IR camera and thermocouples were synchronized. To ensure high emissivity, the vanes that were not black in color were painted with Rust-Oleum® black paint. Calibration was done for
each of the film cooling hole configurations tested. A sample calibration curve is shown in Figure 1.6.

![Typical IR camera calibration curve.](image)

Figure 1.6: Typical IR camera calibration curve.

All experiments of the vane configurations were performed at a range of blowing ratio from 1 to 3 and density ratio of approximately 1 at steady-state condition. When the mainstream and coolant flow rates were set to obtain a particular blowing ratio, they were left to run for 30 minutes to ensure that steady-state condition was achieved. Steady state was considered to be achieved when the change in temperature of each pixel recorded by the IR camera was less than 1% over a minute interval. The blowing ratios tested were 1, 1.5, 2, 2.5 and 3. The temperatures of the mainstream air and secondary air (coolant) were monitored during the experiments using thermocouples. The velocity of air at the inlet of the test section during all experiments was 12.5 m/s. The blowing ratio was set by varying the flow rate of the secondary fluid to the rows of the cooling holes using Brooks® Instrument model 2530 flowmeters (rotameters). The Reynolds number based on
mainstream inlet velocity and axial chord of the vane for all experiments was 175,340. A LABVIEW program was used to acquire experimental data. Data post-processing was done using MATLAB. A summary of the experimental conditions is presented in Table 1.3.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reynolds Number</td>
<td>175,340</td>
</tr>
<tr>
<td>Mainstream Temperature</td>
<td>~ 298 - 302 K</td>
</tr>
<tr>
<td>Secondary fluid Temperature</td>
<td>~ 318 - 339 K</td>
</tr>
<tr>
<td>Mainstream Inlet Velocity</td>
<td>12.5 m/s</td>
</tr>
<tr>
<td>Freestream Turbulence Intensity</td>
<td>8.9%</td>
</tr>
<tr>
<td>Density Ratio</td>
<td>1 ± 0.1</td>
</tr>
<tr>
<td>Blowing Ratio ( (M) )</td>
<td>1 to 3</td>
</tr>
</tbody>
</table>

### 1.4.1. Data Reduction

The adiabatic film cooling effectiveness, \( \eta \), is defined as follows:

\[
\eta = \frac{T_m - T_{aw}}{T_m - T_c} \tag{1.1}
\]

where \( T_m \) is the mainstream temperature, \( T_c \) is the coolant or secondary fluid temperature and \( T_{aw} \) is the adiabatic wall temperature. The mainstream and secondary fluid temperatures were measured using T-type thermocouples and \( T_{aw} \) was determined using the IR camera.

All experiments were carried out at Reynolds number of 175,340 defined as follows:
\[ Re = \frac{U_m C_x}{\nu} \]  \hspace{1cm} (1.2)

where \( U_m \) and \( C_x \) are the mainstream inlet velocity and axial chord of the vane respectively, and \( \nu \) is the kinematic viscosity of air at the temperature of the mainstream air.

The blowing ratio, \( M \), is given by:

\[ M = \frac{\rho_c U_c}{\rho_m U_m} \] \hspace{1cm} (1.3)

where \( \rho \) and \( U \) are the density and flow velocity respectively. The subscripts \( c \) and \( m \) represent the secondary air and mainstream air respectively.

1.4.2. Uncertainty

The accuracy of the Scanivalve ZOC 33 pressure scanner used is \( \pm 0.12\% \) full scale. The accuracy of the rotameters used to measure the flowrate of the secondary fluid in order to determine the blowing ratio is \( \pm 3\% \) full scale. The sequential perturbation method described by Moffat [1.27] was used to obtain experimental uncertainty. In this method, the deviation in the adiabatic effectiveness using the higher value and the lower value of a variable was calculated while keeping the other variables constant. The average of the absolute values of these deviations is the uncertainty due to that variable. The root sum square of the individual uncertainties is the total uncertainty. The thermocouples used in calibration have an error of \( \pm 1.1^{\circ}\)C and considering calibration error, the maximum error in the average wall temperature using the IR camera of \( \pm 4.5 \) K was obtained. The average uncertainty in adiabatic film cooling effectiveness for the experiments was 10\%. The thermocouples are the source of uncertainty in the adiabatic effectiveness calculation.
1.4.3. Conduction Correction

Although the vanes were made of ABS which has low thermal conductivity, conduction within the vane material significantly affects the adiabatic film cooling effectiveness. In order to account for the effect of conduction, computational fluid dynamics (CFD) simulations were performed using ANSYS Fluent. Two simulations were performed for each blowing ratio for the 7-7-7 shaped hole and for the cylindrical hole. Both simulations were representative of the actual vane geometry and linear cascade test section. One simulation was a conjugate heat transfer CFD model using the properties of ABS while the other was performed using an adiabatic surface. An effectiveness correction factor was obtained from the temperature difference between the both cases. The effectiveness correction factor, $\eta_{cf}$, is defined in equation (1.4).

$$\eta_{cf} = \frac{T_{ABS} - T_{adia}}{T_c - T_m}$$

where the numerator in equation (4) is the temperature difference between the two cases and the denominator is the difference in temperature between the secondary air and the mainstream air. Areas just upstream, between and just downstream of a row of holes required the greatest correction. Areas downstream of the second row of holes which are not close to the third row of holes required the least correction. Applying the correction factor to the data results in reduced effectiveness values. Figure 1.7 (a) shows the contour of the raw experimental data, while Figure 1.7 (b) shows the corrected data at $M = 1$ for 7-7-7 shaped hole inclined at 30° compound angle. The laterally averaged effectiveness value for the same case is shown in Figure 1.8. From Figure 1.8, it can be seen that the effect of conduction is significant.
Figure 1.7: Conduction correction for 7-7-7 shaped hole at 30° compound angle at M = 1
(a) Pre-correction contour (b) Post-correction contour.

Figure 1.8: Conduction correction effect on laterally-averaged adiabatic effectiveness for 7-7-7 shaped hole inclined at 30° compound angle at M = 1.

1.5. RESULTS AND DISCUSSION

1.5.1. The Cylindrical Holes

The effect of compound angle on cylindrical holes is presented in this section. The adiabatic effectiveness contours of axial cylindrical holes are presented in Figure 1.9. From Figure 1.9 it can be seen that blowing ratios of 1 and 3 have the greatest cooling effectiveness. Jet lift-off is evident from all the contours as the streaks of coolant are not continuous. The blowing ratio of 1.5 had the
worst performance of all the cases. There is an increase in the adiabatic film cooling effectiveness for blowing ratios from 2 to 3. The large increase in blowing ratio at \( M = 3 \) could be because the coolant has greater momentum.

![Figure 1.9: Adiabatic film cooling effectiveness contours for axial cylindrical holes for (a) M = 1, (b) M = 1.5, (c) M = 2, (d) M = 2.5 and (e) M = 3.](image)

The laterally-averaged adiabatic film cooling effectiveness curves for axial cylindrical holes are shown in Figure 1.10. From Figure 1.10, the greatest peak effectiveness values for the three rows of holes were obtained at \( M = 3 \). As observed from the contour plots in Figure 1.9, the adiabatic film cooling effectiveness is lowest for \( M = 1.5 \) and then increases as \( M \) increases. There
is a remarkable jump in adiabatic film cooling effectiveness values from \( M = 2.5 \) to \( M = 3 \). More investigation is needed to understand why this is so. The interaction between rows of cooling holes is also evident from Figure 1.10. It can be seen that cooling effectiveness greatly diminishes after the first row of rows, but there is greater lateral spread of coolant after the second row of holes. It strongly suggests that the interaction of coolant from the first and second rows of holes produces greater film cooling effectiveness downstream of the second row of holes. This is in agreement with information available in the literature [1,2]. The slope observed just before the third row of holes indicates jet reattachment downstream of the second row of holes and upstream of the third row of holes. This jet reattachment to the vane surface greatly enhances film cooling effectiveness. Higher lateral spread of coolant downstream of the second row of holes are observed for \( M = 1 \) and \( M = 3 \) with \( M = 3 \) having the highest.

![Laterally-averaged adiabatic film cooling effectiveness for axial cylindrical holes.](image)

Figure 1.10: Laterally-averaged adiabatic film cooling effectiveness for axial cylindrical holes.

The adiabatic film cooling effectiveness contours for the 30° compound angle case are presented in Figure 1.11. Generally, as compared to the axial holes, the cooling performance for
the 30° compound angle case is much lower. This strongly suggest that inclining cylindrical holes at 30° compound angle on the pressure surface of the vane results in diminished film cooling performance. As seen from Figure 1.11, the coolant jets attach more to the surface of the vane as compared to axial holes. This effect of compound angle injection is as expected [1.28]. Film cooling effectiveness is greatest for $M = 1$ case. The case with the least performance is the $M = 2$ case. There is increase in film cooling effectiveness as $M$ increases from 2.5 to 3, but the levels are not up to that of the $M = 1.5$ case. It is obvious form Figure 1.11 that even though there are coolant streaks on the surface, jet-lift off occurs. The inclination of the holes at 30° compound angle causes the coolant jets to be significantly blown away from the region of interest, hence the poor film cooling performance when compared to axial holes.

Figure 1.11: Adiabatic film cooling effectiveness contours for cylindrical holes inclined at 30° compound angle for (a) $M = 1$, (b) $M = 1.5$, (c) $M = 2$, (d) $M = 2.5$ and (e) $M = 3$. 
The laterally-averaged adiabatic film cooling effectiveness plots for 30° compound angle cylindrical holes are shown in Figure 1.12. As seen from Figure 1.12, the trend in laterally-averaged effectiveness follows that of the contours in Figure 1.11. The greatest peak effectiveness values at the location of the rows of holes is obtained for \( M = 1 \). Also, the \( M = 1 \) case has the greatest laterally-averaged adiabatic effectiveness values. Downstream of the first row of holes there is a marked decline in laterally-averaged effectiveness for all blowing ratios. This indicates that jet lift-off occurs. Downstream of the second row of holes, the \( M = 1 \) and the \( M = 1.5 \) cases provide better lateral spread of coolant when compared to the other blowing ratios. The effect of cooling hole row interaction cannot be seen downstream of the second row of holes as seen from the cases for the axial hole. This strongly indicates significant jet lift-off and that the coolant is blown away from the ROI. The cooling performance downstream of the second row of holes greatly diminishes for the \( M = 2 \) to the \( M = 3 \) cases. For the \( M = 2 \) case at about \( X/C_x = 0.2 \) to 0.22 and 0.42 to 0.58 no protection is offered by the coolant to the surface. For the \( M = 2.5 \) case at about \( X/C_x = 0.51 \) to 0.56 no protection is offered to the surface by the coolant.
Figure 1.12: Laterally-averaged adiabatic film cooling effectiveness for cylindrical holes inclined at 30° compound angle.

In Figure 1.13, the adiabatic film cooling effectiveness contours for the 60° compound angle case are shown. Generally, it can be seen from Figure 1.13 that film cooling effectiveness increases with blowing ratio with the exception of the $M = 2.5$ case. The interaction between the coolant from adjacent holes in a row of holes is more pronounced and coolant streaks attached more to the surface of the vane as compared to axial holes. As Bogard and Thole [1.2] noted, compound angle injection results in a larger area which the secondary fluid covers downstream of the film cooling hole because of the broader profile of the jet. It is obvious that jet lift-off occurs for all blowing ratios. The inclination of the cylindrical holes at 60° compound angle results in better film cooling performance as compared to the inclination at 30° compound angle. This suggests that there is little benefit obtained from inclining cylindrical holes at 30° compound angle from the adiabatic film cooling effectiveness standpoint. From the adiabatic film cooling effectiveness contours, it
can be seen that axial cylindrical holes offer better protection for the ROI for $M = 1$ and $M = 3$
cases when compared to $60^\circ$ compound angle cylindrical holes for the same blowing ratios.

Figure 1.13: Adiabatic film cooling effectiveness contours for cylindrical holes inclined at $60^\circ$
compound angle for (a) $M = 1$, (b) $M = 1.5$, (c) $M = 2$, (d) $M = 2.5$ and (e) $M = 3$.

The laterally-averaged adiabatic film cooling effectiveness plots for the $60^\circ$ compound angle
cylindrical holes are shown in Figure 1.14 for the different blowing ratios. Again, as observed
from the effectiveness contours in Figure 1.13, the adiabatic effectiveness increases with blowing
ratio with the exception of the $M = 2.5$ case. From Figure 1.14, there is evidence of jet lift-off as
there is rapid decline in adiabatic film cooling effectiveness levels just downstream of the rows of
holes. The peak effectiveness level at the location of the rows of holes was greatest for the $M = 3$
case. This was followed by the $M = 2$ case, although the difference between both cases is not very significant. This could indicate that the optimum blowing ratio for this configuration is $M = 2$. The $M = 2.5$ case had the lowest level of laterally-averaged effectiveness from about $X/C_x = 0.37$ to $0.62$. Jet reattachment downstream of the rows of holes is not as pronounced as that for the axial cylindrical holes shown in Figure 1.10. This is because the curves upstream of the second and third rows of holes in Figure 1.14 are steeper than those in Figure 1.10.

![Area-Averaged Adiabatic Effectiveness for Cylindrical Holes](image)

Figure 1.14: Laterally-averaged adiabatic film cooling effectiveness for cylindrical holes inclined at 60° compound angle.

1.5.2. Area-Averaged Adiabatic Effectiveness for Cylindrical Holes

The plot of area-averaged adiabatic film cooling effectiveness for cylindrical holes as a function of compound angle and blowing ratio is presented in Figure 1.15. From Figure 1.15, it can be seen that axial cylindrical holes had greater area-averaged adiabatic effectiveness values for $M = 1$ and $M = 3$ when compared to all the other cases. The $M = 3$ case for axial cylindrical
holes had the highest area-averaged adiabatic effectiveness value. For the axial holes, the area-averaged value drops from $M = 1$ to $M = 1.5$. It then continues to increase from $M = 1.5$ to $M = 3$. It is evident that axial cylindrical holes at $M = 3$ offers the best protection of the ROI. The 30° compound angle holes had the least area-averaged adiabatic effectiveness values for all blowing ratios except the $M = 1$ case. As noted from the adiabatic effectiveness contours and laterally-averaged adiabatic effectiveness plots, the cylindrical 30° compound angle holes provide the least protection of the ROI. For the 60° compound angle case, the area-averaged effectiveness value increases with blowing ratio except for the $M = 2.5$ case. This follows the trend from the adiabatic effectiveness contours and the laterally-averaged adiabatic effectiveness plots. The area-averaged adiabatic effectiveness value for $M = 3$ for the 60° compound angle case is slightly higher than the $M = 2$ case. Thus, in terms of area-averaged effectiveness for 60° compound angle holes, $M = 2$ seems to be the optimum blowing ratio.

![Figure 1.15: Area-averaged adiabatic film cooling effectiveness for cylindrical holes for the different compound angles and blowing ratios.](image)
1.5.3. The 7-7-7 Shaped Holes

The adiabatic film cooling effectiveness contours for axial 7-7-7 shaped holes are presented in Figure 1.16. It can be seen from Figure 1.16 that the adiabatic film cooling effectiveness increases with blowing ratio. Also, as the blowing ratio increases, the interaction of coolant between the holes in a row of holes increases. In addition, as the blowing ratio increases the interaction between the coolant from the first and second row of holes increases. These lead to increase in adiabatic effectiveness levels. As noted by Bogard and Thole [1.2], when the blowing ratios are higher, two rows of holes which are placed near each other give higher film cooling effectiveness as compared to that obtained from a superposition of the film cooling performance from one row of holes. Shaped holes are known to reduce the momentum of the jet at the exit of the cooling hole, thereby making the coolant to attach to the surface of the vane with enhanced lateral spread of coolant [1.3,1.14,1.29,1.30]. The reduced momentum of coolant, thus the increase in effectiveness levels is obvious from the contours for $M = 1.5$ to 3 as compared to the axial cylindrical holes. Again, it can be observed that as the blowing ratio increases, the coolant footprint also increases. This indicates that the coolant attaches more to the surface of the vane as the blowing ratio is increased. Jet detachment from the vane surface occurs for all blowing ratios, but due to the reduced momentum of the coolant, the coolant sticks more to the vane surface. This is clearly seen when the contours in Figure 1.16 are compared to corresponding contours in Figure 1.9. Jet reattachment to the surface also increases with increase in blowing ratio as seen in Figure 1.16. Axial cylindrical holes provides better film cooling performance than axial 7-7-7 shaped holes in terms of the effectiveness contours only for $M = 1$. 
Laterally-averaged adiabatic effectiveness plots for axial 7-7-7 shaped holes as a function of blowing ratio are presented in Figure 1.17. It can be seen from Figure 1.17 that as blowing ratio increases, the laterally-averaged adiabatic effectiveness values also increase. The laterally-averaged adiabatic effectiveness curves for \( M = 1 \) and \( M = 1.5 \) are nearly identical. Jet lift-off is obvious just downstream of the first row of holes due to the sudden decline in adiabatic effectiveness values. Significant jet reattachment to the vane surface can be observed upstream of the second row of holes as can be seen from the gentle slope of the curves before the sudden rise in adiabatic film cooling effectiveness at the location of the film cooling hole rows. The effect of
the interaction of coolant from the first and second row of holes can be seen downstream of the second row of holes. The decrease in effectiveness levels downstream of the second row of holes is not as sudden as that from the first row of holes and spread of coolant is comparatively more uniform downstream of the second row of holes. Evidence of jet reattachment to the surface of the vane can be seen upstream of the second and third rows of holes. This is indicated by the gentle slope of the curves upstream of the second and third rows of holes before the sudden increase in effectiveness at the locations of the second and third rows of holes. There is marginal increase in the film cooling performance from $M = 2.5$ to $M = 3$.

![Image](image1.png)

**Figure 1.17:** Laterally-averaged adiabatic film cooling effectiveness for axial 7-7-7 shaped holes.

The contours of adiabatic film cooling effectiveness for 7-7-7 shaped holes inclined at 30° compound angle are shown in Figure 1.18. From Figure 1.18, it can be seen that the film cooling effectiveness increases with blowing ratio. Comparing the contours in Figure 1.18 with those for the 30° compound angle cylindrical holes in Figure 1.11, it is obvious the 30° compound angle 7-
7-7 shaped holes performs better than the 30° compound angle cylindrical holes. As the blowing ratio increases, the interaction between coolant from adjacent holes in a row of holes increases and the interaction between coolant from the first and second rows of holes also increases. The coolant footprint increases with increase in blowing ratio as well. Again, the reduced momentum of the jet at the exit of the shaped hole causes the coolant jet to attach more to the vane surface as compared to the cylindrical holes inclined at 30° compound angle. Jet lift-off and reattachment can be observed at all blowing ratios. The amount of jet reattachment downstream of the first and second rows of holes increases with increase in blowing ratio.

Figure 1.18: Adiabatic film cooling effectiveness contours for 7-7-7 shaped holes inclined at 30° compound angle for (a) M = 1, (b) M = 1.5, (c) M = 2, (d) M = 2.5 and (e) M = 3.
In Figure 1.19, the laterally-averaged effectiveness plots for 7-7-7 shaped holes inclined at 30° compound angle are presented for the different blowing ratios. From Figure 1.19 the $M = 1$ case has the least laterally-averaged effectiveness values. Laterally-averaged effectiveness increases with increase in blowing ratio. There is slight difference in the laterally-averaged plots for the $M = 2$ to $M = 3$ cases with the exception being that the peak effectiveness values at the second and third rows of holes show marked difference for these blowing ratios. This could suggest that the optimum blowing ratio for this configuration is $M = 2$. Jet lift-off immediately downstream of the rows of holes can clearly be seen from Figure 1.19. This is especially obvious just downstream of the first row of holes due to the rather rapid decline in film cooling effectiveness values. The decline in effectiveness levels downstream of the second row of holes is not as drastic as that after the first row of holes. This indicates that more coolant attaches to the surface after the second row of holes which could be due to the interaction between the coolant from the first and second rows of holes. Jet reattachment downstream of a row of holes and upstream of the next row of holes can clearly be seen from Figure 1.19 as observed from the higher laterally-averaged effectiveness values. The increase in laterally-averaged effectiveness values due to jet reattachment increases with increase in blowing ratio. Overall, the film cooling performance of the 7-7-7 shaped holes inclined at 30° compound angle is far better than the cylindrical holes inclined at 30° compound angle. Generally, the peak effectiveness values of the 7-7-7 shaped holes inclined at 30° compound angle at the location of the rows of holes are higher than the peak effectiveness values of the other configurations considered so far.
Figure 1.19: Laterally-averaged adiabatic film cooling effectiveness for 7-7-7 shaped holes inclined at 30° compound angle.

The adiabatic film cooling effectiveness contours for 7-7-7 shaped holes at 60° compound angle inclination are presented in Figure 1.20. The holes as shown in Figure 1.20 are slot-like. As can be observed from Figure 1.20 the film cooling performance improves with increase in blowing ratio. Due to the slot-like nature of the holes in a given row of holes, there is increased interaction between the coolant emanating from adjacent holes. This interaction between coolant from adjacent holes in a given row of holes increases as the blowing ratio increases. Also, with increase in blowing ratio, the interaction between coolant from the first and second rows of holes also increase. As the blowing ratio increases, the film cooling effectiveness downstream of the second row of holes increases as well as the coolant footprint. The protection of the vane surface downstream of a row of holes increases with increase in blowing ratio. This indicates that the coolant attaches more to the vane surface downstream of a row of holes for this configuration. Jet lift-off and reattachment can also be observed for the 7-7-7 shaped holes with 60° compound angle.
inclination. As can be seen from Figure 1.20, jet reattachment downstream of a row of holes and upstream of the next row of holes increases with increase in blowing ratio. Jet-reattachment for this configuration is not as pronounced as jet reattachment for the 7-7-7 shaped hole having 30° compound angle inclination.

![Figure 1.20: Adiabatic film cooling effectiveness contours for 7-7-7 shaped holes inclined at 60° compound angle for (a) M = 1, (b) M = 1.5, (c) M = 2, (d) M = 2.5 and (e) M = 3.](image)

The laterally-averaged adiabatic film cooling effectiveness plots for 7-7-7 shaped holes at 60° compound angle inclination are presented for the various blowing ratios in Figure 1.21. It can be seen from Figure 1.21 that the laterally-averaged adiabatic film cooling effectiveness increases as the blowing ratio increases. The 7-7-7 shaped hole inclined at 60° compound angle produced the
very high peak effectiveness levels at the location of the rows of holes as compared to the other configurations tested. This is as a result of the slot-like nature of the film cooling holes. Also, due to the slot-like nature of the film cooling holes, there is better film cooling performance just downstream of the row of holes as indicated by the gentle slope of the curves in these regions. Jet lift-off occurs as can be seen from the low values of laterally-averaged adiabatic effectiveness further downstream of the row of holes. Jet reattachment to the vane surface downstream of a row of holes and upstream of the next row of holes for the 7-7-7 shaped hole at 60° compound angle is not as pronounced as that for the same hole geometry inclined at 30° compound angle. From Figure 1.21, the $M = 1$ case provides no protection to the vane surface from about $X/C_x = 0.46$ to 0.59. The increase in adiabatic film cooling effectiveness from $M = 2.5$ to $M = 3$ is marginal.

Figure 1.21: Laterally-averaged adiabatic film cooling effectiveness for 7-7-7 shaped holes inclined at 60° compound angle.
1.5.4. Area-Averaged Adiabatic Effectiveness for 7-7-7 Shaped Holes

The area-averaged adiabatic film cooling effectiveness plot for 7-7-7 shaped holes comparing the effects of compound angle and blowing ratio is shown in Figure 1.22. Generally, the area-averaged adiabatic effectiveness increases with blowing ratio for all compound angle inclinations. The exception to this trend is the value at \( M = 1.5 \) for the axial hole which is slightly lower than the value for \( M = 1 \) as seen from Figure 1.22. The greatest area-averaged adiabatic effectiveness value is obtained for the axial hole at \( M = 3 \). The holes inclined at 30° compound angle have the highest film cooling effectiveness values for all the blowing ratios except for \( M = 2.5 \) and \( M = 3 \). Even for the blowing ratios of \( M = 2.5 \) and \( M = 3 \) for the 30° compound angle holes, the values of area-averaged adiabatic film cooling effectiveness for these blowing ratios are slightly lower than those for the axial shaped holes. The 60° compound angle holes have the lowest area-averaged effectiveness values for all blowing ratios. This could be attributed to the coolant being significantly blown away from the ROI.

![Figure 1.22: Area-averaged adiabatic film cooling effectiveness for 7-7-7 shaped holes for the different compound angles and blowing ratios.](image)
1.6. CONCLUSIONS

Experiments have been conducted in a subsonic linear gas turbine vane cascade on a scaled-up GE-E3 vane to study the effect of compound angle of a shaped hole on adiabatic film cooling effectiveness on the pressure side of the vane. Steady-state experiments were conducted using infrared (IR) thermography technique. The shaped hole considered in this study is the 7-7-7 shaped hole while the compound angles considered are 0° (axial hole), 30° and 60°. The blowing ratios ranged from 1 to 3. The results obtained for the 7-7-7 shaped holes were compared with those for cylindrical holes having the same configurations and for the same range of blowing ratios. The following conclusions can be drawn from this study:

- Inclining cylindrical holes at 30° compound angle produced the least adiabatic film cooling effectiveness values in the ROI as seen from the contours of adiabatic film cooling effectiveness. This may be due to significant jet lift-off and the jets fail to reattach to the surface in the ROI as can be seen from the laterally-averaged adiabatic effectiveness plots for cylindrical holes at 30° compound angle inclination. The area-averaged adiabatic effectiveness plot of this configuration also shows it has the least film cooling performance of all the configurations tested except for $M = 1$ and 1.5. This strongly suggests that inclining cylindrical holes at 30° compound angle within the region of interest is the least beneficial in terms of adiabatic film cooling effectiveness.

- For the 7-7-7 shaped holes, adiabatic film cooling effectiveness generally increases with increase in blowing ratio. This is due to the shape of the hole exit, which reduces the momentum of coolant at the exit of the hole thereby resulting in the coolant attaching more to the surface of the vane. For all compound angles tested for the shaped hole, there is only marginal increase in film cooling effectiveness from $M = 2.5$ to $M = 3$ in terms of laterally-
averaged adiabatic film cooling effectiveness. This could indicate that the optimum blowing ratio for 7-7-7 shaped holes is $M = 2.5$.

- Jet lift-off occurs for all the configurations of both holes and blowing ratios tested. Jet reattachment downstream of a row of holes and upstream of the next row of holes is not pronounced for the $30^\circ$ and $60^\circ$ compound angle inclination for cylindrical holes. There is evidence of jet reattachment to the vane surface downstream of a row of holes and upstream of the next row of holes for the 7-7-7 shaped holes as seen from the laterally-averaged adiabatic effectiveness plots. This results in improved film cooling effectiveness for the 7-7-7 shaped holes as compared to cylindrical holes in the ROI. This indicates that the reduced jet momentum of coolant from the shaped holes does not allow the jets to be lifted off far above the vane surface.

- The slot-like nature of the 7-7-7 shaped holes inclined at $60^\circ$ compound angle resulted in the greatest adiabatic effectiveness levels at the location of the rows of holes. Also, this configuration resulted in comparatively higher adiabatic effectiveness values just downstream of the rows of holes. However, for this configuration, significant jet-lift off occurs further downstream of the row of holes resulting in lower jet reattachment to the surface upstream of the next row of holes when compared to the other compound angles for the 7-7-7 shaped hole tested. This leads to the 7-7-7 shaped holes at $60^\circ$ compound angle having the lowest area-averaged adiabatic effectiveness values when compared to the other compound angles inclinations for this hole shape.

- The axial 7-7-7 shaped hole at $M = 3$ produced the best film cooling performance in the ROI of all the configurations tested in this study as seen from the adiabatic film cooling effectiveness contours, laterally-averaged and area-averaged adiabatic effectiveness plots.
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Cross-sectional area of cooling hole (m²)</td>
</tr>
<tr>
<td>ABS</td>
<td>Acrylonitrile Butadiene Styrene</td>
</tr>
<tr>
<td>AR</td>
<td>Area ratio, $A_{exit}/A_{inlet}$</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational fluid dynamics</td>
</tr>
<tr>
<td>Cₐ</td>
<td>Axial chord (m)</td>
</tr>
<tr>
<td>Cyl</td>
<td>Cylindrical</td>
</tr>
<tr>
<td>D</td>
<td>Diameter of film cooling hole metering section (m)</td>
</tr>
<tr>
<td>Deg</td>
<td>Degree (°)</td>
</tr>
<tr>
<td>H</td>
<td>Vane span (m)</td>
</tr>
<tr>
<td>IR</td>
<td>Infrared</td>
</tr>
<tr>
<td>L</td>
<td>Length of cooling hole (m)</td>
</tr>
<tr>
<td>M</td>
<td>Blowing ratio (-)</td>
</tr>
<tr>
<td>P</td>
<td>Cooling hole and vane pitch (m)</td>
</tr>
<tr>
<td>R</td>
<td>Radius of the interior edges of diffused outlet (m)</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number (-)</td>
</tr>
<tr>
<td>ROI</td>
<td>Region of interest</td>
</tr>
<tr>
<td>Shp</td>
<td>Shaped</td>
</tr>
</tbody>
</table>
t  Hole breakout width (m)
T  Temperature (K)
U  Velocity (m/s)
X  Streamwise direction
Z  Spanwise direction

Greek Symbols

α  Film cooling hole inclination angle (°)
β  Expansion angle of shaped hole (°)
η  Local adiabatic film cooling effectiveness (-)
ρ  Density (kg/m³)

Subscripts, Accents

adia  Adiabatic
aw  Adiabatic wall
c  Coolant or secondary fluid
cf  Correction factor
fwd  Forward
lat  Lateral
m Mainstream, Metering section

- Lateral average

= Area average
REFERENCES


CHAPTER 2

THE EFFECT OF COMPOUND ANGLE OF A SHAPED HOLE ON VANE PRESSURE SIDE FILM COOLING: COMPARISON OF EXPERIMENTAL AND NUMERICAL RESULTS

2.1. ABSTRACT

Steady-state three-dimensional computational simulations were performed using the realizable k–ε turbulence model in ANSYS Fluent to study the adiabatic film cooling effectiveness on the pressure side of a scaled-up GE-E3 nozzle guide vane in a subsonic linear cascade. The simulations were performed to complement the experimental results presented in Chapter 1. To study the effect of the interaction between coolant from different rows of cooling holes on film cooling, there were three rows of film cooling holes on the pressure side of the vane with each row having P/D = 3. Two hole geometries were tested in this study. They are the 7-7-7 shaped hole and the cylindrical hole. Both hole geometries were tested at three different compound angle orientations of 0° (axial hole), 30° and 60°, at blowing ratios of 1, 2, 3 and density ratio of approximately 1. The Reynolds number based on the axial chord of the vane and the mainstream inlet velocity for all simulations was 175,340. The results of the simulations were compared with the experimental results presented in Chapter 1. At the hole exit and 5D downstream of the hole exit, the adiabatic effectiveness obtained using the 7-7-7 shaped hole increases as the compound angle and blowing ratio increases. This increase is due to the coolant attaching more to the surface of the vane and the greater lateral spread of the coolant. The film cooling performance of the 7-7-7 shaped hole is better than that of the cylindrical hole at the hole exit and 5D downstream of the hole exit for all the compound angles and blowing ratios tested. In addition, comparing
experimental and numerical results, the numerical simulations did not adequately capture jet lift-off from the surface of the vane and did not capture jet reattachment to the vane surface downstream of the hole exit at all. While the experimental and numerical operating conditions were similar, the numerical results do not accurately predict the heat transfer at all points on the vane surface. Better turbulence modeling might help in reducing the inaccuracies.

2.2. INTRODUCTION

Film cooling of gas turbine airfoils is a very crucial for technology for operating gas turbine engines at high efficiency. In modern gas turbine engines, the temperature of the combustion gases from the combustor far exceeds the metallurgical limits of the airfoil material [2.1,2.2]. Reviews of gas turbine film cooling are available in the literature [2.3–2.6]. The first-stage gas turbine vanes experience the greatest heat load from the combustion gases [2.7], therefore they have to be adequately cooled to ensure that they meet their design life span. Film cooling of gas turbine vanes has been extensively studied both experimentally and numerically. Different film cooling hole geometries and configurations have been studied, some studies being [2.8–2.12]. Most of the previous studies on film cooling were conducted on flat plates [2.13].

Sinha et al. [2.14] studied film cooling effectiveness from a row of 7 holes with P/D = 3 with each hole having inclination angle of 35° in a flat plate and L/D = 1.75. They used thin ribbon thermocouples to measure adiabatic surface temperatures. They reported that at lower blowing rates, the centerline effectiveness scales with the mass flux ratio; while at higher blowing rates, jet detachment from and reattachment to the surface become important and scales with momentum flux ratio. Waye and Bogard [2.15] investigated the effect of axial holes in a transverse trench on the adiabatic film cooling effectiveness of the suction side of a turbine vane using infrared (IR)
thermography. Nine trench configurations were tested along with the baseline axial holes. They reported that the best adiabatic film cooling effectiveness performance was obtained using the narrow trench configuration. Schmidt et al. [2.16] studied the effect of the effect of compound angle on adiabatic film cooling effectiveness in a flat plate test facility. They studied a row of round holes, forward-expanded holes both inclined at 60° compound angle and compared the results with those of a row of axial round holes. They noted that at high ratios of momentum flux, inclining the baseline round hole at a compound angle results in greatly increased effectiveness. Also, even greater increase in effectiveness is obtained for the forward-expanded hole with compound angle.

Numerical studies have been performed to understand film cooling physics for cylindrical holes and shaped holes in both streamwise and compound angle orientations, and comparisons were made with experimental data when available [2.17–2.20]. Charbonnier et al. [2.21] performed numerical studies using two commercial computational fluid dynamics (CFD) codes – Fluent and CFX on a vane in a linear cascade wind tunnel and compared their results with experimental data. For wall heat flux and adiabatic film cooling effectiveness studies, they studied four turbulence models using CFX. The models are standard k–ε, standard k–ω, SST k–ω with transition and SST k–ω without transition. They reported that the best results were obtained for SST k–ω model with transition and compared the two CFD codes using this model. They noted that the numerical results confirmed the conclusion from the experimental work that the flow structure within the feeding plenum greatly affects the how the coolant is distributed on the outer surface. Harrison and Bogard [2.22] used the realizable k–ε model with enhanced wall treatment in Fluent to study adiabatic film cooling effectiveness of a narrow trench, wide trench and baseline cylindrical hole in a flat plate. They compared their results with experimental data. They reported that the narrow trench
performed better than the wide trench and the wide trench performed better than the baseline cylindrical hole, as was also determined experimentally. Silieti et al. [2.23] numerically investigated the film cooling effectiveness of a gas turbine endwall using a single fan-shaped cooling hole. They carried out three-dimensional simulations for adiabatic and conjugate heat transfer models at blowing ratio of 1 and coolant-to-mainflow temperature ratio of 0.54. They compared the performance of the realizable k–ε, the SST k–ω and the v²/f turbulence models, and investigated the effect of grid topology using hexahedral-, hybrid-, and tetrahedral-topology meshes. They compared their results with experimental results. They reported that the realizable k–ε model agreed best with experimental data, and noted that results obtained using a hybrid mesh and a hexahedral mesh were identical. Na and Shih [2.24] computationally studied the effect an upstream ramp with a backward-facing step on film cooling effectiveness from a row of film-cooling holes. The simulations were performed using realizable k–ε turbulence model in Fluent. They reported that an upstream ramp with a backward-facing step significantly increased laterally-averaged adiabatic effectiveness. Naik et al. [2.25] experimentally and numerically studied the aerodynamics and film cooling effectiveness behavior of a first stage high lift guide vane as well as its downstream blade. Both the vane and blade had multiple rows of film cooling holes at various axial positions of the airfoils and two different high-speed linear cascades were used for the experiments. They reported that the effect of varying blowing ratio on film cooling effectiveness was relatively little for the vane and a lot greater for the blade.

Shaped holes are the industry standard for cooling gas turbine airfoils [2.5]. Schroeder and Thole [2.26] presented a baseline laidback fan-shaped hole as a basis for comparison of novel film cooling hole geometries instead of comparing them with cylindrical holes since jet lift-off is known to occur for cylindrical holes at high momentum flux ratios. They choose the laidback fan-shaped
hole as the baseline shaped hole because it is the most common in the literature and in industry. This baseline shaped holes has expansion angles of 7° in order to prevent in-hole jet separation. They tested the film cooling performance of the hole in a flat plate facility and reported peak adiabatic effectiveness was obtained at blowing ratio near 1.5 at low and high density ratios. Haydt and Lynch [2.27] investigated the effect of compound angle on the adiabatic film cooling effectiveness of the 7-7-7 shaped hole in a flat plate facility. They tested 0°, 15°, 30°, 45° and 60° compound angle inclinations at two pitchwise hole spacing of P/D = 3 and 6 at blowing ratio range of 1 to 4 and density ratio of 1.2. In addition, they performed steady-state CFD simulations using the realizable k – ε model in Fluent for comparison with experimental data. Haydt et al. [28] carried out experimental and numerical studies of the effect of misalignment between the meter and the diffuser on the adiabatic effectiveness of the 7-7-7 shaped hole. They investigated five meter-diffuser offset directions and two offset sizes. The experiments were performed in a closed-loop wind tunnel using IR thermography. The simulations were performed using the realizable k–ε turbulence model in Fluent. They reported that both experimental and numerical results showed that a fore offset resulted in a little greater film cooling effectiveness in comparison to the baseline 7-7-7 hole. Also, they reported that the aft offset had the worst performance. Dickhoff et al. [2.29] performed numerical simulations in order to study the effect of the various Reynolds-averaged Navier-Stokes (RANS) turbulence models which are available in STAR-CCM+ on film cooling performance of the 7-7-7 shaped hole. For film cooling simulations, the models they considered are realizable k–ε, isotropic k–ω, anisotropic k–ω and v² – f models. Their numerical data was compared with experimental data. They noted that the realizable k–ε model reproduces the flow structures far better than the other turbulence models. Simon et al. [2.30] performed computational studies on the 7-7-7 shaped hole for an incompressible flow with density ratio of 1.5, blowing ratio
of 1.5 and 0.5% freestream turbulence intensity. They used the realizable k–ε model with enhanced wall treatment available in Fluent and a high fidelity solver Lattice-Boltzmann Method (LBM) using Simulia PowerFLOW. They reported that laterally-averaged effectiveness obtained using realizable k–ε model reasonably agrees with experimental results, but the centerline adiabatic effectiveness is overpredicted while on lateral extremities, adiabatic effectiveness is underpredicted. The LBM simulation agreed very well with experimental data. Hossain et al. [2.10,2.31] compared the performance of a row of sweeping jet film cooling holes to that of the 7-7-7 shaped hole on the suction side of a nozzle guide vane (NGV). McClintic et al. [2.32,2.33] in their two-part paper studied the effect of internal crossflow velocity on the film cooling effectiveness of a row of axial and compound angle 7-7-7 shaped holes in a flat plate facility.

Many of the studies on the 7-7-7 shaped hole geometry have been carried out using flat plates. Therefore, there is need for investigations of the film cooling performance of the 7-7-7 shaped hole on a vane surface especially investigating the effect of compound angle of the hole. This study is the second part of the study of the effect of compound angle and multiple rows of the 7-7-7 shaped holes on the adiabatic film cooling effectiveness of the pressure side of a first-stage gas turbine NGV. The first part of this study is the experimental study. This part of the work is the numerical study. Simulations were carried out using the realizable k–ε turbulence model available in ANSYS Fluent using similar boundary conditions as with the experimental study. This turbulence model was chosen because it has been shown, in the literature, to give reasonably good results. As Okita et al. [2.34] noted about recent studies on film cooling, various research groups reported that the realizable k–ε model produced results which reasonably agreed with experimental data. The results of the simulations for the 7-7-7 shaped holes were compared with those for cylindrical hole having the same configurations. The range of compound angles and blowing ratio
studied are 0° to 60° and 1 to 3 respectively. The numerical results were compared with experimental results to determine how well the simulations agreed with experiments.

2.3. COMPUTATIONAL SETUP

Numerical simulations of the subsonic gas turbine vane cascade used in Chapter 1 were performed. In Chapter 1, adiabatic film cooling effectiveness of the pressure side of scaled-up GE–E3 vane was obtained experimentally using IR thermography technique. The experiments were conducted in the subsonic gas turbine vane cascade in the Thermal Energy Research and Management Laboratory (ThERMaL) at North Carolina State University. The details of the experimental setup and procedure have been presented in Chapter 1, and will not be repeated here. The parts of the experimental rig simulated were the transition duct and the test section. The parameters of the vane are presented in Table 2.1. The GE–E3 vane studied has three rows of film cooling holes on the pressure side of the vane with 16 holes in each row. The film cooling holes cover the central 60% of the span of the vane. Two hole geometries were studied. The hole geometries are the 7-7-7 shaped hole [2.26] and the cylindrical hole for comparison. Figure 2.1 shows the design of the 7-7-7 shaped hole while the geometrical parameters of the hole are presented in Table 2.2. As with the experiments, the region of interest (ROI) is defined by X/Cx from 0.1 to 0.68 and Z/H from 0.36 to 0.56. Figure 2.2 shows the schematic of the ROI.
Table 2.1: GE–E3 vane parameters for this study.

<table>
<thead>
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<th>Parameter</th>
<th>Values</th>
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<tr>
<td>Number of Vanes</td>
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<tr>
<td>Hole inlet diameter</td>
<td>1.5 mm</td>
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<tr>
<td>Hole Injection Angle</td>
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<tr>
<td>Compound Angle</td>
<td>0°, 30° and 60°</td>
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<tr>
<td>Hole Spacing (P/D)</td>
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<tr>
<td>Hole length (L/D)</td>
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<tr>
<td>Number of Cooling Hole Rows</td>
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</tr>
<tr>
<td>Number of Cooling Holes in each Row</td>
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<tr>
<td>Normalized Axial Position of Cooling Hole Rows</td>
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<td>Span (H)</td>
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<tr>
<td>Axial Chord (Cₓ)</td>
<td>0.2216 m</td>
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<tr>
<td>Vane Pitch (P)</td>
<td>0.2286 m</td>
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Figure 2.1: The 7-7-7 shaped hole design [2.26].
Table 2.2: Geometric parameters of the 7-7-7 shaped hole [2.26].

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
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<td>Injection Angle, $\alpha$</td>
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<td>$L_m/D$</td>
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<td>$L_{lat}/D$, $L_{fwd}/D$</td>
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<tr>
<td>$L/D$</td>
<td>6</td>
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<td>Laidback Angle, $\beta_{fwd}$</td>
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<tr>
<td>Lateral Angle, $\beta_{lat}$</td>
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<tr>
<td>Coverage Ratio, $t/P$</td>
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<td>Area Ratio, AR</td>
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<td>Sharpness of Inlet at Breakout Edges</td>
<td>Sharp</td>
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<tr>
<td>Rounding of Four Edges Inside Diffuser, $R/D$</td>
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</table>

![Schematic of the ROI](image)

Figure 2.2: Schematic of the ROI.
The simulations were performed using ANSYS Fluent. The solutions to the incompressible three-dimensional steady-state Reynolds-averaged Navier-Stokes (RANS) equations were obtained using the realizable k–ε turbulence model with enhanced wall treatment. ANSYS meshing software was used to generate unstructured mesh. Several studies have shown that the unstructured mesh gives reasonably acceptable results [2.21,2.29,2.35]. Figure 2.3 shows the computational domain and one of the meshes generated. The details of extra refinement of the mesh in the test section and on the vane surfaces are also shown in Figure 2.3. Grid independence study was conducted and the results are presented in Figure 2.4. Figure 2.4 shows the laterally-averaged adiabatic effectiveness on the pressure side of the vane obtained using three different mesh sizes which were 21 million, 34 million and 41 million elements. It can be seen that the laterally-averaged adiabatic effectiveness remains essentially the same using the medium mesh and the fine mesh. Therefore, taking computational cost and accuracy into account, a grid size of about 34 million elements was adequate. The y+ values for all the simulations were below unity. The SIMPLE algorithm was used for pressure-velocity coupling. Spatial discretization for pressure was second order. The second order upwind scheme was used for spatial discretization of momentum, turbulence kinetic energy, turbulence dissipation rate and energy. The convergence criterion was set to residual value of $10^{-5}$ for continuity, $x -, y -, z -$ velocities, turbulence kinetic energy and turbulence dissipation rate while it was set to residual value of $10^{-8}$ for energy. When the solutions converged, the mass and energy imbalances were far below 0.01%.

Numerical simulations were performed for the cylindrical and 7-7-7 shaped holes at compound angles of 0° (axial hole), 30° and 60° for blowing ratios of 1, 2 and 3. The boundary conditions of the numerical simulations closely matched, as much as possible, the experimental boundary conditions. The mainstream and coolant mass flow rates specified in the simulations
matched the corresponding experimental values. The mainstream mass flow rate was constant while the coolant mass flow rate was varied to obtain the different blowing ratios. Table 2.3 shows the parameters for the simulations. Adiabatic and no-slip conditions were imposed on the walls.

Figure 2.3: Fluid domain (a) Selective meshing for the transition duct and the test section (b) Refined mesh on vane surfaces.
Figure 2.4: Grid independence study at $M \sim 0.5$.

Table 2.3: Parameters for the simulations.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reynolds Number</td>
<td>175,340</td>
</tr>
<tr>
<td>Mainstream Temperature</td>
<td>~ 298 - 302 K</td>
</tr>
<tr>
<td>Secondary fluid Temperature</td>
<td>~ 318 - 339 K</td>
</tr>
<tr>
<td>Mainstream Inlet Velocity</td>
<td>12.5 m/s</td>
</tr>
<tr>
<td>Freestream Turbulence Intensity</td>
<td>8.9%</td>
</tr>
<tr>
<td>Density Ratio</td>
<td>1 $\pm$ 0.1</td>
</tr>
<tr>
<td>Blowing Ratio ($M$)</td>
<td>1 to 3</td>
</tr>
</tbody>
</table>
2.3.1. Parameter calculation

The adiabatic film cooling effectiveness, $\eta$, was calculated using equation (2.1):

$$\eta = \frac{T_m - T_{aw}}{T_m - T_c}$$ (2.1)

where $T_m$ and $T_c$ are the mainstream and the coolant (secondary fluid) temperature respectively, while $T_{aw}$ is the adiabatic wall temperature.

All the simulations were performed at Reynolds number obtained from the formula below:

$$Re = \frac{U_m C_x}{\nu}$$ (2.2)

where $U_m$ and $C_x$ are the inlet velocity of the mainstream air and the axial chord of the vane respectively, while $\nu$ is the kinematic viscosity of air at the mainstream air temperature.

The blowing ratio, $M$, is defined as follows:

$$M = \frac{\rho_c U_c}{\rho_m U_m}$$ (2.3)

where $\rho$ and $U$ are the density and flow velocity respectively, and the subscripts c and m indicate the secondary air and mainstream air respectively.

2.4. RESULTS AND DISCUSSION

2.4.1. Cylindrical Holes

The comparison of experimental and numerical results for the cylindrical holes is presented in this section. In Figure 2.5, the adiabatic effectiveness contours for both the experimental and numerical studies are shown for the axial cylindrical hole. From Figure 2.5, it can be seen that there are differences between the experimental and numerical results. Jet lift-off is evident from
the experimental results as jet streaks are not visible in the ROI. Jet streaks are clearly visible in the numerical results. From the computational results it can be seen that adiabatic effectiveness increases with increase in blowing ratio and the effect of the interaction of coolant from multiple rows of holes on adiabatic effectiveness increases with increase in $M$. The $M = 2$ case had significantly lower adiabatic effectiveness than the other blowing ratios for the experimental result. This is not the same for the numerical result.

![Figure 2.5](image)

Figure 2.5: Contours of adiabatic film cooling effectiveness for axial cylindrical holes (a) Experimental, (b) Numerical.

The comparison of the laterally-averaged effectiveness for the experimental and numerical results for axial cylindrical holes is presented in Figure 2.6. From Figure 2.6, it can be seen that the plots of the experimental and numerical results are significantly different. The $M = 3$ plot for the experimental results has the highest peak effectiveness values at the location of the film cooling hole rows of all the cases. This is not so with the numerical results. Jet reattachment to the surface
of the vane downstream of the second row of holes is not captured by the numerical simulations. For the experimental results, jet reattachment downstream of the second row of holes is evident as can be seen by the gradual increase in laterally-averaged effectiveness values before the sharp increase at the location of the third row of cooling holes. The $M = 2$ and $M = 3$ cases for the numerical plots have greater laterally-averaged effectiveness values downstream of the second row of holes than the experimental results. The trends of the curves downstream of the first row of holes is different for the experimental and numerical results. While the experimental curves shown a sharp decrease in effectiveness downstream of the first row of holes and then a gradual increase upstream of the second row of holes, the numerical curves show a gradual decrease downstream of the first row of holes, then an increase before the location of the second row of holes. This increase is more pronounced for the $M = 2$ and $M = 3$ cases. The $M = 1$ cases shows a very slight increase and then decrease in effectiveness downstream of the first row of holes. Generally, the peak effectiveness levels are lower for the numerical plots as compared to the experimental plots except at the location of the third row of holes for the $M = 2$ case in which the peak effectiveness for the numerical plot is a little higher than that of the experimental plot.
The comparison of adiabatic effectiveness contours of the experimental and numerical data is shown in Figure 2.7 for cylindrical holes inclined at 30° compound angle. From Figure 2.7, the adiabatic effectiveness increases with increase in blowing ratio for the numerical results. Also, for the numerical results, the interaction of coolant from the rows of holes results in better film cooling performance as the blowing ratio increases. As seen from the experimental contours, the $M = 1$ case has the best performance followed by the $M = 3$ case and the $M = 2$ case has the least performance. For both experimental and numerical results the jets can be seen to attach more to the surface of the vane. This agrees with findings in the literature that the addition of compound angle to film cooling holes enhances jet attachment to the surface [2.13]. Even though jets attach more to the surface of the vane for holes inclined at 30° compound angle as opposed to axial holes for the experimental results, jet lift-off can be observed for the 30° compound angle inclination because of the discontinuous jet streaks. From the experimental results, significant jet lift-off could
be the reason why the adiabatic effectiveness levels for the 30° compound angle cases are much lower than those for the axial holes. The numerical results do not adequately capture jet lift-off as the experimental results do.

Figure 2.7: Contours of adiabatic film cooling effectiveness for cylindrical holes inclined at 30° compound angle (a) Experimental, (b) Numerical.

The plots of laterally-averaged adiabatic effectiveness for the 30° compound angle case for both experimental and numerical results are presented in Figure 2.8. As seen from Figure 2.8, for the experimental results, the $M = 1$ case had the highest laterally-averaged adiabatic effectiveness values of all the cases followed by the $M = 3$ case. From the experimental results, for the $M = 2$ case, the coolant offered no protection to the vane surface from about $X/C_x = 0.2$ to 0.22 and 0.42 to 0.58. This could be attributed to significant jet lift-off as observed from the contour in Figure 2.7. The numerical plots in Figure 2.8 show trends that are different from the experimental plots. The numerical plots have higher peak adiabatic effectiveness values than the experimental plots at the location of the cooling hole rows except at the first row for the $M = 3$ case. Downstream of the first row of holes, for the experimental results, cooling effectiveness drops sharply and then
increases gradually up to the location of the second row of holes. For the numerical results there is a drop in cooling effectiveness downstream of the first row of holes and then an increase followed by gradual decrease before the sudden increase at the location of the second row of holes for the $M = 2$ and $M = 3$ cases. The $M = 1$ case shows a sharp decrease in effectiveness initially and then a gradual decrease before the increase at the location of the second row of holes. For the numerical results, downstream of the second row of holes, the effect of interaction of coolant from the first and second cooling hole rows on adiabatic effectiveness is obvious as there is significant increase in film cooling effectiveness downstream of the second row of holes. Downstream of the second row of holes, from the experimental results, the $M = 1$ case produced the greatest lateral spread of coolant, while for the numerical result the $M = 3$ case produced the best lateral spread of coolant and the $M = 1$ case produced the least lateral spread of coolant. For the numerical results, downstream of the second row of holes, the effectiveness level for the $M = 1$ case falls continuously. For the $M = 2$ and $M = 3$ cases, it falls then increases a little and then falls. The adiabatic effectiveness levels for the numerical results is generally higher than for the experimental results. This could be due to the numerical results not adequately capturing jet lift-off as evident in the experimental contours of Figure 2.7.
Figure 2.8: Laterally-averaged adiabatic film cooling effectiveness for cylindrical holes inclined at 30° compound angle.

The adiabatic effectiveness contours for the 60° compound angle orientation are presented in Figure 2.9 for both experiments and simulations. From Figure 2.9, adiabatic effectiveness increases with increase in blowing ratio for experiments as well as simulations. From the experimental results, jet lift-off is evident as the coolant streaks from the cooling holes are not continuous. The numerical results also show jet lift-off, but it is not as pronounced in the numerical contours as it is in the experimental contours. The experimental contours show that there is interaction of coolant from adjacent holes in a particular row of holes. The interaction between coolant from adjacent holes in a given row of holes is not clearly pronounced for the numerical results. However, the numerical results show high effectiveness levels between adjacent holes in a given row of holes which suggest that the jets from adjacent holes interact. This interaction
between coolant from adjacent holes in a row of holes leads to higher effectiveness levels as the blowing ratio increases for both experiments and simulations. Inclining the cooling holes at 60° compound angle promotes the interaction between coolant from adjacent rows of holes as can be seen by comparing the contours in Figure 2.9 with those in Figure 2.7 for experiments and simulations.

![Figure 2.9: Contours of adiabatic film cooling effectiveness for cylindrical holes inclined at 60° compound angle (a) Experimental, (b) Numerical.](image)

The experimental plots of laterally-averaged adiabatic effectiveness are compared with the numerical plots for the 60° compound angle inclination in Figure 2.10. From Figure 2.10, the numerical results overpredict the peak effectiveness levels at the location of the rows of holes. This is especially so for the $M = 1$ case with values that are very different from the experimental results. The peak effectiveness values from simulations closely agree with those from experiments at the location of the second row of holes for $M = 2$ and 3. As with experimental results, film cooling for the simulations increases with increase in blowing ratio, the exceptions for the
simulations being downstream of the second row of holes for the $M = 2$ and $M = 3$ cases as well as at the location of the third row of holes, where the $M = 1$ case has the highest peak effectiveness value and the $M = 3$ case has the least. Generally, laterally-averaged effectiveness downstream of the first and second rows of holes is higher for simulations as compared to experiments. Both simulations and experiments show jet lift-off as seen from the sudden decrease in effectiveness levels downstream of any of the rows of cooling holes. For the experiments, the laterally-averaged effectiveness values is not very different for the $M = 2$ and $M = 3$ cases. The simulations also capture this trend. This could indicate that $M = 2$ is the optimum blowing ratio for this configuration. From the experimental plots, jet reattachment downstream of the first and second rows of holes and upstream of the second and third rows of holes is not as pronounced as for the axial cylindrical holes as shown in Figure 2.6. The simulations do not capture this jet reattachment as can be seen in Figure 2.10.
Figure 2.10: Laterally-averaged adiabatic film cooling effectiveness for cylindrical holes inclined at 60° compound angle.

2.4.2. Area-Averaged Adiabatic Effectiveness for Cylindrical Holes

The comparison of the area-averaged adiabatic effectiveness for cylindrical holes for experiments and simulations for the various compound angle orientations and blowing ratios is shown in Figure 2.11. From Figure 2.11, the $M = 3$ case for the experiments had the highest area-averaged adiabatic effectiveness value followed by the $M = 1$ case for axial holes. For axial holes, the simulations underpredict effectiveness values at $M = 1$ and $M = 3$ and overpredict effectiveness value at $M = 2$. From experiments, the area-averaged effectiveness at $M = 2$ for 30° compound angle inclination was the lowest value of all the cases considered. This agrees with the effectiveness contours and laterally-averaged effectiveness plots discussed earlier. For the holes inclined at 30° compound angle, the simulations overpredict the area-averaged effectiveness at $M$
= 2 and $M = 3$, while the value at $M = 1$ is quite close to the experimental value. The simulations overpredict the area-averaged effectiveness value for all blowing ratios for the $60^\circ$ compound angle inclination, although the value at $M = 3$ is quite similar for both the experiment and simulation. The trends for the experiments and simulations for the $60^\circ$ compound angle inclination are comparatively similar.

Figure 2.11: Area-averaged adiabatic film cooling effectiveness for cylindrical holes for the different compound angles and blowing ratios.

2.4.3. The 7-7-7 Shaped Holes

Contours of adiabatic film cooling effectiveness for the experiments and simulations for axial 7-7-7 shaped holes are presented in Figure 2.12. It can be seen from Figure 2.12 that for both experiments and simulations, adiabatic effectiveness increases with increase in blowing ratio. Also, the interaction between coolant from adjacent holes in a given row of holes increases with increase in blowing ratio for the experiments as well as simulations. In addition, the interaction of coolant from adjacent rows of holes leading to greater film cooling effectiveness levels becomes
more pronounced as the blowing ratio is increased as seen from the experimental and numerical contours. From the experiments, jet lift-off occurs at all blowing ratios as the coolant streaks are not continuous. Jet lift-off can also be observed from the simulations as the blowing ratio increases. This is especially evident when the coolant emanating from the first and second rows of holes are considered. From the experimental results, jet reattachment to the surface downstream of a given row of holes and upstream of the next row of holes is evident and this increases as blowing ratio increases. This is not the case for the numerical results. It has been established in the literature that shaped holes reduce the momentum of the jet at the exit of the hole and thus allow the secondary fluid to attach more to the surface being cooled [2.36]. The reduced momentum of coolant from the holes causes significant jet reattachment downstream of the row of holes although jet lift-off occurs as seen from the experimental contours. The coolant footprint for the axial 7-7-7 shaped holes is much broader than that for axial cylindrical holes for both experiments and simulations which is as expected [2.37].
Figure 2.12: Contours of adiabatic film cooling effectiveness for axial 7-7-7 shaped holes (a) Experimental, (b) Numerical.

In Figure 2.13 the laterally-averaged adiabatic film cooling effectiveness plots for the experiments and simulations are presented. For the experimental results, laterally-averaged adiabatic effectiveness increases with increase in blowing ratio. The peak effectiveness values at the locations of the rows of cooling holes for the $M = 1$ case is overpredicted by the simulations. The peak effectiveness value for the $M = 2$ case is similar to experimental result at the location of the second row of holes, it is underpredicted at the location of the first row of holes and overpredicted at the location of the third row of holes. The simulations underpredict the peak effectiveness values at the location of the rows of film cooling holes for the $M = 3$ case. From the experimental plots, jet lift-off can be observed just downstream of the rows of holes because of the sudden decrease in effectiveness levels just downstream of the holes. From the numerical results, the decrease in effectiveness just downstream of the rows of holes for the $M = 1$ and $M = 2$ cases is not as pronounced as that for the experimental results. For the $M = 3$ case for the
simulations, after the first row of holes there is a slight decrease in effectiveness and then an increase before a gradual decrease upstream of the second row of holes. Downstream of the second row of holes for this $M = 3$ simulation case, there is steeper decrease in effectiveness initially and then the decrease is comparatively gentle later. From Figure 2.13 it can be seen that the simulations did not capture jet reattachment downstream of particular row of holes and upstream of the next row of holes. From the experimental plots, jet reattachment to the vane surface is evident by the increased effectiveness levels upstream of the second and third rows of holes.

![Graph](image)

Figure 2.13: Laterally-averaged adiabatic film cooling effectiveness for axial 7-7-7 shaped holes.

The contours of adiabatic effectiveness for the $30^\circ$ compound angle case are presented in Figure 2.14. As seen from Figure 2.14, adiabatic effectiveness increases with increase in blowing ratio for the experiments as well as the simulations. Also, the film cooling performance of the $30^\circ$ compound angle inclination for the 7-7-7 shaped hole is better than that of the axial hole seen Figure 2.12 for all blowing ratios for both experiments and simulations. From the experimental and numerical contours, it can be seen that the $30^\circ$ compound angle inclination facilitates the
interaction of coolant from adjacent holes in a given row of holes. Also, a broader coolant footprint is obtained. Inclining the 7-7-7 shaped hole at 30° compound angle produced far better film cooling performance than that of the 30° compound angle cylindrical hole for both experiments and simulations. Schmidt et al. [2.16] showed that the combination of compound angle and hole shaping significantly improved adiabatic effectiveness at high ratios of momentum flux. Jet lift-off occurs for all blowing ratios from the experimental contours because the coolant streaks are not continuous. However, it can be seen that the coolant footprint on the surface of the vane increases leading to higher adiabatic effectiveness levels when compared to the axial 7-7-7 shaped holes in Figure 2.12. Also, because of the reduced momentum of the jets at the exit of the cooling holes, the jets that are lifted off tend to stay rather close to the surface of the vanes thus leading to jet reattachment to the surface of the vane downstream of a row of holes and upstream of the next row of holes as seen in the experimental contours in Figure 2.14. Jet reattachment to the vane surface increases with increase in blowing ratio. Jet lift-off is also captured by the numerical results in Figure 2.14 which increases as the blowing ratio increases. Jet lift-off is obvious from the numerical results by considering the coolant footprint from the first and second rows of holes as the blowing ratio increases. Jet reattachment to the vane surface cannot be observed from the numerical contours as seen in the experimental contours.
Figure 2.14: Contours of adiabatic film cooling effectiveness for 7-7-7 shaped holes inclined at 30° compound angle (a) Experimental, (b) Numerical.

The laterally-averaged adiabatic effectiveness plots for the 7-7-7 shaped holes inclined at 30° compound angle are shown in Figure 2.15. From the experimental results in Figure 2.15 it can be observed that adiabatic effectiveness increases with increase in blowing ratio. The peak effectiveness values for all blowing ratios at the location of the rows of film cooling holes for the experimental results are higher than the corresponding results for the axial hole in Figure 2.13 except at the location of the third row of holes for the $M = 1$ case. Thus, it can be seen that the 30° inclination greatly enhances film cooling effectiveness. Jet lift-off can be seen from the experimental plots by the rapid decrease in effectiveness levels downstream of row of holes. Although jet lift-off also occurs for the simulations, the decrease in effectiveness levels downstream of a row of holes for the simulations is not as rapid as that for the experiments. From Figure 2.15, the simulations overpredict adiabatic effectiveness for the $M = 1$ case. For the $M = 2$ and $M = 3$ cases the simulations underpredict the peak adiabatic effectiveness values at the location
of the cooling hole rows except at the third row of holes for the $M = 2$ case. Downstream of the second row of holes, the simulations overpredict adiabatic effectiveness levels. The simulations do not capture jet reattachment to the vane surface downstream of a given row of holes and upstream of the next row of holes as captured by the experimental plots. For the experimental plots, jet reattachment to the vane surface increases as the blowing ratio increases.

Figure 2.15: Laterally-averaged adiabatic film cooling effectiveness for 7-7-7 shaped holes inclined at 30° compound angle.

Contours of adiabatic effectiveness for the 60° compound angle inclination for both experiments and simulations are presented in Figure 2.16. As seen in Figure 2.16, adiabatic effectiveness increases as the blowing ratio increases for the experiments and simulations. Due to the slot-like nature of the film cooling holes at each row of holes, adiabatic effectiveness just at the exit of the film cooling holes is very high. For both experiments and simulations, the slot-like nature of the cooling holes in a given row of holes causes significant interaction between coolant
from adjacent holes in a row of holes resulting in very high effectiveness levels at the location of the row of holes. For this compound angle inclination, the coolant appears to be blown away from the region of interest after exiting the film cooling holes leading to the diminished film cooling performance downstream of a row of cooling holes as observed from the experimental and numerical contours. However, adiabatic effectiveness increases as blowing ratio increases downstream of the second row of holes due to the interaction of coolant from the first and second rows of holes. Like in the previous compound angle inclination considered for the 7-7-7 shaped hole, jet reattachment to the surface occurs downstream of a given row of holes and upstream of the next row of holes, and it increases with increase in blowing ratio as observed from the experimental contours. The numerical contours did not show jet reattachment to the surface of the vane.

![Figure 2.16](image)

Figure 2.16: Contours of adiabatic film cooling effectiveness for 7-7-7 shaped holes inclined at 60° compound angle (a) Experimental, (b) Numerical.
The plots of laterally-averaged adiabatic effectiveness for the 60° compound angle orientation are shown in Figure 2.17 for the experiments as well as simulations. From Figure 2.17 it can be seen that the peak adiabatic effectiveness values at the locations of the film cooling hole rows are quite close for the experiments and simulations for the $M = 2$ and $M = 3$ cases. For the $M = 1$ case, the peak effectiveness value at the location of the cooling hole rows from the simulation is overpredicted for the first and third row of holes, and the value is quite similar to the experimental value for the second row of holes. The slot-like nature of the holes at the locations of the rows of holes results in the comparatively higher peak effectiveness values for the 60° compound angle orientation. Generally, from both the experimental and numerical plots, adiabatic effectiveness increases as the blowing ratio increases with the exception being downstream of the second row of holes for the simulation at $M = 3$. Jet lift-off just after each row of holes is indicated by the rapid decline in adiabatic effectiveness levels for both experiments and simulations. However, the decrease in effectiveness just after the location of film cooling hole rows for the simulations is not as rapid as that for the experiments resulting in the simulations having greater laterally-averaged adiabatic effectiveness values downstream of the film cooling hole rows when compared to the experiments. For the $M = 1$ experimental plot, no protection is offered by the coolant to the vane surface from about $X/C_x = 0.46$ to 0.59. The $M = 1$ numerical plot does not capture this. For the experimental plots, jet reattachment to the vane surface downstream of a given row of holes and upstream of the next row of holes increases as the blowing ratio increases. This can be seen by the increase in adiabatic effectiveness just upstream of the second and third rows of holes. Again, the simulations did not capture jet reattachment to the surface of the vane.
Figure 2.17: Laterally-averaged adiabatic film cooling effectiveness for 7-7-7 shaped holes inclined at 60° compound angle.

2.4.4. Area-Averaged Adiabatic Effectiveness for the 7-7-7 Shaped Holes

The area-averaged adiabatic effectiveness plot for the different compound angles inclinations and blowing ratios for the 7-7-7 shaped holes for experiments and simulations is presented in Figure 2.18. From Figure 2.18, it can be seen that area-averaged adiabatic effectiveness increases as the blowing ratio increases for both experiments and simulations with the exception of the simulation for 60° compound angle orientation at $M = 3$. The simulations underpredict area-averaged effectiveness for the axial holes and the holes inclined at 30° compound angle, while it overpredicts the values for the 60° compound angle orientation. The underpredicted values could be because the simulations did not capture jet reattachment to the surface of the vane. The 60° compound angle orientation produced the lowest experimental area-averaged adiabatic effectiveness values for all the blowing ratios tested. The numerical results for this compound angle gave higher adiabatic effectiveness values for $M = 1$ and $M = 2$ and a reasonably close value
for $M = 3$. As noted from the contours in Figure 2.16 and the plots in Figure 2.17, the coolant appears to be blown away from the region of interest for the 60° compound angle orientation. This is especially obvious from the experimental contours and plots. This could, therefore, result in the experimental area-averaged effectiveness values for this configuration being the lowest of all the configurations tested. The higher numerical area-averaged adiabatic effectiveness values for the 60° compound angle inclination could be because the simulations did not adequately capture jet lift-off. The experimental and numerical area-averaged adiabatic effectiveness values are very close for the 30° compound angle orientation at $M = 1$ and for the 60° compound angle orientation at $M = 3$.

Figure 2.18: Area-averaged adiabatic film cooling effectiveness for 7-7-7 shaped holes for the different compound angles and blowing ratios.
2.4.5. Adiabatic Effectiveness at the Hole Exit and 5D Downstream of the Hole Exit

The contours of adiabatic effectiveness on a plane normal to the vane surface at the exit of the film cooling hole which is at \( X/C_x = 0.31 \) and that on a plane 5D (five hole diameters) downstream of the exit of the film cooling hole at \( X/C_x = 0.344 \) are presented in this section for all the blowing ratios. The contours are overlaid with streamlines of vorticity. The lateral distance on the vane surface is one pitch which is \( Z/H \) from 0.42 to 0.46 with the center of the hole for the axial cylindrical hole being at \( Z/H = 0.44 \). This same pitch was used for both cylindrical and 7-7-7 shaped holes for the different compound angle orientations considered. Also included in this section are the plots of local adiabatic effectiveness at the hole exit and 5D downstream of the hole exit for the same pitch as for the contours, that is, from \( Z/H = 0.42 \) to 0.46 for the cylindrical and 7-7-7 shaped holes for the various compound angles and blowing ratios. The exact same pitch was used for the cylindrical and 7-7-7 shaped hole so that the effect of blowing ratio and compound angle on adiabatic film cooling effectiveness for both hole geometries can be directly compared. All results presented in this section are the results of numerical simulations.

The contours of adiabatic effectiveness for \( M = 1 \) for both cylindrical and 7-7-7 shaped hole are presented in Figure 2.19. From Figure 2.19, for the axial cylindrical hole, the coolant emanating from the film cooling hole has the counter-rotating vortex pair. This is indicated by the mushroom shape of the jet as seen in the adiabatic effectiveness contour. Jets in cross flow have counter-rotating vortex pairs [2.38,2.39]. The interaction of the coolant and the mainstream causes the vortex on the plane as shown for the axial cylindrical hole. As the compound angle increases for the cylindrical hole, the counter rotating vortex pair is distorted, hence weakening the vortex as seen for the 30° and 60° compound angle cases at the exit of the film cooling hole. Five hole diameters downstream of the exit of the cooling hole, at \( X/C_x = 0.344 \), the vortex on the plane is
above the surface for the axial cylindrical hole and the axial cylindrical hole has relatively high adiabatic effectiveness level 5D downstream of the exit of the hole. This could be the reason why axial cylindrical holes have comparatively high adiabatic effectiveness at $M = 1$. At 5D downstream of the hole exit for the cylindrical hole at 30° and 60° compound angle inclinations, the coolant jets appear to be blown away from the surface as there is increased mixing with the mainstream. This could be why film cooling effectiveness levels downstream of the second row of holes for cylindrical holes at 30° and 60° compound angle inclinations at $M = 1$ are lower as compared to levels for axial cylindrical holes. As can be seen from the contours for the 7-7-7 shaped holes, the shaped holes diminish the strength of the counter rotating vortex pair of the coolant jet at the exit of the film cooling holes, thus the vortex on the normal plane seen for the corresponding cylindrical holes is not generated. For the shaped holes the jets are more attached to the vane surface both at the exit of the cooling hole and 5D downstream of the exit of the cooling hole, and the level of attachment to the surface increases as blowing ratio increases. From Figure 2.19, it can be seen that considering the normal plane both at the cooling hole exit and 5D downstream of the cooling hole exit, jet lift-off occurs both for the cylindrical and the 7-7-7 shaped hole. The coolant from the 7-7-7 shaped hole attaches more to the vane surface both at the hole exit and 5D downstream of the hole exit as compared to the cylindrical hole for the different compound angles. Thus, the 7-7-7 shaped hole, generally, offers better lateral spread of coolant on the vane surface as Figure 2.19 shows. The performance of the 7-7-7 shaped hole is better than that of the cylindrical hole 5D downstream of the hole exit.
Figure 2.19: Contours of adiabatic effectiveness on the normal plane at the exit of the hole and 5D downstream of the hole exit at $M = 1$ for (a) cylindrical hole, (b) 7-7-7 shaped hole.
The plots of local adiabatic film cooling effectiveness at the exit of the film cooling hole and 5D downstream of the exit of the film cooling hole for the distance of one pitch on the vane surface at $M = 1$ are shown in Figure 2.20. At the location of the hole, the plot for the axial cylindrical hole has a narrow peak. The peak of the plots for the cylindrical hole gets broader with the addition of compound angle. In other words, the peak for the $60^\circ$ compound angle inclination plot is broader than the peak for the $30^\circ$ compound angle inclination for the cylindrical hole. The addition of compound angle to cylindrical holes results in better lateral spreading of the coolant and lowers coolant jet penetration into the mainstream [2.12]. The 7-7-7 shaped hole has better lateral spread of coolant at the exit of the hole. The cooling effectiveness level at the exit of the hole for the shaped hole improves as the compound angle is increased. Thus, for the 7-7-7 shaped hole with compound angle, there is better lateral spread of the coolant on the vane surface and the coolant attaches more to the surface when compared to the corresponding cylindrical hole. Cooling effectiveness level 5D downstream of the film cooling hole exit reduces significantly. This is attributed to the diffusion of the coolant jet into the mainstream. Overall, the 7-7-7 shaped hole provides greater effectiveness levels than the cylindrical holes 5D downstream of the hole exit. Therefore, it can be seen that coolant from the 7-7-7 shaped hole attaches more to the surface of the vane than that from the cylindrical hole 5D downstream of the cooling hole exit.
The contours of adiabatic effectiveness for $M = 2$ for the cylindrical and 7-7-7 shaped hole are shown in Figure 2.21. For the cylindrical holes, it can be seen that with the increase in blowing ratio from $M = 1$ to $M = 2$, the vortex on the plane at the exit of the film cooling hole becomes stronger. Also, while there was no vortex on the plane at the exit of the cooling hole for the $M = 1$ simulation for the different compound angles of the 7-7-7 shaped hole, vortices are observed on the plane at the exit of the film cooling hole for the $M = 2$ simulations, except for the 60° compound angle inclination. Due to the slot-like nature of the film cooling holes for the 60° compound angle case, the jets emanating from the 7-7-7 shaped hole are ejected very close to the surface of the vane. The vortices observed for the $M = 2$ simulations for the shaped hole is because of the larger coolant momentum flux which increases the strength of the counter-rotating vortex pair at the exit of the hole. For the cylindrical holes, the vortices seen on the plane 5D downstream of the hole exit at $M = 2$ is stronger than those for $M = 1$ for the axial and 30° compound angle orientations. This is due to greater mixing of the coolant with the mainstream and could be the reason for the lower laterally-averaged effectiveness values for these orientations at $M = 2$ when compared to $M$
= 1. Also, for axial and 30° compound angle 7-7-7 shaped hole, vortices are observed on the plane 5D downstream of the cooling hole exit at $M = 2$ and they were not observed at $M = 1$. These vortices are due to greater dilution of the coolant jets with the mainstream which reduces adiabatic effectiveness levels 5D downstream of the hole exit. However, since at $M = 2$ the cooling jets have more momentum, levels of adiabatic effectiveness downstream of the cooling holes are higher than at $M = 1$. For the 60° compound angle orientation for both cylindrical and 7-7-7 shaped hole, the jets attach more to the surface of the vane. The jet for the shaped hole attaches more to the surface than the cylindrical hole at this compound angle orientation due to the slot-like nature of the hole. No vortices are formed on the plane 5D downstream of the hole exit for both holes at 60° compound angle inclination, which results in generally higher effectiveness levels close to the vane surface for this compound angle orientation.
Figure 2.21: Contours of adiabatic effectiveness on the normal plane at the exit of the hole and 5D downstream of the hole exit at $M = 2$ for (a) cylindrical hole, (b) 7-7-7 shaped hole.
The local adiabatic film cooling effectiveness plots for $M = 2$ at the exit of the film cooling hole and 5D downstream of the exit of the hole for the distance of one pitch on the surface of the vane are presented in Figure 2.22. Generally, at the exit of the film cooling hole for the cylindrical as well as the 7-7-7 shaped hole, the peak of the plots are narrower when compared to those at $M = 1$ for all but the shaped hole inclined at $60^\circ$ compound angle. Again, as the compound angle increases the lateral spread of the coolant increases at the exit of the film cooling hole. The lateral spread of the 7-7-7 shaped hole inclined at $60^\circ$ compound angle is very high and fairly uniform at the hole exit. This can be attributed to the slot-like nature of the film cooling hole. Overall, at the exit of the film cooling hole, the shaped hole inclined at $60^\circ$ compound angle provided the best film cooling performance on the vane surface. The 7-7-7 shaped hole generally performed better than the cylindrical holes 5D downstream of the exit of the hole. It can be seen that adding compound angle to cylindrical holes increases film cooling performance 5D downstream of the hole exit. The improvement in adiabatic effectiveness for the cylindrical hole increases as the compound angle is increased. The greatest peak effectiveness value 5D downstream of the hole exit was obtained with the shaped hole inclined at $30^\circ$ compound angle, although the adiabatic effectiveness level for this compound angle orientation for the shaped hole drops below that of the $60^\circ$ compound angle inclination for most of the pitch. The 7-7-7 shaped hole inclined at $60^\circ$ compound angle has the best film cooling performance 5D downstream of the hole exit when compared to all the other configurations.
Figure 2.22: Local adiabatic effectiveness at the hole exit and 5D downstream of the hole exit for one pitch at $M = 2$.

Figure 2.23 shows the contours of adiabatic effectiveness for $M = 3$ for the cylindrical as well as the 7-7-7 shaped hole. At $M = 3$, the momentum of the jet emanating from the film cooling hole is higher than those for the other two blowing ratios. This leads to the jet from the film cooling hole having stronger counter-rotating vortex pair both for the cylindrical and the 7-7-7 shaped hole. As seen for the contours at $M = 1$, only the axial and 30° compound angle cylindrical hole had the vortex on the normal plane at the exit of the hole. Due to the higher momentum of the jet at $M = 3$ compared to at $M = 1$, all but the shaped hole inclined at 60° compound angle had the vortex on the normal plane at the exit of the film cooling hole. These vortices are generated as a result of the interaction between the jet and the freestream. The slot-like nature of the shaped hole at 60° compound angle orientation makes the jet emanating from the hole attach more to the surface of the vane as compared to the other configurations, thereby preventing the formation of the vortex on the normal plane at the exit of the cooling hole. For the 7-7-7 shaped hole inclined at 60° compound angle, the lateral spread of the coolant both at the exit of the hole and 5D downstream of the hole exit is better than for the other configurations. The addition of compound angle and
hole shaping has been shown to improve film cooling effective at high blowing ratios [2.16,2.40]. The film cooling performance 5D downstream of the cylindrical hole inclined at 60° compound angle is high. Inclining the cylindrical hole at 60° compound angle results in the coolant attaching more to the surface of the vane and also increases the lateral spread of the coolant on the vane. These effects result in the improved film cooling performance 5D downstream of the hole exit for this configuration as seen in Figure 2.23. As can be observed on the normal plane 5D downstream of the hole exit for both the axial cylindrical and shaped holes as well as both the cylindrical and shaped hole inclined at 30° compound angle, the strength of the vortex on the plane is weaker for the shaped hole as compared to the cylindrical hole. This could be why the shaped hole has better film cooling performance 5D downstream of the hole exit as compared to the cylindrical hole. Generally, as can be observed from Figure 2.23, the shaped hole performs better than the cylindrical hole for all the configurations tested both at the exit of the hole and 5D downstream of the hole exit.
Figure 2.23: Contours of adiabatic effectiveness on the normal plane at the exit of the hole and 5D downstream of the hole exit at $M = 3$ for (a) cylindrical hole, (b) 7-7-7 shaped hole.
The local adiabatic film cooling effectiveness plots at $M = 3$ at the exit of the film cooling hole as well as 5D downstream of the exit of the hole for the distance of one pitch on the vane surface are shown in Figure 2.24. As can be seen from Figure 2.24, the peaks of the plots of adiabatic effectiveness at the exit of the hole is narrower than the peaks of the plots at $M = 1$ and $M = 2$. This can be attributed to the increased jet momentum at $M = 3$. At the exit of the 7-7-7 shaped hole inclined at 60° compound angle, the value of adiabatic effectiveness across the pitch is fairly uniform which is due to the slot-like nature of the hole. At $M = 3$ as with at $M = 1$ and $M = 2$, the addition of compound angle increases the lateral spread of coolant from both cylindrical and shaped holes at the exit of the hole. It can be seen that the 7-7-7 shaped hole performs better than the cylindrical hole 5D downstream of the hole for the different compound angle orientations. Also, it can be seen that the addition of compound angle to the shaped hole generally increases adiabatic effectiveness 5D downstream of the hole exit. Again, as noted earlier, this is due to the reduced jet momentum as a result of hole shaping which allows the coolant to attach more to the surface of the vane. Also, with the addition of compound angle, the jets attach more to the surface of the vane and the lateral spread of the coolant is increased. These result in the improved performance of the 7-7-7 shaped hole over the cylindrical hole 5D downstream of the hole exit. It should also be noted that the addition of compound angle improves the film cooling performance of the cylindrical hole 5D downstream of the hole exit. From Figure 2.24, it can be seen that the 7-7-7 shaped hole inclined at 60° compound angle had the best film cooling performance both at the hole exit and 5D downstream of the hole exit when compared to all the other configurations.
Figure 2.24: Local adiabatic effectiveness at the hole exit and 5D downstream of the hole exit for one pitch at $M = 3$.

2.5. CONCLUSIONS

Steady-state numerical simulations have been performed using realizable $k$–$\varepsilon$ turbulence model in ANSYS Fluent to study how the compound angle of a shaped hole influences adiabatic film cooling effectiveness on the pressure side of a scaled-up GE-E3 vane in a subsonic linear cascade. The shaped hole considered in this study is the 7-7-7 shaped hole inclined at 0° (axial hole), 30° and 60° compound angles. There were three rows of film cooling holes each having 16 holes on the pressure side of the vane. The results of the simulations of the shaped holes were compared to results of the simulations of cylindrical holes for the same configurations. For each hole type and compound angle orientation, three blowing ratios were examined – they are 1, 2 and 3. The results of the numerical simulations were compared with the experimental results presented in Chapter 1.

Conclusions drawn from the present study are as follows:

- The addition of compound angle to the 7-7-7 shaped hole significantly improves film cooling effectiveness at the exit of the hole and 5D downstream of the hole exit when compared to the cylindrical holes. This is because shaped holes reduce the momentum of
coolant at the exit of the hole and the addition of compound angle to the shaped hole causes the coolant to attach more to the surface of the vane and increases the lateral spread of coolant on the vane surface.

- For both the 7-7-7 shaped hole and the cylindrical hole, the lateral spread of coolant at the hole exit and 5D downstream of the hole exit generally increases as the compound angle increases.

- Comparing the adiabatic effectiveness contours and laterally averaged adiabatic effectiveness plots for the experiments and numerical simulations, it can be seen that the numerical simulations did not adequately capture jet lift-off from the vane surface. Jet lift-off can be observed from the sudden decline in adiabatic effectiveness just downstream of the film cooling hole row in the laterally-averaged adiabatic effectiveness plots. While the experimental results generally showed sharp decline in adiabatic effectiveness just downstream of the hole rows, the numerical results did not.

- The numerical simulations did not capture jet reattachment to the surface of the vane after jet lift-off from the surface of the vane while the experimental results show jet reattachment to the surface of the vane.

- While the experimental and numerical results generally follow the same trends, the differences between the experimental and numerical results are quite significant. These differences are obvious from the adiabatic effectiveness contours, the laterally-averaged adiabatic effectiveness plots and the area-averaged adiabatic effectiveness plots. Therefore, steady-state CFD simulations using realizable $k-\varepsilon$ turbulence model cannot be solely relied upon as a design tool for gas turbine vane film cooling.
NOMENCLATURE

A  Cross-sectional area of cooling hole (m$^2$)

ABS  Acrylonitrile Butadiene Styrene

AR  Area ratio, $\frac{A_{exit}}{A_{inlet}}$

CFD  Computational fluid dynamics

$C_x$  Axial chord of the vane (m)

Cyl  Cylindrical

D  Diameter of film cooling hole metering section (m)

Deg  Degree (°)

Expt  Experimental

H  Vane span (m)

IR  Infrared

L  Length of film cooling hole (m)

$M$  Blowing ratio (-)

NGV  Nozzle guide vane

Num  Numerical

P  Film cooling hole and vane pitch (m)

R  Radius of the interior edges of diffused outlet (m)

RANS  Reynolds-averaged Navier-Stokes
\( Re \)  
Reynolds number (-)

ROI  
Region of interest

Shp  
Shaped

t  
Hole breakout width (m)

T  
Temperature (K)

U  
Velocity (m/s)

X  
Streamwise direction

Y  
Normal direction on vane surface

Z  
Spanwise direction

Greek Symbols

\( \alpha \)  
Film cooling hole inclination angle (°)

\( \beta \)  
Expansion angle of shaped hole (°)

\( \eta \)  
Local adiabatic film cooling effectiveness (-)

\( \rho \)  
Density (kg/m\(^3\))

Subscripts, Accents

adia  
Adiabatic

aw  
Adiabatic wall
c Coolant or secondary fluid
fwd Forward
lat Lateral
m Mainstream, Metering section
- Lateral average
= Area average
REFERENCES


CHAPTER 3

THE EFFECT OF COMPOUND ANGLE OF A SHAPED HOLE AND FREESTREAM TURBULENCE ON GAS TURBINE VANE PRESSURE SIDE FIM COOLING

3.1. ABSTRACT

Film cooling of gas turbine airfoils is very critical for efficient operation of gas turbine engines. The effect of compound angle and freestream turbulence on the adiabatic effectiveness of the pressure side of a first-stage scaled-up GE-E3 vane was experimentally investigated in this study using infrared thermography technique. The effect of multiple rows of holes on the pressure side of the vane was investigated as well with each row of holes having P/D = 3. Steady-state experiments were performed in a subsonic gas turbine linear vane cascade. The experiments were conducted at density ratio of approximately 1, for a range of blowing ratios of 1.5 to 2.5, at three different compound angles of 0°, 30° and 60°. The turbulence intensities investigated are 8.9% and 18.2% with turbulence length scales of 0.8D and 3.8D respectively. All experiments were conducted at Reynolds number based inlet freestream velocity and the axial chord of the vane of 175,340. The film cooling performance of the 7-7-7 shaped hole was compared to that of the cylindrical hole for the same test conditions. The higher freestream turbulence mostly resulted in lower film cooling performance for the cylindrical holes and 7-7-7 shaped holes due to greater jet lift-off from the surface of the vane and lower reattachment of coolant jets to the vane surface.
3.2. INTRODUCTION

Modern gas turbine engines are operated at very high rotor inlet temperatures (RITs) so as to increase the efficiency of the engines. These temperatures far exceed the permissible limits of the materials used in making components of the engine [3.1]. Film cooling, in addition to other technologies, is used to ensure that gas turbine engines can operate efficiently at such high temperatures. Film cooling in gas turbine applications has been studied extensively for over four decades. Reviews of gas turbine film cooling are available in the literature [3.1–3.5]. Many film cooling hole shapes have been studied both experimentally and numerically. Some examples of film cooling holes studied are fan shaped [3.6], laidback fan-shaped [3.7], converging-slot hole or console [3.8,3.9], sweeping jet [3.10], antivortex [3.11], and cylindrical holes that are embedded in transverse trenches [3.12,3.13]. Bunker [3.3] noted that shaped film cooling holes are the industry standard. Laid-back fan shaped holes are the most common in industry and the literature [3.7]. Many factors affect film cooling of gas turbine airfoils [3.1,3.4] of which freestream turbulence is one of them. Freestream turbulence in the areas where film cooling is used in a gas turbine engine is about 10 to 20% [3.14].

Ames [3.15,3.16] in a two-part paper investigated the effect of turbulence on vane heat transfer and adiabatic effectiveness using a four-vane subsonic cascade. Low and high turbulence levels having turbulence intensities of 1% and 12% respectively were studied. In the first part, the effect of velocity ratio and turbulence on heat transfer was studied for one row and two staggered rows of holes on the pressure and suction surfaces. Also, for the two turbulence levels, the effect of heat transfer downstream of showerhead injection on both the pressure and suction surfaces were studied. In the second part of the paper, the corresponding measurements of adiabatic effectiveness were obtained. It was reported that Stanton number augmentation ratios were greater
in the laminar regions as compared to the turbulent regions of the flow. Also, it was reported that
turbulence had moderate effect on film cooling of the suction surface and significant effect on film
cooling of the pressure surface, especially for lower velocity ratios.

Mayhew et al. [3.14] studied the effect of freestream turbulence on the adiabatic effectiveness
of a flat plate using liquid crystal thermography technique. They studied a row of three straight
cylindrical holes, with P/D = 3 and density ratio near unity for three blowing ratios of 0.5, 1 and
1.5. The two turbulence intensities they studied were 0.1% and 10%. They reported that at low
blowing ratios, high freestream turbulence reduces area-averaged adiabatic effectiveness due to
greater mixing of the cooling air with the mainstream air. At high blowing ratios, on the other
hand, high freestream turbulence increases area-averaged adiabatic effectiveness as a result of
greater mixing between the cooling air and mainstream air which entrains some coolant within the
mainstream air and mixes it with the air close to the surface. Mayhew et al. [3.17] studied the effect
of freestream turbulence on the film-cooling performance of a flat plate with one hole having
injection and compound angles of 30° and 45° respectively. They used thermochromic liquid
crystal thermography technique. They noted that high freestream turbulence reduces adiabatic
effectiveness at low blowing ratio, whereas it increases adiabatic effectiveness at high blowing
ratio. Schmidt and Bogard [3.18] also reported the same findings in their study of how freestream
turbulence and surface roughness affects film cooling performance. In addition, their study showed
a greater range of optimum momentum flux ratios for high freestream turbulence combined with
surface roughness, which is representative of the actual conditions of turbine operation, when
compared to range obtained from past low freestream turbulence investigations.

Jenkins et al. [3.19] experimentally investigated the joint effects of film cooling and high
turbulence on how a simulated hot streak disperses while passing over a scaled-up nozzle guide
vane (NGV). Mainstream turbulence and film cooling significantly affected the hot streak attenuation. Cutbirth and Bogard [3.20,3.21] in their two-part paper studied the effect of showerhead blowing on the film cooling performance on the pressure side of a turbine vane. One row of film cooling holes inclined at 45° compound angle was used on the pressure side of the vane. Results were presented for low mainstream turbulence in the first part of the paper while they were presented for high mainstream turbulence in the second part. The low and high mainstream turbulence intensities were 0.5% and 20% respectively. They obtained measurements of adiabatic effectiveness as well as the flow and thermal fields above the vane surface. They carried out flow visualization using titanium dioxide seed particles. In the first part of the study, they reported lower peak values of adiabatic effectiveness with combined showerhead and pressure side blowing downstream of the pressure side row of holes than values obtained with a superposition prediction. From measurements of velocity field, this discrepancy was mainly due to the high turbulence caused by showerhead injection which possibly causes significant dispersion of the jets of coolant on the pressure side of the vane. They reported that due to the large scale of the high mainstream turbulence of 7D, in the second part of the study, the turbulence leads to lateral oscillation of the coolant jets instead of jet break-up. A wider distribution of adiabatic effectiveness is the result of the lateral fluctuation.

Saumweber et al. [3.22] noted that previous investigations on freestream turbulence effects on film cooling did not consider the shape of the hole exit. They experimentally studied the effect of hole shape and freestream turbulence on film cooling performance using a flat plate test facility. They studied a row of three scaled-up cylindrical, fan-shaped and laidback fan-shaped holes with L/D = 6 and P/D = 4 using infrared (IR) thermography. Their tests were conducted at turbulence intensities which range from 3.5 to 11%, integral length scale at a particular turbulence intensity
up to 3.5D, and for blowing ratios which range from 0.5 to 1.5 for the cylindrical holes and from 0.5 to 2.5 for the shaped holes. They reported that for cylindrical holes, increasing the freestream turbulence intensity reduces adiabatic film cooling effectiveness at low to moderate blowing ratios while effectiveness increased at high blowing ratios. The increase in effectiveness was attributed to better lateral spreading of the coolant in addition to a reduced tendency of the coolant detaching from the surface. They noted that increasing the turbulence level for the shaped holes resulted in reduced film cooling performance at all blowing ratios. Hayes et al. [3.23] investigated the effect of freestream turbulence and blowing ratio on the film cooling performance of the anti-vortex film cooling hole in a flat plate facility using infrared thermography technique. They reported that at high blowing ratios, high freestream turbulence improves span-averaged and centerline film cooling effectiveness. They also reported that further increase in turbulence intensity at blowing ratios of 1.5 and 2 resulted in lower cooling effectiveness.

Schroeder and Thole [3.7] introduced the laid-back fan shaped hole, called the 7-7-7 shaped hole used in the present study. They introduced the 7-7-7 shaped hole as a baseline shaped hole for comparison with new hole geometries since shaped holes are known to perform better than cylindrical holes. They chose the laidback fanshaped hole as the baseline shaped hole geometry because it is most commonly used in industry and the literature. The 7-7-7 shaped hole has conservative expansion angles of 7° to prevent jet separation inside the hole. They reported that increasing the freestream turbulence to 5.4% at $M = 0.5$ and 1 resulted in up to 10% reduction in area-averaged adiabatic effectiveness whereas there was no reduction in area-averaged adiabatic effectiveness at higher blowing ratios. Schroeder and Thole [3.24] obtained flowfield and adiabatic effectiveness measurements of the 7-7-7 shaped hole in a flat plate facility. They investigated the effect of freestream turbulence intensity up to 13.2%. They reported that elevated freestream
turbulence increased fluctuations of velocity around the coolant jet thus resulting in greater spreading of coolant laterally. Haydt and Lynch [3.25] studied the effect of compound angle of the 7-7-7 shaped hole using a flat plate both experimentally and numerically. Their experimental results indicated greater lateral coolant spread with increasing compound angle. Haydt et al. [3.26] carried out experimental and numerical investigations to determine how a meter-diffuser offset affects the adiabatic effectiveness using the 7-7-7 shaped hole in a flat plate test facility. McClintic et al. studied how internal cross flow velocity affects the film cooling performance of a row of axial [3.27] and compound angle [3.28] 7-7-7 shaped hole in a flat plate facility using infrared thermography technique. Hossain et al. [3.29–3.31] conducted studies in which the film cooling performance of a row of sweeping jet film cooling holes was compared to that of a row the 7-7-7 shaped hole on the suction surface of a NGV using infrared thermography technique.

Many of the studies of the 7-7-7 shaped holes have been conducted using flat plate test facilities. In this study the 7-7-7 shaped hole is experimentally tested on the pressure side of a gas turbine first-stage NGV. This is the first study to determine how compound angle of the 7-7-7 shaped hole and freestream turbulence affect the adiabatic effectiveness on the pressure side of a gas turbine vane to the best of the author’s knowledge. Also, this study is unique because the effect of multiple rows of film cooling holes was considered as well. The film cooling performance the 7-7-7 shaped holes is compared with that for cylindrical holes under identical experimental conditions.
3.3. EXPERIMENTAL SETUP

All experiments were conducted in the subsonic gas turbine vane cascade which is located in the Thermal Energy Research and Management Laboratory (ThERMaL) at North Carolina State University. The schematic diagram of the rig which is an open-loop linear cascade wind tunnel is presented in Figure 3.1. A pressure blower having a maximum flow rate of 3000 CFM supplies air to the tunnel. A 20-HP motor with a maximum speed of 3450 R.P.M. powers the blower. The air flow rate into the wind tunnel is controlled by the PowerFlex 70 Adjustable Frequency AC Drive.

![Figure 3.1: The Subsonic Gas Turbine Vane Cascade.](image)

Air passes through a diverging section, a straight section and then enters a converging section. Thereafter, the air goes through a transition duct before entering the test section. The test section is a linear cascade with three vanes and two passages. The vanes are scaled-up GE–E3 first-stage vanes made from acrylonitrile butadiene styrene (ABS), a low-thermal conductivity material. Two tailboards were used to achieve periodic flow around the central vane. Mainstream air at about 25
–29°C is supplied via the blower and the secondary fluid, which is heated air, is heated to about 45 – 66°C using an inline heater. The laboratory compressor supplies the secondary fluid and it is passed through two buffer tanks before being heated. The secondary fluid enters a plenum from which it then enters into rotameters which feed each row of film cooling holes. The flow rate of the secondary fluid required to achieve a given blowing ratio is controlled by the rotameters.

The middle vane, which is modular, is changed so as to study the various configurations of film cooling holes. The vane parameters are presented in Table 3.1. The metering section of all film cooling holes has diameter (D) of 1.5 mm. The film cooling hole geometries employed in this study are the 7-7-7 shaped hole introduced by Schroeder and Thole [3.7] and the cylindrical hole. The 7-7-7 shaped hole design is shown in Figure 3.2 while the geometric parameters of the 7-7-7 shaped hole are presented in Table 3.2. The angle of inclination of the film cooling holes to the vane surface is 30° and the L/D of the holes is 6. Three compound angles were studied. The compound angles were 0° (axial hole), 30° and 60°. The L/D of 6 for the 7-7-7 shaped holes at 30° and 60° compound angles orientations could not be obtained because of the curvature of the pressure surface. However, the diverging section of the shaped hole for these compound angles, had the same profile as the 7-7-7 hole and the metering section had the same length as the axial 7-7-7 hole. There were three film cooling hole rows on the pressure side of the vane located at normalized axial positions, X/Cx of 0.15, 0.3 and 0.65 with 16 holes in each row. The holes occupied the middle 60% of the vane span. The schematic diagrams of the vanes studied are presented in Figure 3.3. The region of interest (ROI) encompasses X/Cx from 0.1 to 0.68 and Z/H from 0.36 to 0.56. A quartz window in the transition duct enabled the region of interest to be viewed. Figure 3.4 shows the schematic of the ROI.
Table 3.1: GE–E3 vane parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
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<tbody>
<tr>
<td>Number of Vanes</td>
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</tr>
<tr>
<td>Hole inlet diameter</td>
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<tr>
<td>Hole Injection Angle</td>
<td>30°</td>
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<tr>
<td>Compound Angle</td>
<td>0°, 30° and 60°</td>
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<tr>
<td>Hole Spacing (P/D)</td>
<td>3</td>
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<tr>
<td>Hole length (L/D)</td>
<td>6</td>
</tr>
<tr>
<td>Number of Cooling Hole Rows</td>
<td>3</td>
</tr>
<tr>
<td>Number of Cooling Holes in each Row</td>
<td>16</td>
</tr>
<tr>
<td>Normalized Axial Position of Cooling Hole Rows (X/C&lt;sub&gt;x&lt;/sub&gt;)</td>
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</tr>
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<tr>
<td>Axial Chord (C&lt;sub&gt;x&lt;/sub&gt;)</td>
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<tr>
<td>Vane Pitch (P)</td>
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</table>
Figure 3.2: Design of the 7-7-7 shaped hole [3.7].
Table 3.2: The 7-7-7 shaped hole geometric parameters [3.7].

<table>
<thead>
<tr>
<th>Parameter</th>
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<tr>
<td>$L_{lat}/D$, $L_{fwd}/D$</td>
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<tr>
<td>$L/D$</td>
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<td>Laidback Angle, $\beta_{fwd}$</td>
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<tr>
<td>Lateral Angle, $\beta_{lat}$</td>
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<tr>
<td>Sharpness of Inlet at Breakout Edges</td>
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<tr>
<td>Rounding of Four Edges Inside Diffuser</td>
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<tr>
<td>Diffuser, $R/D$</td>
<td>0.5</td>
</tr>
</tbody>
</table>
Figure 3.3: Schematic diagrams of the vanes investigated (a) axial cylindrical hole, (b) cylindrical hole with 30° compound angle, (c) cylindrical hole with 60° compound angle, (d) axial 7-7-7 shaped hole, (e) 7-7-7 shaped hole with 30° compound angle, (f) 7-7-7 shaped hole with 60° compound angle.

Figure 3.4: The region of interest (ROI).
3.4. EXPERIMENTAL PROCEDURE AND DATA REDUCTION

Pressure taps and Kiel probes connected to the Scanivalve ZOC 33 pressure scanner were used to measure the static and total pressures respectively. Measurements of the static and total pressures at inlet of the test section, as well as freestream turbulence were obtained at 0.25C\textsubscript{x} upstream of the vane’s leading edge. The design of the vane cascade is such that 0.25C\textsubscript{x} upstream of the leading edge of the vane the turbulence intensity is 18.2\% with length scale of 3.8D. Wire meshes were installed at different locations within the rig resulting in breakdown the vortical structures so as to obtain turbulence intensity of 8.9\% with length scale of 0.8D at the same upstream location of 0.25 C\textsubscript{x} for measurements at the lower turbulence intensity level. At the exit of the vane, measurements were obtained at 0.2C\textsubscript{x} downstream of the vane’s trailing edge. Measurements of the static pressure at the inlet of the test section confirmed that the flow at the inlet was uniform. Again, static pressure measurements were obtained at 50\% of the span of the vane on the pressure side. The subsonic gas turbine cascade in this study was benchmarked with another GE-E3 vane cascade. The vane surface Mach number distribution on the pressure surface of the vane in the present study was compared with that in the study of Ramesh et al. [3.32] and shown in Figure 3.5. It can be seen that there is very good agreement in the data obtained from both studies.
Figure 3.5: Comparison of the vane pressure surface Mach number distribution.

The FLIR A6750sc IR camera was used to measure vane’s surface temperature. A quartz window placed in the transition duct permitted measurement of the vane’s pressure side temperature using the IR camera. Calibration was required since the vane’s pressure side temperature measurement was obtained through the quartz window using the IR camera. During calibration, three thermocouples were used. Each thermocouple was placed close to the exit of a film cooling hole in each of the three rows of film cooling holes. The temperatures measured by the thermocouples and the temperatures of the pixels of the IR camera’s image which correspond to the locations of the thermocouples were synchronized, so as to obtain the relationship between true surface temperature of the vane and that determined using the IR camera. Calibration was performed for the whole temperature range encountered in all the experiments conducted. The vanes which were not black in color were painted with Rust-Oleum® black paint so as to obtain
high emissivity. Calibration was performed for all the film cooling hole configurations studied. Figure 3.6 shows a sample calibration curve.

![Calibration Curve](image)

Figure 3.6: Typical IR camera calibration curve.

For each configuration of the cooling holes, experiments were performed at blowing ratios of 1.5, 2 and 2.5 at density ratio of approximately 1. All experiments were performed at steady-state condition. The mainstream inlet velocity was constant in all the experiments at 12.5 m/s which gives Reynolds number based on mainstream inlet velocity and the vane’s axial chord of 175,340. The secondary fluid (coolant) flow rate to the individual film cooling hole rows was controlled by Brooks® Instrument model 2530 flowmeters (rotameters). A given blowing ratio was obtained by controlling the coolant flow rate. Thermocouples were used to monitor the mainstream air and secondary air (coolant) temperatures during the experiments. For all experiments, when the blowing ratio was set by setting the coolant flow rate, the experiments were allowed to run for 30 minutes so that steady-state condition is reached. Steady-state condition was considered to be attained when the variation in temperature of the individual pixels of the IR camera image was
lower than 1% in the span of one minute. Data was acquired using a LABVIEW program, and post-processed using MATLAB. The experimental conditions are summarized in Table 3.3.

Table 3.3: Experimental conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reynolds Number</td>
<td>175,340</td>
</tr>
<tr>
<td>Mainstream Temperature</td>
<td>~ 298 - 302 K</td>
</tr>
<tr>
<td>Secondary fluid Temperature</td>
<td>~ 318 - 339 K</td>
</tr>
<tr>
<td>Mainstream Inlet Velocity</td>
<td>12.5 m/s</td>
</tr>
<tr>
<td>Freestream Turbulence Intensity</td>
<td>8.9%, 18.2%</td>
</tr>
<tr>
<td>Density Ratio</td>
<td>1 ±0.1</td>
</tr>
<tr>
<td>Blowing Ratio (M)</td>
<td>1.5 to 2.5</td>
</tr>
</tbody>
</table>

3.4.1. Data Reduction

The definition of the adiabatic film cooling effectiveness, $\eta$, is presented below:

$$\eta = \frac{T_m - T_{aw}}{T_m - T_c}$$  \hspace{2cm} (3.1)

where $T_m$ and $T_c$ are the mainstream and secondary fluid (coolant) temperatures respectively, while $T_{aw}$ is the adiabatic wall temperature. T-type thermocouples were used to obtain the mainstream and coolant temperatures and the IR camera was used to obtain $T_{aw}$.

The Reynolds number of all experiments was 175,340 and is defined thus:

$$Re = \frac{U_m C_x}{v}$$  \hspace{2cm} (3.2)
where $U_m$ is the mainstream inlet velocity and $C_x$ is axial chord of the vane, and $\nu$ is the kinematic viscosity of air at the mainstream air temperature.

The blowing ratio, $M$, is defined as follows:

$$ M = \frac{\rho_c U_c}{\rho_m U_m} $$  \hspace{1cm} (3.3)

where $\rho$ is the fluid density and $U$ is the fluid velocity. The subscripts c and m indicate the secondary air and mainstream air respectively.

### 3.4.2. Uncertainty

The Scanivalve ZOC 33 pressure scanner used has accuracy of $\pm 0.12\%$ full scale. The rotameters used to meter the secondary fluid so as to determine the blowing ratio have accuracy of $\pm 3\%$ full scale. The experimental uncertainty was obtained using the sequential perturbation method of Moffat [33]. In this method, the deviation in the adiabatic effectiveness using the higher value and the lower value of a variable was calculated while keeping the other variables constant. The average of the absolute values of these deviations is the uncertainty due to that variable. The root sum square of the individual uncertainties is the total uncertainty. The error of the thermocouples used for calibration is $\pm 1.1^\circ C$. When calibration error is accounted for, the maximum error in the average temperature of the vane surface using the IR camera was $\pm 4.5 \text{ K}$. For the experiments, the average uncertainty in adiabatic effectiveness was $9\%$. The thermocouples are the source of uncertainty in the adiabatic effectiveness calculation.
3.4.3. Conduction Correction

The vanes were produced using ABS, a material having low thermal conductivity. However, conduction within the vane has a significant effect on adiabatic effectiveness. Computational fluid dynamics (CFD) simulations were performed using ANSYS Fluent so as to account for conduction within the vane material. Two simulations at each blowing ratio for the cylindrical hole as well as the 7-7-7 shaped hole were carried out. The simulations were typical of the vane geometry and linear cascade test section used in this study. While one simulation was a conjugate heat transfer CFD model with the vane having the properties of ABS, the other was conducted using a vane with an adiabatic surface. The temperature difference between the two cases was used to determine an effectiveness correction factor.

The definition of the effectiveness correction factor is presented in equation (3.4)

\[ \eta_{cf} = \frac{T_{ABS} - T_{adia}}{T_c - T_m} \]  

(3.4)

The temperature difference between both cases is the numerator in equation (3.4) while the temperature difference between the secondary air and mainstream air is the denominator. The areas requiring the greatest correction are those just upstream and just downstream of the rows of holes, as well as areas between adjacent holes in a row of holes. The areas downstream of the second row of holes not near to the third row of rows needed the least correction. Reduced effectiveness values were obtained when the correction factor is applied to the data. The raw experimental data contour is presented in Figure 3.7(a) and the corrected data is shown in Figure 3.7(b) for the 7-7-7 shaped hole inclined at 30° compound angle at \( M = 2 \) for \( Tu = 18.2\% \). The laterally-averaged adiabatic effectiveness plots for the same case is presented in Figure 3.8. Figure 3.8 shows the significant effect of conduction.
Figure 3.7: Conduction correction for 7-7-7 shaped hole at 30° compound angle at $M = 2$, $Tu = 18.2\%$ (a) Pre-correction contour (b) Post-correction contour.

Figure 3.8: Effect of conduction on laterally-averaged adiabatic effectiveness for 7-7-7 shaped hole at 30° compound angle inclination at $M = 2$ for $Tu = 18.2\%$. 
3.5, RESULTS AND DISCUSSION

3.5.1. Cylindrical Holes

The contours of adiabatic effectiveness for axial cylindrical holes for Tu = 8.9 and 18.2% are shown in Figure 3.9. From Figure 3.9, it can be seen that the adiabatic effectiveness is greater at $M = 1.5$ for Tu = 18.2% when compared to at $M = 1.5$ for Tu = 8.9%. The higher freestream turbulence enhances adiabatic effectiveness at $M = 1.5$ due to mixing between the coolant and the mainstream which allows coolant entrained within the mainstream to return to the surface [3.34]. Evidence of increased mixing between the coolant and the mainstream due to higher freestream turbulence can clearly be seen from the contour for $M = 1.5$ resulting in higher adiabatic effectiveness in the ROI. For the $M = 2$ and 2.5 cases, the adiabatic effectiveness is greater at Tu = 8.9% than at Tu = 18.2%. For these blowing ratios, the higher freestream turbulence causes greater lift-off of coolant jets from the surface of the vane thereby resulting in the reduced film cooling performance. For Tu = 8.9%, the film cooling performance downstream of the second row of holes increases as the blowing ratio increases and the interaction between coolant from adjacent holes in a given row of holes increases with increase in blowing ratio. Generally, significant jet lift-off from the vane surface occurs for all blowing ratios and both levels of turbulence intensity, as seen from the contours in Figure 3.9.
Figure 3.9: Adiabatic effectiveness contours for axial cylindrical holes for (a) $M = 1.5$, (b) $M = 2$, (c) $M = 2.5$.

The laterally-averaged adiabatic effectiveness plots for axial cylindrical holes for $Tu = 8.9\%$ and $18.2\%$ are shown in Figure 3.10. As is shown in Figure 3.10, the laterally-averaged adiabatic effectiveness for $Tu = 18.2\%$ is greater than for $Tu = 8.9\%$ at $M = 1.5$. The higher turbulence intensity level has higher peak laterally-averaged adiabatic effectiveness at the locations of the rows of holes at $M = 1.5$. Again, this is due to enhanced mixing of the coolant and the mainstream as a result of the higher freestream turbulence level. This greater mixing leads to entrainment of coolant within the mainstream, then some of the coolant was pushed back to the surface of the vane. For both turbulence intensity levels, it can be seen that significant jet lift-off occurs immediately downstream of the rows of film cooling holes. This can be seen from the rapid
decrease in laterally-averaged adiabatic effectiveness levels just downstream of the rows of holes. Jet reattachment to the vane surface can be observed upstream of the second and third rows of holes for all blowing ratios and both levels of turbulence intensity, as seen by the positive slope of the curves upstream of the second and third rows of holes. Generally for \( \text{Tu} = 8.9\% \), the laterally-averaged adiabatic effectiveness increases as blowing ratio increases. For \( \text{Tu} = 18.2\% \), the laterally-averaged effectiveness values are nearly the same at \( M = 2 \) and 2.5. This could indicate that for the higher freestream turbulence level, increasing the blowing ratio causes no changes to the film cooling performance of the axial cylindrical holes. The laterally-averaged effectiveness values at \( M = 2 \) and 2.5 are lower for \( \text{Tu} = 18.2\% \) than for \( \text{Tu} = 8.9\% \) and for these blowing ratios, more coolant reattaches to the vane surface upstream of the second and third row of holes for \( \text{Tu} = 8.9\% \) when compared to \( \text{Tu} = 18.2\% \). Thus it can be seen that for the higher freestream turbulence level, greater jet lift-off from the vane surface occurs as blowing ratio increases. As noted in the contours in Figure 3.9 for \( \text{Tu} = 8.9\% \), it can be seen in the plots in Figure 3.10 that the interaction of coolant from adjacent holes in a given row of holes increases as the blowing ratio increases for that turbulence level. The best performance downstream of the first and second rows of cooling holes in terms of lateral coolant spread was obtained at \( M = 1.5 \) for \( \text{Tu} = 18.2\% \). It can, therefore, be seen that the higher freestream turbulence levels leads to reduced film cooling performance as the blowing ratio increases.
The contours of adiabatic effectiveness for cylindrical holes inclined at 30° compound angle for Tu = 8.9% and 18.2% are presented in Figure 3.11. Generally, as can be seen from Figure 3.11, the higher turbulence intensity level produces better film cooling performance at $M = 2$ and 2.5 while the lower turbulence intensity level produces better film cooling performance at $M = 1.5$. Jet lift-off from the vane surface can be observed at all blowing ratios and for both freestream turbulence intensity levels. For Tu = 8.9%, the $M = 1.5$ case has the best film cooling performance followed by the $M = 2.5$ case and then the $M = 2$ case. As is seen from Figure 3.11, the adiabatic effectiveness levels are generally low for both levels of turbulence intensity which is attributable to significant lift-off of coolant jets from the vane surface.
Figure 3.11: Adiabatic effectiveness contours for cylindrical holes inclined at 30° compound angle for (a) $M = 1.5$, (b) $M = 2$, (c) $M = 2.5$.

The plots of laterally-averaged adiabatic effectiveness for cylindrical holes inclined at 30° compound angle are shown in Figure 3.12. From Figure 3.12, it can be seen that at $M = 1.5$, the laterally-averaged adiabatic effectiveness for $Tu = 18.2\%$ is lower than that for $Tu = 8.9\%$. For $M = 2$ and 2.5 at $Tu = 18.2\%$, the laterally-averaged adiabatic effectiveness values are higher than for the corresponding blowing ratios at $Tu = 8.9\%$. This increase in laterally-averaged effectiveness values at the higher freestream turbulence intensity level for $M = 2$ and 2.5 could be due to greater mixing between the coolant and the mainstream, some of the coolant is entrained within the
mainstream which then returns back to the surface of the vane thereby increasing adiabatic effectiveness. The trends of the curves for both turbulence intensity levels are quite similar as the $M = 2$ case had the lowest film cooling performance for both levels. The $M = 2$ case at $Tu = 8.9\%$ had the lowest laterally-averaged adiabatic effectiveness values. For this case, at $X/C_x$ of about 0.2 to 0.22 and 0.42 to 0.58, the coolant is completely blown away from the vane surface as it offers no protection to the surface. Also, for $M = 2.5$ at $Tu = 8.9\%$, from about $X/C_x = 0.51$ to 0.56 the coolant offers no protection to the vane surface. At $Tu = 8.9\%$, the $M = 1.5$ case had the best lateral distribution of coolant on the vane surface of all the blowing ratios for that turbulence intensity level. Significant lift-off of coolant jets occurs just downstream of the hole exits for both turbulence intensity levels and all blowing ratios. This can be seen from the sudden decrease in laterally-averaged adiabatic effectiveness values just downstream of the hole exits. Just downstream of the hole exits at $Tu = 8.9\%$, the jets attach more to the surface at $M = 1.5$ than at $M = 2$ and 2.5. Considering $Tu = 18.2\%$, the $M = 2.5$ case had the best lateral spread of coolant of all the other two blowing ratios. For both turbulence intensity levels and at all blowing ratios, jet reattachment to the vane surface downstream of the first and second rows of holes and upstream of the second and third rows of holes is not very pronounced. This strongly suggest that the coolant is mainly blown away from the ROI thereby leading to the lower adiabatic effectiveness levels shown in the contours in Figure 3.11.
Figure 3.12: Laterally-averaged adiabatic effectiveness for cylindrical holes inclined at 30° compound angle.

The contours of adiabatic effectiveness for cylindrical holes inclined at 60° compound angle are presented in Figure 3.13. Generally, as can be seen from Figure 3.13, the film cooling performance at $\text{Tu} = 18.2\%$ is lower than that at $\text{Tu} = 8.9\%$ for all blowing ratios. The interaction of coolant emanating from the first and second rows of holes is greatest for the lower turbulence intensity level at $M = 2$ of all the cases tested for this configuration. At $\text{Tu} = 8.9\%$, as the blowing ratio increases the interaction between the coolant from adjacent holes in a given row of holes increases with the exception being the $M = 2.5$ case. Clearly, as seen from Figure 3.13, high freestream turbulence significantly reduces the film cooling performance for cylindrical holes inclined at 60° compound angle. Jet lift-off from the surface of the vane can be observed for both levels of turbulence intensity and for all blowing ratios and it is more significant at $\text{Tu} = 18.2\%$. 
The $M = 2$ case at $Tu = 8.9\%$ has the best film cooling performance of all the cases tested for cylindrical holes inclined at $60^\circ$ compound angle.

Figure 3.13: Adiabatic effectiveness contours for cylindrical holes inclined at $60^\circ$ compound angle for (a) $M = 1.5$, (b) $M = 2$, (c) $M = 2.5$.

The plots of laterally-averaged adiabatic effectiveness for cylindrical holes inclined at $60^\circ$ compound angle are shown in Figure 3.14. From Figure 3.14, it can be seen that the higher turbulence intensity level has much lower film cooling performance at all blowing ratios when compared to the lower turbulence intensity level. For $Tu = 18.2\%$, the curves for $M = 1.5$ to 2.5 are nearly identical. This could indicate that for the $60^\circ$ compound angle orientation, for the higher level of turbulence intensity, film cooling performance is not a strong function of blowing ratio at $M = 1.5$ and above. Generally, for $Tu = 8.9\%$, the laterally-averaged adiabatic effectiveness
increases with increase in blowing ratio except at $M = 2.5$. Jet reattachment to the vane surface downstream of the first and second rows of holes and upstream of the second and third rows of holes can be clearly observed at $\text{Tu} = 8.9\%$. This can be seen by the slope of the curves just upstream of the second and third row of holes before the large increase in adiabatic effectiveness at the locations of the rows of holes. Jet reattachment to the vane surface is not very pronounced at $\text{Tu} = 18.2\%$ as it is at $\text{Tu} = 8.9\%$. Also, coolant jet lift-off is greater at $\text{Tu} = 18.2\%$ due to the steeper slopes of the curves just downstream of the rows of holes when compared to at $\text{Tu} = 8.9\%$. These effects of more coolant being lifted off the surface and less returning to the surface resulted in the higher turbulence intensity level having lower film cooling performance when compared to the lower turbulence intensity level. For $\text{Tu} = 18.2\%$, the coolant offers no protection to the surface at $M = 1.5$ and 2 at $X/C_x$ of about 0.43 to 0.59 and at $M = 2.5$ at $X/C_x$ of about 0.4 to 0.6. This further indicates that the coolant is blown away from the surface of the vane. In addition, for $\text{Tu} = 18.2\%$, at $M = 2.5$ the coolant offers no protection to the vane surface at $X/C_x$ of about 0.24 to 0.25.
Figure 3.14: Laterally-averaged adiabatic film cooling effectiveness for cylindrical holes inclined at 60° compound angle.

3.5.2. Area-Averaged Adiabatic Effectiveness for Cylindrical Holes

The plot of area-averaged adiabatic effectiveness for cylindrical holes for the two levels of turbulence intensity and for the blowing ratios and compound angle orientations is shown in Figure 3.15. From Figure 3.15, it can be seen that the highest area-averaged adiabatic effectiveness in the ROI of all the cases tested was obtained at $M = 2$ for the 60° compound angle inclination for $Tu = 8.9\%$. The higher turbulence intensity level resulted in lower area-averaged adiabatic effectiveness for the axial holes except at $M = 1.5$, while it resulted in higher area-averaged adiabatic effectiveness for the 30° compound angle orientation except at $M = 1.5$. The higher freestream turbulence level resulted in much lower area-averaged adiabatic effectiveness for the holes inclined at 60° compound angle as compared to the lower freestream turbulence level. The film
cooling performance of the holes inclined at 60° compound angle at \( \text{Tu} = 18.2\% \) is the lowest for all blowing ratios of all the cases for cylindrical holes tested except at \( M = 2 \).

Figure 3.15: Area-averaged adiabatic effectiveness for cylindrical holes for the different compound angles, blowing ratios and freestream turbulence intensities.

3.5.3. The 7-7-7 Shaped Holes

The contours of adiabatic effectiveness for axial 7-7-7 shaped holes for the two levels of turbulence intensity are presented in Figure 3.16. From Figure 3.16, the film cooling performance at \( \text{Tu} = 18.2\% \) is much lower than that at \( \text{Tu} = 8.9\% \). Generally, at \( \text{Tu} = 8.9\% \), adiabatic effectiveness increases as blowing ratio increases. As the blowing ratio increases at \( \text{Tu} = 8.9\% \), the interaction of coolant from adjacent holes in a given row of holes as well as the interaction of coolant from the first and second rows of holes increases. At \( \text{Tu} = 18.2\% \), the film cooling performance at \( M = 1.5 \) is higher than at \( M = 2 \) and 2.5. It seems that the axial orientation of the 7-7-7 shaped hole on the vane surface causes the coolant to be lifted off from the diffuser section of
the cooling hole as the blowing ratio increases for the higher freestream turbulence level thereby resulting in lower film cooling performance as compared to the lower turbulence level. The better performance for $M = 1.5$ at $Tu = 18.2\%$ compared to the other two blowing ratios for that turbulence level could be due to the reduced momentum of coolant jets at the hole exits which allows the coolant to attach more to the vane surface. The lower film cooling performance at $Tu = 18.2\%$ can be attributed to greater lift-off of coolant jets from the surface of the vane due to the higher turbulence level. Also, jet reattachment to the vane surface downstream of the first and second rows of holes at $Tu = 18.2\%$ is significantly lower than at $Tu = 8.9\%$.

Figure 3.16: Adiabatic effectiveness contours for axial 7-7-7 shaped holes for (a) $M = 1.5$, (b) $M = 2$, (c) $M = 2.5$. 
In Figure 3.17 the plots of laterally-averaged adiabatic effectiveness for axial 7-7-7 shaped holes at both turbulence intensity levels are shown. As can be seen from Figure 3.17, the laterally-averaged adiabatic effectiveness increases with increase in blowing ratio for \( \text{Tu} = 8.9\% \). The laterally-averaged adiabatic effectiveness of the axial holes at \( \text{Tu} = 18.2\% \) is substantially lower than at \( \text{Tu} = 8.9\% \) for all blowing ratios. Thus, it can be seen that the higher freestream turbulence level greatly reduced the film cooling performance. At \( \text{Tu} = 18.2\% \), the \( M = 1.5 \) case had the best laterally-averaged adiabatic effectiveness of the blowing ratios tested while the laterally-averaged adiabatic effectiveness at \( M = 2 \) and 2.5 was nearly identical. This could indicate that at the higher freestream turbulence level, the film cooling performance is unchanged at blowing ratios from 2 and above. The better film cooling performance at \( M = 1.5 \) for \( \text{Tu} = 18.2\% \) when compared to the other blowing ratios can be attributed to the reduced momentum of coolant at the exits of the holes.

As noted in the discussion of the contours in Figure 3.16, for the axial orientation of the 7-7-7 shaped hole, more coolant is blown away from the exit of the hole (the diffuser section of the hole) at \( \text{Tu} = 18.2\% \) for all blowing ratios when compared to at \( \text{Tu} = 8.9\% \). For both freestream turbulence intensity levels, jet-reattachment to the vane surface upstream of the second and third rows of holes can be observed. This can be observed by the positive slope of the curves upstream of the rows of holes before the sharp increase in laterally-averaged adiabatic effectiveness levels at the location of the rows of holes.
Figure 3.17: Laterally-averaged adiabatic effectiveness for axial 7-7-7 shaped holes.

The adiabatic effectiveness contours for the 7-7-7 shaped holes inclined at 30° compound angle for the two turbulence intensity levels are presented in Figure 3.18. As can be seen in Figure 3.18, the adiabatic effectiveness increases as blowing ratio increases for $\text{Tu} = 8.9\%$. At $\text{Tu} = 8.9\%$, the interaction of coolant from adjacent holes in a row of holes and the interaction of coolant from the first and second rows of holes increases with increase in blowing ratio. The film cooling performance for $\text{Tu} = 18.2\%$ is considerably lower than that for $\text{Tu} = 8.9\%$ at all blowing ratios. The lift-off of coolant jets from the surface of the vane is much more pronounced at $\text{Tu} = 18.2\%$ and coolant reattachment to the surface is much lower at $\text{Tu} = 18.2\%$ as compared to at $\text{Tu} = 8.9\%$. These factors result in the adiabatic effectiveness at $\text{Tu} = 18.2\%$ being much lower than that at $\text{Tu} = 8.9\%$. 
Figure 3.18: Adiabatic effectiveness contours for 7-7-7 shaped holes inclined at 30° compound angle for (a) M = 1.5, (b) M = 2, (c) M = 2.5.

In Figure 3.19 the plots of laterally-averaged adiabatic effectiveness for the 7-7-7 shaped holes inclined at 30° compound angle are presented. As can be seen in Figure 3.19, the laterally-averaged adiabatic effectiveness increases with increase in blowing ratio at Tu = 8.9%. At Tu = 8.9%, the curves for M = 2 and 2.5 are nearly identical except at the film cooling hole exit locations. This could indicate that at Tu = 8.9% the optimum blowing ratio is 2. The film cooling performance for Tu = 8.9% is much better than that for Tu = 18.2%. The laterally-averaged adiabatic effectiveness for Tu = 18.2% at all the blowing ratios was nearly identical. This could indicate that at the higher freestream turbulence level film cooling performance is independent of blowing ratio. As can be seen from Figure 3.19, more coolant is blown away from the vane surface at Tu = 18.2%.
in comparison to at $\text{Tu} = 8.9\%$. Coolant reattachment to the surface of the vane occurs for both turbulence intensity levels at all blowing ratios upstream of the second and third rows of holes. However, coolant reattachment to the vane surface is more pronounced at $\text{Tu} = 8.9\%$ than at $\text{Tu} = 18.2\%$.

Figure 3.19: Laterally-averaged adiabatic effectiveness for 7-7-7 shaped holes inclined at 30° compound angle.

The adiabatic effectiveness contours for the 7-7-7 shaped holes inclined at 60° compound angle are presented in Figure 3.20. From Figure 3.20, it can be observed that the film cooling performance at all blowing ratios is superior for $\text{Tu} = 8.9\%$ as compared to $\text{Tu} = 18.2\%$. This shows that higher freestream turbulence is detrimental to the film cooling performance of the 7-7-7 shaped holes inclined at 60° compound angle. Higher freestream turbulence sweeps away the coolant from the vane surface for the 60° compound angle orientation due to the slot-like nature.
of the holes in a row of holes thereby offering reduced protection of coolant in the ROI. As is seen from Figure 3.20, jet lift-off from the vane surface is greater at $Tu = 18.2\%$ in comparison to $Tu = 8.9\%$ for all blowing ratios. For $Tu = 8.9\%$, the interaction between coolant from adjacent holes in a row of holes and the interaction between coolant from the first and second rows of holes increases as blowing ratio increases, thus resulting in improved film cooling performance as blowing ratio increases. A similar trend is observed for $Tu = 18.2\%$.

![Figure 3.20: Adiabatic effectiveness contours for 7-7-7 shaped holes inclined at 60° compound angle for (a) $M = 1.5$, (b) $M = 2$, (c) $M = 2.5$.](image)

The plots of laterally-averaged adiabatic effectiveness for the 7-7-7 shaped holes at 60° compound angle orientation for the two turbulence intensity levels are shown in Figure 3.21. As seen in Figure 3.21, the film cooling performance for all the blowing ratios at $Tu = 18.2\%$ is much lower than at $Tu = 8.9\%$. Also it can be seen that jet lift-off from the vane surface at the locations
of the rows of holes is greater at $Tu = 18.2\%$ than at $Tu = 8.9\%$, as observed from the slope of the curves just downstream of the first and second rows of holes, leading to the lower film cooling performance for $Tu = 18.2\%$ at all blowing ratios. The values of peak laterally-averaged adiabatic effectiveness at the locations of the rows of holes are higher at $Tu = 8.9\%$ than at $Tu = 18.2\%$. This indicates less penetration of the coolant into the mainstream at the hole exits for the lower turbulence intensity level and greater dilution of the coolant within the mainstream at the higher turbulence intensity level. The slot-like nature of the holes for the 7-7-7 shaped hole at 60° compound angle allows more coolant to be attached to the surface of the vane downstream of the rows of holes at $Tu = 8.9\%$. The boarder spread of coolant just downstream of the film cooling hole rows at $Tu = 8.9\%$ can be seen by the gentle slope of the curves following the peak effectiveness values. Due to greater jet lift-off at $Tu = 18.2\%$ noted previously, the slopes of the curves just downstream of the rows of holes are steeper for $Tu = 18.2\%$ for all blowing ratios. The coolant provides no protection to the surface of the vane further downstream of the second row of holes at $Tu = 18.2\%$ from $X/C_x$ of about 0.4 to about 0.58 for all blowing ratios. More coolant reattaches to the surface of the vane upstream of the second and third row of holes at $Tu = 8.9\%$ in comparison to at $Tu = 18.2\%$. 
Figure 3.21: Laterally-averaged adiabatic film cooling effectiveness for 7-7-7 shaped holes inclined at 60° compound angle.

3.5.4. Area-Averaged Adiabatic Effectiveness for 7-7-7 Shaped Holes

The plot of area-averaged adiabatic effectiveness for 7-7-7 shaped holes for the two turbulence intensity levels and for the blowing ratios as well as compound angle orientations is shown in Figure 3.22. As seen in Figure 3.22, the area-averaged adiabatic effectiveness increases with increase in blowing ratio for Tu = 8.9%. The 7-7-7 shaped hole inclined at 60° compound angle for Tu = 18.2% has the least area-averaged adiabatic effectiveness of all the configurations tested except at $M = 2.5$ in which the value was slightly higher than that of the axial hole at Tu = 18.2%. The very low film cooling performance of the hole inclined at 60° compound angle can be attributed to significant jet lift-off from the vane surface for the shaped hole inclined at 60° compound angle discussed previously considering the contours in Figure 3.20 and plots in Figure
3.21. The axial hole at $M = 2.5$ for $Tu = 8.9\%$ had the highest area-averaged adiabatic effectiveness of all the configurations tested. Generally, the area-averaged adiabatic effectiveness at $Tu = 8.9\%$ is much greater than at $Tu = 18.2\%$ for all the configurations and blowing ratios tested. As discussed previously, higher freestream turbulence leads to greater coolant jet lift-off from the surface of the vane, and coolant reattachment to the vane surface is much reduced at the higher freestream turbulence level. Therefore, the area-averaged adiabatic effectiveness is the ROI is much lower at $Tu = 18.2\%$ than at $Tu = 8.9\%$.

Figure 3.22: Area-averaged adiabatic effectiveness for 7-7-7 shaped holes for the different compound angles, blowing ratios and freestream turbulence intensities.
3.6. CONCLUSIONS

In this study, steady-state experiments were conducted in a subsonic linear gas turbine vane cascade using infrared thermography technique. The effect of compound angle of the 7-7-7 shaped hole and freestream turbulence intensity on the adiabatic film cooling effectiveness on the pressure side of a scaled-up GE-E3 vane was investigated. The range of blowing ratio tested is 1.5 to 2.5 and the compound angles considered are 0° (axial hole), 30° and 60°. The experiments were conducted at Reynolds number based on the mainstream inlet velocity and axial chord of the vane of 175,340 and at density ratio of approximately 1. Results obtained using the 7-7-7 shaped holes were compared with those obtained using cylindrical holes for the same test configurations. It should be noted that the baseline results for this research are the results obtained at turbulence intensity of 8.9%. Thus, a strong benchmark at this level of turbulence intensity to compare with previous studies is not available. All trends were compared on the basis of the results for turbulence intensity of 8.9%. Conclusions that can be drawn from this study are as follows:

- The M = 1.5 case at Tu = 18.2% resulted in far better lateral spread of coolant on the vane surface for axial cylindrical holes.
- The cylindrical holes inclined at 30° compound angle had much lower film cooling performance compared to the other configurations studied.
- The higher freestream turbulence led to substantially reduced film cooling performance for cylindrical holes inclined at 60° compound angle.
- For the cylindrical and 7-7-7 shaped holes, the higher freestream turbulence primarily results in much greater jet lift-off from the vane surface at all blowing ratios. Also, jet reattachment to the vane surface was much lower at the higher freestream turbulence level.
in comparison to the lower level. These result in significantly reduced adiabatic effectiveness levels in the ROI.

- The peak laterally-averaged effectiveness values at the locations of the rows of holes are greater at all blowing ratios for Tu = 8.9% in comparison to Tu = 18.2% for cylindrical holes inclined at 60° compound angle and for the shaped holes. This indicates dilution of the coolant by the mainstream at the film cooling hole exits due to higher freestream turbulence for all blowing ratios.

**NOMENCLATURE**

- **A** Cross-sectional area of cooling hole (m²)
- **ABS** Acrylonitrile Butadiene Styrene
- **AR** Area ratio, $A_{exit}/A_{inlet}$
- **CFD** Computational fluid dynamics
- **Cₚ** Axial chord of the vane (m)
- **Cyl** Cylindrical
- **D** Diameter of film cooling hole metering section (m)
- **Deg** Degree (°)
- **H** Vane span (m)
- **IR** Infrared
- **L** Length of film cooling hole (m)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M$</td>
<td>Blowing ratio (-)</td>
</tr>
<tr>
<td>NGV</td>
<td>Nozzle guide vane</td>
</tr>
<tr>
<td>$P$</td>
<td>Film cooling hole and vane pitch (m)</td>
</tr>
<tr>
<td>$R$</td>
<td>Radius of the interior edges of diffused outlet (m)</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number (-)</td>
</tr>
<tr>
<td>ROI</td>
<td>Region of interest</td>
</tr>
<tr>
<td>Shp</td>
<td>Shaped</td>
</tr>
<tr>
<td>$t$</td>
<td>Hole breakout width (m)</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature (K)</td>
</tr>
<tr>
<td>$Tu$</td>
<td>Freestream turbulence intensity</td>
</tr>
<tr>
<td>$U$</td>
<td>Velocity (m/s)</td>
</tr>
<tr>
<td>$X$</td>
<td>Streamwise direction</td>
</tr>
<tr>
<td>$Z$</td>
<td>Spanwise direction</td>
</tr>
</tbody>
</table>

**Greek Symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>Film cooling hole inclination angle (°)</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Expansion angle of shaped hole (°)</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Local adiabatic film cooling effectiveness (-)</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density (kg/m³)</td>
</tr>
</tbody>
</table>
Subscripts, Accents

adia  Adiabatic
aw    Adiabatic wall
c     Coolant or secondary fluid
fwd   Forward
lat   Lateral
m     Mainstream, Metering section
-     Lateral average
=     Area average
REFERENCES


RECOMMENDATIONS FOR FUTURE RESEARCH WORK

The following recommendations are presented for future research work:

- Two variables are important in the determination of vane film cooling performance. They are adiabatic effectiveness and the heat transfer coefficient. While adiabatic effectiveness was determined in this work using heated coolant and cold mainstream, the heat transfer coefficient as well as the adiabatic film cooling effectiveness can be determined by using heated mainstream and cold coolant. This will help to gain a full understanding of the performance of the 7-7-7 shaped holes in comparison to cylindrical holes.

- By using heated mainstream and cold coolant, conduction correction to obtain adiabatic effectiveness can be performed experimentally. This is a better approach than using computational fluid dynamics (CFD) for conduction correction. Thus, conduction correction can be done experimentally in future studies.

- In this work all investigations were performed on the pressure side of the vane. Experiments should also be performed on the suction side as well as the leading edge of the vane.

- Overall cooling effectiveness can be determined by using a conducting vane. This will help to fully assess the film cooling performance of the 7-7-7 shaped holes when compared to cylindrical holes.

- In actual gas turbine engines, the flow at the exit of the vane is transonic. Experiments should be performed in a transonic cascade as this is more representative of actual gas turbine engine conditions.

- Novel cooling hole geometries and cooling concepts can be tested using the subsonic gas turbine linear vane cascade.
• Experimental techniques like particle image velocimetry (PIV) and laser-Doppler velocimetry (PDV) should be used to obtain a deeper understanding of the physics of film cooling using the 7-7-7 shaped holes and cylindrical holes on the surfaces that are representative of the vane.

• In actual gas turbine airfoil film cooling, the coolant is supplied via crossflow and not via a plenum. The effect of injecting the coolant via crossflow in comparison to using a plenum should be investigated.

• The effect of variation of hole spacing on the film cooling performance of the vane should be studied.

• The density ratio in actual gas turbine engines is approximately 2. The density ratio in this study, as well as many other studies in the literature is approximately 1. The effect of actual engine-representative density ratio of approximately 2 on the film cooling performance of the 7-7-7 shaped holes compared to the cylindrical holes on the vane surface should be studied.

• As seen from the numerical results, the simulations did not predict coolant jet reattachment to the surface of the vane. Therefore, modeling efforts should be directed at enhancing the capability of the realizable k-ε turbulence model with enhanced wall treatment in ANSYS Fluent to predict coolant reattachment to the surface.

• The k-ω shear stress transport (SST) turbulence model has also been used in studies on gas turbine film cooling reported in the literature. The performance of the realizable k-ε turbulence model with enhanced wall treatment on the vane film cooling should be compared with that of the k-ω shear stress transport (SST) turbulence model.