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PRELIMINARY STUDY ON THE IMPACT OF IMPINGEMENT ON THE EFFECTIVENESS OF FILM COOLING IN THE PRESENCE OF GAS PATH PRESSURE GRADIENT

by

ANIL SUBASH BABU PERAVALI
B.S. Indian Institute of Technology Madras, 2004

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in the Department of Mechanical, Materials, and Aerospace Engineering in the College of Engineering and Computer Science at the University of Central Florida Orlando, Florida

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ABSTRACT

Impingement is the most commonly used method of cooling in the hot stages of gas turbines. This is often combined with film cooling to further increase the cooling performance. The mainstream flow where in the coolant films discharge often has large stream wise pressure variations. All existing studies on coupled film and impingement cooling concentrated on the effect of the film depletion on the impingement heat transfer. This study investigates the impact of impingement on film cooling, where the jets impinging on a flat plate are depleted through arrays of film cooling holes in the presence of pressure gradient in the main gas path. The main characteristic of the test setup is that there is an impingement wall on the backside of the film effusion wall. The fluid used for both impingement flow and main flow is air. The impingement flow is heated as opposed to the usual practice of heating mainflow, and the array of film holes are configured under the impingement jet hole arrays such that there is no direct impingement on the film holes. The static pressure variations and Mach number (0.01 to 0.3) in the mainstream underneath the flat plate are controlled by inserts with varying flow area. The detailed temperature distribution on the film-covered surface is measured using the Temperature Sensitive Paint (TSP) technique, and film cooling effectiveness is calculated from the measurements. Results are presented for averaged impingement jet Reynolds numbers of 5000 and 8000. The effect of impingement on film effectiveness is studied by comparing the results from the two cases: one where film flow is directly supplied from a plenum and the other where the post-impingement flow is depleted through film effusion holes. The results are presented for cylindrical film cooling holes which are inclined at angles of 20 deg and 30 deg with respect to
the target plate surface. The variation of the effectiveness of the film hole arrays along the mainstream are studied in detail. It is observed that the impingement through jet effects the pressure distribution on the target plate with film holes, which in turn affects the blowing rates of each row. The change in the blowing ratios because of a different pressure distribution on the impingement side of the target plate causes the effectiveness to change. From the results it is observed that the farther rows of impingement are affected by the pressure distribution underneath the film holes and have more flow through the film cooling rows, this increases the inlet flow of the films which increase the blowing ratios and in turn decreases the effectiveness of the film cooling holes. The pressure distribution and the change of effectiveness are studied in detail.
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CHAPTER 1: INTRODUCTION

1.1 Impingement Cooling

Impingement cooling is a mechanism of heat transfer by means of collision. This phenomenon can be achieved when fluids come in contact with hot or cold surface by impingement. Around the impact region, the boundary layer is very thin, and hence heat can be transferred easily. Engineering applications of jet impingement include annealing of metal, tempering of glass, drying of paper, cooling of electronic equipment, freezing of tissue in cryosurgery, anti-icing system for forward facing surfaces of civil aircraft, cooling of gas turbine components and the outer wall of combustors.

Gas turbines are commonly used in power stations to generate electricity. There are many applications of impingement cooling in gas turbines, including the cooling of the critical gas turbine parts, e.g. the outer wall of combustors and the inner wall in the leading edge region of gas turbine blades. If a jet impinges on a plate and another fluid is in contact with other side of the plate, effective heat transfer will occur between the two fluids across the thickness of the plate. A much smaller surface area is needed than it would have been without jet impingement, for the same amount of heat transfer. This lays the foundation of an impingement cooler, which is basically a heat exchanger. Jet impingement heat transfer has been an extensive area of research in determining both the peak, local and average heat transfer for different configurations of jet and surfaces. Of all the factors that influence the impingement heat transfer crossflow is one of the main factors as shown in Figure 1. Many laboratory studies suffer from this impact of
crossflow degradation. In real engines, crossflow is depleted through film holes on the target surface hence the degradation can be brought down as can be seen in Figure 2.

![Figure 1: Array Impingement Crossflow](image1)

![Figure 2: Film Extraction of the Crossflow](image2)

1.2 Film Cooling

To combat and avert failure of turbine blades in gas turbine engines resulting from these excessive operating temperatures, film cooling has been incorporated into blade designs. Film cooling is a technique that has been used in many systems to protect component surfaces exposed to high temperature gas streams. Applications have been widespread, particularly in gas
turbines, where combustor liners, turbine shrouds and blades and other hot parts of the engine have used air, bled off from the compressor outlet, as the coolant film. Such techniques reduce the thermal stresses that tend to occur with an increase of inlet temperature to the first-stage turbine of high-performance gas turbine systems. In film cooling, cool air is bled from the compressor stage, ducted to the internal chambers of the turbine blades, and discharged through small holes in the blade walls. This air provides a thin, cool, insulating blanket along the external surface of the turbine blade. The basic physics and flow of film cooling can be seen in Figure 3.

![Figure 3: Flow through a Film Cooling Hole (J C Han 1999)](image)

1.3 Impingement Coupled with Film Cooling

In the hot stages of gas turbines, impingement is often combined with film cooling to further increase the cooling performance as well as providing external thermal protection. However, the mainstream flow where the coolant films discharge often has large streamwise pressure variations. Streamwise pressure variations affect the film cooling effectiveness of the holes and also have an impact on the impingement heat transfer on the other side of the surface.
1.4 Objectives

This study investigates a simplified model where the jets impinging on a flat plate are depleted through arrays of film cooling holes. The objectives of the study include the study of the impact of the jet impingement on the film cooling effectiveness in the presence of pressure variation in the mainstream. The film holes are aligned under the jet hole arrays such that there is no direct impingement on the film holes as shown in Fig 4. The pressure variations in the mainstream on the discharge-side of the film and on the other side of the flat plate are controlled by inserts that produce Mach numbers ranging from 0.01 to 0.3 as seen in Fig 5. The detailed temperature distribution on the film-covered surface is measured using temperature sensitive paint (TSP) technique, and film effectiveness is calculated from the measurements. Results are presented for averaged jet Reynolds numbers of 5000 and 8000 in the case where the impingement/film supply air is from the jet plate and the mass flow rate is kept constant while testing the case of direct impingement. The blowing ratios are calculated using the mass flow rates and pressure measurements on both sides of the effusion plate. The results are presented for cylindrical film cooling holes which are inclined at angles of 20° and 30° with respect to target plate surface. The variation of the effectiveness of the film hole arrays along the mainstream are studied in detail.

The cases to be considered and compared are listed below.

Hole inclinations (α) considered:

- α = 20°
- α = 30°
Average jet Reynolds numbers ($Re_j$) considered:

- $Re_j=5000$
- $Re_j=8000$

Impingement cases compared:

- Direct impingement on the target plate
- Impingement through the jet plate onto the target plate

Figure 4: Alignment of target film holes with respect to impingement jet holes
Figure 5: Inserts to generate the required mach no. under the film holes

Figure 6: Coupled Configuration Tested in the Present Study
CHAPTER 2: LITERATURE REVIEW
AND FOUNDATION FOR PRESENT STUDY

2.1 Impingement Cooling

Impingement heat transfer is considered as a promising heat transfer enhancement technique. Among all convection heat transfer enhancement methods, it provides a significantly high local heat transfer coefficient. At the surface where a large amount of heat is to be removed/added, this technique can be employed directly through a very simple design involving a plenum chamber and orifices. In gas turbine cooling, jet impingement heat transfer is suitable for the leading edge of a rotor airfoil, where the thermal load is highest and a thicker crosssection enables accommodation of a coolant plenum and impingement holes. Other applications include cooling of the combustor chamber wall, steam generators, ion thrusters, tempering of glass, electronic devices cooling and paper drying. There are typically two compartments in a jet impingement configuration. These two chambers are separated by a perforated surface. One of these chambers is pressurized, and the outer experiences the impingement effects. There are several arrangements possible with cooling jets, and different aspects need to be considered before optimizing an efficient heat-transfer design. The shape of the jet nozzle, the layout of jet holes, the shape of confinement chambers and the shape of target surfaces have a significant effect on the heat-transfer coefficient distribution.
2.1.1 Hydrodynamics of Impinging Flow

Figure 7: Comparison of flow regions in (a) impinging jet (b) free jet (J C Han 1999)

The flow field of impinging jets from single round and slot nozzles can be divided into three regions as shown in Figure 7 the free jet region, the stagnation flow region, and the wall jet region. The velocity field of an impinging jet is also shown in the figure. The critical jet Reynolds number $Re_j$ (based on nozzle diameter and nozzle exit velocity), which distinguishes laminar jets from turbulent jets, is about 3000. In gas turbine component cooling, the jet nozzle diameters are quite small, but the Reynolds number can be quite large owing to the high velocity of the jet. In such applications, the jet developing from the nozzle is generally turbulent, and the turbulence intensity at the core can be as high as 25%. The high velocity coupled with high velocity fluctuations, increases turbulent mixing, and hence significantly increases the heat transfer capability.
Multiple jets in arrays are used to cover larger areas. Studies have focused on the geometric arrangement of jets. The geometry of regular, inline jet holes arrays can be characterized by non dimensional parameters, x/d, y/d and z/d, where x/d and y/d represent the jet-to-jet spacing in streamwise and spanwise direction respectively and z/d represents the jet-to-target plate spacing. All three parameters are normalized by the jet nozzle diameter (d). Florschuetz et al. [1] and Koopman and Sparrow [2] showed the effect of crossflow on jet arrays. Mass flowing out of a jet, after impingement, moves in the cross-jet flow direction and can alter the performance of neighboring jets. The multiple jet arrangement can be either inline or staggered. They plotted Nu at different Reynolds numbers varying the jet-to-target spacing (Figure 9). The Reynolds number (Re) is the ratio of inertial forces ($v_0 \rho$) to viscous forces ($\mu / L$) and is used to determine whether a flow will be laminar or turbulent. Nusselt number (Nu) is defined as the ratio of the convective heat transfer and conduction heat transfer.

Figure 8: Impingement cooling in the first stage turbine inlet guide vane (J C Han 1999)
2.1.2 Effect of Jet-to-Jet Spacing

Effect of surrounding jets for large jet to jet spacing is presented by Hollworth and Berry [3]. Their results showed a significant dependency on the wall-to-jet spacing, and for a small wall to jet spacing the Nusselt number increased toward a secondary peak. Jung-Yang San et al. [4] investigated optimum jet to jet spacing of heat transfer for staggered arrays of impinging air jets. They concluded that for a small spacing from jet to jet if z/d and Re are kept constant, an increase of the spacing results in a reduction of the jet interference. A further increase of the spacing, Nu will descend due to the fountain effect and an increase of the corresponding heated area. If the spacing is still increased strength of the jet fountain will gradually decrease and Nu will rise again, and as spacing exceeds a certain value, the Nu will approach the value for a single jet impingement (Figure 10).
2.1.3 Effect of Crossflow

In addition to the above factors, the cross flow is another important parameter that affects jet impingement heat transfer performance. Cross flow is the spent jet flow upstream of the local jet and reinforced by the local jet flow after its impingement. Strong crossflow is not a desirable factor in impingement heat transfer because crossflow tends to push the impinging air flow downstream and dilutes the impinging jet intensity. Florschuetz et al. [1] also studied the effect of initial crossflow on impingement heat transfer. Their results show that the initial cross flow lowers the impinging heat transfer performance. As the initial crossflow rate increases, convective heat transfer will be more dominated by the crossflow. However for a smaller crossflow, i.e. for crossflow velocity less than 10% of the jet flow velocity, the cooling is better overall and the Nusselt number increases. Figure 7 gives a comparison of Nusselt number for different exit flow conditions at various Reynolds numbers.

Figure 10: Effect of jet to jet spacing on Nu (Jung-Yang San et al. [4])
2.1.4 Effect of Coolant Extraction

Figure 12 shows a typical flow arrangement in an airfoil with coolant impingement and extraction. The coolant is at a higher pressure than the hot-gas side, and therefore the coolant passes through the channels from the coolant side to the hot gas side. Hollworth et al. [5] studied heat transfer from arrays of impinging jets with spent fluid removal through inclined holes on the impingement. They considered both inline and staggered configurations with respect to the impinging jets. As seen in the Figure 13 staggered arrangement shows the heat transfer
coefficient is higher for a closer spacing. Ekkad et al. [6] studied the heat transfer enhancement in the presence of coolant extraction from their three different exit configurations. They compared their study with the case with no film holes. Figure 13 shows the effect of coolant extraction on span-averaged Nu for different exit directions. From all the studies that have been done on this region it was concluded that the film extraction mainly helps in the weakening of the crossflow and thus increases the impingement heat transfer.

![Figure 12: Cooling passages in the leading edge of a turbine airfoil (J C Han 1999)](image)
2.2 Film Cooling

The development of gas turbines is characterized by increasing the compressor pressure ratio and the turbine inlet temperature in order to increase specific power and efficiency. However, near the physical limits, great care is necessary to protect highly loaded parts of the high-pressure turbines because a small increase in metal temperature can lower the engine lifetime drastically. The use of film cooling is a means to intensify the convection cooling at
critical locations on the blade. With film cooling, air is blown out from the surface to be cooled and mixes with the hot gas flow near the wall so that a film of cooler gas is formed which is effective over a certain region downstream from the point of blowing.

The heat exchange between hot gas and the film-cooled wall depends upon numerous parameters, e.g., the blowing rate, the ratio of momentum, the boundary layer thickness, the relative hole spacing, the number and the spacing of the rows as well as the hole arrangement (staggered or inline), the injection angle and the shape of the holes. In addition to these typical film cooling parameters, there are general heat transfer parameters like Reynolds number, temperature ratio, pressure gradient, roughness, free stream turbulence and wall curvature, which may change their influence due to film cooling. Figure 14 shows the schematic of the general film cooling concept.

![Figure 14: Flow through a film hole (JC HAN 1999)](image)
2.2.1 Effect of Pressure Gradient

The literature contains some contradictory results about the influence of pressure gradient on film cooling. In 1985, Kruse [7] found that a favorable pressure gradient slightly decreases effectiveness, while an adverse pressure gradient slightly increases effectiveness with small blowing rate. Except thickening of the boundary layer, no influence on the wall was found with high blowing rates. He concluded that pressure gradient significantly affects the film cooling effectiveness only at low blowing rates and small hole spacing. In 1991, Teekaram et al. [8] experimented with a row of 30 deg holes to compare the effect of favorable, zero and adverse pressure gradients on the effectiveness. They concluded that the effect of main stream pressure gradients is minimal at low injection rates where the cooling jets are still attached to the surface. At these conditions, the physical processes are similar to those for two dimensional slot injection. After the maximum effectiveness, which may be associated with jet separation, the film cooling effectiveness is markedly affected in the vicinity of injection. They observed an increase in effectiveness with a streamwise favorable pressure gradient relative to zero and adverse pressure gradients. However, far downstream they couldn’t find any consistent variation of the film cooling effectiveness with pressure gradient.

In 1995, Bogard et al. [9] imposed a pressure gradient which represents the suction side of a gas turbine on the main flow and tested the changes of the effectiveness. They found out that when the cooling jets did not completely detach, the application of the pressure gradient improved the lateral spread of the jets immediately downstream of the film cooling holes and increased the decay rate of the laterally averaged effectiveness. However, the overall changes in effectiveness were small. Laterally averaged Effectiveness was plotted for different Blowing ratios (M) and Momentum ratios (I) (Figure 15).
2.2.2 Full-Coverage Film Cooling

Full-coverage discrete film cooling offers an effective cooling technique of the combustor wall and airfoils. The coolant air passes through multiple rows of discrete holes cooling the inside wall and providing film to protect the gas-side surface. It is also called effusion or transpiration cooling. Andrews et al. [10] investigated the various effects of parameters on full-coverage film cooling. They presented the effects of hole length on film effectiveness to a surface covered by full-coverage film cooling. Martiny et al. [17] also investigated film effectiveness on a plate with full-coverage film cooling. They used a test plate with a series of staggered film hole rows. The holes were all angled 17° along the streamwise direction. The spacing between holes in each row was s/d = 4.48, and the distance from row to row was p/d = 7.46.
Figure 16 shows the spanwise averaged film effectiveness along the test plate for the first four hole rows. The effect of blowing ratio is also shown. It appears that film effectiveness increases rapidly with increasing film attachment as coolant spreads both streamwise and spanwise. The coolant accumulation effect is significant at lower blowing ratio. Full-coverage film cooling is a very effective cooling technology used in cooling combustor walls. The limitation of cooling mass flow for providing cooling in airfoils is the main reason for the lack of using full-coverage film cooling in airfoils.

![Figure 16: Spanwise-averaged film effectiveness for full coverage film cooling [17]](image)

2.3 Foundation For Present Study

One of the characteristics of impingement cooling in most (but not all) gas turbines is that it is coupled with the film cooling, where the post-impingement flow is depleted through film holes. However, just a study of the coupled case as in Figure 17 is not enough and even may be
misleading when the main flow path has significant stream-wise variation in static pressure, as discussed below.

Figure 17: Impingement/Film hybrid cooling system

Figure 18 shows a typical coupled cooling system where the coolant supply from the compressor hits the inner walls and the spent air provides film cooling through the leading edge holes. Moreover, the resultant coolant film after exiting the holes generally experiences significant variation in static pressure in the main stream (Figure 19). As shown in this figure, the main flow goes through the converging passage in a typical vane with accompanying drop in static pressure and increase in flow Mach number. In this case, with multi-row, full-coverage film cooling, different rows of film holes experience different sink pressures in the main stream and hence the flow distribution among the film rows and in the post-impingement chamber gets seriously altered. Thus, it is expected that the extent by which jet impingement impacts subsequent film cooling will be different in the presence of mainstream pressure variation, and that is the objective of this study.
Figure 18: E3 stage-1 HPT vane, inner band-cooling design (Halita et al., 1982)

Figure 19: E3 stage-1 HPT vane, outer-band-cooling design (Halita et al., 1982)
The present experimental study was conducted to investigate the effect of the presence of large pressure variation and impingement/effusion film cooling on the effectiveness of film cooling holes. This technique combines two cooling schemes, jet impingement cooling and film cooling; hence the inner surfaces of hot components, such as the combustor wall or blade surface are cooled by the impingement of cooling air, and the outer surfaces which contact with hot gases are protected by effusion film cooling. The impingement/effusion cooling technique has been investigated and developed by some researchers since 1980’s [11].

Ekkad et al. (1999) [12] studied the heat transfer enhancement in the presence of coolant extraction for their three different exit configurations. They compared their study with the case with no film holes. The effects of coolant extraction on span-averaged Nu for different exit directions were plotted. From their study, they concluded that the film extraction mainly helps in the weakening of the crossflow and thus increases the impingement heat transfer.

Hollworth and Dagan [5] and Hollworth et al. [11] measured average and local heat transfer coefficients on the effusion surface, and reported that arrays with staggered vents consistently yield higher heat transfer rates than do the impinging jets on the solid plates. Nazari and Andrews [13] studied film cooling performance with the effects of the number of holes for impingement/effusion cooling. Cho and Goldstein [14] investigated the effect of hole arrangements on local heat/mass transfer characteristics inside the effusion / target plate. They found that the high transfer rate is approximately 45% -55% higher than that for the impingement cooling alone. Cho and Rhee [15] and Rhee et al. [16] also investigated heat/mass transfer and flow characteristics of an impingement/effusion cooling system with various experimental conditions, such as gap distance, Reynolds number, hole arrangement and size.
As opposed to all the existing impingement/effusion studies which focus on the effect of film depletion on the impingement heat transfer, the present study focuses on the effect of impingement on the film effectiveness in the presence of mainstream pressure variation. This technique would greatly help in determining if any impingement pattern is implemented on the coolant side how it would affect the film cooling effectiveness. This study also gives a broader view to look at optimization of the impingement and film cooling hole pattern while designing a cooling system.
CHAPTER 3: EXPERIMENTAL SETUP

All the tests of this study are conducted on the Impingement Film coupling (IFC) rig located at the Engineering Field lab located on the main campus of the University of Central Florida. A flexible test set up is designed that allows easy change of orifice plate (which contains the impingement orifices), easy change of side boundaries of the post-impingement plenum to fit any given configuration, and easy change of the target plate (so that holes for post-impingement depletion and film cooling holes can be implemented). This flexibility makes the experimental setup adaptable for all gas turbine impingement and film hole coupling configurations. The IFC rig can be mainly divided into two sections, Vertical tunnel (Figure 20) and the horizontal duct (Figure 26). The vertical tunnel is mainly used for impingement flow. The horizontal duct is used as the crossflow or mainflow in this case. The fluid used in both the impingement flow and mainflow is air. The impingement flow section is an open-loop duct where the air is supplied by a compressor, and the mass flow rate entering the tunnel can be controlled using a regulator. The mainflow is supplied by a suction blower which has a rating of 645 CFM. The setup is explained in detail in the following sections.

3.1 Impingement Tunnel

Figure 20 shows sketch of the impingement tunnel. It can be mainly divided into four parts, Air supplying components, heating components, flow recovery section and test section.
3.1.1 Air Supplying Components

The air supply for the impingement vertical duct is supplied by a compressor which is connected to a tank in parallel that can hold up to 200 psi. The air from the compressor is passed through a dehumidifier to remove the humidity of the air. This dehumidified air is stored in the tank. The pipe line from the tank runs through an oil filter which is equipped with a carbon-activated oil vapor separation filter to filter out the oil vapor from the air. This is necessary because if the oil from the compressor enters the test section, it can cause a deposit on the
Temperature sensitive paint (TSP) surface and can disturb the paint surface and have a direct impact on the readings. A regulator is installed at the beginning of the piping for the tunnel right after the oil filter. This enables us to set the required mass flow rate through the impingement tunnel. The inlet mass flow rate at the regulator is measured using a Scanivalve inline venturi meter. A pressure tap and a thermocouple are installed at the Scanivalve to calculate the mass flow rate at those particular conditions.

### 3.1.2 Piping, Heater and Insulation

The air after being set at the required mass flow rate conditions is then passed through an 18 KW Sylvania inline air heater which has the highest temperature rating of 1400°F. Figure 21, shows the performance rating on the heater. The heater is installed on the ground. If operated correctly, the heater can operate continuously for 5000 hrs or more. One must always have sufficient airflow through the heater before applying power. Otherwise, the element will overheat very quickly and burn out. A thermocouple cannot detect temperatures if there is no flow, so it is important to turn on flow before applying power. A thermocouple is installed one inch after the exit of the heater, the other end of which is connected to the electric heater feedthrough system. The two important factors for the heater are that the element wire should not go above 1066°C for maximum heater life and there must be sufficient flow through the heater so that the epoxy seal at the feed thorough does not become overheated.

Since the heater is controlled using a temperature controller, the response time of the thermocouple and temperature controller become critical. The thermocouple should be an exposed junction style, maximum wire diameter of 0.020”", and should be located within an inch
of the heater exit and should be located at/or slightly beyond the center of the pipe. The limiting factor for flow is the ability to sense the air temperature. At low flow rates, the thermocouple response will lag far behind the element temperature and there is a potential of overheating the element wire. The overshoot in the element temperature will be more critical if the heater is being run near its maximum temperature than if the heater is being run at relatively low temperatures. As a rough figure for thermocouple sensing, a minimum flow of 200 SCFH at lower temperatures and 400 SCFH at higher temperatures would work. The cross section of the heater can be seen in Figure 22.

![Figure 21: Performance rating of the heater (OSRAM SYLVANIA LTD.)](image1)

![Figure 22: Inline heater cross-section (OSRAM SYLVANIA LTD.)](image2)
The 2” steel piping starts from the exit of the heater on the ground to the entrance of the vertical tunnel. Since the hot air from the heater has to pass through piping, the flow recovery and flow straighteners, which are all made of steel, before reaching the test section a lot of heat is lost through all the steel which is directly exposed to the outside air. To avoid the heat loss and speed up the hot air to reach the test section, the entire test section is insulated starting from the exit of the heater to the start of the test section. The insulation used is a 1” thick foam insulation which can insulate up to 400 deg C.

### 3.1.3 Pressure Recovery and Flow Straightening Components

The entrance to the tunnel is of rectangular cross-section with a distributor, which is a rectangular plate with holes so it can distribute the flow from the pipe throughout the rectangular crosssection. The flow is then made to pass through a diffuser for pressure recovery. Honeycombs are set after the distributor for flow distribution and flow straightening. Then the hot air is made to pass through a 3:1 contraction nozzle which is then connected to the plenum with the jet plate with impingement holes. The whole flow conditioning section is insulated and sealed at all the connections to make sure there is no heat or mass loss through it. Figure 23 shows the parts which are discussed in this section.
After exiting the nozzle the hot gas or the impingement flow enters the plenum where the test section is bolted. The jet plate, which contains the impingement holes, is installed to the test section. As shown in Figure 24 the legs of the test section are used to hold the crossflow duct in place. A belt design is implemented such that the distance between the target plate and the jet plate can be changed by moving the belts upwards or downwards.
Figure 24: Cross-section of the test section

Figure 25: Test section with Camera and lights
3.2 Mainflow Tunnel/Section

The Mainstream section can be similarly divided into four parts (Figure 26) which are the suction blower, duct connecting the blower to the test section, cross-flow duct, and the flow conditioning section.

![Schematic of mainstream/crossflow duct](image)

**Figure 26: Schematic of mainstream/crossflow duct**

3.2.1 Suction Blower

The main flow under the target plate is provided using a suction blower (Figure 27). The blower has a volume flow rate of 645 SCFM. The inlet is connected to a transition piece which transfers the circular inlet to a 12” by 4” rectangular duct. Figure 27 shows the crosssection of the blower and Figure 28 shows the performance rating of the blower.
Figure 27: Suction blower (Spencer Turbine Co.)

Figure 28: Suction Blower performance rating (Spencer Turbine Co.)
3.2.2 Duct Connecting the Blower to the Test Section

The Duct connects to the crossflow duct on one side and the blower on the other side. Split vanes are installed on the 90° curve edges to reduce the pressure drop while the flow is curving. A height adjustable section is installed just before the crossflow duct to move with the crossflow duct while changing the distance between the jet and target plates. Figure 29 shows the flexible section which is connected to the crossflow duct to adjust the vertical height of the duct.

![Duct connecting to the Mainstream/crossflow duct](image)

**Figure 29: Duct connecting to the Mainstream/crossflow duct**

3.2.3 Crossflow Duct

Crossflow duct is the section where the target plate with film cooling holes is installed on. The duct as seen in Figure 30 is installed with foam inserts on either side, to control the Mach number and pressure variation. A slot is made in the duct to accommodate the target plate
with film cooling holes. The bottom side of the duct is made of acrylic, so the TSP on the other side of the target plate can be seen for the camera which is fixed underneath the duct. The inserts are made of foam and machined in the shape of required pressure variation along the flow. A tape is used to cover the machined surface of the inserts, so the flow will see a smooth surface. The tape is painted black so the light which is used to excite TSP won’t reflect from the tape.

![Figure 30: Crossflow Duct](image)

### 3.2.4 Flow Conditioning Section

The other side of the crossflow duct is connected to a 2-d, 3:1 contraction nozzle which is in turn connected to a flow straightening section on the other end. One honeycomb layer and a mesh are used to help straighten the flow.

Figure 31 shows the full assembly of the mainflow installed.
Figure 31: Full assembly of the experimental setup


CHAPTER 4: DESIGN AND PROCEDURE

4.1 Design of Impingement (Jet) and Film (Target) Plates

4.1.1 Jet Plate

The impingement/film supply flow from the main rig impinges on the target plate through the jet plate. The end of test section is a 12” by 12” section where the jet plate is bolted. The Jet plate design considered for this test is shown as in the Figure 32. The Jet plate is a 12” by 12” steel plate with straight holes of diameter 2 mm. The x/d measured in the streamwise direction is 5; y/d measured in the spanwise direction is 8.

For the case where the impingement is through the jet plate, the z/d is maintained as 3, where z is the distance between the jet and target plate. This is to maintain a similarity with Florschuetz [1], where the study is done with a configuration of (x/d, y/d, z/d) maintained at (5, 4, 3). If a need to extend the present study to the impingement side is planned, it will be easier to proceed with Florschuetz results as reference. The heat transfer coefficient values of Florschuetz are also used to do a simple FLUENT analysis on the temperature drop of the impingement flow through the film holes. This is discussed in detail in the results section.

For the case without the jet plate a rectangular portion is cut in the jet plate so the impingement is directly on the target plate as shown in the Figure 33. This is needed to support the target plate between the crossflow duct and the test section. Gaskets and weather strips are used to make a similar slot on the target plate and these two slot sections are matched while installing the plates.
Figure 32: Cross section of the jet plate used for impingement

Figure 33: Cross section of the Jet plate used for the direct film supply case
4.1.2 Target Plates

The target plates are the plates with the film cooling holes. Rohacell WF200 was the material used to make these target plates. Rohacell is polymethacrylimide- (PMI-) hard foam that is used as a core material for sandwich constructions. It shows outstanding mechanical and thermal properties. In comparison to all other foams, it offers the best ratio of weight and mechanical properties as well as highest heat resistance. The WF 200 product is the hardest Rohacell available in the market at the present instance. The main reason for the use of Rohacell in this study is its thermal conductivity value which is as low as 3.3x10^{-02} Watts/m C. This acts as a perfect insulation and it prevents any heat from the hot air that is impinging on the plate to the other side of the plate. Fluent results and calculations which were done to estimate the heat loss through the plate are discussed in detail in the Results section. So, while calculating the Effectiveness of the film holes, the recovery temperature value can be taken as the actual stagnation temperature which depends on the velocity of the mainflow at that particular row. The recovery temperature measurements are done with the film holes blocked and the impingement flow let out from the sides. The heat loss through the target plate can be neglected, and the recovery temperature is close to the stagnation temperature of the mainstream flow. The procedure of measuring and estimating the recovery temperature is discussed in detail in the procedure and results chapter.

Table 1 gives the mechanical property sheet of Rohacell
Table 1: Mechanical properties of Rohacell WF200

<table>
<thead>
<tr>
<th>Properties</th>
<th>Unit</th>
<th>ROHAC ELL® 200 WF</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>Kg/m³</td>
<td>205</td>
</tr>
<tr>
<td></td>
<td>lbs./cu.ft.</td>
<td>12.81</td>
</tr>
<tr>
<td>Compressive strength</td>
<td>MPa psi</td>
<td>9.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1,305</td>
</tr>
<tr>
<td>Tensile strength</td>
<td>MPa psi</td>
<td>6.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td>986</td>
</tr>
<tr>
<td>Flexural strength</td>
<td>MPa psi</td>
<td>12.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1,740</td>
</tr>
<tr>
<td>Shear strength</td>
<td>MPa psi</td>
<td>5.0</td>
</tr>
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<td></td>
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<td>725</td>
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<tr>
<td>Elastic modulus</td>
<td>MPa psi</td>
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<tr>
<td></td>
<td></td>
<td>50,750</td>
</tr>
<tr>
<td>Shear modulus</td>
<td>MPa psi</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td></td>
<td>21.750</td>
</tr>
<tr>
<td>Elongation at break</td>
<td>%</td>
<td>3.5</td>
</tr>
<tr>
<td>Heat distortion resistance</td>
<td>°C °F</td>
<td>200 392</td>
</tr>
</tbody>
</table>

Two plates with hole angles of 30 deg and 20 deg, respectively, are considered for comparison in this study. The target plates are designed such that there is no direct impingement of jets on the film holes. A staggered design as shown in Figure 34 was implemented. x/d and y/d are similar to the jet plate. The film hole diameter was kept the same as the jet holes diameter i.e., 2mm. Five rows of film holes are designed to study the effect of pressure gradient along the crossflow. The inserts are designed such that they guide the crossflow with required pressure.
gradient. More information about the design of the inserts and the variation of pressure and Mach number along the mainflow is discussed in the next section. Figure 35 shows the alignment of film holes with respect to the jet holes.

Figure 34: Crosssection of the target plate with film holes

Figure 35: Alignment of film holes with the impingement holes
Kapton tape is glued on the top and bottom sides of the target plates, to make the surfaces smooth. TSP is painted on the bottom of the target plate where the film holes exit on the surface of the Kapton tape. Figure 36 shows the target plate with Kapton tape glued and TSP painted on the back side.

![Figure 36: Back side of the target plate](image)

4.1.3 Design of Inserts

The Inserts which guide the crossflow are made of light weight sturdy foam. The main function of the inserts is to vary the cross-section, which will in turn vary the Mach number
along the film cooling hole arrays. The design of the inserts is based on the Mach number variation required. Figure 37 shows the alignment of the film holes with the inserts. The Mach number variation calculated along the inserts and the value at each hole row is plotted in Figure 38. The inserts are bolted to the crossflow duct underneath the target plate. The Mach number variation along the film hole rows is kept constant for both 20 deg and 30 deg plates for ease of comparison of the results. The hatched portion in Figure 37 shows the alignment of the inserts with respect to the holes. Kapton tape is used along the surface of the inserts for smoothness of the flow. Black paint is applied on the Kapton tape, so it won’t reflect the light on the paint, while taking the actual test pictures.

Figure 37: Alignment of inserts with film hole exits
4.2 Experimental Procedure

4.2.1 Adiabatic Temperature Measurement

Adiabatic wall temperature distribution is measured by allowing mainstream flow as well as impingement flow. The mass flow rate is set to obtain average jet Reynolds numbers, Re = 5000 and 8000 for the case where the impingement/film air supply is through the jet plate. For the case where the hot air is directly impinged on the target surface, the same mass flow rate is maintained. The temperature of impingement flow at the plenum should be maintained to facilitate proper temperature difference at the exit of film holes with respect to main flow. The experimental procedure followed for this test is a straightforward one. The compressor is switched on and when the tank collects enough air, the valve which controls the flow through impingement tunnel is set to the required flow level. The heater is switched on and set at a
temperature of 600 deg F. As there is a huge heat loss through the entire tunnel, it takes around two to three hours for the temperature to reach the required level. A three-way flow valve is arranged in the steel piping to divert the flow when needed. When the temperature is steady, the flow is diverted and the crossflow duct which contains the inserts and the target plate is installed. The camera and the lights are installed below the crossflow duct. The TSP coated underneath the target plate is captured using high-resolution IR camera. Reference pictures are taken without main flow or impingement flow. The hot air is directed back to the impingement tunnel, and the suction blower which supplies the main flow is switched on. When the plenum temperature is constant, pictures are taken again. The methodology of processing the TSP images is discussed in the next section. Figure 39 shows a brief setup to measure the adiabatic temperature on the surface painted with TSP while the jet are impinging on the target surface through jet plate.

![Figure 39: Adiabatic temperature measurement](image)
4.2.2 Recovery Temperature Measurement

Recovery temperature distribution is measured by blocking all the film holes and maintaining the same conditions as the adiabatic temperature test. The hot air impinged on the target surface is let out from the sides. The mainflow is allowed to run below the target plate. TSP is coated on the bottom side of the target plate. This recovery temperature is used in calculating the effectiveness of the film holes. A similar approach was used by Bogard [18] while calculating the effectiveness. Figure 40 shows a brief setup to measure the recovery temperature on the surface painted with TSP while the film holes are blocked and the jets impinging is let out from the sides.

![Recovery temperature measurement](image)

**Figure 40:** Recovery temperature measurement
4.2.3 Discharge Coefficient ($C_D$) Measurement

The discharge coefficient ($C_D$) is a ratio that compares the observed amount of flow going through a hole, or number of holes, to the predicted flow. The $C_D$ is measured by using the Bernoulli equation and calculating the mass flow rate through the film holes and comparing it to the actual flow going through. In the present study $C_D$ measurement is done for the direct impingement case, since the pressure distribution is uniform on the target plate. The inserts in the mainstream were removed to keep the sink pressure constant. Same conditions as the adiabatic temperature tests are maintained. The pressure and the temperature in the plenum, and the flow rate through the venturi are measured for various pressure ratios. The step by step procedure of calculating $C_D$ and the curves plotted are discussed and presented in the latter sections.

4.3 Instrumentation

The following instrumentations were used in this study to capture and maintain the required conditions for testing.

4.3.1 Temperature Measurement

4.3.1(a) Temperature Sensitive Paint

Uni-coat TSP formulated by ISSI is used in this study. The TSP is supplied in aerosol cans and is very easy to apply. Painting flow rate is determined by the nozzle tip of the aerosol can. Therefore the nozzle should be protected carefully and cleaned with alcohol after every painting to prevent clogging of the nozzle. The aerosol can should be shaken vigorously for 10 minutes every time before painting starts. This shaking time should be extended for a new TSP can. TSP is applied in layers; usually a well painted surface will consist of 8 to 10 layers where
each layer means one lateral movement from right to left and left to right. Enough time should be
given, between the layers, for TSP to dry. After painting the required number of layers the TSP is
heat treated in a oven which is set at 100 degree C. The painted surface can be buffed using fine
grit sanding sponge to improve the surface finish. After heat treated above 100 ºC for 30 minutes
the temperature sensitivity of the paint is about 0.9%/ºC. The TSP painted surface is smooth. An
optical 590nm long pass filter is also used on the camera to separate the excitation light and
emission light from the paint.

TSP incorporates luminescent molecules in paint together with a transparent polymer
binder. Light of the proper wavelength is directed at the painted model to excite the luminescent
molecules. The sensor molecules become excited electronically to an elevated energy state. The
molecules undergo transition back to the ground state by several mechanisms. The predominant
mechanism is radiative decay (luminescence). Sensor molecules emit luminescent light of a
longer wavelength than that of the excitation light. The appropriate filters can separate
excitation light and luminescent emission light, and the intensity of the luminescent light can be
determined using a photo detector. The excited energy state can also be deactivated by
quenching processes. Through two important photo-physical processes known as thermal- and
oxygen-quenching, the luminescent intensity of the paint emission is inversely proportional to
local temperature and pressure.

For TSP, polymer binders are not oxygen permeable and hence TSP is not pressure
sensitive. In principle, a full spatial distribution of the surface temperature can be obtained by
using TSP technique. Figure 41 shows a typical TSP set up.
For a more thorough description and theoretical explanation of the properties of TSP, please refer to the study by Liu et al (2003) [21] as well as Liu, 2006 [19].

4.3.1(b) Thermocouples

Accurate measurement of temperature is one of the most common and vital requirements in industrial instrumentation. It is also one of the most difficult objectives to achieve. Unless proper temperature measuring techniques are employed, serious inaccuracies of reading can occur, or otherwise useless data can result. The thermocouple is by far the most widely used temperature sensor for industrial instrumentation. Type E thermocouples are used in this study to make temperature measurements at different locations. Four thermocouples are used in the test rig to capture the temperatures at their particular locations. As shown in the Figure 42, T1 is placed at the venturi location at the inlet of the flow which is used in calculating the mass flow rate entering the impingement rig. T2 is placed in the plenum to set the required temperature for

Figure 41: Typical TSP setup [21]
impingement flow. T3 is placed between the spacing between jet and target plate, this temperature reading is used in calculating the effectiveness of the film holes. T4 is placed in the mainflow to read the temperature in the crossflow duct just before the inserts.

Figure 42: Thermocouple locations in the Test rig
4.3.1(c) CCD Camera

A high resolution 14-bit CCD (Charged Couple Device) camera was utilized for this study. It is a PCO-1600 CCD camera provided by Cooke Corporation with spatial resolution of 1200 by 1600 pixels. The image data is transferred via an IEEE 1394 (“firewire”) cable and firewire PCI card to a data collection PC. “CamWare” software provided by Cooke Corp. is used in the Windows operating system to control initialization, exposure time and image acquisition. The acquired image data are processed using MATLAB. The camera must be thermo-electrically cooled and have a high quantum efficiency at the paint emission wavelengths. The choice and quality of the scientific-grade camera dictate the measurement accuracy. Figure 43 shows the camera and the tungsten light used in the present study.

Figure 43: CCD camera and Tungsten light (The Cooke Corporation)
4.3.1(d) Tungsten Light Source

LED-based illumination source (peak wavelength at 464 nm) was selected as the excitation light for the TSP. The stability of the source provided by ISSI is about 1% after 10 minute warm up. The excitation spectrum of the LED is shown in Figure 44.

![Excitation Spectrum of LED Source](image)

Figure 44: Spectra of LED source (Liu, 2006)

4.3.2 Pressure Measurement

4.3.2(a) Pressure Transducer

A Pressure Transducer supplied by Scanivalve corp. is used to measure the static pressure at various locations in the whole rig. Pressure taps, 1/16” in diameter are used to connect to the transducer port. A voltmeter is calibrated with the transducer to digitally display the pressure value. The range calibrated on the voltmeter is +/- 5psi. It is used to measure the gauge pressure in the plenum and at the venturi location.
4.3.2(b) Pitot Tube

The Pitot tube (Figure 45) measures a fluid velocity by converting the kinetic energy of the flow into potential energy. The conversion takes place at the stagnation point, located at the Pitot tube entrance. A pressure higher than the free-stream (i.e. dynamic) pressure results from the kinematic to potential conversion. This static pressure is measured by comparing it to the flow's dynamic pressure with a differential manometer.

![Figure 45: Pitot tube (www.efunda.com)](image)

The velocity of the flow can be obtained by,

\[ v = \sqrt{\frac{2(p_2 - p)}{\rho}} \]
4.3.2(c) *Venturimeter*

A venturi Meter is a device used to measure the flow speed of a fluid in a pipe. The schematic of the venturimeter is show in the Figure 46.

![Figure 46: Schematic of Venturimeter](image)

In the upstream cone of the Venturi meter, velocity is increased, pressure is decreased. The pressure drop in the upstream cone is utilized to measure the rate of flow through the instrument. Velocity is then decreased and pressure is largely recovered in the down stream cone. A venturimeter supplied by PRESO is used in this study to measure the mass flow rate entering the impingement tunnel. A thermocouple and pressure tap are installed at the entrance of the 2” 10 venturimeter to calculate the physical parameters of the gas entering the venturi. The calibration curve provided by the company is shown in the Figure 47.
Figure 47: Calibration curve for the 2" 10 Venturimeter (Preso Flow Metering Equipment)
CHAPTER 5: ANALYSIS

5.1 Discharge Coefficient

Discharge coefficient \( (C_D) \) is a parameter that is used to express the performance of a nozzle or a hole, and is defined as the ratio of the mass flow rate through the hole to the mass flow rate through the hole for isentropic flow from the same inlet state to the same exit pressure. Discharge coefficient depends on both the local geometry and the flow conditions upstream and downstream of the hole. The discharge coefficient \( (C_D) \) is a ratio that compares the observed amount of flow going through a hole, or number of holes, to the predicted flow, based on the compressible flow equation for a specific physical area.

The discharge coefficient is assumed to be constant for both the impingement cases (with and without jet). It is calculated in this study using the Bernoulli’s equation and the compressible flow properties.

5.1.1 \( C_D \) Calculation

Pressure distribution along the inserts, below the target plate, is calculated using the Mach no. values based on the area at the crosssection of each hole exit rows.

\[
\frac{P}{P_t} = (1 + \frac{\gamma - 1}{2} M^2)^{\frac{\gamma}{\gamma - 1}}
\]
Where $P$ is the pressure, $t$ indicates the total conditions, $\gamma$ is the specific heat ratio and $M$ is the Mach number.

Since the maximum Mach number of the film holes tested is around 0.25 incompressible flow equations can be used to calculate the $C_D$ for the film holes. The pressure distribution using the above equations is used in the Bernoulli equation to get the velocity at each row exit.

$$P + \frac{1}{2} \rho v^2 + \rho gh = \text{const}$$

Where $p$ is the pressure, $\rho$ is the density, $V$ is the velocity, $h$ is elevation and $g$ is the gravitational acceleration.

This gives

$$\frac{P_2 - P_1}{\rho} = \frac{V^2}{2} \quad \text{(neglecting} \ h)$$

Mass flow rate through each row is calculated from the velocities obtained from the above equation

$$m_n = \rho \times V \times A \times N_n$$

Where $\rho$ is the density, $V$ is the velocity, $A$ is the area of the holes, and $N$ is the number of holes in each row.

Total mass flow rate is obtained by summing all the flow rates. Discharge coefficient is calculated by using the calculated mass flow rate and the mass flow rate that is measured through the venturimeter at the inlet of the tunnel. It is assumed that the $C_D$ is constant for impingement through jet and direct impingement cases.
5.2 Calculation of Blowing Rate

Blowing rate is defined as the ratio of the mass flow rate per unit area of the coolant to that of the mainstream. It is given by the equation

\[ M = \frac{\frac{m_c}{A_c}}{\frac{m_m}{A_m}} = \frac{(\rho_c \cdot U_c \cdot A_c)}{(\rho_m \cdot U_m \cdot A_m)} = \frac{\rho_c U_c}{\rho_m U_m} \]

For the present study, \( \rho_c \) is the density of the hot gas that is impinging on the target plate. It is calculated based on the temperature and pressure measured at the plenum. \( \rho_m \) is the density of the main/crossflow which is taken as the constant throughout the flow since the temperature variation is very low. Blowing rate of each hole in each row is calculated using the formula:

\[ M = \frac{\dot{m}_{film} / A_{hole}}{(\rho u)_{x}} \]

For the direct impingement case, the blowing ratios are calculated based on the inlet mass flow rate and the flow rate of the main/crossflow measured. It is assumed that the air impinging is uniform through the chamber. Average velocity is calculated based on the mass flow rate and the area of holes. The mass flow rate of the main/crossflow measured through the Pitot tube and the area ratio variation along the inserts is used to get the velocity variation along the film rows. This velocity ratio multiplied by the density ratio gives the Blowing rates for each row of holes.

For the second case where the impingement is through jet plate, the mass flow rate going through each film row is back calculated based on the pressure measurement made in the
chamber at each row and the $C_D$ calculated in the direct impingement case. The arrangement of the pressure taps with respect to the film holes inlet is shown in the Figure 48. Using those mass flow rate values the velocities and blowing rates are calculated.

Figure 48: Pressure taps installed in the chamber between jet and target plates
5.3 Film Cooling Effectiveness

Cooling effectiveness ($\eta$) of film holes is a non dimensional number given by:

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

Where $T_m$ is Mainflow Temperature, $T_{aw}$ is adiabatic wall Temperature and $T_c$ is the Coolant temperature.

This definition, which involves the mainstream temperature, applies only if there are no conduction effects on the test section as a result of cooling, heat leakage, or other sources of thermal noise that would cause a temperature difference between the mainstream and the TSP surface, or render the conditions downstream of the cooling holes non-adiabatic. An additional consideration is the fact that the test section is exposed to a significantly high velocity flow which means there will also be a recovery temperature due to viscous dissipation. This recovery temperature is used instead of the mainstream temperature in the calculation of the effectiveness. The recovery temperature and the mainstream temperature along the film hole are shown in Figure 49.

So the ratio changes to,

$$\eta = \frac{T_r - T_{aw}}{T_r - T_c}$$

Figure 49: Recovery temperature along a surface
The procedure followed to measure the recovery temperature is explained in detail in the experimental procedure section.

**5.4 TSP Data Reduction and Analysis**

The raw data obtained in this study are pictures captured by the CCD camera. These pictures are imported to MATLAB software, where they are processed using the calibration curve. The calibration test is done in a small steel cube chamber where a small 1” by 1” coupon is painted with TSP. It is installed on a heater which is connected to a power supply. A thermocouple is soldered to the test coupon which is used to measure the temperature of the coupon. The light is adjusted so that the intensity is uniform over the whole coupon. Pictures are taken at intervals of 5° C, where the temperature is maintained by controlling the power supply. At each point, four pictures are taken and averaged to negate the effect of temperature change (if any) and the movement of the camera (if any). All the pictures are imported to MATLAB, and the intensity of the whole coupon is averaged to one value (I). I is non-dimensionalised with a reference intensity value (Ir). θ is defined as 

$$\frac{T - 22}{100}$$

where 100 is the range of the TSP. I/Ir and θ are plotted, and the calibration curve as shown in Figure 50 obtained is used to back calculate the θ distribution on the TSP surface from the intensity ratios of the actual tests.
For the actual tests, all the pictures were imported in MATLAB and the intensity values were obtained at each pixel. A point is fixed on both the reference and test pictures and the pixel location is compared to check for the movement of the camera. The pictures are shifted according to the displacement of the point. This has to be done to align each reference point to the corresponding point in the test pictures. Then the ratio of the intensities at each pixel is calculated and the calibration curve is applied. The resulting temperature values give the temperature distribution along the whole TSP area. These are used in calculating the effectiveness value at each pixel.
5.5 Error Analysis

The uncertainty of the measurements in the present study is as follows

- Precision interval of TSP is found to be ±0.93°C with 95% level of confidence (Quan Liu 2006)

- The uncertainty in the effectiveness values was on an average less than 3%, while the highest value was calculated to be less than 9%.

- Uncertainty of $C_D$ values is less than 5%
CHAPTER 6: RESULTS AND DISCUSSION

6.1 CD Tests

Discharge coefficient tests were performed on both the 20° and 30° plates with impingement/film supply directly impinging on the plates and without inserts on the mainstream side. The results are presented as a curve between pressure ratio and the CD calculated. The same density ratio is maintained by keeping the impingement and the mainstream temperature same as the temperature tests. Figure 51 and Figure 52 show the CD data obtained from the tests for the 20° and 30° plates respectively. The usual trend of any CD curve is that it starts from zero and increases to a constant value. This test is done to ensure that all the pressure ratios tested fall in the constant CD line.

![CD curve for 20 deg plate.](image)

Figure 51: CD curve for 20 deg plate.
From the curves it can be seen that for both 20° and 30° plates the $C_D$ value is pretty constant. The $C_D$ values at the pressure ratios in the actual test are used in calculating the blowing rates for the case with the impingement through jet plate.

### 6.2 Heat Loss through Rohacell Plate:

A simple analysis is done to determine the heat loss through the Rohacell plate. CFD is used to determine the amount of temperature loss of the impingement jet in the film hole. Figure 53 shows the model setup to analyze in Fluent. Heat transfer coefficient (htc) calculated using Florschuetz[1] formula for staggered jet were applied on the top surface, which is around 300 for Rej of 5000, and a free flow htc value of around 100 is applied on the mainstream surface. It is observed that the temperature drop is less than one degree from the inlet to the outlet of the hole, which is believable because of a very low thermal conductivity value of Rohacell.
The temperature storage capacity of the material is calculated using the formula

\[ \Delta T_{\text{stored}} = \frac{k \Delta TA}{L \left( \frac{m}{c_p} \right)} \]

The calculated value is around 0.6° C. From the above two analysis, the assumption of the target plate as a good insulator is very much relevant.
6.3 Mach Number Calculations along the Inserts:

The Mach number variations along the mainstream inserts are calculated both analytically using the area ratio and computationally using CFD.

Figure 54: FLUENT results of velocity variation along the inserts

Figure 55: Mach number comparison
As shown in Figure 54 and 55, the Mach numbers calculated computationally and analytically are very close, with a maximum error of 10% at one row. This error is due to the design error of the inserts in CFD.

### 6.4 Recovery Temperature Tests

The procedure for testing the recovery temperature ($T_r$) is discussed in detail in the experimental procedure chapter. The entire recovery temperature tests yielded similar results within one or two degrees difference. Only the 20° plate is tested for the recovery test since all the parameters are same for both the plates. This is cross checked with the TSP data of the adiabatic temperature results before the film cooling rows start. All the values fall within $+/- 1.5^\circ$ C range. Since the plates are made of Rohacell, which has a very low conductivity value, this assumption is valid. These values are used while calculating the effectiveness of the film holes.

Table 2 gives the values of the $T_r$ for the 20deg Plate.

**Table 2: Recovery temperature values for the 20deg plate**

<table>
<thead>
<tr>
<th>Case</th>
<th>Re$_j$</th>
<th>$T_r$</th>
</tr>
</thead>
<tbody>
<tr>
<td>20deg (impingement through jet plate)</td>
<td>8000</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>5000</td>
<td>29.2</td>
</tr>
<tr>
<td>20deg (direct impingement)</td>
<td>8000</td>
<td>28</td>
</tr>
<tr>
<td></td>
<td>5000</td>
<td>28.3</td>
</tr>
</tbody>
</table>
Table 3 gives the values of the $T_r$ for the 30deg Plate.

<table>
<thead>
<tr>
<th>Case</th>
<th>Re$ _j$</th>
<th>Tr</th>
</tr>
</thead>
<tbody>
<tr>
<td>30deg (impingement through jet plate)</td>
<td>8000</td>
<td>28.5</td>
</tr>
<tr>
<td></td>
<td>5000</td>
<td>28.8</td>
</tr>
<tr>
<td>30deg (direct impingement)</td>
<td>8000</td>
<td>29.1</td>
</tr>
<tr>
<td></td>
<td>5000</td>
<td>28.9</td>
</tr>
</tbody>
</table>

**6.5 Adiabatic Temperature Tests**

**6.5.1: Blowing rates**

Each row of film holes has different pressure ratios because of the inserts along the mainstream. The pressure on the impingement side with the jet plate is measured using pressure taps as discussed in the previous chapter. The sink pressures are calculated based on the compressible flow equations and the Mach number calculated along the inserts based on the area ratio and CFD results. From the static pressure values measured both between the jet and target plate and along the inserts, the pr ratio is obtained. The $C_D$ value is used to find out how much the actual through the flow through the hole is and Blowing rates are calculated using the mass flow rate values. The $C_D$ of each row is calculated from the $C_D$ curves obtained from the tests (based on the pressure ratio at that particular row). For the case without the jet plate, the average
velocity is calculated based on the mass flow rate entering the tunnel. Blowing rates are calculated based on the average velocity and the velocities calculated on the mainstream side.

Tables 3 to 6 show the blowing ratio (M) distribution along the mainstream. From the tables it can be seen that the blowing ratios vary from row to row. For the case where the impingement is through jet it can be observed that the blowing ratios increased at the last rows when compared to the case where the impingement is directly on the target surface. The blowing ratio at the first rows goes down. This is mainly because of the pressure distribution on the impingement side is not uniform while the supply is from the jet plate. This has an impact on the effectiveness of the rows.

**Table 4: Blowing ratios for 30deg plate at Re\textsubscript{j}=5000**

<table>
<thead>
<tr>
<th>(pu)\textsubscript{a} (psia)</th>
<th>(P\textsubscript{static})\textsubscript{a} (psia)</th>
<th>Pr ratio</th>
<th>For same total film flow but without jet plate</th>
<th>For average Re\textsubscript{j}, impingement = 5000</th>
</tr>
</thead>
<tbody>
<tr>
<td>(pu)\textsubscript{f}</td>
<td>M</td>
<td>(pu)\textsubscript{f}</td>
<td>M</td>
<td></td>
</tr>
<tr>
<td>25.09</td>
<td>14.67</td>
<td>1.02</td>
<td>74.36</td>
<td>2.96</td>
</tr>
<tr>
<td>30.94</td>
<td>14.66</td>
<td>1.03</td>
<td>74.36</td>
<td>2.40</td>
</tr>
<tr>
<td>40.13</td>
<td>14.65</td>
<td>1.04</td>
<td>74.36</td>
<td>1.85</td>
</tr>
<tr>
<td>56.97</td>
<td>14.61</td>
<td>1.05</td>
<td>74.36</td>
<td>1.31</td>
</tr>
<tr>
<td>93.06</td>
<td>14.53</td>
<td>1.06</td>
<td>74.36</td>
<td>0.80</td>
</tr>
</tbody>
</table>
Table 5: Blowing ratios for 30deg plate at $Re_j=8000$

<table>
<thead>
<tr>
<th>$(pu)_a$</th>
<th>$(P_{static})_a$ (psia)</th>
<th>Pr ratio (with jet plate)</th>
<th>For same total film flow but without jet plate</th>
<th>For average $Re_j$, impingement = 8000</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>$(pu)_f$</td>
<td>M</td>
</tr>
<tr>
<td>25.09</td>
<td>14.67</td>
<td>1.09</td>
<td>131.65</td>
<td>5.07</td>
</tr>
<tr>
<td>30.94</td>
<td>14.66</td>
<td>1.10</td>
<td>131.65</td>
<td>4.11</td>
</tr>
<tr>
<td>40.13</td>
<td>14.65</td>
<td>1.107</td>
<td>131.65</td>
<td>3.17</td>
</tr>
<tr>
<td>56.97</td>
<td>14.61</td>
<td>1.116</td>
<td>131.65</td>
<td>2.23</td>
</tr>
<tr>
<td>93.06</td>
<td>14.53</td>
<td>1.127</td>
<td>131.65</td>
<td>1.37</td>
</tr>
</tbody>
</table>

Table 6: Blowing ratios for 20deg plate at $Re_j=5000$

<table>
<thead>
<tr>
<th>$(pu)_a$</th>
<th>$(P_{static})_a$ (psia)</th>
<th>Pr ratio (with jet plate)</th>
<th>For same total film flow but without jet plate</th>
<th>For average $Re_j$, impingement = 5000</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>$(pu)_f$</td>
<td>M</td>
</tr>
<tr>
<td>25.09</td>
<td>14.67</td>
<td>1.027</td>
<td>74.36</td>
<td>2.62</td>
</tr>
<tr>
<td>30.94</td>
<td>14.66</td>
<td>1.034</td>
<td>74.36</td>
<td>2.13</td>
</tr>
<tr>
<td>40.13</td>
<td>14.65</td>
<td>1.04</td>
<td>74.36</td>
<td>1.64</td>
</tr>
<tr>
<td>56.97</td>
<td>14.61</td>
<td>1.05</td>
<td>74.36</td>
<td>1.15</td>
</tr>
</tbody>
</table>

69
The pixel by pixel distribution of the temperature is obtained from the post processing of the TSP images as discussed in the experimental procedure chapter. Figures 56 and 57 show the adiabatic temperature distribution on the surface for 20deg film holes for both the Rej cases where the impingement supply is directly on the target surface. Effectiveness contours are plotted from the temperature distribution by applying the effectiveness formula at each pixel. Figures 58 and 59 show the effectiveness distribution on the test surface. All the unwanted portion of the contour region is cleared and each pixel row of the effectiveness contour is averaged to get the laterally averaged effectiveness value at each pixel row.
Figure 56: Temperature distribution for 20deg plate at Rej=5000

Figure 57: Temperature distribution for 20deg plate at Rej=8000
Figure 58: Effectiveness contour for 20deg plate at Rej=8000

Figure 59: Effectiveness contour for 20deg plate at Rej=5000
The laterally averaged effectiveness ($\bar{\eta}$) and the centerline effectiveness ($\eta_c$) values are plotted for all the cases tested. The average jet Reynolds numbers of 5000 and 8000 are tested for each plate. Figures 43 and 44 show the $\bar{\eta}$ values plotted along the mainstream for the 30° plate. As seen in the Figure 60, for the first two rows M values is so high that film supply through jet plate or direct supply doesn’t make a difference, since the jets lift off. So the coverage is very low. As we reach rows 3, 4 and 5 the blowing ratio (M) values reduce, so we can observe an increase in $\eta$ values. In the presence of no pressure gradients, flow from 30deg holes usually lifts off at M values around 1.5 and the optimum value of M is around 0.5. This can be observed in the laterally averaged profiles. As the M values reach 1.5 a significant difference can be seen in the $\bar{\eta}$ values because of significant attachment. At the fifth row (last row) there is a huge difference in the $\bar{\eta}$ values of the two cases. Similar trend can be observed in Figure 61, which is the $\bar{\eta}$ plot for 30deg holes at Rej=8000. Figures 62 and 63 show the $\bar{\eta}$ values plotted for the 20° plate. In all the plots X=0 indicates the start of the film cooling rows.

As seen in figure 62 and 63 after the fifth row the $\bar{\eta}$ values of the two impingement cases intersect. This is observed only for the 20deg plate. This might be since the flow is accelerating this whole time, the film might be stretched and destroyed and hence we see a higher effectiveness for higher M value. This is not further investigated, as this is out of scope for the present study.
Figure 60: Laterally averaged effectiveness for 30deg at $Re_j=5000$

Figure 61: Laterally averaged effectiveness for 30deg at $Re_j=8000$
Figure 62: Laterally averaged effectiveness for 20deg at Re_j=5000

Figure 63: Laterally averaged effectiveness for 20deg at Re_j=8000
Similar plots are drawn for the centre line effectiveness. Each plot is divided into three sections. First part is from exit of first row to the entrance of third row, second part is from exit of third row to the entrance of fifth row, third part is from the exit of the fifth row to the rest of the section. The effect of impingement which is observed in the laterally averaged effectiveness can also be seen in the centerline effectiveness curves. First three rows have a similar effectiveness, and a difference in the effectiveness values can be observed at the fifth row.

Figure 64: Centerline for the film hole rows

Figure 65: Centerline effectiveness for 20deg at $Re_j=5000$
Figure 66: Centerline effectiveness for 20deg at Re_j=8000

Figure 67: Centerline effectiveness for 30deg at Re_j=5000
6.6 Comparison With Previous Study

Lutum et al [20] has studied the effect of L/D on the effectiveness of a single film rows angled at 35deg. His results can be used to make a simple comparison between the centerline effectiveness values. Figures 69 and 70 show the centerline effectiveness values plotted for M values of 0.8 and 1.5 respectively. For the present study centerline effectiveness after the last row is plotted for comparison. From the figures it can be seen that the effectiveness values are more for the present studies for almost the same conditions, this is mainly because of the coverage effect and also may be because of the acceleration of the mainstream. The films exiting from the
previous rows add to the flow coming out of the last row and this enhancement can be seen in the effectiveness plots.

Figure 69: Present study vs. Lutum et al. [20] (M=0.8)

Figure 70: Present study vs. Lutum et al. [20] (M=1.5)
CHAPTER 7: CONCLUSION & SCOPE FOR FUTURE WORK

7.1 Conclusions

The following conclusions can be derived from the present work

- Two cylindrical film cooling hole configurations at angles 20° and 30° were tested for the effect of impingement on the coolant supply side, in the presence of pressure gradient in the mainstream side.

- Measurements of discharge coefficient and cooling effectiveness were made for both the configurations.

- The resulting trends for the effect of impingement on the film cooling effectiveness were compared.

- The current findings were compared with the existing literature for the effect of full coverage.

- The pressure ratios tested in this study are high enough for the $C_D$ to be considered to be constant.

- As expected the cooling effectiveness along the main flow increases as the flow builds up and also the blowing ratio gets lower. The last row with a single hole is the one to gain the most.

- The main effect of impingement is on the pressure distribution in the chamber between the jet and target plates, which affects the blowing ratios of the film hole rows.
The pressure at the downstream of the chamber is higher because there is more flow through the last rows of impingement. This is the main reason for the blowing ratio to decrease in the first rows and increase in the downstream rows.

It is seen in the Averaged lateral effectiveness ($\overline{\eta}$) that the effectiveness of the downstream rows decrease in the presence of the impingement through jet as compared to the $\overline{\eta}$ values for the direct impingement case.

**7.2 Scope for Future Work**

The cases studied in this research have a lot of scope for improvements and changes

- Different hole angles can be studied, in most of the gas turbines $35^\circ$ is the most commonly used angle for film cooling.
- The shape of the holes is yet another factor that can be changed and investigated.
- A different pressure gradient (like adverse gradient as opposed to favorable gradient used in this study) can be implemented and the effect it has on the film cooling effectiveness can be studied.
- The impingement jet configuration can be changed and its effect on the film effectiveness can be studied.
- The effect of the pressure gradient on the impingement heat transfer can be studied.
- The optimal Blowing rates of the film holes will be different in the presence of the pressure gradients. These blowing rates for different hole angles and various pressure gradients can be investigated.
REFERENCES


