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Enhanced Cold-Side Cooling Techniques for Lean Burn Combustor Liners

Graham Peacock
A Doctoral Thesis
Submitted in partial fulfilment of the requirements for the award of Doctor of Philosophy
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Executive Summary

In order to meet the increasingly strict emissions targets required in modern civil aviation, lean burn combustors are being pursued as a means to reduce the environmental impact of gas-turbine engines. By adopting a lean air/fuel mixture NO\textsubscript{x} production may be reduced. The increase in proportional amount of high pressure air entering directly into the combustor reduces the amount available for cooling of the combustor liner tiles. A reduced mass of air places restrictions on the porosity of cooling arrays, requiring a departure from applications of pedestal and slotted film cooling typically used to cool double skin combustor liners. An alternative approach applied to lean burn combustors places impingement and effusion arrays on the cold and hot skins respectively for cooling of both sides of the hot liner skin. Although impingement cooling is well established as a means of promoting forced convection cooling, there are many areas on a liner tile where cooling behaviour is not well characterised. Additionally, film cooling reduces combustive efficiency and increases the production of NO\textsubscript{x} and CO, prompting interest in reducing its use in combustor cooling.

The research for this thesis has focussed on investigations into current and proposed geometries to identify methods to enhance cold side cooling in lean burn applications. A fully modelled combustor liner tile has been used for investigation into the impact of structural and pressure blockages on cold side cooling performance of an impingement-effusion array using a transient liquid crystal technique to measure heat transfer performance. Research has found structural blockages can reduce heat transfer performance to \(~60\%\) of typical values, with crossflow development due to pressure blockage producing similar reductions in Nusselt values to \(\sim70\%\) of typical.

A second investigation explored enhanced cooling geometries combining a distributed impingement feed over roughened channels of pedestals at variable height \((H/D)\) and pitch \((P/D)\). A newly proposed ‘Shielded Impingement’ concept combines full height pedestals, to protect impingement jets from developing crossflow, with quarter height pedestals for turbulence enhancement of crossflow cooling. The research has found that Shielded Impingement geometries displayed the strongest cooling performance of all tested designs due primarily to increased downstream Nusselt numbers. Pressure losses were comparable to short pedestal geometries, with little apparent effect of full height pedestals. Low pressure losses mean that application to extended channels in line with the full tile geometry is possible.
Preface

The work detailed within this thesis was conducted over the period October 2006-December 2012 at Loughborough University. Funding for the work was provided by Rolls-Royce. This document contains approximately 69000 words and 160 figures and is the result of my own work except where indicated in the text.

Throughout the course of this work I have had the good fortune to have the assistance and guidance of many people at Loughborough University whose contributions deserve acknowledgement for helping me (eventually) complete this thesis. Thanks must first of all go to my supervisor, Dr Steven Thorpe for his support, guidance and of course patience throughout this work. Fellow student Thomas Melia for being my regular sounding board and for proving it is possible to write up in your spare time. I would like to thank technicians Neil Thorley and Les Monk for their work on the test facilities used for conducting the experiments; the mesh heater is a work of art. Finally I would like to thank everyone who has helped me get through the last six months to put this thesis together, particularly Steph for being my keeper and Mark Brend for his assistance in the eleventh hour.
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Nomenclature

\( A \) \hspace{1cm} \text{Area}
\( C_D \) \hspace{1cm} \text{Discharge coefficient}
\( C_{por} \) \hspace{1cm} \text{Porosity coefficient}
\( C_p \) \hspace{1cm} \text{Specific heat capacity (J/kgK)}
\( D \) \hspace{1cm} \text{Diameter}
\( f \) \hspace{1cm} \text{Loss Coefficient / Friction factor (}= \Delta p/2\rho U^2\))
\( h \) \hspace{1cm} \text{Heat transfer coefficient (W/m}^2\text{K)}
\( H/D \) \hspace{1cm} \text{Pedestal height to diameter ratio}
\( I \) \hspace{1cm} \text{Current}
\( k \) \hspace{1cm} \text{Thermal conductivity (W/mK)}
\( L/D \) \hspace{1cm} \text{Jet hole length to diameter ratio}
\( \dot{m} \) \hspace{1cm} \text{Mass-flow rate}
\( M \) \hspace{1cm} \text{Mach number}
\( Nu \) \hspace{1cm} \text{Nusselt number (}= hD/k\))
\( \bar{Nu} \) \hspace{1cm} \text{Average Nusselt number}
\( p \) \hspace{1cm} \text{Pressure}
\( P/D \) \hspace{1cm} \text{Pedestal pitch to diameter ratio}
\( Pr \) \hspace{1cm} \text{Prandtl Number}
\( \dot{q}'' \) \hspace{1cm} \text{Heat flux (W/m}^2\text{)}
\( t \) \hspace{1cm} \text{time}
\( T \) \hspace{1cm} \text{Temperature}
\( U \) \hspace{1cm} \text{Velocity}
\( V \) \hspace{1cm} \text{Voltage}
\( x \) \hspace{1cm} \text{Streamwise (axial) distance}
\( X/D \) \hspace{1cm} \text{Streamwise pitch to diameter ratio}
\( y \) \hspace{1cm} \text{Spanwise (circumferential) distance}
\( Y/D \) \hspace{1cm} \text{Spanwise pitch to diameter ratio}
\( z \) \hspace{1cm} \text{Height between hot and cold skins}
\( Z/D \) \hspace{1cm} \text{Jet to Target Spacing (Jet height to diameter ratio)}
Greek Letters

\[ \alpha \] Sonic velocity

\[ \beta = h\sqrt{\varepsilon}/\kappa \]

\[ \kappa \] Material Thermophysical properties \( (= \sqrt{\rho c k}) \)

\[ \mu \] Dynamic Viscosity \( (kg/ms) \)

\[ \nu \] Kinematic Viscosity \( (m^2/s) \)

\[ \tau \] Time-constant

\[ \rho \] Density

\[ \Theta \] Non-dimensional temperature \( (T_w - T_0)/(T_{aw} - T_0) \)

Often used subscripts

\[ 0 \] Ambient, baseline, initial

\[ aw \] Adiabatic wall

\[ c \] Crossflow

\[ e \] Entrance to working section

\[ f \] Film

\[ g \] Gas

\[ j \] Jet

\[ lc \] Liquid crystal

\[ m \] Mainstream

\[ mb \] Mixed bulk

\[ p \] Pedestal

\[ w \] Wall

\[ \infty \] Free-stream value

\[ chan \] Channel

\[ imp \] Impingement

\[ eff \] Effective

\[ equiv \] Equivalent
Chapter 1  Introduction

Initial applications of gas-turbine engines for aircraft propulsion date back to the 1930s, with parallel development by von Ohain in Germany and Whittle in England. A simple layout of a gas-turbine engine is illustrated in Figure 1.1 [1] in which the three primary sections of the engine can be seen: the compressor, combustion chamber and turbine stages. Through the compressor section, work is done to the fluid, with the pressure elevated as it passes through a series of rotational and stationary blades. In the combustion chamber, the compressed air is mixed with fuel and ignited, accelerating the fluid into the turbine. The hot gas jet emanating from the combustion chamber turns the turbine blades, which operate on the same shaft as the compressor. Through this, the power generated is used to drive the compressor. Modern engines typically utilise two shafts to which the low and high pressure turbines and compressors are attached.

Early engine designs were geared primarily towards military applications, with the original turbojet engines designed for operation at high-speed but experiencing short lifespan; poor reliability and high fuel consumption. Once the potential for use in civil aircraft was realised the turbojet was replaced by the turbofan, in which a portion of the air passing through the low pressure compressor bypasses the core of the engine and mixes with the hot jet stream exhaust from the turbine. This reduces the jet velocity and temperature, increasing the overall propulsive efficiency. Over the years, continual development of gas-turbine engines for civil aviation has been driven by a desire to increase performance and reduce operational costs through increased propulsive efficiency, power output and lifespan. The ideal thermodynamic cycle of gas-turbine engines consists of four stages illustrated in Figure 1.1 [1]: Isentropic compression (1-2’), constant pressure heat addition (2’-3) isentropic expansion (3-4’) and constant pressure heat rejection (4’-1). In practice the ideal conditions cannot be matched, with non-isentropic compression (1-2) and expansion (3-4), performed over the compressor and turbine stages respectively; while the heat addition within the combustion chamber is performed at near constant pressure. The efficiency of this cycle depends only on the pressure ratio and the nature of the gas, while the specific work depends also on the maximum cycle (turbine inlet) temperature [2]. By increasing the overall pressure ratio and the turbine entry temperature (TET), the thermal efficiency of the engine and the specific thrust may both be increased [3] and this has led to a gradual rise in overall pressure ratio and TETs over the years.
An undesirable consequence of high TET is an increased formation of pollutants, specifically oxides of Nitrogen (NOx). Due to regulations placed by bodies such as the ICAO on the allowable level of emissions, controlling the levels of pollutants produced by an engine is of vital importance to manufacturers. In addition to NOx, the other main products of incomplete combustion, Carbon Monoxide (CO) and unburned hydrocarbons (UHC) are also regulated. This has prompted a significant push in the consideration of emissions.

As regards NOx production, the flame temperature in the combustor has been identified as the single most important factor affecting its production, with the rate of formation having been shown to increase exponentially with temperature [2] as is illustrated in Figure 1.2 [4]. The most effective means of reducing NOx production would be to reduce flame temperature, as emissions have been shown to more than halve in response to a reduction of TET from 1900K to 1800K [2]. An alternative means of reducing NOx is to alter the air/fuel ratio. As shown in Figure 1.3 the concentrations of different pollutants may be altered through changes to the air/fuel mix. An increase or decrease in the air/fuel mix away from stoichiometric conditions can be seen to reduce NOx emissions but increase emissions of CO and UHC; however, the relative levels of variation are not equal. Thus operating at a leaner air/fuel ratio can significantly reduce overall pollutant production.

Aircraft gas turbine engines typically employ rich burn combustion systems, in which only a relatively small portion of the air (~30%) that has passed through the compressor stages enters into the combustion chamber, giving a rich air/fuel ratio which is desirable due to the higher specific temperature rise promoted than with leaner mixes [2]. For lean burn combustion to be implemented the proportion of compressed air entering into the combustion chamber is increased to ~70%, with the leaner air/fuel ratio reducing the level of NOx emissions with comparatively small effects on CO and UHC production.

1.1 Lean Burn Combustor Liner Cooling

Due to the high TET that is favoured in many gas turbine engines, temperatures in the combustion chamber can approach 2000K for civil applications. This is far in excess of the maximum allowable temperature of the Nickel alloys from which turbine blades and the combustor liner are manufactured and have been specifically selected for their high thermal resistance. As a result it is necessary to cool the combustor liner and turbine blades to a suitably low temperature to resist the associated heat fatigue effects such as creep and oxidation that impact upon the component durability.
1.1.1 Combustor Liner Cooling Current Practice

Gas-turbine combustors commonly feature double skin liner geometries in which an inner liner that acts as a heat shield is encased within an outer barrel that forms the structural shell of the combustor. The inner liner protects the outer, cold skin from the high temperatures occurring within the combustion chamber; however, due to the high temperatures within the combustion chamber it must be constantly cooled to avoid structural damage, reducing the life of the engine. Current methods of cooling combustor liners may be divided into hot and cold side cooling techniques. Hot side cooling methods involve injecting a stream of air into the combustion chamber that forms a protective film of cool air over the combustor wall. Typically in combustor liner applications, the films of coolant are fed through slots as illustrated in Figure 1.4. Cold side cooling methods involve promoting forced convection in order to increase turbulence and accordingly, the heat transfer rate in these regions. Forced convection methods involve the use of impingement jet arrays and pedestal/rib roughened passages. The former of these methods promotes turbulence by forcing the coolant through an array of small holes such that it enters the cavity between the hot and cold skins at highly elevated velocity, increasing turbulence, and accordingly heat transfer coefficient ($h$). In pedestal/rib roughened passages, protrusions are positioned within the channel that increase heat transfer rate by promoting increased turbulence and also increasing the surface area of the liner wall, thus presenting a greater area over which heat exchange may occur. Typical current practice for cooling a combustor liner utilises a combination of hot and cold side cooling methods, in which coolant is fed into the cavity ‘cooling channel’ between the hot and cold skins through small numbers of large impingement holes. The coolant is then passed through arrangements of pedestals for cold side cooling and injected into the main combustion chamber through large slots. An illustration of this typical current practice is shown in Figure 1.5.

1.1.2 Complications for Cooling Lean Burn Combustor Liners

In order to achieve the relatively low air/fuel ratio in traditional rich burn combustion a high proportion (~70%) of the air that has passed through the high pressure compressor bypasses the fuel injector with the remainder of the fluid (~30%) being used for primary combustion. In lean burn combustion this ratio is switched, with ~70% entering the fuel injector. This leaves ~30% remaining, more than halving the amount of air available for use as coolant. The smaller available mass of air necessitates a change from current practice,
with a reduced porosity cooling array required. This has led to a change of arrangement for lean burn combustion systems with the combination of pedestals and slots illustrated in Figure 1.5 most commonly replaced by arrays of impingement and effusion holes for cold side forced convection cooling and hot side film cooling respectively [5]. Such a configuration has been used by Rolls-Royce PLC on the lean burn Affordable Near-Term Low Emissions (ANTLE) engine [6]. Impingement cooling has very high potential to improve heat transfer performance but typically results in large spatial variations in $h$, falling with increasing distance from impingement location. Impingement jets in large arrays are also severely weakened by crossflow [7] within the cavity between hot and cold skins, perpendicular to the impingement jet direction. Crossflow results from accumulation of spent jet air migrating towards the exit of a channel although the presence of effusion holes in the target plate should eliminate this.

The application of an impingement-effusion strategy to lean burn, low NOx combustor liner cooling has several complications. Using a double skin arrangement for combustor lines typically requires the use of structural components to connect the inner and outer skins. Cooling of these components is essential to maintaining the life span of the component and with an impingement-effusion cooling arrangement, cooling of these components requires dedicated jets. Typical arrays of impingement jets use circular holes, arranged normal to the target surface and with film cooling holes present on the target plate much flow quickly exits through the hot skin. Accordingly, for fastening studs to be cooled requires the use of dedicated, angled impingement jets directly targeting them, although angled jets have been reported as providing weaker cooling performance than normal jets [8]. In traditional, rich burn cooling arrangements of pedestals, the flow of the coolant through the channel acts to cool the fastening components in the same fashion as heat is transferred from the pedestals. An additional problem presented by the presence of fastening studs for impingement jet cooling is that a reduced jet presence in the vicinity of the fasteners can negatively impact upon surface cooling performance.

1.1.3 Impingement over Roughened Channels

Although typical cooling arrangements for application to lean burn combustor liners involve the use of impingement-effusion arrays, alternative arrangements have been considered in which the role of film cooling is reduced. This is in response to reports of film cooling having an adverse effect on the production of NOx and CO emissions [9]. In the
absence of film cooling, significantly improved cold side cooling is required if adequate cooling of the liner tile is to be achieved. Due to the extremely high temperatures within the combustion chamber, the complete removal of film cooling may not be achievable; however, a reduction in the amount of coolant injected into the combustion chamber may be achieved.

A main area of investigation for improved cold side cooling performance is the use of impingement jet cooling in which the jets feed into channels roughened with pedestals or ribs. Various studies into such geometries have identified a potential for improved performance over traditional arrays of impingement jets into smooth passages, but at generally only for high crossflow cases and typically at significantly increased pressure loss.

### 1.2 Format of Thesis

For this thesis an experimental investigation is conducted into providing enhanced cold-side cooling of double skin combustor liners for application to a Rolls-Royce Environmentally Friendly Engine (EFE) illustrated in Figure 1.6. To achieve this aim, two main lines of investigation were followed for this thesis:

1. An evaluation of the cooling performance of an impingement-effusion cooling array applied to a fully representative model of a lean burn EFE combustor liner tile.

2. An investigating into enhanced cold side-cooling performance in the absence of effusion cooling for a geometry representative of the fully modelled tile.

Chapter two reports the findings of a review into published literature, primarily focussing on different cooling methods applicable to combustor liners with a review also conducted of different measurement techniques. Chapter three gives an expanded review of the adopted transient liquid crystal measurement technique. Chapters four, five and six detail the design of the experimental facilities for the two main lines of investigation that were followed for this thesis. The measurement and analysis techniques used for both experimental investigations are described in Chapter four and the design of the two test facilities are described in Chapters five and six. Chapters seven and eight detail the experimental results from the two test facilities, whilst in Chapter nine a comparison of the results from the two branches of investigation is conducted. The conclusions of this thesis are reported in Chapter ten.
1.3 Key parameters for scaling of heat transfer experiments

The investigation into enhanced cold-side heat transfer performance conducted for this thesis involves extensive experimental measurements characterising aerodynamic and heat transfer performance of different double-skin cold side cooling concepts. For the high temperature and pressure conditions occurring within a gas-turbine combustor to be recreated under laboratory conditions would be highly costly and would also present a very demanding measurement environment. As such, experiments have been developed for operation at near atmospheric conditions, for which aerodynamic and thermal conditions from the engine application must be accurately scaled. Assuming there is geometric similarity between engine and model arrangements, equivalent conditions may be modelled by matching relevant dimensionless groups. Aerodynamic and heat transfer behaviour depends on the viscosity \( \mu \); specific heat capacity \( C_p \); thermal conductivity \( k \); density \( \rho \) and velocity \( U \) of the fluid; the absolute and relative temperatures of the fluid and heat transfer surfaces \( T_g, T_w \) and \( \Delta T \) and also the size of the model. In addition, the wall heat-flux \( q'' \) and heat transfer coefficient \( h \) govern heat transfer between the model and the fluid. To ensure that conditions in the experimental models match those present within the combustor these listed variables may be arranged into dimensionless parameters that if all are matched indicate equivalent conditions. Various such parameters have been used for establishing appropriate scaling of experimental conditions and as means of comparing measurements. As these terms will be referred to throughout this thesis a brief introduction of each is included here, outlining their definition and relevance to the work.

**Reynolds number**

Reynolds number \( (Re) \) is the ratio of inertial to viscous forces and is a vital similarity parameter for correctly scaling the viscosity of a flow.

\[
Re = \frac{\rho U x}{\mu}
\]  

Maintaining consistent Reynolds numbers in aerodynamic scaling is vital for correct modelling of viscous flow behaviour, as the resulting boundary layer has a strong effect on heat transfer performance. Although the degree of influence varies with geometry, the presence of a power relationship between Reynolds and Nusselt number \( (Nu, \text{ for definition see below}) \) is well acknowledged and correlations relating \( Nu \) and \( Re \) have been derived for a variety of different applications. One such example correlation relating \( Nu \) to \( Re \) and \( Pr \) is
the Dittus-Boelter equation for fully developed turbulent flow inside a smooth pipe, valid for \( Re \geq 10^4 \).

\[
Nu = 0.023Re^{0.8}Pr^n
\]  

(1.2)

The value of \( n \) in this equation depends on the direction of heat transfer; if the wall is being heated \( n = 0.3 \), if the fluid is being heated \( n = 0.4 \). Additionally the correlation between \( Nu \) and \( Re \) is different for laminar and turbulent flows with \( Nu \propto 0.5 \) in the case of laminar flow. Relationships specific to particular cooling arrangements and experimental conditions have been developed by various researchers. These may be used to draw comparisons against experimental results and are discussed later in the thesis.

**Prandtl number**

Prandtl number (\( Pr \)) is the ratio of fluid momentum and thermal diffusivities.

\[
Pr = \frac{c_p \mu}{k}
\]  

(1.3)

The components of \( Pr \) all increase with temperature; however, due to the nature of these variations the resultant impact of temperature on \( Pr \) is small for most gases. Using the \( Pr \) of air as an example, the value decreases from 0.715 at 0ºC to 0.68 at 400ºC, only a 5% reduction over a temperature increase of 400ºC. The value of \( Pr \) is similarly unaffected by variations in pressure. Prandtl number is frequently a component of correlations relating heat transfer performance to different flow conditions, with the previously mentioned Dittus-Boelter equation, but one example. The stability of Prandtl number with changes in pressure and temperature meant that no special consideration was required for it to be matched when scaling experimental facilities.

**Mach number**

Mach number (\( M \)); the ratio between the flow velocity and the speed of sound, is important for scaling compressibility of high velocity flows.

\[
M = \frac{U}{\alpha}
\]  

(1.4)

With high speed flows, compression of the fluid occurs around the object increasing the flow density. If the relative speed between the object and the fluid is low, typically less than \( M=0.3 \) then the density of the fluid remains constant. Engine conditions provided for the EFE combustor indicated velocities would all fall below this level; consequently holding a
fixed Mach number is not necessary for correct scaling so long as velocities within the experimental facilities remain lower than the threshold value of $M=0.3$.

**Nusselt Number**

When measuring heat transfer performance it is necessary to present the data in a non-dimensional form such that it may be used for comparisons with engine conditions as well as other investigations. The heat transfer coefficient ($h$) between a wall and the surrounding fluid, obtained from Newton’s Law of Cooling, equation (1.5) is used for this purpose.

$$h = \frac{q''}{(T_\infty - T_w)}$$  \hspace{1cm} (1.5)

The heat transfer coefficient relates the heat flux to the temperature difference between the heat transfer surface and the fluid with which it is in contact. Using experimental temperature measurements to evaluate $h$ allows measurements of heat transfer performance from different applications and experiments to be compared, whether under ambient or engine conditions and for experiments. The nature of the heat transfer coefficient is such, that its value is unaffected by the direction of heat flow, having the same value whether heat is transferred to or from the fluid. Heat transfer coefficient is not non-dimensional, possessing units of $W/(m^2K)$. As such its value varies with temperature difference and size of area under consideration. The heat transfer coefficient is of importance however as it is a component of the non-dimensional parameter, Nusselt number ($Nu$) typically used to measure heat transfer performance. This is the ratio of convective to conductive heat transfer coefficient and is related to $h$ by the equation:

$$Nu_x = \frac{hx}{k_g}$$ \hspace{1cm} (1.6)

$Nu$ is calculated using the characteristic length, which throughout this thesis is defined as the diameter of impingement holes or pedestals. Experimental measurements if heat transfer coefficients for this thesis have been converted to Nusselt numbers in order to enable comparison of results between different test facilities. Additionally, average Nusselt numbers ($\bar{Nu}$) may be obtained by integrating the values over a particular area, typically used to span or area averaged values.
**Biot Number**

Biot number (\(Bi\)) describes the boundary condition for thermal conduction in a solid body. The definition of \(Bi\) shown in Equation (1.7) can be seen to take a similar form to that of \(Nu\) (Equation (1.6)); however, the thermal conductivity (\(k_w\)) used to calculate \(Bi\) is that of the solid, whereas it is the fluid thermal conductivity (\(k_g\)) that is used to calculate \(Nu\).

\[
Bi = \frac{hx}{k_w}
\]  

(1.7)

\(Bi\) is a measure of the relative resistance to heat flow of the inside of the solid to that of the adjacent fluid [4]. A small \(Bi\) indicates that resistance to thermal conduction in the body is significantly smaller than the heat transfer resistance at its boundary with the reverse true for high values of \(Bi\). The techniques used in this thesis to measure \(h\) (detailed in Chapter 3) employ assumptions of one-dimensional heat transfer to a semi-infinite wall. To maximise the accuracy of these assumptions a large \(Bi\) is desirable. This may be achieved through selection of a material with low thermal conductivity for the heat transfer surface.
1.4 Figures

Figure 1.1: Illustration of a simple gas turbine system and basic gas turbine cycle indicating non-isentropic compression and expansion [1]

Figure 1.2: Exponential variation of NOx production with increasing combustion flame temperature
Figure 1.3: Effect of varying fuel/air ratio on production of NOx, CO and UHC

Figure 1.4: Conventional combustor cooling arrangement incorporating double skin geometry and slot cooling [10]
Figure 1.5: Traditional pedestal arrangement as applied to double-skin combustor liners.

Figure 1.6: General arrangement of EFE lean burn combustion chamber.
Chapter 2  Literature Review

Gas-turbine combustor liners commonly feature a double skin arrangement in which an outer barrel forms the structural shell of the combustor, encasing an inner liner that acts as a heat shield, protecting the outer, cold skin from the high temperatures occurring within the combustion chamber. In order to maintain the life-span of these components and contain the hot gases, various approaches have been employed for cooling the hot skin of the liner, with specific methods targeting both the hot and cold sides of the liner. Methods focussed on cooling the hot-side of the inner liner typically focus on shielding the skin from the hot temperatures resulting from combustion, typically by projecting a film of cool air along the wall. Cold-side methods focus on enhancing turbulence of the coolant between the hot and cold skins, thus improving the rate of heat pickup from the liner by increasing the heat transfer coefficient. This thesis targets analyses of the heat transfer performance of various different approaches to cooling combustor liners. One branch of investigation considers the impact on heat transfer performance of aerodynamic and structural blockages for a cooling geometry representative of those encountered in current Rolls-Royce gas-turbine engines. The second branch of investigation targets the design of original, enhanced cooling methods in which no hot side cooling is employed. In order to inform the design of the experimental facilities required by the two investigations, a review was conducted of published literature investigating various relevant cooling methods. In addition to researching different cooling approaches, a review of different measurement techniques was also conducted. The findings of the literature review are included in this Chapter.

2.1 Cooling Methods

The focus of this thesis into enhanced cooling techniques appropriate for cooling the combustor liner within a lean burn gas-turbine engine, is directed at improving cold-side heat transfer performance, specifically the cold side of the hot skin for combustors featuring a double-skin arrangement. Cooling methods in the combustor liner are similar to those adopted for turbine blade applications, with typical forced convection methods featuring impingement jet cooling, or pedestal/rib roughened passages to augment heat transfer cold-side heat transfer. Cold-side cooling techniques are generally used in combination with film cooling in which the hot side of the liner is cooled. This involves the spent air being fed through slots or holes and ejected onto the hot surface. As was discussed in Chapter 1, significantly less fluid is available for cooling lean burn combustion systems than in rich burn
engines, due to the amount of air from the high pressure compressor which bypasses the combustion chamber and is thus available for cooling purposes being reduced from ~70% in rich burn applications to ~30% in lean burn. In order to ensure adequate cooling, this reduced supply must be used more efficiently. For this thesis investigations are conducted into the performance of a traditional cooling array adapted for lean burn conditions, and also original patterns that provide improved performance in the absence of film-cooling. Research into different cooling techniques and the parameters that influence performance was conducted to provide sufficient knowledge to perform this task and an overview of the relevant findings is reported in this Chapter.

2.1.1 Film Cooling

Although the investigations into liner cooling for this thesis are concerned exclusively with cold-side cooling performance, an understanding of hot-side cooling methods is required. Direct cooling of the hot-side of the inner liner is fundamental to many cooling strategies; however, one branch of investigation for this thesis is into the possible application of zero-effusion cooling arrangements. As such it is necessary to understand the traditional role of film cooling and the implications of operating without it, both positive and negative.

Film cooling protects the surface of gas-turbine components from exposure to hot gas conditions by introducing a secondary, coolant fluid through holes or slots positioned at various locations along the component surface. The coolant forms a protective film, as illustrated in Figure 2.1, reducing the gas temperature at the wall and the resulting thermal loading on the surface. For a theoretical scenario in which the cooling film remains fully attached to the surface, with no dispersion into the mainstream, the film temperature, $T_f$ would have equal temperature to that of the coolant, $T_c$. The performance of film cooling is often related to this ideal case using equation (2.1) as the definition of film effectiveness, $\eta$. A maximum effectiveness of 1 would represent no mixing and all the coolant remaining close to the surface. The general case however, is of rapid mixing of coolant with the hotter mainstream resulting in a larger value of $T_f$ than $T_c$, reducing the effectiveness below 1.

$$\eta = \frac{(T_m - T_f)}{(T_m - T_c)} \quad (2.1)$$

The performance of film cooling is governed by the degree to which the surface heat load is reduced, characterised by the ratio of heat flux with and without film cooling $\dot{q}'' / \dot{q}_0''$ (net heat flux reduction). This may be expressed as shown in equation (2.2) [11]. Equation
(2.2) expresses net heat flux reduction in terms of the film effectiveness and heat transfer coefficients with and without film cooling. Film cooling is quantified in terms of these parameters, and a full understanding of the performance requires knowledge of all three [12].

\[
\frac{\dot{q}''}{\dot{q}_0'} = \frac{h}{h_0} \left( 1 - \eta \frac{T_m - T_c}{T_m - T_w} \right)
\]  

(2.2)

Injecting cooling jets through the liner and into the mainstream flow generally increases turbulence levels within the boundary layer [13]. Increasing the turbulence in such a manner also increases the degree of mixing of the coolant with the hotter gases within the combustion chamber. Consequently, the heat transfer coefficient increases above that occurring in the absence of film cooling. From equation (2.2) it can be seen that this has a negative impact on the net heat flux reduction; however, through high effectiveness and temperature ratios, an overall reduction in heat flux may be achieved. Although film cooling can significantly reduce the liner wall temperature, injecting coolant into the combustion chamber can have a negative influence on emissions and combustion efficiency as ejecting coolant into the combustion chamber acts to worsen the radial temperature profile at combustor outlet [10], with this effect enhanced by the increased mixing of the cooling and main-stream flow. Additionally, experimental results from combustor tests conducted by Rolls-Royce PLC indicated an increase in combustive efficiency and a reduction in NOx output in response to a reduction in the number of effusion holes [6].

2.1.2 Pedestal and Dimple Cooling Performance

Positioning cylindrical pins (pedestals) within a channel can increase the rate of heat transfer in two ways relative to that of a plain, smooth channel. By presenting an obstruction to the flow, a wake is produced that increases turbulence of the downstream flow, increasing the local heat transfer coefficient and influencing the flow and heat transfer of the downstream pins. The use of pins also improves heat transfer by increasing the surface area in contact with the coolant and is a method used by both Pratt and Whitney and Rolls-Royce in combustor liners and turbine blades.

The heat transfer performance of pedestal arrays has been shown to be influenced by the pitch-to-diameter (P/D) ratio, with changes to the streamwise (X/D) and spanwise (Y/D) spacing observed to have different effects. An investigation by Metzger et al. [14] into equilaterally spaced pedestal arrays identified an increase in $\bar{Nu}$ for smaller values of $P/D$, with these findings supported by the findings of Metzger and Haley [15]. Further
investigation into the independent effects of streamwise and spanwise spacings by Metzger et al. [16] identified that $\overline{Nu}$ increased with a reduction in either $X/D$ or $Y/D$, but with a stronger trend with streamwise spacing. It was however found that the independent effects of each on pressure loss were different, with friction factor found to be more dependent on $Y/D$ than $X/D$. The influence of height-to-diameter ($H/D$) of pedestals has been investigated by Chyu et al. [17] who identified that overall array-averaged heat transfer increases with $H/D$, with most of the contribution to heat transfer coming from the pedestals for $H/D > 2$. In addition to the improved heat transfer performance an increase in $H/D$ was found to result in increased pressure losses.

As the heat transfer augmentation provided by pedestal arrays is a function of heat transfer from both the end-wall and pedestals the relative contributions of each is of interest. Investigations into the heat transfer performance of pedestal arrays have consistently indicated $Nu$ displayed by the pedestals ($Nu_p$) to be larger than that measured for the end-wall ($Nu_w$); however the difference in levels has been found to be inconsistent across studies [16]. Independent investigations of pedestal arrays at $P/D = 2.5$ and $H/D = 1$ by Chyu et al. [18] and Metzger et al. [16] found $Nu_p$ to be ~20% and ~100% higher than $Nu_w$ respectively. For the larger $H/D$ values investigated by Chyu et al. [17] $Nu_p$ was seen to increase further with respect to $Nu_w$ to levels ~30-70% higher. An investigation into the effect $P/D$ on the ratio of $Nu_p$ to $Nu_w$ by Lyall et al. [19] indicated an increased dominance of pedestal heat transfer for larger values of $P/D$; however, this appeared to be due primarily to an increase in end-wall heat transfer for closer pedestal spacing against relatively stable values of $Nu_p$.

Studies have been carried out into both full and partial channel height pedestals; with a similar approach also adopted in which rounded dimples were recessed into the plate. A review of different cooling methods by Ligrani et al. [20] reviewed the relative heat transfer and pressure loss characteristics of different arrays. In terms of heat transfer performance, it was shown that arrangements of dimples increase $\overline{Nu}$ by a smaller amount than for arrays of full height pins. Collated data showed $\overline{Nu}/Nu_0$ ranging from 2.0 to 3.8 for full height, circular pedestals [21] [22]; but only from 1.8 to 2.8 for dimples [23-28]. The dimpled arrangements were however, shown to have the advantage of producing significantly lower pressure losses. Investigations into geometries featuring arrays of pedestals with tip clearance [29] identified a reduction in levels of $\overline{Nu}/Nu_0$ for increasing distance between the
tip and the upper wall, but with friction factors reducing in line with $\frac{\bar{N}u}{Nu_0}$. A separate investigation into the impact on heat transfer performance of the pedestals themselves indicated similar reductions in pressure losses, but identified $Nu$ as being much more stable, indicating potential benefits of adding tip clearance. For a comparison of overall performance, Ligrani et al. [20] considered the amount of heat transfer augmentation, $\frac{\bar{N}u}{Nu_0}$ produced by each geometry and offset it against the additional pressure loss impaired, expressed in terms of friction factor $(f / f_0)$ to evaluate performance in terms of $\frac{\bar{N}u}{Nu_0} / (f / f_0)$, referred to as the Reynolds analogy performance parameter. This performance parameter was used to compare the different geometries, with the accumulated data compared in Figure 2.2. This shows that the highest values of $\frac{\bar{N}u}{Nu_0} / (f / f_0)$ were seen for geometries featuring dimples on one wall of a channel with values up to 1.8. The lowest values were shown to typically occur for pedestal geometries, with typical values ranging from $<0.1 \rightarrow 0.3$. This performance parameter represents a good measure of performance for applications in which the available pressure loss is restricted.

### 2.1.3 Rib Turbulated Cooling

Rib turbulated cooling works on the same principles as pin fin cooling, by promoting turbulence and increasing the surface area in contact with the coolant. Rib turbulators generally promote a greater increase in heat transfer coefficient than pedestals [11] but present a greater obstruction to the flow and are associated with greater pressure losses. When compared using the performance parameter in Figure 2.2 it can be seen that rib turbulators generally perform slightly better for a given pressure loss [20]. This is reflected in engine applications, where for channel cooling, ribs are generally preferred if space is available with pins being reserved for tighter passages where ribs cannot be placed. Various studies have shown rib cooling performance is heavily influenced by channel aspect ratio [30] and rib pitch [31]. The orientation of the ribs whether perpendicular or angled to the flow has also been shown to have a large influence on both heat transfer augmentation and associated pressure losses [32].

### 2.2 Jet Impingement Cooling

In jet impingement cooling, fluid is drawn through an array of holes creating jets with momentum significantly higher than the surrounding flow above the target plate. The high momentum jets impinge upon a hot surface creating a highly turbulent flow over the
Figure 2.3 [33] shows the stagnation region and boundary layer development resulting from a single impingement jet. As can be seen, the coolant initially flows as a free jet creating a stagnation region directly below the jet hole. After reaching the target plate the jet behaves similarly to a wall jet with a boundary layer developing radially from the point of impingement. It can be seen that the boundary layer velocity profile differs from a typical viscous boundary layer as a point of inflection occurs. This produces a high near-wall thin boundary layer velocity that combined with turbulent mixing increases the heat transfer coefficient significantly. The resulting flow structure produces a high level of heat transfer augmentation in the near jet region, but with a rapid deterioration with radial distance from jet centre. Jet impingement cooling is an effective, but uneven, means of increasing local heat transfer coefficient that experiences large variations in heat transfer coefficient over the surface that potentially lead to over and under-cooling in particular locations. Additionally, in large arrays of jets; spent air builds up and results in mean channel flow that acts to divert downstream jets and reduce performance.

A wide range of research has been carried out into improving the overall average cooling performance of impingement arrays. As with all turbulence enhancing cooling methods, jet impingement cooling is highly dependent on $Re$. Individual studies generally look at altering one of numerous contributing geometric parameters and analysing performance over a range of Reynolds numbers. Characteristics include the distance between individual jets; plate separation; hole shape and angle to target plate. Additionally, the effect of crossflow upon jet performance has been frequently studied. Findings have highlighted a strong interaction between different characteristics on performance.

### 2.2.1 Jet Discharge Coefficient

The behaviour of flow within an impingement jet hole is characterised by its geometry and the flow conditions around its inlet and exit. Fluid entering sharp edged holes has been shown to contract, producing a vena contracta as shown in Figure 2.4. This flow contraction results the hole presenting an effective area smaller than the geometric open area, defined by the discharge coefficient ($C_D$). This parameter is important as a change in effective area can affect the mass flow rate through a hole, which can impact upon the porosity of a cooling array.

$$A_{eff} = C_D A$$  \hspace{1cm} (2.3)
Hole geometry has been shown to have a large influence on \( C_D \), with the \( L/D \) being a key parameter. As is illustrated in Figure 2.4, for short holes, the flow contraction continues beyond the exit of the hole, but as \( L/D \) increases, the vena contracta occurs within the hole and flow commences to diverge before the exit, leading to the discharge coefficient increasing. As \( L/D \) is increased further flow within the hole continues to expand from the throat and for \( L/D > 2.0 \) the flow has been shown to reattach to the edges of the hole for a measure of flow recovery [34]. A maximum \( C_D \) has been identified as occurring at \( L/D = 2 \) with a gradual, linear reduction occurring as \( L/D \) increases [35]. Due to its effect on the flow pattern occurring within the hole, Lichtarowicz et al. showed the length-to-diameter ratio to have a significant effect on the \( C_D \) of short holes, with similar results derived by McGreehan and Schotsch [36]. Lichtarowicz suggested that for \( L/D > 1.5 \) the strong dependence on \( C_D \) is avoided. In addition to the impact of hole geometry, \( C_D \) has been shown to be affected by the up and downstream conditions and the flow through it, with \( Re \) and the pressure ratio across a hole both having been shown to affect its value. In terms of nozzle \( Re \), Lichtarowicz et al. [35] identified its effect to be negligible for \( Re > 2 \times 10^4 \), this relationship was also observed by Florschuetz et al. [37]. For value of \( Re \) below \( 2 \times 10^4 \) the discharge coefficient is variable, increasing with \( Re \) and gradually stabilising as it approaches \( Re = 2 \times 10^4 \). Florschuetz et al. also identified that low levels of crossflow (\( G_c/G_j < 0.5 \)) in the channel to which the jet flow is injected has no influence on the orifice discharge coefficient. It was shown however that \( C_D \) decreases as the level of crossflow rises.

2.2.2 Jet-to-Target Plate Spacing

The jet-to-target spacing (\( Z/D \)) has been shown to be a contributing factor to performance of an impingement jet array. Several studies have observed a \( Z/D \) of 1 to be the approximate spacing for maximum average heat transfer performance in arrays of circular air jets [38-39]. Altering \( Z/D \) has been shown to alter the profile as well as the magnitude of the surface \( Nu \) resulting from the jet. For \( Z/D = 1 \), Huber and Viskanta observed secondary peaks occurring away from the jet centre [38] at \( X/D \approx 0.5 \) and \( X/D \approx 1.6 \). The inner peak was attributed to fluid accelerating from the stagnation region, and to turbulence generated by the shear layer around the jet circumference. The outer peak has been shown to be the result of boundary layer transition to turbulent flow. A review of similar studies [33] [40-41] indicated that both secondary peaks observed by Huber and Viskanta became less pronounced with increased \( Z/D \). While similar behaviour was observed by Cho and Rhee
it was found that for $Z/D \geq 2$ $Nu$ in the stagnation region increased but with the secondary peaks reducing. For $Z/D > 6$ $Nu$ in the stagnation region was observed to start declining, leading to an overall reduction in $\overline{Nu}$. Overall, smaller values of $Z/D$ were observed to produce higher averaged heat transfer performance; however, the benefit of this behaviour was questioned as improvements were observed primarily in the near-jet region with $Nu$ at the mid-point between jets significantly lower for values of $Z/D < 1$. A study by Viskanta [33] into jet behaviour for very small jet-to-target spacings ($Z/D<1$) supported the findings regarding the secondary peaks, indicating that the prominence of the outer secondary peak increased as separation was reduced. It was also observed that boundary layer transition occurred earlier for a smaller $Z/D$ spacing, with the outer peak moving closer to the stagnation point and absorbing the inner peak.

### 2.2.3 Jet-to-Jet Spacing

The cooling performance of impingement jet arrays is largely dependent on the jet-to-jet spacing ($X/D$) due to the mutual interaction of the flow from neighbouring jets, shown in Figure 2.5 [42]. It has been reported that the performance of an individual jet is greater than that for a given jet of equivalent geometry in an array [38] [43]. This reduction in heat transfer performance is due to jet interaction between neighbouring jets, occurring due to interference of the flow before impingement and to the collision of wall-jets, producing a fountain region as illustrated in Figure 2.5. An investigation into this behaviour was carried out by San and Lai [44]. It was shown that for small $X/D$ arrays, interaction is dominated by interference of adjacent jets before impingement, whilst the jet fountain results from interaction of wall jets in arrays with larger jet spacing. It has been also been shown that $Z/D$ influences the degree of jet interaction within an array, with larger separations allowing a greater degree of interaction. Huber and Viskanta’s study [38] shows that closer jet spacing reduces the cooling performance of the individual jets within an array due to increased interaction. It was; however, also shown that the close jet proximity means that $\overline{Nu}$ is highest for small $X/D$. For a constant $Re$ to be achieved at a denser jet spacing an increase in mass flow rate of air would be required. In applications where the supply of coolant is limited, as with lean-burn combustion, it is not practical to have closely packed jets and efforts to optimise the performance of jets within the array is desirable. Jet-to-jet spacing has been shown to not only affect local heat transfer performance but also affect crossflow development. In an investigation into the effect of $X/D$ upon performance in extended
channels, Goodro et al. [45] observed that for arrays of $X/D$ of 8 and 12, spatially $\overline{Nu}$ generally decreased along the channel length for the denser jet array, but remained roughly constant for the $X/D=12$ case. This indicates that a sparser jet arrangement may aid in combating crossflow development.

2.2.4 Crossflow of Spent Jet Air

Crossflow occurs in impingement jet arrays in the space between the jet holes and the surface plate as air from spent impingement jets propagates towards the channel exit. With sufficient strength a crossflow is capable of deflecting a jet away from the impingement surface, thereby reducing its cooling performance. Crossflow is often considered to be the primary contributor to reduced performance in large arrays, particularly in the downstream region [7]. Factors affecting crossflow build-up have consequently been widely studied. Figure 2.6 illustrates how crossflow develops within an impingement jet array. A beneficial aspect of crossflow can be seen in the turbine blade example, where a pin fin array is positioned downstream of the impingement jets to take advantage of the crossflow.

The method by which spent impingement jets exit the cooling channel has a significant effect on crossflow development, and consequently the performance of the array. Huang et al. [46] investigated the effect of various flow orientations in which all the spent air exited from the ends of the cooling channel. Ekkad et al. [47] employed a similar scenario but with effusion holes added to allow flow through the target plate. The orientations for these investigations are shown in Figure 2.7 with the heat transfer measurements of the different orientations, expressed as span-averaged $\overline{Nu}$ shown in Figure 2.8. The findings of both studies indicated that orientation 2 provided improved cooling compared with orientations 1 and 3. This behaviour was attributed to the smaller crossflow that came from allowing the spent air to exit the channel from both sides. Ekkad et al. observed that the presence of effusion holes resulted in a large reduction in $\overline{Nu}$, of order 20%, for orientation 2. It was theorised that this effect resulted from the effusion holes enabling a portion of the air to exit directly through the plate, impairing impingement jet performance due to premature removal of the coolant from the channel. The same reduction in cooling performance was not seen with the other arrangements and similar values of $\overline{Nu}$ were observed for orientations 1 and 3 as in the case without effusion holes. This consistency of heat transfer performance was attributed to the local removal of coolant through effusion holes impairing impingement performance as with orientation 2, but with this being offset by the retardation of the
crossflow, reducing its negative impact on jet performance. A study by Hollworth and Dagan [48] into the effect of jet alignment in impingement/effusion arrays upon cold side performance supported this theory. Experimental results showed that configurations of impingement jets in-line with an effusion array had weaker performance than that of a staggered array. This was mainly attributed to flow from the impingement jets positioned directly above the effusion holes bypassing the impingement plate. Whilst this effect is less pronounced with the staggered array the impingement performance is still impaired.

2.2.5 Hole Shape and Inclination

The majority of impingement jet arrays employ circular holes angled normal to the heat transfer surface. A major reason for this is ease of manufacture, as changes in shape and angle are much more complex to produce. The effect of different hole shapes and orientations has been investigated widely and been found to have significant effects upon the heat transfer performance of jets, with the effects found to be highly dependent on jet-to-target spacing. The influence of $Z/D$ was observed to become more pronounced with increasing $Re$ [49]. The magnitude of stagnation region $Nu$ values ($Nu$ at the jet touchdown location) has been observed for $Z/D \leq 6$ for rectangular [50], elliptical [51-53], hyperbolic [49], and lobed-holes [54]. The influence of the hole geometry was found to increase for very close plate separation with the largest effects observed in the case $Z/D \leq 1$. It was observed by Gulati et al. [50] that the nozzle profile not only affected $Nu$, but also the associated pressure loss. For the same $Re$, the circular hole produced a smaller pressure loss than the square and rectangular holes also investigated. This was consistent over the entire investigated $Z/D$ range (0.5-12). A final observation from the various studies was that of the different geometries investigated, the square nozzle exhibited the most similar behaviour to the circular jet, suggesting axi-symmetry may be a factor in jet performance.

A study into the performance of $\pm 45^\circ$ jets by Ekkad et al. [8] observed that for angled jets overall $\overline{Nu}$ was 12-20\% lower than for normal jet arrays, with performance dependent on jet direction. Jets angled in the same direction as the inlet flow exhibited elevated $Nu$ numbers to opposing jets, regardless of exit direction from the cooling channel. Although an overall reduction was observed it was also found that $Nu$ distributions were more uniform for angled jets than for orthogonal jets.
2.2.6 Correlations of Impingement Jet Performance

Much work has been conducted over the years to establish correlations predicting the heat transfer performance of impingement jets based upon various governing parameters. One such correlation has been derived by Goldstein et al. [55] for surface Nusselt numbers occurring beneath a single impingement jet. Goldstein’s correlation, equation (2.4) expresses surface Nusselt number as a function of the jet Reynolds number and the jet-to-target spacing. The variation of $Nu$ with increasing radial distance from the jet centre is also modelled through this correlation.

$$\frac{Nu}{Re^{0.76}} = \frac{(24 - |Z/D - 7.75|)}{533 + 44(R/D)^{1.285}}$$  \hspace{1cm} (2.4)

Due to the typical application of impingement jet cooling involving large arrays of jet holes and considering that individual jets have been reported to exhibit greater heat transfer performance than equivalent jets in an array [38] [43] correlations of performance for a single jet are of limited use. Due to the number of parameters that have been identified as influencing the performance of impingement arrays, including jet-to-jet spacing ($X/D$ and $Y/D$); jet-to-target spacing ($Z/D$) and the level of crossflow present; such correlations are difficult to derive. One widely regarded correlation has been developed by Florschuetz et al. [56] for modelling the performance of impingement jet arrays issuing into channels with developing crossflow. A range of variables and coefficients are used to calculate surface Nusselt number. Florschuetz et al.’s correlation is displayed in equation (2.5). The coefficients, $A$ and $B$ and the exponents, $m$ and $n$ in equation (2.5) are dependent on the geometric parameters, $X/D$, $Y/D$ and $Z/D$ and are calculated through equation (2.6). The coefficient $C$ and exponents, $n_x$, $n_y$ and $n_z$, derived by Florschuetz et al. are listed in Table 2.1. It was observed that different behaviour is exhibited for staggered and inline arrays of impingement holes with separate variables in Table 2.1 corresponding to each.

$$Nu = A \, Re_j^m \{1 - B[(z/D)(G_c/G_j)^n]\} Pr^{1/3}$$ \hspace{1cm} (2.5)

Where:

$$A, m, B \text{ and } n = C(X/D)^{n_x}(Y/D)^{n_y}(Z/D)^{n_z}$$ \hspace{1cm} (2.6)
Table 2.1: Constants for use in Florschuetz et al.’s correlation equation (2.6) for averaged Nusselt number beneath arrays of inline or staggered jets impinging into a channel with crossflow [56]

Florschuetz’s correlation was derived from experiments conducted over a wide range of values for the dependent parameters. It is reported as being valid for jet Reynolds numbers from $2.5 \times 10^3$ to $7 \times 10^4$; jet-to-jet spacings of $X/D$ from 5 to 15 and $Y/D$ from 4 to 8; jet-to-target spacing of $Z/D$ of 1 to 3. It was also derived for ratios of crossflow/jet velocity of 0 – 0.8 and a valid range of channel aspect ratios of $(0.625 \leq x_n/y_n \leq 3.75)$.

2.2.7 Combined Impingement and Roughened Surface Cooling

The negative performance and cost aspects of diverting air for cooling purposes in many gas-turbine engine applications has driven investigations into more efficient use of cooling flow for which the primary avenue of investigation is to utilise impingement jets exhausting into roughened passages featuring ribs, pedestals or dimples. A selection of different arrangements encountered is shown in Figure 2.9. In a review of studies into different rib-enhanced impingement jet cooling geometries, conducted by Andrews et al. [5] an associated improvement of $\overline{Nu}$ with the amount of blockage presented by the surface features (up to a maximum investigated blockage of 50%) was observed, but only under high crossflow conditions. It was also found that large channel obstructions could weaken the performance of the impingement jet array. The addition of roughness features to the target plate increased pressure blockages in the channel, leading to reduced flow through the early stage jets, at channel inlet as feed air instead diverted to the final row jets closest to channel
exit, thereby bypassing the roughness features and reducing cooling performance at the leading edge, but with improved cooling performance at the rear of the array.

The relative strengths of the crossflow and jet velocity have also been defined in terms of blowing ratio, \( \frac{\dot{m}_c}{\dot{m}_j} \) the ratio of crossflow to impingement mass flow rate. Several studies investigating the influence of the blowing ratio on cooling performance of rib-enhanced impingement systems have been reviewed. Over a range of blowing ratios, \( \frac{\dot{m}_c}{\dot{m}_j} = 0.5 \rightarrow 1.5 \) investigations were carried out into a variety of geometries including solid [57-58] and broken [59] ribbed turbulators of different angles and pin fins. The combined results of the studies support the observations of Andrews et al. that the influence of the roughness features increases with blowing ratio. Comparisons of the effect of blowing ratio upon the different geometries showed a much larger deterioration of \( Nu \) for impingement over smooth passages than for roughened channels. At relatively low blowing ratios of 0.5, smooth channels were observed to provide comparable or superior \( Nu \) to ribbed or pinned channel, but for blowing ratios of 1 or greater, with the cross mass flow rate exceeding the jet mass flow rate, roughened channels were seen to have greater \( Nu \) as the influence of the ribs and pins became more dominant.

The studies of Rhee et al. [57] [60] into the impact upon the pressure drop of the system of adding ribs or pins to impingement channels found that the addition of ribs more than doubled the pressure drop within the channel, accounting for up to 25% of the total pressure losses; a value that would increase for longer jet arrays. A final observation that was made from the study was that the amount of blockage presented by ribs and pins was not the only influencing factor on cooling performance. The profile and positioning of the ribs and pins in respect to the impingement jets was observed to have a significant impact upon \( Nu \) and the system pressure loss. Of the ribbed geometries investigated, it was found that ribs angled at 45° to the flow direction provided improved cooling performance to straight ribs, perpendicular to the flow [57]. This improved heat transfer performance was observed to occur at the expense of increased pressure losses. Of the pin finned channels, it was identified that altering the relative position of pins to appear just upstream or downstream of an impingement jet affected cooling performance [60]. At \( \frac{\dot{m}_c}{\dot{m}_j} = 1 \) it was found that positioning pins upstream of the jet improved \( Nu \) compared with downstream pins due to blockage of crossflow in the stagnation region. The improved performance was somewhat localised however, as the wake produced by the pin fin was observed to wash away the wall
jet, reducing $Nu$ in the inter-jet region. Although combining impingement jet cooling with surface roughness features indicated potential for improved cooling performance, many papers identified arrangements in which their presence weakened regional or overall heat transfer performance, particularly in the case of low crossflow [5] [57-58]. As such, careful consideration is needed in the design of combined cooling geometries to ensure beneficial performance is provided.

2.2.8 Summary of Current Combustor Liner Cooling Practices

Current combustor liners employed by Rolls-Royce PLC typically involve arrays of pedestals fed by large upstream impingement holes. Portions of the coolant are injected through slots into the main-combustion chamber for film cooling purposes. A representative arrangement of current practice is illustrated in Figure 1.4. Interest in both impingement-effusion arrays and reduced reliance on film cooling has guided research into different geometries for this thesis; however manufacturing and cost limitations restrict the scope of possible arrangements. Current manufacturing processes place limitations on the size and shape of jet holes that may be used, with additional complexity associated with angled effusion holes over the more traditional slots used in the combustor liner.

2.3 Heat-Transfer Measurement Techniques

In order for the heat transfer performance of cooling geometries to be experimentally measured, an appropriate experimental technique must be used. Due to the difficulty and cost of recreating the high temperatures and pressures encountered with a gas-turbine combustor liner experiments for this thesis are conducted at ambient conditions. As such a method of evaluating heat transfer performance is required that may be expressed in non-dimensional terms so that it may be equated back to engine conditions. Typical parameters used are Nusselt number, heat-transfer coefficient and temperature ratios [11].

2.3.1 Heater Methods with Thermocouples

Numerous different techniques have been developed to suit different applications and experimental conditions. Thermocouple techniques equate direct measurements of temperatures captured from the heat-transfer surface, to heat-transfer coefficients. One such technique involves the application of thin-foil heaters to a low conducting surface which when supplied with electrical current acts as a constant heat flux surface. Using this
boundary condition in conjunction with thermocouple measurements of wall and gas temperature enables the heat-transfer coefficient to be measured. This method provides low-resolution heat-transfer coefficient measurements with typical uncertainty of ±5 to ±10% [11]. This approach has been used by Han [31] for heat transfer-measurements in rib roughened passages. A similar method instead using copper heaters to obtain regionally averaged heat-transfer measurements has been employed in rib roughened passages [32] and beneath impingement jet arrays [61].

2.3.2 Mass-Transfer Analogy Techniques

An alternative approach that has been used is the mass-transfer analogy approach, in which the mass-transfer of a surface coating due to evaporation or sublimation is equated to heat transfer performance. This measurement approach has the benefit of high-resolution results and no heat losses. The mass transfer technique most frequently encountered during the course of the literature review was the naphthalene sublimation technique, in which mass-transfer is evaluated by measuring the local and bulk naphthalene vapour densities. This technique has been used in investigations into heat/mass transfer performance of impingement arrays over smooth channels with effusion holes [41] [62-63] roughened passages, featuring ribs [57] and pedestals [60] with effusion holes.

2.3.3 Optical Techniques

Optical techniques use observations of the test surface to determine the heat transfer coefficient, with various modes used to visualise the temperature. Two such methods for this are the use of infra-red cameras and thermochromic liquid crystals, with both capable of producing high resolution, full surface maps of heat-transfer performance, characterised by Nusselt number. Using infra-red thermography, direct measurements of wall temperature can be made which are subsequently analysed to calculate $Nu$. The equipment required for infra-red thermography is expensive, with the cost increasing for more accurate measurement equipment. For accurate cameras, uncertainty can be as low as ±1% [11].

A range of different liquid crystal thermography techniques exist, in which thermochromic liquid crystals (TLCs) with carefully calibrated colour response are used to obtain full surface temperature maps, which are subsequently analysed to calculate surface heat transfer coefficient. Liquid crystals have been frequently used for surface temperature measurement in heat transfer experiments as they provide a repeatable response and their
colourplay can be easily recorded with a video camera [64]. Different experimental approaches can be broadly separated into steady-state and transient methods, with both methods requiring the appearance of individual colour bands of the crystal to be tracked. Typically, tracking follows the yellow or green band, although an alternative method developed by Camci et al. uses the hue variation of the TLC to track surface temperature in steady-state [65] and transient [66] applications. The accuracy of a given liquid technique is dependent upon both the experimental conditions and the degree of sophistication in processing the crystal colour play. An uncertainty in the region of ±6% results from the most straightforward TLC experiments [67], although this may be reduced by thorough calibration of measurement equipment, using more of the colour response data and by refining the assumptions to reduce analytical errors.

The ability to generate high resolution, full surface $Nu$ maps meant that an optical method was preferred for experiments for this thesis, with the high equipment costs for infra-red thermography resulting in liquid crystal thermography being selected. The choice of liquid crystal thermography prompted further research to be conducted into this measurement technique in terms of application and means to reduce uncertainty. The findings of this review are included in Chapter 3.

### 2.4 Closure

Research conducted into previously published literature on cooling techniques has highlighted a strong interdependency of different parameters on the heat transfer performance of different cooling approaches. Heat transfer performance is frequently characterised not only in terms of heat transfer augmentation, but also the amount of pressure loss resulting from the change in geometry. Pedestal arrays, which are the standard means of cold-side cooling in current Rolls-Royce combustor liners have been identified as displaying very poor heat transfer to pressure loss characteristics. Impingement jet cooling has potential for very high heat transfer augmentation, but performance is reduced under the influence of crossflow and in the case of coolant extraction through film holes on the target plate. The impact of crossflow has been identified as the single largest contributor to reduced heat transfer performance of large arrays of impingement jets. Consequently, various geometries featuring impingement jets in combination with roughened passages have been investigated, in order to improve performance in the presence of crossflow, with mixed results being reported. While heat transfer performance can be improved by combining methods, it has been generally seen
to occur at the expense of additional pressure losses. The most promising results were identified for impingement over pedestal arrays as improvements in heat transfer performance were typically generated for less pronounced increases in pressure drop than for ribbed passages. Various studies indicated that while performance benefits could be achieved, in many cases, the addition of roughness features reduced surface heat transfer performance.

An initial review into different heat transfer measurement techniques resulted in liquid crystal thermography being selected as the chosen method. This initial investigation is followed by a thorough review of different liquid crystal measurement methods in order to establish the experimental method to be used for this thesis. This is reported in Chapter 3.
2.5 Figures

Figure 2.1: Illustration of film-cooling process for hot-side cooling [68]

Figure 2.2: Comparison of thermal performance parameters for heat transfer augmentation techniques [20]
Figure 2.3: Flow effects resulting from a single impingement jet [33]

Figure 2.4: Flows through short and long sharp edged holes [69]
1. Lift jet flow
2. Jet impingement region
3. Wall jet flow
4. Fountain formation region
5. Fountain up-wash flow
6. Wall jet interaction stagnation line
7. Entrainment
8. Ground plane
9. Blocking surface

Figure 2.5: Schematic illustration of twin-jet impingement flow [42]

Figure 2.6: Illustration of crossflow development in a) test rig [46] and b) turbine blade [7]
a) Figure 2.7: Test setup investigating effect of exit orientation on heat transfer rate:
(A) With film-cooling holes [46]  (B) Without film cooling holes [47]

b) Figure 2.8: Streamwise Nusselt number variation for impingement jets into channels with [46] and without [47] film cooling holes, from experimental setups illustrated in Figure 2.7
Figure 2.9: Selected rib and pin roughened impingement jet arrays [6] [60]
Chapter 3 Transient Liquid Crystal Thermography

Various different techniques are used for the measurement of heat transfer, with particular approaches suitable for different purposes. A review of these methods was conducted in Chapter 2. For the experimental investigations of this thesis, liquid crystal thermography has been selected as the most appropriate method due to the ability to generate high resolution maps of surface heat transfer coefficient \( (h) \), and the repeatability of the method without reapplying the liquid crystal paint. The method was preferred over infra-red thermography, which can generate full surface heat-transfer maps of comparable resolution, as liquid crystal thermography could be conducted using a standard digital video camera.

The choice of liquid crystal thermography as a means of measuring heat transfer necessitated further research to be conducted into factors that influence the colour response of thermochromic liquid crystal (TLCs). Additionally, different approaches for using TLCs to measure heat transfer performance were researched to identify the most appropriate application for the experiments conducted for this thesis. The specific method employed is a transient single colour capturing technique in which the surface \( h \) is determined by tracking the green response of the TLCs in response to a near step change in temperature. The decision to adopt this measurement technique necessitated further research into the associated analytical approaches and sources of error in order to assess the most appropriate method by which captured transient liquid crystal data may be analysed to provide accurate results in the different experimental facilities. A number of different studies were encountered in which alternative methods of measuring heat transfer performance from liquid crystal data captured using this technique were proposed.

3.1 Thermochromic Liquid Crystal Properties

Thermochromic liquid crystals are named due to the fact that under certain temperature conditions they react to produce a colour response. At a particular wavelength, TLCs reflect a single wavelength of light. As the transition of colours is sharp and precise, the crystal colour response may be designed such that the appearance of red, green and blue (RGB) colours may be fixed to appear at specific temperatures [11]. In addition to fixing the response temperatures of the TLC; the bandwidth, defined as the temperature range for transition from the appearance of red to the appearance of blue, may also be calibrated. Calibrated crystals are often divided into two different classifications based upon their
bandwidth. A complete transition over a range greater than 5°C is classified as wide-band. Conversely, a TLC is considered to be narrow band if it has a bandwidth of less than 2°C. Various factors have been identified that can affect TLC calibration, both in terms of intensity of response and temperature at which a particular colour is perceived.

3.1.1 Application of TLC to Surface

The colour response of raw TLCs has been shown to degrade due to chemical contamination or exposure to ultraviolet light [70]. In order to combat this, TLCs are encapsulated and mixed with a sprayable binding material for application onto a surface using an airbrush. In order to ensure consistent and uniform colour response it is necessary to ensure a smooth and even coat. Too thin a coating has been shown to affect colour quality; however a film too thick may experience a different surface temperature than the wall to which it is applied [71].

3.1.2 Impact of Viewing/Illumination Angle

The colour response of TLCs is sensitive to both viewing and illumination angles at which it is viewed, as well as the difference in angle between the light source and the target. The perceived colour of a TLC depends on the lighting and viewing arrangement; the spectrum of the main light source and the background light [70]. The influence of viewing angle differs greatly for narrow and wide-band crystals. For narrow-band crystals Camci et al. [65] identified viewing angles less than 40° from the normal to have a negligible effect on perceived colour. A minor variation of perceived hue with viewing angle was observed for hue values greater than 0.38, equivalent to the green-to-blue transition [71]. The degree of difference becomes more pronounced at 45° viewing angle, but with no divergence below this hue level even at this angle. Wide band crystals have been shown to be highly dependent on viewing and lighting angle. A variation of only 5° in the light/viewing angle can result in up to a 10% variation in the perceived hue, relative to full bandwidth of the crystal [70].

3.1.3 Hysteresis Effects

The reaction temperatures of both narrow and wide-band crystals in response to heating and cooling were investigated by Anderson and Baughn [72] with hysteresis effects observed to result in TLCs displaying a different response when being heated than when cooled. The intensity of the RGB response and the temperature at which the peak intensities of each band
of colour occurred were both observed to change for transitions occurring due to heating and cooling. When cooled from above their colour play range rather than heated below the start temperature the intensities of the individual colour responses decreased, with a greater reduction occurring as the peak temperature prior to cooling increased. The decrease in peak colour intensity was shown to be non-constant and non-proportional resulting in a shift in hue angle as well as changes to intensity. The temperatures at which the peak green and red responses occur were observed to increase in the event of the wall temperature marginally exceeding the top of the crystal response range. A subsequent decrease in reaction temperature was seen as the peak temperature prior to cooling increases, this behaviour is shown in Figure 3.1 for all the crystals tested. Anderson and Baughn observed that the hysteresis effects could become permanent as the temperature greatly exceeded the final response temperature. The combined hysteresis effects highlight difficulties in liquid crystal applications involving the cooling of the crystals. Through careful experimentation, hysteresis damage may be avoided, by terminating heating following full reaction of the crystals. Additionally, the effects on single-band tracking transient liquid crystal thermography are limited as it is the temperature, rather than the hue or intensity that is of importance and that value remains unchanged.

### 3.2 Comparison of Liquid Crystal Measurement Techniques

Liquid crystal thermography techniques can be broadly divided into steady-state and transient methods, although each distinct category consists of a range of different measurement approaches. An overview of the different measurement approaches are outlined in this section.

#### 3.2.1 Steady-State Thermography

Steady-state liquid crystal methods typically employ a heated surface technique, in which the TLC is applied to a heat transfer surface manufactured from material with good insulating properties (high specific heat capacity, low thermal conductivity) over a fine electrically conductive coating as shown in Figure 3.2 [73]. The thin conductive film is used to electrically heat the plate and the presence of only low conduction in the heater, plastic and insulation, mean that the surface boundary condition is very close to uniform heat flux. By adjusting the voltage of the electrical heater, the surface heat flux can be altered, allowing the surface temperature to be raised or lowered. Local values of $h$ may be calculated by adapting
equation (3.1) to this specific application, equation (3.2) and solving for constant heat flux [74].

\[
h = \frac{\dot{q}_c''}{(T_w - T_g)}
\]  

(3.1)

Where the local convective surface heat flux, \( \dot{q}_c'' \), is given by:

\[
\dot{q}_c'' = \frac{IV}{A} - \varepsilon \sigma (T_w^4 - T_g^4)
\]  

(3.2)

The heated-coating method has been carried out using both narrow [67] and wide-band [65] [70] crystals. Sabatino et al. discussed a comparison of the relative merits of the different approaches [75]. Performing a constant flux experiment using narrow band crystals, has the advantage that only one calibration point is required for the TLC temperature/hue response to be determined. The narrow bandwidth requires a large number of images to be taken for complete mapping of a test surface. Resolution has also been shown to be strongly dependent on the size of the local temperature gradients. In the presence of small spatial temperature variations this will reduce the accuracy of the technique. Using wide-band crystals overcomes the disadvantages of the narrow band experiment as the entire surface temperature distribution can be established from a single image. In order for this method to work however, an intensive calibration experiment is required to accurately resolve the crystal response. This makes experiments highly sensitive to any variation in the previously outlined conditions affecting perception of crystal colour response, most notably changes to illumination intensity or angle. As such, it is necessary to maintain constant light levels and camera position. If either of these parameters is changed then recalibration is required. This makes it difficult and time consuming to carry out multiple experiments. This calibration process results in steady-state experiments having durations of 2-3 hours. These factors can render steady-state techniques somewhat unwieldy. An additional undesirable aspect of steady-state liquid crystal thermography is the need to calculate the heat losses that occur throughout the experiment due to the lack of an adiabatic wall. This adds further to the complexity of the method.

3.2.2 Transient Thermography

Transient liquid crystal thermography was first demonstrated by Ireland and Jones [76]. The basic method uses a solution of Fourier’s one-dimensional conduction equation, developed by Schultz and Jones [77] to calculate \( h \) from the reaction of a TLC in response to
a step-change in temperature. This approach has since been used for numerous cold side heat transfer measurement experiments, encompassing a diverse range of geometries including, rib-roughened passages [78]; impingement-jet arrays [46] as well as combined impingement/rib methods [58] and also for film-cooling heat transfer measurement [79].

Transient techniques differ from steady-state methods in that the colour play of TLCs is recorded in response to a sudden, defined temperature change from an initial state of thermal equilibrium. The sudden change in temperature may be achieved either using a heated fluid or a heated wall [74]. Heating the fluid was originally accomplished using switching valves [76] [80], although a more recent development uses a high current mesh heater to rapidly and uniformly heat the flow directly [81]. Transient tests are comparatively rapid, with typical durations under a minute. Duration is limited as assumptions used by the technique render the method invalid if heat conducts through to the rear of the target plate. To allow for sharp response peaks to be observed, narrow band crystals are generally used, rather than the wide-bandwidth crystals typically associated with steady-state methods. One benefit of this is that calibration of narrow band crystals is less sensitive to viewing angle and light intensity [74]. Calibration is still required, but only to identify the exact response temperatures at which the crystals experience peak intensity which vary with slurry concentration and coating thickness. A benefit of the short test duration and less-sensitive calibration is that it is possible to carry out multiple tests in close succession, so long as the initial temperature equilibrium is reasserted. The short testing times also means that heat-losses due to the absence of an adiabatic wall need not be calculated. For these reasons, transient experimental setups are generally preferred over steady-state and will be considered for this experiment. An additional reason for the choice is that the short test duration and less demanding calibration are more suited to use on large test rigs, where it is required to move the camera to different positions.

A typical basic arrangement for the test section of a transient liquid crystal experiment is shown in Figure 3.3, with the TLC applied to a surface plate constructed from a material of low thermal diffusivity. This experimental setup requires the surface temperature to be initially matched to that of the fluid, such that $T_{w} = T_{g}$ before a change in fluid temperature is activated. This method uses the resulting surface temperature transient to determine $h$ across the plate, most commonly by evaluating the time taken for a crystal reaction to be observed. The use of a surface with low thermal diffusivity (for example Perspex) alongside
test durations of under a minute allows a one-dimensional assumption to be used, due to minimal lateral conduction.

### 3.3 Fourier Solution Method for HTC Calculation

As suggested by the name, transient liquid crystal thermography requires the transient colour play of narrow band liquid crystals to be used in order to calculate surface $h$. Numerous studies have been identified in which transient liquid crystal thermography was used to calculate surface heat transfer coefficient. The generally established analysis method for interpreting transient crystal data uses a solution of Fourier's one-dimensional conduction equation (3.3), for a semi-infinite wall with temperature $T = T(x, t)$ where $x$ is the direction of heat transfer, normal to the plane of the wall and where $\alpha$ is the thermal diffusivity ($k/\rho c$) of the wall material.

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} \tag{3.3}$$

Using this approach, $h$, is determined by evaluating the time taken for the liquid crystal to elicit a given response. The analysis has most commonly been carried out by approximating the wall temperature variation to be the result of a step change in the temperature of the fluid and solving iteratively using equation (3.4) [77] where $T_0$ is the initial, uniform temperature of the wall and gas.

$$\theta = 1 - e^{\beta^2} \text{erfc}(\beta) \tag{3.4}$$

Where:

$$\theta = \frac{T_w - T_0}{T_{aw} - T_0} = \frac{\theta_w}{\theta_{aw}} \tag{3.5}$$

$$\beta = \frac{h \sqrt{\ell}}{\kappa} \tag{3.6}$$

And:

$$\kappa = \sqrt{\rho c k} \tag{3.7}$$

### 3.4 Assumptions of Fourier Solution

Derivation of equation (3.4) requires several assumptions to be made concerning the experimental conditions and the modes of heat transfer between the fluid and the surface. Discrepancies between experimental conditions and the ideal scenario considered in this
equation will reduce the accuracy of calculated values of $h$. The combined effect of these assumptions may result in significant error in the value of $h$ calculated by this method. While measures may be taken to evaluate and restrict the magnitude of uncertainty, the fundamental assumptions limit the accuracy of the calculation. The validity of this method and the resultant sources of error are discussed within this section. A number of papers have been identified that have proposed alternative solutions than equation (3.4) to the one-dimensional conduction equation, (3.3). Several of the alternative methods suggested are derived from more general solutions to the equation.

### 3.4.1 One-Dimensional Heat Transfer

Because this analysis method uses a solution of the one-dimensional conduction equation it is necessary to assume that all heat transfer between the fluid and the wall occurs normal to the plane of the plate. The presence of any in-plane surface temperature differential results in lateral conduction, introducing a source of uncertainty into the calculated value of $h$. It has been shown in previous experiments that with significant lateral variation of temperature, errors in the calculated $h$ can be as large as 15-20% [82]. Jet impingement cooling in particular is noted for providing uneven cooling, with large gradients occurring in the regions between the jets. The impact of conduction errors may be significantly reduced by limiting the temperature differentials across the plate, most easily achieved by restricting the magnitude of the temperature step used in the experiment, while restricting the duration of the test will also limit the amount of lateral conduction. By selecting a material with low thermal conductivity for the target plate, conduction through the plate can be further restricted. A method to correct for errors from lateral conduction was proposed by Kingsley-Rowe et al. [83]. This approach uses an adapted version of the 1D analytical solution to provide an approximate 2D solution. Equation (3.4) is used as a starting point, to generate a quasi-2D form of the solution. This 2D approximation, establishes a correction parameter that may be used to adjust the 1D result to account for lateral conduction. An evaluation of the method was provided by Kingsley-Rowe et al. [83] using a numerical method for comparison of the corrected and uncorrected data. Using the 1D and quasi-2D forms of the solution, one-Dimensional ($B_{i_1}$) and corrected ($\overline{B}_{i}$) Biot numbers were calculated. Using these values and a known true value of $B_{i}$ ($B_{i}^{*}$) non-dimensional error in $B_{i}$ before and after correction were calculated respectively as $\varepsilon_1 = \frac{B_{i_1}/B_{i}^{*} - 1}{\varepsilon_1 = \frac{\overline{B}_{i}/B_{i}^{*} - 1}{\varepsilon_1}$. The true Biot number $B_{i}^{*}$ was a known polynomial
function of normalised lateral position $\lambda$. Figure 3.4 shows that there is a significant reduction in error from $Bi_1$ to $\tilde{Bi}$ when compared with the numerically obtained value. While this paper claims a greatly improved accuracy, there would be little confidence in corrected values obtained using the method without the numerical data against which to compare it.

### 3.4.2 Semi-Infinite Plate Assumption

The derivation of the solution to equation (3.4) is based on the assumption that the heat transfer surface may be approximated to a semi-infinite plate, it is assumed to have infinite thickness in the $x$-direction for the purpose of transmitting heat. For this assumption to be valid, it is necessary that the heat input from the fluid does not fully permeate through the plate, with the rear of the wall maintaining constant temperature. This is achieved by selecting a material with high density and specific heat capacity, and low thermal conductivity for the target plate. A measure of validation for this assumption has been defined by Schultz and Jones [77] using equation (3.8) to calculate the plate penetration time, that is the time taken for the heat to penetrate the wall thickness to such an extent that the rear face of the wall increases by a defined temperature, $0.01(T_w - T_0)$. The penetration time ($t_p$) presents an upper limit for the duration of the experiment during which the semi-infinite assumption remains valid and beyond which heat starts to exit the rear face of the plate, introducing significant error.

$$t_p = 0.10d^2 \frac{\rho c}{k}$$  \hspace{1cm} (3.8)

Where $d$ represents the wall thickness.

### 3.4.3 Step Temperature Change

It is assumed in equation (3.4) that the coolant gas undergoes a step change in temperature $\Delta T$, from an initial value $T_0$ to an elevated steady-state value $T_{aw}$. In practice implementing an instant transition in fluid temperature is not possible; however, very rapid temperature transitions may be used by adopting specialised means of heating the flow. Mesh heaters and flow switching are favoured methods of simulating a step-change. As the step-change assumption cannot be precisely matched, for the transition period in which temperature increases from the initial equilibrium condition, to the final steady-state temperature, the temperature differential between the fluid and the wall as used in equation (3.4) is overestimated. The impact of this error will increase, the further the actual
temperature profile varies from the ideal step case. The resultant error due to this assumption will also be greater for high $h$ with rapid crystal transition. This effect would reduce as the crystal transition time increased, due to the decreasing significance of the initial lag in fluid temperature.

Several research papers investigating improved liquid crystal thermography measurement techniques have sought methods to account for a gradual, rather than step, increase in temperature. Using an appropriate model to map the fluid temperature transient will enable $(T_{aw} - T_w)$ to be matched much more precisely than is achieved with a simple step assumption. As such, $h$ would be calculated with a significantly reduced error. A number of the papers that have been identified use different approximations to model the fluid temperature transient resulting from the activation of a heated flow. In one such paper by Gillespie et al. [84] the gas temperature is modelled as increasing exponentially in relation to a time-constant, $\tau$. This exponential temperature rise occurs as shown in equation (3.9) from the initial value $T_{aw} = T_0$ at $t = 0$, towards a steady-state value, $T_{aw} = T_{a,\infty}$ as $t$ tends to infinity.

$$T_{aw}(t) = T_{aw,0} + (T_{aw,\infty} - T_{aw,0})(1 - e^{-t/\tau})$$  \hspace{1cm} (3.9)

For this case an analytical solution was obtained using a solution of equation (3.3) for the exponential temperature rise defined by equation (3.9). A rearrangement of the solution derived by Gillespie et al. is shown in equations (3.10)-(3.12):

$$\theta = \frac{T_w - T_0}{T_{aw,\infty} - T_0} = g(\beta, \beta_\tau)$$  \hspace{1cm} (3.10)

Where:

$$g(\beta, \beta_\tau) = 1 - \frac{1}{1 + \beta_\tau^2} e^{\beta^2} \text{erfc}(\beta) - e^{-t/\tau} \frac{\beta_\tau^2}{1 + \beta_\tau^2}$$

$$\times \left\{1 + \frac{1}{\beta_\tau} \left[\frac{1}{\pi} \sqrt{\frac{\tau}{\tau}} + \frac{2}{\pi} \sum_{n=1}^{\infty} \frac{1}{n} e^{-n^2/4} \sinh \left( n \sqrt{\frac{\tau}{\tau}} \right) \right]\right\}$$  \hspace{1cm} (3.11)

And

$$\beta_\tau = \frac{h \sqrt{\tau}}{\kappa}$$  \hspace{1cm} (3.12)

Gillespie et al.’s method offers greater capacity to model a temperature rise based on measured conditions. Varying the magnitude of $\tau$ may significantly alter the gradient of the
temperature transient as illustrated in Figure 3.5. It can be seen that the speed at which the fluid reaches its steady state value increases with smaller values of τ and indeed, the case where τ = 0 results in equation (3.10) reducing to the step-change solution. The constant may therefore be used to fit the modelled transient to the measured experimental data by selecting an appropriate value of τ such that the modelled response most closely matches the experimental temperature data. While in many cases the exponential approximation may provide a closer match to the actual fluid temperature increase than the step change assumption; there are many applications for which it would not be appropriate. A method was developed by Newton et al. [85] that was adapted from the earlier work by Gillespie et al. to consider the temperature transient as a series of exponential functions, allowing greater potential to more closely model the actual conditions. Using this method, the equation for the fluid temperature can be presented as:

\[
T_{aw}(t) = T_0 + \sum_{j=1}^{m} \Delta T_{aw,j} \left(1 - e^{-t/\tau_j}\right)
\]  

(3.13)

\(T_{aw,j}\) and \(\tau_j\) are the constant amplitude and time constant for each term. This approach provides greater freedom to match the modelled temperature transient to the experimental conditions by altering the number of terms and the respective amplitude and time constants. Figure 3.6 illustrates an example of using an iterative approach to match a measured temperature transient, with the addition of each exponential term bringing a closer fit between the two. For the case shown, the improved ability of the more complex models can be seen, with the exponential model offering a marked improvement from the step assumption and the exponential series improving further. Using equation (3.13) to model the fluid temperature, a general solution was derived for the wall temperature, satisfying Fourier’s equation for \(j = 1 \rightarrow m\). This is given by equation (3.14) which was developed from a similar theory to that used by Gillespie et al. with the form it takes meaning that it may be used in place of equation (3.10). The exponential solution may be considered as a special case of the exponential series, where \(m = 1\).

\[
\Theta = \frac{T_w - T_0}{T_{aw,\infty} - T_0} = \sum_{j=1}^{m} \frac{\Delta T_{aw,j}}{T_{aw,\infty} - T_0} g(\beta, \beta_{\tau_j})
\]  

(3.14)

One final method of modelling the fluid temperature rise, is to treat it as a series of step changes in temperature, each over small increments of time. The basic principle of this method may be seen in Figure 3.7 [86]. Using this approach is a relatively simple model that
may be easily made to match any measured temperature profile. The accuracy of the method is largely linked to the number of time steps used to approximate the transient. Using a large number of infinitesimal steps allows for the greatest potential accuracy. By its nature, the step-series method will result in a temperature transient model that is an underestimation of the actual case as is clearly illustrated in Figure 3.7. The resultant error from this will be significantly lower than for the large overestimation present within the step-solution. The step-series method requires that simulated time steps be incorporated into equation (3.4) such that \( h \) is calculated taking into account the increased number of smaller step rises in temperature. A method known as Duhamel’s superposition theorem is used to produce equation (3.15) which may be used to iteratively calculate \( h \).

\[
T_w - T_0 = \sum_{j=1}^{m} \left[ 1 - \exp \left( \frac{h^2(t - t_j)}{\kappa^2} \right) \right] \text{erfc} \left( \frac{h \sqrt{(t - t_j)}}{\kappa} \right) \left[ \Delta T_{aw(j, j-1)} \right]
\]  

(3.15)

3.4.4 Mixed Bulk Analysis Method

The local heat transfer coefficient calculated from the transient method, using equation (3.4) and its variants, by its nature, has its reference temperature based on the inlet temperature of the rig/cooling geometry, rather than the local bulk mean temperature. In the case of long cooling channels associated with rib or pin-fin arrays and side-feed jet impingement, a significant drop in gas temperature will occur along the channel length. As the local bulk mean temperature varies with both location and time, the degree of difficulty involved in prediction or measurement of the temperature renders it impractical for this to be done. Several authors have investigated theoretical methods to account for the variation of fluid temperature across long channels.

A very simple method of accounting for streamwise bulk temperature variation is to use linear interpolation between measured temperatures recorded by thermocouples positioned at upstream and downstream locations, generally at the source of the heated flow and at cooling geometry exit. Linear interpolation has been employed by Clifford et al. [87] and also in cooling of two-pass channels by Ekkad and Han [78], with interpolation between measured temperatures at the midpoint of each pass. This technique has the advantage of very simple, low intensity processing; however the simple temperature assumption provides little confidence of the degree to which the actual local bulk temperature is matched. Additionally, while the inlet temperature may be reasonably modelled from a point location, in the
downstream region a single temperature measurement may not necessarily be representative of the bulk flow at that location due to the complex flow patterns produced by passage through the cooling geometry.

\[
\left( \rho C_p U \right)_g A \left( T_e(\tau) - T_{mb}(\tau, x) \right) = \int_0^p \int_0^x dq \]

\[= \int_0^p \int_0^x h_e \left( T_e(\tau) - T_w(\tau, x, y) \right) dx \, dy \quad (3.16)
\]

In addition to interpolation between measured temperatures, several methods have been developed based upon the principle of thermal energy balance, assuming initially uniform temperature. This may be expressed over an integral volume, (equation (3.16), Figure 3.9a) or a differential volume over an integral region, (equation (3.17), Figure 3.9b) [88].

\[
\left( \rho C_p U \right)_g A \left( T_{mb}(\tau, x) - T_{mb}(\tau, x + dx) \right) \]

\[= \left[ \int_0^p h_{mb}(x, y) \left( T_{mb}(\tau, x) - T_w(\tau, x, y) \right) dy \right] dx \quad (3.17)
\]

Methods have been outlined by Chyu et al. [88] for using the energy balance equation to calculate the local mixed bulk temperature based upon a single transient measurement of temperature at channel inlet. The transient marching method analyses the energy balance along the channel length using equation (3.17). To calculate \( T_{mb,0} \) and \( h_{mb,0} \), a temperature transient measured at channel inlet is used to calculate \( h_e \) from \( T_e(\tau) \) using equation (3.4) or one of the variations outlined within the present chapter. Both of these values are equal to the mixed bulk equivalents at \( x = 0 \). Based upon the assumption of time-constant \( h \), the calculated \( h_{mb,0} \) may then be used to calculate the wall temperature transient at \( x = 0 \). The transient mixed bulk temperature for \( x = 1 \) may be calculated using equation (3.17), with \( T_{mb,0}, T_{w,0} \) and \( h_{mb,0} \) used as inputs. The calculated value of \( T_{mb,1}(\tau) \) may then be used to determine \( h_{mb,1} \). By repeating these steps along the entire length of the channel, a time-variant streamwise mixed bulk temperature may be determined and used to calculate \( h_{mb} \) at all locations. This process can be simplified by assuming a negligible variation of bulk temperature with time and adopting a steady-state marching technique. For this case the \( \tau \) term is removed from \( T_{mb}(\tau, x) \) and \( T_w(\tau, x, y) \) in equation (3.17). Using this simplification, \( h \) is calculated for a time averaged mixed bulk temperature, with the inlet temperature used as an initial reference. Time averaged bulk temperatures and \( h \) are then calculated in a similar marching method using differential control volumes. This simplification increases the error
of the heat transfer calculation but significantly reduces the amount of processing required to obtain a solution.

A second approach that has been employed to calculate $h$ from the local mixed bulk temperature using the energy balance principle was developed by Metzger and Larson [89] and has been since used in a number of studies since, including Wang et al. [90] and Chambers [1]. This approach considers that solutions for $h$ based upon equation (3.4) use temperatures measured at a single upstream location, $x = 0$, with $h_e$ calculated under these conditions as defined in equation (3.18). Calculation of $h_{mb}$ based upon local mixed mean fluid temperature takes the form shown in equation (3.19) and combination of the two equations enables the ratio of the two coefficients to be expressed in terms of the local and upstream temperatures, equation (3.20).

$$\dot{q}/A = h_e(T_e - T_w)$$  \hspace{1cm} (3.18)

$$\dot{q}/A = h_{mb}(T_{mb} - T_w)$$  \hspace{1cm} (3.19)

$$\frac{h_{mb}}{h_e} = \frac{(T_e - T_w)}{(T_{mb} - T_w)}$$  \hspace{1cm} (3.20)

Using the energy balance principle in the streamwise direction, the mixed bulk mean temperature at any streamwise location in a duct can be expressed using equation (3.21) and combining this with equation (3.20) allows the mixed bulk heat transfer coefficient ($h_{mb}$) to be determined based upon heat transfer coefficient values calculated from the inlet temperature ($h_e$) using equation (3.22).

$$T_{mb} = T_e - (T_e - T_w) \int_A \frac{h_e dA}{mc_p}$$  \hspace{1cm} (3.21)

$$\frac{h_{mb}}{h_e} = \frac{1}{1 - \int_A \frac{h_e dA}{mc_p}}$$  \hspace{1cm} (3.22)

Equation (3.22) depends upon the assumption of a spatially isothermal wall thermal boundary condition which is not strictly true. Analyses of the accuracy of this method have been carried out by Chyu et al. [88] and Tsang et al. [91]. Both papers proposed variations to the
equation to remove the assumption of the spatially isothermal wall assumption. Tsang et al. [91] proposed equation (3.23), with the $\psi$ term accounting for spatial variations in wall temperature given by equation (3.24).

$$h_{mb} = \frac{1}{1 - \psi \int_A \frac{h_e dA}{mc_p}}$$  \hspace{1cm} (3.23)

Where:

$$\psi = 1 - \frac{T_w^* - T_w}{T_e - T_w}$$  \hspace{1cm} (3.24)

$$T_w^* = \frac{1}{A} \int \frac{h_e}{h_e} T_w dA$$  \hspace{1cm} (3.25)

It was reported that in practice $\psi$ falls close to unity as the wall temperature variation, $T_w^* - T_w$ is much smaller than the temperature difference between the fluid and the wall, $T_e - T_w$. It was identified that the equation based upon the isothermal wall assumption is valid up to $20D_h$ ($20 \times$ the hydraulic diameter) before the $\psi$ term starts to vary significantly from unity ($\psi$ was observed to have values of 1.08 and 1.10 at distances $40D_h$ and $60D_h$ respectively). A similar term given in equation (3.26), was independently developed by Chyu et al. [88] with the same conclusion that the equivalent $\psi$ term may be disregarded for shorter channels in which case equation (3.22) provides an appropriate solution.

$$\psi = 1 - \frac{\bar{T}_w - T_w}{\frac{\rho c_p U}{h_e} \frac{A}{P_x} (T_e - T_w) + (T_w - T_w)}$$  \hspace{1cm} (3.26)

Where:

$$\bar{T}_w(\tau, x) = \frac{\int_0^\rho \int_0^x h_e(x, y)T_w(\tau, x, y) dx dy}{h_e}$$  \hspace{1cm} (3.27)

A comparison by Chyu et al [88] of the different methods for accounting for variable mixed bulk temperature over channel length identified that the transient marching and isothermal wall analogue methods produced near identical results, with $h_{mb}$ initially matching $h_e$ close to the inlet region. As distance from channel inlet increases it was shown that $h_e$ reduces, whilst $h_{mb}$ remains more constant. Comparisons with the steady-state marching and linear interpolation methods suggest that linear interpolation overestimates $h$ in the downstream region, whilst the steady-state assumption results in a lower $h_{mb}$ being calculated over the entire channel length. It would appear that depending upon channel length, the best approach of measuring $h$ based upon the local mixed bulk gas temperature is
to use equation (3.22) or (3.23) depending on whether the channel length is more or less than $20D_h$.

### 3.4.5 Constant Heat Transfer Coefficient

The experimental method being considered requires the assumption of a constant heat transfer, irrespective of the analysis method being used. As the heat transfer is the parameter being calculated; it is considered constant provided flow conditions within the test section remain unaltered. It has been shown by Butler and Baughn [92] that $h$ may not be constant for some cases using forced convection in which $h$ is dependent upon the surface temperature distribution. This is related to the fact that in free convection, $h$ is dependent on the difference between the fluid and surface temperatures. If this behaviour is reflected in forced convection methods, then $h$ will change as the wall temperature converges towards the fluid temperature. The effect of any variation of $h$ with time may be restricted, similarly to the case of lateral conduction error, by limiting the magnitude of the fluid temperature increase. This would reduce the change in $T_{aw} - T_w$ through the duration of the experiment and would consequently reduce any change in $h$.

An alternative method to account for variable values of $h$ would be to use multiple liquid crystals, with colour responses calibrated to different temperatures. This would provide multiple solutions to equation (3.4), theoretically all of the same value for constant $h$. Comparison of solutions would allow an evaluation of the internal consistency of the calculated values of $h$ for a changing wall temperature by compiling data from the different crystals and analysing trends.

### 3.5 3D Numerical Solution

An alternative approach for measuring $h$ was used by Lin and Wang [82]. Their study proposed the use of a 3D numerical approach using an inverse transient conduction scheme. This method is fundamentally different from those previously outlined which are variations on the basic solution of Fourier’s one-dimensional conduction equation, introduced in equation (3.4). The analytical approach of this technique consists of two distinct stages. The first of these involves direct solution of the heat conduction problem using the general governing equation (3.28). The second stage iteratively solves for $h$ at the front of the plate, using a correction process to converge on the actual value from initial estimates of the extreme limits.
\[ \rho c \frac{\partial T}{\partial t} = \nabla \cdot (k \nabla T) + S \]  

(3.28)

Where:

\[ \nabla = \frac{\partial}{\partial x} i + \frac{\partial}{\partial y} j + \frac{\partial}{\partial z} k \]  

(3.29)

And \( S \) is the interior heat source

In this analysis process the entire plate is considered as the computational domain with a number of discrete grid points used for simulation of this, as shown in Figure 3.8.

Variations on the general governing equation (3.28) that satisfied the different heat transfer boundary conditions within the system were also used. Three different categories were identified.

Type 1. Given boundary temperature. Lin and Wang set the rear face as experiencing this boundary condition, considering the system to be a semi-infinite wall.

Type 2. Given boundary heat flux. A special case of this boundary condition was used by Lin and Wang for the four side walls, in which the surface heat flux was defined as \( q'' = 0 \).

Type 3. Given heat transfer coefficient. Lin and Wang applied this boundary condition to the front face, but it was required to initially assume a value and iterate towards the correct heat transfer coefficient.

The correction process used to calculate \( h \) used the Newton root-finding method to gradually converge towards the actual value from initial assumptions. This was done by using two initial values of \( h \). These initial estimates were \( h_1 \) and \( h_2 \), which were respectively set to values significantly smaller and larger than actually anticipated such that the real value would fall between the two extremes.

\[ h' = h_1 + \frac{h_2 - h_1}{t_2 - t_1} (t - t_1) \]  

(3.30)

Equation (3.30) was used to produce a revised value of \( h \) that would fall between the two initial guesses. This would replace \( h_2 \) and as shown in Figure 3.10 would gradual converge towards the heat transfer coefficient corresponding to the identified crystal reaction
time. A comparison was made by Ling and Wang of the relative performance between the 3D numerical method that had been developed and the traditional 1D method. Figure 3.11 shows that the 1D model consistently over-calculates $h$, with this being attributed to the lateral conduction resulting from temperature gradients being unaccounted for. This error was observed to peak at the maxima and minima, at a value of 15-20%, with the mean error being calculated as 12%. This method has the benefit of fully considering the effect of conduction through the heat transfer surface; however, this method requires extremely intensive processing. This presents a major obstacle to its use, especially when attempting to measure $h$ over a large area as is the intent of this project. A brief viability study was carried out into the possible application of this method, but it was apparent that the processing time involved meant it would not be a practical solution method.

### 3.6 Full Crystal History Method

A weakness in the recognised transient liquid crystal methodology is that very little of the liquid crystal data is used to determine the heat transfer coefficient, with only the time of the peak response being used. An alternative method has been proposed that uses the entire calibration curve of the normalised intensity vs. temperature.[93]. This technique is based upon the assumption that the adiabatic wall temperature may differ from the measured fluid temperature, with a degree of spatial variation across the surface of the heat transfer plate. To account for this, calibration data of the crystal response against temperature is used to generate a look up table of ideal intensity plots from possible combinations of $T_{aw}$ and $h$. The ideal intensity plots are generated by solving equation (3.4) for the appropriate values of $T_{aw}$ and $h$ to generate the theoretical wall temperature transients that would result from those conditions. The calibration data is then used to produce time plots of theoretical crystal response that are compared against the recorded crystal intensity time plots. By matching the recorded and theoretical data, the actual local $T_{aw}$ and $h$ may be found.

This method has good potential to make fuller use of the crystal data than just identifying the peak intensity, with the full bandwidth of the reaction used. This method was considered worth pursuing as a method of analysing liquid crystal data and an investigation into the use of the full crystal history was carried out using crystal data captured on the Sector-Rig. It was found that uncertainty in the local fluid temperature transient profile was such that when processing the data, numerous combinations of $T_{aw}$ and $h$ were found to produce matching crystal responses. As a result it was considered that without more detailed
knowledge of the exact gas temperature distribution, there was insufficient in the results provided using this approach for it to be implemented.

### 3.7 Uncertainty in Calculated Heat Transfer Coefficient

Uncertainty is an unavoidable part of experiments, as no measurement equipment is 100% accurate. In order for experimental results to provide meaningful information it is essential that the uncertainty of any published results be quoted. A general definition for the uncertainty of experimental results is the range of values in which there is a 95% confidence that the actual value will lie, referred to as the 95% confidence range. A well-established technique for evaluating uncertainty is provided by the Kline-McClintock method [94] that defines the uncertainty $u$, of a result $R$, as a function of independent variables $x_1, x_2, x_3, ..., x_n$, such that:

$$R = R(x_1, x_2, x_3, ..., x_n) \quad (3.31)$$

If the total uncertainty of the result is taken to be $u_R$, with individual uncertainties of $u_1, u_2, u_3, ..., u_n$ then the total uncertainty is given by equation (8.3).

$$u_R = \sqrt{\left(\frac{\partial R}{\partial x_1} u_1\right)^2 + \left(\frac{\partial R}{\partial x_2} u_2\right)^2 + \left(\frac{\partial R}{\partial x_3} u_3\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} u_n\right)^2} \quad (3.32)$$

#### 3.7.1 Nature of Uncertainty

When carrying out the uncertainty analysis for an experiment the presence of two different categories of uncertainty must be accounted for. These are random uncertainty and bias. Figure 3.12 shows a graphical representation of how these two different types of uncertainty manifest. Bias is a constant uncertainty that results in measured data having values offset a constant amount from the actual value. Bias results from systematic errors within an experiment and its components. Measures may be taken to identify the level of individual bias for each source through calibration/diagnostic experiments. Although the bias uncertainty may be reduced in this manner, there will generally be a limit to how precise a level of calibration may be achieved. As such, some degree of bias uncertainty will remain. Random uncertainty is simply a randomly occurring fluctuation around a correct, central value. With sufficient recorded data, random fluctuations would gradually build up towards a normal distribution from the correct about this central point. Random uncertainty in measurements generally results from noise in the measured data, resulting from physical
conditions or limitations to instrumentation. The level of random uncertainty is the potential amplitude of the error.

### 3.7.2 Minimisation of Uncertainty through Experimental Practice

Considering the approach outlined by the Kline-McClintock method with reference to the solution equation (3.4) and its variants; the uncertainty in \( h \) may be considered to be a combination of the uncertainties of \( \beta \) and \( \Theta \). Considering equations (3.5) and (3.6) the uncertainty of \( h \) may be expressed in terms of \( \beta, t \) and \( \kappa \).

\[
\begin{align*}
\sigma_h &= \sqrt{\left(\frac{\partial h}{\partial \beta} \sigma_\beta \right)^2 + \left(\frac{\partial h}{\partial t} \sigma_t \right)^2 + \left(\frac{\partial h}{\partial \kappa} \sigma_\kappa \right)^2} \\
(3.33)
\end{align*}
\]

The uncertainty of \( \Theta \) may be expressed in terms of \( \beta \), and the measured temperatures, \( T_w, T_i \) and \( T_{aw} \), assuming that uncertainties in the different temperature measurements are unconnected [95].

\[
\begin{align*}
\sigma_\Theta &= \sqrt{\left(\frac{\partial \Theta}{\partial T_w} \sigma_{T_w} \right)^2 + \left(\frac{\partial \Theta}{\partial T_i} \sigma_{T_i} \right)^2 + \left(\frac{\partial \Theta}{\partial T_{aw}} \sigma_{T_{aw}} \right)^2} \\
(3.34)
\end{align*}
\]

\[
\begin{align*}
\sigma_\Theta &= \sqrt{\left(\frac{\partial \Theta}{\partial \beta} \sigma_\beta \right)^2} \\
(3.35)
\end{align*}
\]

A method was developed by Yan and Owen [95] to unify the expressions for uncertainty, equations (3.33) to (3.35) into a single equation. In this analysis the random uncertainties in \( t \) and \( \kappa \) are considered negligible, reducing equation (3.33) to the form used in equation (3.36). \( u_h \) is expressed as a function of the uncertainties in the temperature measurements linked by a factor \( \Phi_h \), referred to by the authors as amplification factor.

\[
\begin{align*}
\left(\frac{u_h}{h}\right) &= \Phi_h \left(\frac{u_T}{\Theta_{aw}}\right) \\
(3.36)
\end{align*}
\]

Where:

\[
\begin{align*}
\Phi_h &= \frac{\{2(1 - \Theta + \Theta^2)\}^{1/2}}{\sqrt{2\beta}\{\beta(\Theta - 1) + \pi^{-1/2}\}} \\
(3.37)
\end{align*}
\]
Equations (3.36) and (3.37) were the result of rearranging equation (3.35) in terms of \( u_\beta \) and solving for the case where the measured temperatures have equal uncertainties \( u_T \). Yan and Owen identified that \( \Theta \) is related only to \( \beta \) and as such the amplification factor is independent of \( h \), varying only with non-dimensional temperature, as shown in Figure 3.13. The amplification factor can be seen to remain relatively level in the range \( 0.3 \leq \Theta \leq 0.7 \), with a minimum value observed by Yan and Owen of \( \phi_h = 4.4 \) at \( \Theta \approx 0.5 \). Beyond the specified range it can be seen that \( \phi_h \) increases rapidly as the non-dimensional temperature approaches the limits of 0 and 1. This is useful behaviour to consider when selecting crystal reaction temperatures and the magnitude of the step change. To ensure minimum uncertainty experimental conditions should be defined such that:

\[
\theta_w \approx \frac{1}{2} \theta_{aw}
\]

The method of Yan and Owen was adapted to develop expressions for the amplification factor that could be applied to calculate uncertainty for the exponential and exponential series methods of measuring \( h \) [96] that have been discussed within this chapter.

### 3.8 Closure

Different analysis approaches that have been investigated within this chapter were explored to identify the most appropriate method for analysing experimental data.

It was considered that solutions obtained using Schultz and Jones’ solution equation (3.4) would incur too great an error due to the inaccurate consideration of the step-assumption. Of the temperature approximation methods, the step-series approach has been chosen due to improved accuracy over the step-assumption. Uncertainty of the approach may be minimised by taking high frequency gas temperature measurements and using very small time steps. The method was preferred over the exponential series approach, as the gas-temperature approximation may be taken directly from recorded data and the same solution method may be applied at all different locations and conditions. It was considered that for different locations and different Reynolds cases, the gas temperature transient may change. This would require specific tailoring of modelled profiles if using the exponential or series method, for comparable levels of accuracy. To account for decreasing fluid temperature along the channel length in the Sector-Rig, due to heat loss to the walls a version of the mixed bulk analysis method developed by Metzger and Larson [89] will be used. The
method in question uses the energy balance analogy for a spatially isothermal wall to determine \( h \) based upon the local bulk mean gas temperature based upon an \( h \) calculated from a known inlet temperature. The full intensity history method was considered, but deemed inappropriate for this purpose due to an inability to ascertain the local gas temperature to a sufficiently tight range at specific points. This approach output numerous possible combinations of \( T_{aw} \) and \( h \) that could produce the same plot, resulting in unacceptable uncertainty in the technique.
3.9 Figures

Figure 3.1: Hysteresis effects on crystal reaction temperatures [72].

Figure 3.2: Liquid crystal/heater packaging arrangements [73].
Figure 3.3: Example application and orientation of liquid crystal paint to impingement surface in transient techniques [74].

Figure 3.4: Comparison of Biot number error with $\lambda$ for the corrected and uncorrected values.
Figure 3.5: Effect of time constant, \( \tau \) on modelled temperature transient.

<table>
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<th>Number of terms</th>
<th>( T_{a,0}(^\circ C) )</th>
<th>( T_{a,j}(^\circ C) )</th>
<th>( \tau_j ) (s)</th>
<th>( \Delta T_a(^\circ C) )</th>
<th>( S(\Delta T_a)(^\circ C) )</th>
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<tr>
<td>( m = 0 ) (step)</td>
<td>48.80</td>
<td>48.80</td>
<td>0.394</td>
<td>1.329</td>
<td>1.963</td>
</tr>
<tr>
<td>( m = 1 )</td>
<td>47.72</td>
<td>48.80</td>
<td>0.540</td>
<td>0.923</td>
<td>1.426</td>
</tr>
<tr>
<td>( m = 1' )</td>
<td>48.80</td>
<td>48.80</td>
<td>0.540</td>
<td>0.923</td>
<td>1.426</td>
</tr>
<tr>
<td>( m = 2 )</td>
<td>48.65</td>
<td>19.64</td>
<td>0.123</td>
<td>6.14</td>
<td>0.008</td>
</tr>
<tr>
<td>( m = 3 )</td>
<td>49.06</td>
<td>17.01</td>
<td>0.096</td>
<td>5.08</td>
<td>1.33</td>
</tr>
</tbody>
</table>

Figure 3.6: Example of exponential series approximation to experimental gas temperature rise [85].
Figure 3.7: Fluid temperature response and step-series approximation [86].

Figure 3.8: Computational grid system used by [82]

Figure 3.9: Energy balance over (a) an integral volume and (b) a differential volume [88]
Figure 3.10: Method to correct estimated heat transfer coefficient [82]

Figure 3.11: Comparison of 3D and 1D heat transfer coefficient calculation methods [82]

Figure 3.12: Nature of different types of uncertainty.
Figure 3.13: Variation of $\phi_h$ with $\theta$ for step-change case, illustrating impact on uncertainty. [95]
Chapter 4 Experimental Procedure and Analysis

Technique

4.1 Overview

The primary focus of this thesis is an investigation of improved methods for cooling the cold side of combustor liners in lean burn gas-turbine applications. With this aim, two different experimental facilities have been developed, targeting separate branches of investigation. The first of these experimental facilities, referred to as the Full-Tile Test Facility, features a large scale model of a complete EFE (Environmentally Friendly Engine) combustor liner tile featuring a cooling geometry comprising impingement jets over a smooth plate featuring an effusion cooling array. This was aimed in part at evaluating the impact of localised structural blockages on the cooling performance of the tile in the affected regions and also investigating the impact on cold-side cooling performance of pressure blockages to flow exiting through the hot skin. This second area of investigation for the full-liner tile model is in response to observed static pressure fluctuations occurring within the combustion chamber as a result of lean burn combustion [6]. In addition to investigating the implications of structural and pressure blockages on heat transfer performance, an evaluation into the general, unobstructed array is of interest. As many Nusselt-Reynolds correlations of heat transfer performance are calculated from benchmark tests of regular arrays, an evaluation of the performance measured from a complete representative tile could establish any deviations from such performance predictions derived within a more controlled environment.

The second test facility, referred to as the Sector-Rig Test Facility, has been developed to measure the heat transfer performance of an array of impingement jets ejecting into pedestal roughened channels. By altering select parameters of the pedestal array in the cooling channel, the aim is to identify the possible benefits of combining impingement and pedestal cooling and to establish an optimised design for comparison against traditional cooling methods such that the potential of combining cooling techniques could be assessed. The ultimate objective of the parametric study is to evaluate whether the tested geometries could potentially be applied to a cooling tile similar to that tested on the Full-Tile Test Facility and whether the results indicate that double skin combustor liners employing zero effusion cooling methods may be pursued as a viable option in lean burn gas-turbine engines.
Both facilities are concerned with measuring surface heat transfer coefficient \( (h) \) over a geometry representative of a combustor liner tile, using a transient liquid crystal approach. As a result, the two test facilities feature numerous fundamental similarities, associated with the common measurement technique that they employ. In order to avoid repetition, the experiment design is split into three chapters. Two rig specific chapters outline the design and scaling of the individual facilities, including the development of the cooling geometries under investigation. Included within the present chapter is an overview of the objectives of the two independent test facilities used for capturing data and details the common measurement and analysis techniques used for both test facilities. The design of the Full-Tile Test Facility is detailed in Chapter 5 and the design of the Sector-Rig Test Facility is detailed in Chapter 6.

4.1.1 Full Tile Test Facility

In order to evaluate different cooling techniques and the influence of aerodynamic and geometric variations on performance, investigations into heat transfer performance are typically focussed on idealised models based on representative sectors of engine specific applications. Such experiments are necessary for systematic investigation into the impact of different parameters on cooling performance and provide essential information; however, various features present on complete tiles are not necessarily represented. The combustor liners that are the focus of this thesis have been specified as using a double skin arrangement, typical manufacture of such components requires fastening studs to connect the inner and outer skins (illustrated below in Figure 5.3). The presence of these studs present obstructions within the cavity between the hot and cold skins and in the case of impingement cooling, due to the dead space they take up on the liner wall in which no jet holes may be positioned. Further complications that result from the presence of the studs is the obstacle they present to the jet airflow within the cooling channel as well as the requirement for additional, angled jets dedicated to their cooling. Studs are one of several different blockage features present within a combustor liner, with the presence of various ports (see below in Figure 5.3) also presenting similar obstruction to tile-cooling. By reducing the full tile to a model comprising an idealised representative sector, the various complexities of cooling performance arising from the presence of these blockage features may be lost. In order to assess cooling performance in an engine representative setting, accounting for the effects of blockages, a fully modelled combustor liner tile needs to be considered. For this purpose heat transfer measurements from a complete combustor liner tile model are captured using the Full Tile
Test Facility. Full details of the design of the Full Tile Test Facility and the cooling geometry under investigation are included in Chapter 5.

4.1.2 Sector-Rig Parametric Study

The negative impact of film cooling on the radial temperature profile at combustor outlet [10] coupled with experimental results from combustor tests conducted by Rolls-Royce PLC indicating increased combustion efficiency and reduced NOx emissions in response to a reduction in effusion cooling [6] has led to a desire to reduce or remove it as a form of cooling the combustor liner. In the absence of film cooling, significantly improved cold side cooling is required if adequate cooling of the liner tile is to be achieved. Due to the extremely high temperatures within the combustion chamber, the complete removal of film cooling may not be possible; however, a reduction in the amount of coolant injected into the combustion chamber may be achievable. In order to study zero-effusion cooling performance in lean burn combustion, an investigation was proposed into the addition of surface roughness features to a representative sector of the impingement plate, in the cavity between the hot and cold skins targeting increased cold-side convective performance. The use of ribs and pins increases the surface area available for heat transfer and interacts with the wall jet created by the impingement jet to establish regions of secondary impingement.

A detailed review of enhanced cooling methods, included in Chapter 2 referenced studies conducted into combined cold-side cooling methods that highlighted potential improvements in heat transfer performance for impingement jets targeting plates featuring various arrangements of ribs [57-58] and pedestals [57] [60]. In a review of studies into different rib-enhanced impingement jet cooling geometries, conducted by Andrews et al. [5] an associated improvement of $\bar{Nu}$ with the amount of blockage presented by the surface features (up to a maximum investigated blockage of 50%) was observed, but only under high crossflow conditions, with cooling performance weakened at low crossflow.

In order to investigate the potential application of impingement-arrays over pedestal roughened channels for use in lean burn combustor liners, the Sector-Rig Test Facility has been developed. Using this facility the cooling performance of an impingement array representative of that used on the EFE full tile model featured in the Full Tile Test Facility is investigated in response to the presence of a variety of different pedestal arrays on the target. The Sector-Rig Facility has been developed to accommodate a parametric study into the effects of changing pedestal height-to-diameter ratio ($H/D$) while maintaining constant
channel height and also the effect of altering pedestal spacing, \((P/D)\). Further investigation was conducted into the effect of utilising select pedestals to shield impingement jets from the developing crossflow, aimed at improving downstream heat transfer performance for limited additional pressure losses. Full details of the design of the Sector-Rig Test Facility and the range of cooling geometries under investigation for the parametric study are included in Chapter 6.

### 4.2 Experimental Procedure for Surface Heat Transfer Measurement

Within this section the procedure for measuring surface \(h\) using a green-band tracking transient liquid crystal thermography method is detailed. An overview of the method is included in Section 4.3; however for full details of the theory behind the adopted technique readers may refer to Chapter 3.

Operation of the experimental facilities is performed with the aid of Labview for capturing data from thermocouples and pressure transducers, and MATLAB, which interfaces with the camera and captures video data, recording directly to a PC. Temperature and differential pressure measurements captured by Labview are used to establish conditions within the rig. These data are recorded for post-experiment analysis to calculate Reynolds numbers \((Re)\), friction-factor and fractional pressure-drops. Surface and gas temperature data are also used to produce maps of surface heat transfer coefficient. The operating procedure detailed below is common to both Full Tile and Sector-Rig Test Facilities.

Prior to running a test the video camera is connected to the lab PC and accessed using the MATLAB image acquisition software (imaqtool). The camera is positioned to give the desired field of view, ensuring that the entire measurement area may be captured without compromising image resolution by attempting to capture too large an area and that the LED used to indicate mesh heater activation is also visible. To ensure adequate intensity of colour response a source of light directly focussed on the target plate is required. Although transient liquid crystal thermography is more resilient to changes in light intensity than steady state methods, light from other sources is blocked to maintain consistent illumination throughout a test to improve intensity and quality of the captured colour transition. As it is necessary to view the target plate through the walls of the rig, the camera and light source must be arranged such that reflections from the surface do not interfere with the captured image. The
use of narrow-band crystals renders the crystal calibration insensitive to the relative viewing and lighting angle [65] [70], allowing a degree of freedom in this operation. The Labview program used for running the rig is activated and measurements are taken of ambient temperature and pressure. The pressure transducers are connected to the rig at the correct tapping locations. The fan is activated and set to the required speed for matching $Re$ for the current test. The front panel of the Labview code displays live measurements of $Re$ calculated from the captured pressure and temperature data. These values are estimates used for guiding testing, with recorded data saved in its raw form and processed later using MATLAB. As the power required by the mesh heater to generate a given temperature change depends on mass flow rate, this is set with the fan running at speed, prior to testing. Once the power supply is set the heater is deactivated and the fan is left running to allow thermal equilibrium to be re-established, using thermocouple readings to monitor fluid and wall temperature.

After preparation has been carried out, capturing data is a brief and straightforward procedure. With the fan running at design speed, the Labview code is activated and data capture is triggered. Video recording is then triggered from the imaqtool, and the power supply to the mesh heater manually activated immediately after capturing is triggered. The video recording captures 1200 frames at 25Hz, a duration of 48 seconds. Once capturing is complete, the power supply is deactivated and data capture in Labview is stopped. The rig is kept running at speed and the internal temperature monitored until thermal equilibrium is re-established in preparation for the next test. Ambient conditions and the amount of heat input to the rig alter the amount of time for conditions to reset, but typically tests may be carried out at intervals of 20-30 minutes.

4.3 Heat Transfer Measurement

Surface $h$ is measured using a green-band tracking transient liquid crystal thermography method, a technique first demonstrated by Ireland and Jones [76]. The analysis procedure used to determine $h$ from captured temperature data detailed within this section is based upon an adaptation of the basic method utilising a solution of Fourier’s one-dimensional conduction equation, developed by Schultz and Jones [77], detailed fully in Chapter 3.
4.3.1 Liquid Crystal Analysis Technique

Measurements of the distribution of heat transfer coefficient (and thereby Nusselt number) are made using transient liquid crystal thermography, tracking the transition of the green band of the crystal response. This is an optical technique that allows spatially resolved Nusselt number distributions to be obtained on complex models. The method employs a Perspex model of the cooling geometry that is spray coated with an encapsulated liquid crystal mixture backed with a sprayed layer of matt-black paint to provide contrast against which the crystal colourplay may be observed. A video recording system captures the liquid crystal colour transition, with the recording fed straight to a PC using MATLAB’s imaqtool. In-situ liquid crystal calibration is required for accurate testing as the colour response varies with mixture quantities and the thickness of the coating. A fast response mesh heater, capable of heating the inlet gas be tens of degrees, covers the entire inlet section and is used to simulate a step-change in the temperature of fluid entering the test facility to initiate the wall temperature transient.

Data captured during a transient liquid crystal experiment is stored and may be analysed after testing is complete. This allows multiple tests to be carried out consecutively and analysed upon completion of the series of tests. The method by which liquid crystal colourplay is analysed to obtain surface heat transfer data uses a solution of Fourier’s one-dimensional conduction equation [77]. The analytical method used for heat transfer calculation is discussed below. In response to a change in fluid temperature imparted by the mesh heater, the heat transfer surface experiences a temperature transient as shown in Figure 4.1, rising from the initial equilibrium state towards a stable value determined by the fluid temperature and the local heat transfer coefficient. In the transient liquid crystal technique, the crystal colourplay resulting from this wall temperature transient is analysed to calculate the heat transfer coefficient. As the wall temperature reaches certain prescribed values, the crystal reacts with a defined colour response. The locations of the peak intensities that have been superimposed over the temperature transients in Figure 4.1 indicate the times at which the wall has reached the crystal reaction temperatures. Data reduction of the recorded crystal colourplay is performed using Matlab and spatially resolved heat transfer coefficients are calculated using a solution of Fourier’s one-dimensional conduction equation. The data reduction method is discussed later in this chapter.
4.3.2 Liquid Crystal Materials and Calibration

The liquid crystals used for the experiments are manufactured by Hallcrest Ltd. and are narrow-band (~2°C green colourplay range) with red transition temperatures of nominally 30, 35 and 40°C. The actual transition temperatures are calibrated in-situ against surface mounted T-type thin-foil thermocouples. A typical calibration of the green response for a mixture featuring equal quantities of the three crystals is shown in Figure 4.2. The calibration process uses a slow progression of wall temperature by making incremental increases of power to the mesh heater and therefore the gas-temperature in order to keep the wall close to thermal equilibrium. Thermal conditions within the rig are allowed to stabilise and a still image is taken of the crystal, as is a surface temperature reading using the thin-foil thermocouple. The green intensity response of the crystal at the thermocouple location is extracted from the series of still images over the full range of colour transition for all three crystals. This data is used to produce a plot of intensity vs. thermocouple measured wall-temperature as shown in Figure 4.2. The colour response of a particular encapsulated crystal has been shown to vary for different applications, with the particular mix quantities and coating thickness affecting magnitude and temperature range of the reaction. To ensure heat transfer coefficients are accurately calculated, individual calibrations are performed for all crystal regions, with multiple calibrations for particularly large application zones.

4.3.3 Use of Mesh Heater for Step Temperature Change

Mesh heaters, invented by Gillespie *et al.* [97] are a means of rapidly heating the flow entering the test section, by positioning a finely woven mesh across the entire inlet area of the rig and energising it with an electrical current. The positioning of these within the two test facilities are illustrated below in Figures 5.1 for the Full-Tile Facility and 6.1 for the Sector-Rig Facility.

\[ \eta = \frac{T_{\text{downstream}} - T_{\text{upstream}}}{T_{\text{mesh}} - T_{\text{upstream}}} \]  

Mesh heaters operate at a high conductive efficiency, as defined by equation (4.1), although this has been shown to be dependent upon the air speed through the mesh and the thickness of the mesh wires [81]. The influence of both of these variants can be seen in Figure 4.3. To maximise efficiency the mesh heaters for both the tile and Sector-Rigs are positioned in large area inlet regions and constructed from a stainless steel plain weave mesh with wire diameter of 40 microns at 250 threads per inch. On energising the mesh heater all
of the temperature of the flow entering the working section is rapidly elevated by typically 30 K relative to ambient temperature. This transition from zero to full load occurs in approximately 0.2 seconds and allows a near step-change in rig inlet flow temperature to be achieved. A typical plot of the variation in flow temperature entering the tile is shown in Figure 4.4, illustrating the proximity to a step-change in temperature that can be achieved using the mesh heater.

The mesh heaters are powered by direct current supplies, delivered through copper bus bars soldered along the mesh edges. The bus bars provide a uniform current density and hence spatially uniform heater power. To enable maximum power to be obtained from the power supply, the mesh must be correctly sized such that it has resistance matching the Voltage and current of the source as illustrated in Table 4.1. This may be done by tailoring the length and width of the heater mesh to provide a particular resistance calculated using equation (4.2) to values of 0.045Ω and 0.026Ω for the Tile and Sector-Rigs respectively.

<table>
<thead>
<tr>
<th>Test Rig</th>
<th>Maximum Voltage</th>
<th>Maximum Current</th>
<th>Maximum Power ((P = IV))</th>
<th>Resistance Target: (R = V/I)</th>
<th>Actual: (R = \frac{\rho L}{A_{\text{mesh}}})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sector-Rig</td>
<td>60V</td>
<td>200A</td>
<td>12kW</td>
<td>0.30Ω</td>
<td>0.28Ω</td>
</tr>
<tr>
<td>Tile-Rig</td>
<td>50V</td>
<td>1000A</td>
<td>50kW</td>
<td>0.05Ω</td>
<td>0.045Ω</td>
</tr>
</tbody>
</table>

Table 4.1: Mesh heater power supply details.

\[ R = \frac{\rho L}{A_{\text{mesh}}} \quad \text{(4.2)} \]

Where: \(\rho = \text{resistivity} \ (7.2 \times 10^{-7} \Omega \cdot m)\)

\[ A_{\text{mesh}} = A_{\text{wire}} \times n_{\text{wire}} = \pi (2 \times 10^{-5})^2 \times \left(\frac{250}{25.4} \times \text{width}\right) \]

As all of the electrical power supplied to the mesh is used to heat the air, the available temperature rise that a mesh heater produces may be closely predicted using equation (4.3) [98]. A comparison of the measured and calculated \(\Delta T\) revealed a maximum deviation of 2.5%.

\[ T_{\text{downstream}} - T_{\text{upstream}} = \Delta T = \frac{IV}{\dot{m}c_p} \quad \text{(4.3)} \]

This relationship allows a maximum achievable temperature rise that may be achieved with each power supply to be predicted for a given mass flow rate. Based upon the actual
resistances of the mesh heaters in the two test facilities, the useful power supplied to each is 45\(kW\) for the Tile-Rig and 11\(kW\) for the Sector-Rig. These correspond to maximum temperature rises of 35\(^{\circ}\)C and 40\(^{\circ}\)C respectively at mass flow rates corresponding to the engine maximum pressure condition supplied by Rolls-Royce PLC [6].

The large power supply to mesh heaters requires measures to be taken to avoid overheating. In the absence of airflow through the mesh, the wires become very hot and are at high risk of severe damage. As a safety measure, a connection is made to the power supply, in which the heater will not activate until the mass flow rate through the rig reaches a defined level and cuts off automatically if the heater deactivates. During heater operation, overheating of the portion of mesh sandwiched between the bus bars is a potential hazard to the rig casing. This enclosed portion of the heater can reach very high temperatures and in order to protect the Perspex rig, the bus bars are insulated using Tufnol, a Polyimide glass fabric laminate with thermal endurance up to temperatures of 150\(^{\circ}\)C. Fully casing the heater within Tufnol strips prevents heat penetrating through to the Perspex for the experiment durations used within transient testing.

### 4.3.4 Indication of Experiment Start Time

The analysis method used to evaluate the crystal data is based on identifying the wall temperature transient in response to a near step change in gas temperature. Accurate readings of gas temperature transient are required for calculation of \(h\) as is an accurate indication of the time taken for crystal green responses to be displayed. It is essential that these separately logged data sets are synchronised. As the transient liquid crystal experiment commences upon activation of the mesh heater, this is used to provide an indication of the start time that can be identified visually in the camera recording and is also recorded through the NI data acquisition equipment. This is achieved by connecting an LED to the heater power supply such that it illuminates at the moment the heater is energised. Positioning the LED within the field of view of the camera provides a visual indication of mesh heater activation. By also tracking the voltage supply to the LED using the same Labview code used for taking rig aerothermal measurements, the mesh heater activation time is identifiable within a channel of the data written to the test log. This allows accurate synchronisation of the video recording and the Labview data file to a degree of certainty limited by the frame rate of the camera.
4.4 Instrumentation

The test conditions within the Tile-Rig are monitored using various static pressure tappings and thermocouple sensors. Data acquisition from these sensors is accomplished using National Instruments SCXI system signal conditioning and digitisation equipment, connected to the PC via a USB using a NI-SCXI-1600, 16-bit digitiser.

4.4.1 Pressure Measurement

Differential pressure measurements are required in the test rigs to determine pressure drops at strategic locations and also for evaluating flow velocity, using equation (4.4) in order to determine mass flow rate and $Re$ for the rig, as well as to calculate loss coefficients across different rig stages.

$$U = \sqrt{\frac{2\Delta p}{\rho}}$$  \hspace{1cm} (4.4)

The pressure measurements within the rigs are captured using differential pressure transducers, supplied by Furness Controls and Sensor Technics. These are connected to the PC using a NI-SCXI-1143 device, an 8-channel Butterworth low-pass filter. A NI-SCXI-1304 BNC connector block is used to attach the transducers to the SCXI unit. Pressure transducers have been selected with ranges of ±50, 250 and 700 mm H$_2$O to accommodate the different stage pressure drops present within the rigs. Detailed transducer arrangements are included in Chapters 5 and 6 for the Full-Tile and Sector-Rig Facilities respectively.

Calibration of the pressure transducers is performed prior to testing whereby the voltage output in response to a known differential pressure, referenced to a digital manometer, is recorded across the full operable range, positive and negative. The gradient of the voltage response of a transducer is consistent with time; however the voltage at zero pressure difference has been shown to vary. In order to ensure accurate calibration during testing zero pressure voltage readings are taken prior to each test and used with the calibrated gradients to measure pressure difference.

4.4.2 Temperature Measurement

The application of the transient liquid crystal methodology requires knowledge of the gas temperature history at inlet to the working section. Fine-wire exposed junction thermocouples are used for this purpose (Omega Engineering, type K, wire diameter
0.05\text{mm}). The gas thermocouples are suspended across the inlet to particular impingement holes across the two test facilities. Heat transfer measurements on the Tile-Rig are divided into five Measurement Regions, with gas (and surface) thermocouples positioned within each region. To establish any variation in impingement jet temperature resulting from the use of a side feed inlet channel, gas-thermocouples in the Sector-Rig were positioned over holes at different streamwise locations. Full details of the thermocouple positions are included in the relevant rig design chapters. Placing the junction at impingement hole inlet situates the thermocouple in an appropriate location to establish the inlet thermal boundary conditions, and also ensures a sufficiently fast temporal response due to the elevated flow velocity in this region. The typical variation in flow temperature during a transient test shown in Figure 4.4 illustrates the near-step change of \(-35\) K following mesh heater activation at \(-9\) seconds with a stability of \(\pm 1\) K for the remainder of the test.

Measurements of the surface temperature are required for in-situ calibration of the rig and to evaluate thermal equilibrium between the fluid and surface prior to commencing a test. Thin foil thermocouples are used for this purpose (Omega Engineering, type T, foil thickness 0.013\text{mm}). Thermocouples are attached to the plate at each different measurement region, positioned away from the impingement jet locations in regions of lower \(h\) to ensure the wall is close to thermal equilibrium through the calibration. This allows in-situ calibration to be performed more easily than if positioned in a region of high heat transfer.

Gas and surface thermocouples are connected to the PC using a NI-SCXI-1120 8-channel isolation amplifier. Individual thermocouples are connected to the unit using a NI-SCXI-1328 isothermal thermocouple input with built in cold-junction compensation. Calibration of the gas and surface thermocouples is carried out using a platinum resistance thermometer (PRT). Thermocouple readings are taken under steady-state ambient conditions and compared against the PRT readings. The offset of the thermocouple temperature readings from the PRT recorded value is used to identify and remove the bias error as shown in Figure 4.5.

4.5 Data Analysis Method

For recorded liquid crystal colour data to be used to measure \(h\) the recorded data must be processed and analysed. The data reduction technique used on both test rigs is described within this section. The analysis code detailed in this section is the original work of the author, using the solution to Fourier’s one-dimensional conduction employed by Ekkad and
Han [86] in which the gas temperature rise is modelled as a series of infinitesimal steps to solve for \( h \) based upon recorded gas temperature transients and liquid crystal data. A detailed discussion of the theory behind the analysis has been outlined in Chapter 3.

\[
T_w = T_0 + \sum_{j=1}^{m} \left[ 1 - \exp \left( \frac{h^2(t - t_j)}{\kappa^2} \right) \times \text{erfc} \left( \frac{h \sqrt{(t - t_j)}}{\kappa} \right) \right] [\Delta T_{aw(j,j+1)}] \tag{4.5}
\]

Liquid crystal colour changes are recorded directly to PC from a video camera, using MATLAB’s imaqtool to write the video as an ‘avi’ file. The digitised video comprises three signals; the red, green and blue (RGB) intensities. Using MATLAB, the captured colour data is written into a 4D array of spatially and temporally resolved colour intensity for the three different colour channels. The green signal provides the clearest defined peaks and hence the green colour data is extracted from the avi file and used for tracking the surface temperature transient. A typical intensity history for three narrowband crystals combined for application onto the measurement surface is shown in Figure 4.1. The colour intensity increases as the crystals approach their reaction temperatures. Calibration of the liquid crystals, as shown in Figure 4.2, relates the intensity peaks for the crystals to defined surface temperatures.

Green intensity histories are obtained for all spatial locations within defined measurement regions, a typical example is shown in Figure 4.6. The heat transfer areas under investigation comprise typically 2-3x10^5 pixels; this requires the crystal colourplay analysis and heat transfer calculation to be automated. Two distinct processes have been developed; the first identifies the presence of crystal colour intensity peaks; if a complete response is identified then the time at which the peak green intensity is reached is recorded. The second process uses the reduced crystal data to produce surface \( Nu \) maps by iteratively solving equation (4.5) at all locations. All analysis is performed in MATLAB with series of interacting script files used to govern the calculations.

The code to identify green response of the liquid crystals, referred to as the ‘Peak Finder’ code consists of several script files, tasked with processing the video files to extract the green signal and also to identify the start time of the experiment. For the start time to be identified, a red LED is placed in the camera field of view. The LED is linked to the mesh heater power supply, illuminating at the moment the mesh heater is energised providing a visual indication of heater activation time. The LED location is input to the Peak Finder for each measurement region, and by tracking the red signal at that location, the start time can be identified. Still images of the LED taken from the video file are provided for the user, and
confirmation of LED activation is required for the code to continue. This is the final user input required by the script. With time $t_0$ identified, the raw green crystal intensity is extracted from the video file and written to a 3D matrix, recording the intensity time history for every pixel location. The unfiltered, unsmoothed intensity plots are used to identify locations where no crystal reaction occurs in the regions beneath pins or effusion holes, studs, ports) and this is used to produce a rudimentary mask by recording the non-responsive locations for later reference.

The extracted data is conditioned by taking an average of a 5x5 array of green intensity response centred on the pixel in question. MATLAB is used to apply a Butterworth low pass filter [99] to remove high frequency noise from the signal and the intensity response is normalised to a range from 0-1 by dividing the entire signal over the maximum observed intensity. A comparison of an example raw and a conditioned intensity plot is shown in Figure 4.7. The initial mask definition is used to remove pixels with corrupt or absent crystal histories from the average, allowing uncorrupted crystal data to be obtained from around features and the edge of crystal regions. The term ‘corrupt’ crystal histories refers to locations where crystal data has been captured, but a confident assessment of reaction time cannot be measured. This was most commonly the result of reflections; with Figures 7.24-26 (Chapter 7) illustrating areas where reflections from internal rig components have affected the experimental results. Absent crystal data occur at the edge of measurement regions and in positions where liquid crystal could not be applied (effusion hole and pedestal locations). In areas where crystal responses are removed, the 5x5 location is only used if more than one-third of the pixels (9 from the 25) provide valid data. If less than nine active pixels are available in a given region then it is assigned as not-a-number (NaN). A drawback to averaging multiple pixels to smooth the data is that the resultant average intensities of neighbouring pixels share some of the same data. The size of individual pixels ~0.8 mm in the largest case, ensures that a high degree of resolution is maintained in the output heat transfer data.

The Peak Finder code analyses the green intensity data to identify the time at which the different peaks occur, using a fully automated logic process. The tile and Sector-Rigs used different solver functions, as the Tile-Rig has three different liquid crystals combined on the heat transfer surface whilst the Sector-Rig features two. The logic used by the codes is the same, although the process is significantly less complex in the two crystal case. Large spatial variations in heat transfer coefficient make fully automated identification of the location of
three crystal peaks difficult as full transition of all crystals may occur in less than 5 seconds in the regions of high \( h \); or more than 30 seconds may pass before the first crystal peaks in regions of low \( h \). In order to account for this, the solver first identifies the peak with highest maximum intensity, regardless of which crystal produces it, although it was identified through practice that the final crystal typically exhibited the highest peak. Due to this, it was necessary to ensure that the identified green response had reached maximum intensity and was not in transition by verifying that the peak had not occurred in the last 5% of the test duration. This was an arbitrary scale chosen that was found to give positive results, with minimum drop out of data. Typically, due to the peak values in slow transitions being less defined the confidence of a peak value occurring at a time greater than this would be low.

If the peak was found to occur beyond the cut-off time it was recorded as an incomplete reaction and isolated; otherwise, the reaction time \( t_{\text{peak}} \) was recorded as a complete reaction and the peak isolated. Isolation of the peaks refers to mapping forwards and back from the peak reaction time to find the limits of the response. The peak finder then searches backward and forwards in the intensity-time series to identify additional peaks until all have been identified as present or absent. The times at which the different crystals reach peak intensity are recorded in independent matrices and saved for use in the Nusselt calculation portion of the code. In the case of incomplete colour transition in which a green response is not elicited from all crystals a NaN value is recorded for the unresponsive crystal.

The ‘Nusselt Calculation’ code produces maps of surface \( h \) using data from the reactions of an individual crystal and the recorded gas temperature history. The code uses the thermophysical properties of the Perspex, \( \kappa = 567 \, \text{W} \cdot \text{s}^{0.5} / \text{K} \cdot \text{m}^2 \) and crystal calibrations specific to the measurement region under investigation to produce maps of surface \( h \), which are then converted to \( Nu \). Heat transfer coefficient is calculated iteratively, with the reaction time and an initial estimate of \( h = 150 \, \text{W} / \text{m}^2 \text{K} \), input to equation (4.5) to calculate \( T_w \) based upon those parameters. This calculated \( T_w \) is compared to the calibration temperature \( T_{lc} \). The iterative solver uses the logic shown in equation (4.6) to solve for \( h \) for an error of \( \pm 0.1^\circ \text{C} \), corresponding to an error in \( h \) of \( \pm 0.5\% \). Once \( h \) has been calculated for a particular location, the assumed value of \( h \) is reset and the process repeated.

\[
\text{If: } |T_{lc} - T_w| \geq 0.1
\]

\[
h_{\text{new}} = h + h \left( \frac{T_{lc}}{T_w} - 1 \right)
\]
The transient adiabatic wall temperature input to equation (4.5) is taken in the case of the Tile-Rig from a thermocouple at impingement exit above the measurement region; for the Sector-Rig, thermocouple measurements are taken from just downstream of the mesh heater. The thermocouple measurements are captured at a rate of 1 kHz. Recorded data are averaged over a time period of 0.04s in order to smooth the measurements and match the 25 Hz capture rate of the camera. This is the time period used in equation (4.5). The temperature transient used in the solver code is synchronised to the video recording using the voltage supplied to the LED. This voltage is recorded alongside pressure and temperature data using Labview, and a step change in the LED voltage indicates the moment of heater activation. This allows the visual and sensory data sets to be synchronised to a level of certainty limited by the camera frame rate.

4.5.1 Mixed Bulk Correction Method

The heat transfer coefficients calculated using the outlined analysis codes are based on readings from a single thermocouple at working section inlet for use as the driving fluid temperature within the solution equations. The plenum fed arrangement used within the full Tile-Rig and the use of locally positioned thermocouples means that the measured temperature can be assumed to incur negligible heat losses due to the distance of only 23mm from hole exit to the target plate. The side feed arrangement used within the Sector-Rig facility however, will result in a reduction in fluid temperature with distance from the heater due to heat loss to the endwalls. For this region an additional analysis code has been developed that calculates $h$ based upon the local mixed bulk gas temperature using the surface maps of $h$ calculated from the temperature at channel inlet.

This analysis is performed using the technique developed by Metzger and Larson [89] discussed in Chapter 3. Equation (4.7) is used within the analysis code, with the area to which heat losses may occur approximated as $\bar{y}$ which is the nominal perimeter of the channel susceptible to heat loss, the exposed perimeter. Rohacell spacers at the sides of the channel and the areas obstructed by pedestals are considered adiabatic and therefore heat transfer is considered to occur only to the Perspex walls of the hot and cold skins. The exposed perimeter, $\bar{y}$ is defined in equation (4.8), for which $y$ is the width of the channel, $n$ is the number of pedestals and $D$ is pedestal diameter. Note that the subscripts refer to the presence of full, half and quarter height pins with full pins counted twice as these obstruct heat losses to the surface at the base and the tip.
\[
\frac{h_{mb}}{h_e} = \frac{1}{1 - \frac{\bar{y}}{c_p} \int_0^L \frac{h_e(x)}{\bar{m}(x)} dx}
\]  
(4.7)

\[
\bar{y} = 2y - (2n_f + n_{h,q}) \left( \frac{\pi D_{pin}^2}{4l} \right)
\]  
(4.8)

The impingement array feed complicates the equation as fluid is gradually injected into the cooling channel, with mass flow rate increasing with each subsequent row of impingement holes. The presence of the pedestal array within the cooling channel also means that pressure drop within the cooling channel results in variable mass flow rate through the different jet rows. Row pressure drop is estimated using interpolation from measurements taken at inlet, midpoint and exit to the feed and cooling channels. This is in turn used to calculate the mass flow rate injected through each jet row, from which a streamwise variation of mass flow rate in the direction of the channel exit, \( \bar{m}(x) \) is plotted. Heat loss to the walls of the inlet channel is considered small when compared with losses in the cooling channel, and as such these are not considered in the analysis. In practice average heat transfer coefficient in the inlet channel, approximated using the Dittus-Boelter equation and compared to measured results has been found to be equivalent to \( \sim 10\% \) of those occurring within the cooling channel.

**4.6 Closure**

The experimental technique outlined in this chapter by which aerodynamic and heat transfer measurements were recorded is applicable to both the Full-Tile and Sector-Rig test facilities that were introduced at the start of the chapter. As both facilities employ the same technique to measure surface \( h \) many of the same features are shared between the facilities with similar instrumentation and hardware used. In addition to the equipment, the analysis code designed for translation of the captured liquid crystal colour data to maps of surface \( h \) which may be converted to \( Nu \) is used to interpret the data from both test facilities.

Design of the two test facilities is included in Chapter 5 for the Full Tile-Rig and Chapter 6 for the Sector-Rig. Although many of the common features have been introduced here parameters specific to the two rigs, such as placement of instrumentation are included in the relevant design chapters.
4.7 Figures

Figure 4.1: Temperature transient resulting from a step change in temperature and the resulting colour response of 30, 35 and 40° C crystals applied to the surface.

Figure 4.2: A typical liquid crystal green-response calibration obtained in-situ on the Tile-Rig.
Figure 4.3: Mesh convective efficiency as a function of flow speed, including a prediction of changing wire diameter.

Figure 4.4: A typical gas temperature history during a transient heat transfer test on the Tile-Rig (mesh heater turned on at ~9 seconds).
Figure 4.5: Ambient temperature thermocouple signals (a) uncorrected readings. (b) referenced to PRT reading.

Figure 4.6: A typical video frame obtained during a liquid crystal heat transfer test (obtained from Sector-Rig).

Figure 4.7: Comparison of raw intensity plot with filtered and averaged intensity centred to the same location.
Chapter 5 Full Tile Test Facility

Development of the full tile test facility was undertaken with the aim of characterising the aero-thermal performance of an impingement-effusion cooling system within a complete lean-burn combustor liner tile geometry. By modelling the tile in its entirety the impact of blockage features (fastening studs, igniter ports) upon cooling performance may be determined and the ability to cool these components may be evaluated. Additionally, restrictor plates positioned at effusion exit may be used to simulate pressure blockages that result from instabilities in the combustor internal static pressure field allowing the impact of such blockages on the flow distribution and convective cooling performance of typical lean-burn combustor tile geometries to be assessed. The experimental procedure for this facility has been detailed in Chapter 4.

5.1 Overview

The full tile test facility features a laboratory-based model of a complete combustor tile built to a geometric scale-factor of 7.5. The modelled tile, based on a representative outer annulus cooling tile from early EFE [6] combustor designs, measures approximately \(1.2 \times 1\) m in plan form. In order to simplify manufacture, the tile model is flat and therefore does not simulate the curvature present in the annular engine geometry; this however, is considered to represent a minor effect on the coolant flow. An impingement-effusion cooling system was adopted for the tile inspired by a previous pattern used in ANTLE [6] as no confirmed EFE cooling pattern existed at the time of rig design.

A schematic diagram of the Tile-Rig Test Facility is shown in Figure 5.1 and a photograph of the working section is shown in Figure 5.2. Air is drawn through the tile model by a centrifugal fan at up to 1.5 \(m^3/s\) and an overall tile pressure drop of up to 9 kPa. The flow enters the tile through a large rectangular inlet across which is suspended a particulate filter. As shown in Figure 5.1 the air then passes through a mesh heater before reaching the impingement (cold skin) holes of the tile model. The mesh heater (detailed further in the following section of this chapter) is important to the application of the transient liquid crystal method for measuring the heat transfer coefficient. Heat transfer performance is measured using liquid crystal thermography. The working section and connecting duct of the rig are constructed from Perspex, as seen in Figure 5.2, so as to allow optical access of the crystal colourplay using a video camera. After passing through the tile model, the airflow is
collected in a large plenum where it is turned through 90° and passes to the inlet of the metering plenum before the air is passed to the fan and is discharged into the laboratory. The exhaust flow is directed upwards towards an extractor in order to enable heated spent air to exit the test chamber. The Tile-Rig also includes a means for varying the static pressure distribution at the exit of the hot skin plate. This has been achieved by employing additional flow restrictor plates in three axially arranged regions downstream of the hot skin.

The Tile-Rig is essentially plenum-fed and does not simulate the impact of crossflow that would normally be present in the combustor outer annulus. The design includes facility to adopt a side-fed layout through an alternative coolant feed system. However, at the time the rig was manufactured the definition of this annulus (and the potential impact of blockage within the annulus) was not sufficiently defined to justify the setup. A plenum fed approach is preferred to avoid heat losses in the feed channel and to provide improved confidence of uniform temperatures across the plate area. The test conditions within the Tile-Rig are monitored using various static pressure tappings and thermocouple sensors. Data acquisition from these sensors is accomplished using National Instruments SCXI system signal conditioning and digitisation equipment.

5.2 Cooling Geometry

At the design stage of the full Tile-Rig no cooling geometry had been defined for the EFE combustor liner tiles. As such an original pattern was developed to match the tile geometry for predefined coolant flow conditions. In order to ensure an appropriate pattern was adopted, a cooling geometry from a similar engine, ANTLE [6], was used as a starting reference. The sizing and definition of the impingement-effusion geometry used on the Tile-Rig is detailed within this section.

5.2.1 Tile Definition

The physical dimensions of the full liner tile were directly modelled upon an EFE combustor outer liner tile as shown in Figure 5.3 (only hot skin for clarity). The EFE combustor features 18 tiles arranged circumferentially and in order to model the liner tile as a flat plate, the tile width was converted from polar to Cartesian units. The flattened tile has dimensions as shown in Table 5.1. The overall depth listed corresponds to a separation between the hot and cold skins of the outer liner of 3.10mm.
Table 5.1: EFE outer combustor tile dimensions [6]

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length:</td>
<td>168.70 mm</td>
</tr>
<tr>
<td>Width:</td>
<td>126.90 mm</td>
</tr>
<tr>
<td>Thickness:</td>
<td>1.40 mm</td>
</tr>
<tr>
<td>Overall depth</td>
<td>4.50 mm</td>
</tr>
</tbody>
</table>

Each tile features several structural components between the inner and outer liners. In addition to the studs and spacers present on each tile, igniter ports are present on certain tiles. A decision was made to use the arrangement shown in Figure 5.3 and model the tile featuring all the possible obstructions. In order to accurately model the tile, efforts were made to model the blockage features as closely as possible on the engine design. A sectional view of a stud is shown in Figure 5.4 with dimensions taken from the model provided by Rolls-Royce. The overall diameter of 4.5 mm includes a 1 mm deep thread surrounding a 43 mm central pillar. The stud has a fillet radius of 1.7 mm at the base. Modelling of the ports present within this geometry required the surrounding barriers shown in Figure 5.3 to be modelled. To ensure the flow fields within the hot-cold skin cavity are modelled correctly flow through the hollow central area of each port is blocked using Perspex cylinders. Simulation of the ports does not continue either side of the hot and cold skins.

5.2.2 Array sizing

Whilst the tile model was in development, no defined cooling geometry had been adopted for EFE. This necessitated the design of an appropriate jet array for use in the model based upon source data and a similar previous cooling geometry (Figure 5.5). EFE tile inlet conditions for the upper and lower engine limits were provided by Rolls-Royce; this data is displayed in Table 5.2.

<table>
<thead>
<tr>
<th></th>
<th>Engine upper limit</th>
<th>Engine lower limit</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_o$</td>
<td>48</td>
<td>5</td>
<td>bar</td>
</tr>
<tr>
<td>$T_o$</td>
<td>900</td>
<td>500</td>
<td>K</td>
</tr>
<tr>
<td>$\rho$</td>
<td>15.5</td>
<td>3.48</td>
<td>kg/m$^3$</td>
</tr>
<tr>
<td>$\mu$</td>
<td>39.0×10$^{-6}$</td>
<td>27.3×10$^{-6}$</td>
<td>Ns/m$^2$</td>
</tr>
<tr>
<td>$C_p$</td>
<td>1120</td>
<td>1025</td>
<td>J/kgK</td>
</tr>
<tr>
<td>$\Delta p/P_o$</td>
<td>2.8</td>
<td>2.4</td>
<td>%</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>6.55</td>
<td></td>
<td>kg/s</td>
</tr>
</tbody>
</table>

Table 5.2: Coolant flow characteristics for EFE combustor tile.
The pressure drop identified in Table 5.2 is the total across both skins. In order that pressure instabilities within the combustor would not lead to hot gas ingestion it was specified that a minimum proportion of this must be reserved for the hot skin. This was defined as 20% of the total liner pressure drop. For the calculation of impingement and effusion jet velocities it was assumed that the entire pressure drop occurs only across the skins, with any losses between the two considered negligible.

\[
U = \sqrt{\frac{2\Delta p}{\rho}} \quad (5.1)
\]

The jet velocity calculations allowed a total jet array area to be calculated using equation (5.2), with the number of holes being given subsequently by equation (5.3).

\[
A_{\text{array}} = \frac{m}{\rho U C_D} \quad (5.2)
\]

\[
\frac{A_{\text{array}}}{A_{\text{hole}}} = \text{number of holes} \quad (5.3)
\]

The strong dependency of jet \( C_D \) on hole geometry and flow conditions has been discussed in Chapter 2; however to correctly size the array open area, an estimate of \( C_D \) must be made as no measurement can be taken. Based upon the tile thickness of 1.4 mm and a typical impingement jet diameter <1mm, the jets may be treated as long, sharp edged holes. For similar geometries discharge coefficients of \( \sim 0.8 \) were found. In cases with no initial crossflow Florschuetz et al. [100] identified \( C_D = 0.82 \) and Hay et al. [101] observed values of \( C_D = 0.81-0.82 \). The findings of Lichtarowicz et al. [35] and McGreehan and Schotsch [36] also suggested a \( C_D \) in this region. Additionally, for typical impingement jet Reynolds numbers encountered, the \( C_D \) may be considered to be invariant with Re.

<table>
<thead>
<tr>
<th>Section</th>
<th>Pressure drop %</th>
<th>Velocity (ms(^{-1}))</th>
<th>Total mass flow rate (kg.s(^{-1}))</th>
<th>Array area (mm(^2))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impingement</td>
<td>2.24%</td>
<td>1.075</td>
<td>118</td>
<td>0.364</td>
</tr>
<tr>
<td>Effusion</td>
<td>0.56%</td>
<td>0.269</td>
<td>59</td>
<td>0.364</td>
</tr>
</tbody>
</table>

Table 5.3: Calculation of impingement and effusion total open area.

The total open area of impingement and effusion holes required to match the mass flow rate and pressure drop of the upper engine limit provided by Rolls-Royce plc. is presented in Table 5.3. The mass flow rate for the tile is based upon the assumption that the total coolant is divided evenly between the 18 sectors.
5.2.3 Development of Cooling Pattern

With the broad parameters set, impingement and effusion jet arrays were required to provide full cooling coverage of the tile. In order to facilitate this process, a previously received cooling geometry from the ANTLE combustor liner, was used as a starting point. The impingement and effusion geometries are shown in Figure 5.5. The ANTLE cooling geometry featured impingement and effusion holes predominantly with diameters of 0.7mm and 0.75mm respectively. These were arranged into two distinct inline arrays on the impingement plate, with a more densely packed array in the front segment, followed by a sparser pattern at the rear. This was reflected in the effusion jet arrangement, but with staggered, rather than inline arrangements of jets. All of the effusion film cooling holes were angled, with a modal value of 19º to the horizontal. It can also be seen clearly in Figure 5.5(b) that the general array was altered to account for gaps that would appear in the film coverage due to the presence of the central igniter port.

It was decided to adopt these particular features of the ANTLE design, dividing the array into 2 segments and using the same size and angles for the jets. With the open hole areas calculated in Table 5.3, this would correspond to an impingement array of 650 holes of diameter 0.7mm and an effusion array of 1130 holes of 0.75mm, angled at 19º. Given the tile dimensions, an impingement hole diameter of 0.7mm corresponds to an impingement gap-to-diameter ratio of $Z/D=4.4$. The full array that has been developed for the impingement and effusion plates can be seen in Figures 5.6 and 5.7 respectively, each plate having the same dimensions. The circled segments are shown in more detail in Figures 5.8 - 5.12 to provide information regarding the jet arrangement in the arrays and surrounding the blockage features. Table 5.4 is included to summarise the relevant array data.

Care was taken in the construction of the arrays to ensure that by design, no impingement jets would be positioned directly above an effusion jet. For this to be achieved the pitches had to be carefully selected due to the number of holes in each plate required to match the defined porosity being irregular. In the spanwise direction the jet-to-jet spacing of the impingement and effusion jets was matched, but this was not the case in the streamwise direction. This resulted in impingement and effusion relative jet positions that varied with rig location, but never coincided.
### Standard Jet Arrays

<table>
<thead>
<tr>
<th>Array</th>
<th>Spanwise</th>
<th>Streamwise</th>
<th>No. of Jets</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Pitch (mm)</td>
<td>No. of rows</td>
<td>Pitch (mm)</td>
</tr>
<tr>
<td>Impingement</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Narrow</td>
<td>4.6</td>
<td>26</td>
<td>4.7</td>
</tr>
<tr>
<td>Wide</td>
<td>5.2</td>
<td>23</td>
<td>5.4</td>
</tr>
<tr>
<td>Stud, Igniter Port</td>
<td>Angular jets positioned around features.</td>
<td>See Figures 5.9-5.10 for details.</td>
<td>47</td>
</tr>
</tbody>
</table>

**Total number of impingement holes:** 657

| Effusion            |          |            |             |             |           |
|                     |          |            |             |             |           |
| Narrow              | 4.6      | 26         | 2.7         | 25          | 652       |
| Wide                | 5.2      | 23         | 6.2         | 22          | 411       |
| Igniter Port        | Additional jets positioned around igniter port. | See Figure 5.12 for details. | 67          | 67          |           |

**Total number of effusion holes:** 1130

Table 5.4: Summary of cooling array data for impingement and effusion plates.

The use of the radially positioned cooling holes around the blockage features, shown in Figures 5.9 and 5.10, is to limit breakdowns in cooling in areas where holes are omitted due to the local obstruction of structural components. The jets were positioned in an effort to avoid any locations on the plate being under-cooled and also to provide direct cooling of the studs connecting the hot and cold skins. As can be seen in Figure 5.10, jet arrangement around the igniter port involved replacement of the baseline rectangular impingement array with circumferentially positioned jets, to prevent local hot-spots. For the jets around the studs, angled holes were positioned for direct component cooling. A sectional view of the near-stud geometry is shown in Figure 5.4. Three angled impingement holes, with inlet 42.5 mm from stud centre are positioned around the stud, separated by angles of 120°. Different angled holes have been used in an effort to evaluate the effect of the jet incidence upon the cooling performance. The two different angles affect the point on the stud with which the jet makes first contact. Two of the stud jet holes are at an angle of 50° and one is at an angle of 30°. The introduction of the angled impingement alters the gap from jet exit to stud-impingement from the distance of 3.1 mm (at engine scale) of the normal impingement holes. At engine scale, the jet issuing from the 50° hole impinges on the fillet at the base of the stud, at a distance of 2.9 mm from hole exit; the jet from the 30° hole impinges on the stud in the top half of the cooling channel, targeting the formation of a wall jet down the stud length. The shallower angle of the 30° hole results in a much closer jet-to-target spacing of ~1 mm.
Although hot side cooling was not considered in the remit of this experiment, efforts were made to consider the position of the film cooling holes for consistency. In a similar manner to the local feature impingement arrays, an arrangement of effusion holes surrounding the igniter port was proposed. The arrangement, shown in Figure 5.12 was used in an effort to avoid breakdowns in film cooling coverage in the region of the igniter port.

5.3 Geometric Scaling

For heat transfer performance of the cooling tile to be evaluated, a scaled model of the cooling geometry is required for testing at ambient conditions. The large size of the full tile geometry necessitates a compromise to be made when selecting the scale of the model. A large scale factor is desirable to allow matching of aerodynamic conditions, particularly the engine Reynolds and Prandtl numbers. A large scale factor also facilitates transient liquid crystal measurements by reducing the heat transfer coefficient and restricting crystal transition speed. Manufacturing considerations however, place an upper bound on the scale factor that can be used before the tile model becomes too large to construct or house.

A scale factor of 7.5 was chosen for the model, allowing Reynolds matching across the full range of specified engine conditions. Key dimensions of the engine and model tile and coolant properties are shown in Table 5.5.

<table>
<thead>
<tr>
<th></th>
<th>Engine upper limit</th>
<th>Engine lower limit</th>
<th>Tile-Rig</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geometric scale factor</td>
<td>1.00</td>
<td></td>
<td>7.50</td>
<td></td>
</tr>
<tr>
<td>Impingement hole diameter</td>
<td>0.70</td>
<td>5.25</td>
<td></td>
<td>mm</td>
</tr>
<tr>
<td>Effusion hole diameter</td>
<td>0.75</td>
<td>5.63</td>
<td></td>
<td>mm</td>
</tr>
<tr>
<td>Tile Area</td>
<td>0.019</td>
<td>1.08</td>
<td></td>
<td>m²</td>
</tr>
<tr>
<td>Tile thickness</td>
<td>1.40</td>
<td>12.0</td>
<td></td>
<td>mm</td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>0.36</td>
<td>0.05</td>
<td>1.31</td>
<td>0.27</td>
</tr>
<tr>
<td>(\Delta P/P_o)</td>
<td>2.8</td>
<td>2.4</td>
<td>7</td>
<td>0.3</td>
</tr>
<tr>
<td>(U)</td>
<td>118</td>
<td>74</td>
<td>117</td>
<td>22</td>
</tr>
<tr>
<td>(Re)</td>
<td>33,000</td>
<td>7,000</td>
<td>33,000</td>
<td>7,000</td>
</tr>
<tr>
<td>(Pr)</td>
<td>0.70</td>
<td>0.71</td>
<td>0.71</td>
<td>0.71</td>
</tr>
</tbody>
</table>

Table 5.5: Combustor liner dimensions and coolant conditions under engine operating conditions and for model.

In order for similarity of engine aerodynamic conditions to be achieved by the model at atmospheric conditions it is required that the Reynolds and Prandtl numbers of the coolant are matched. By setting conditions such that Reynolds matching is achieved the Nusselt number
will also be matched between the engine and the model. Due to the low Mach number of the cooling flows, M~0.3 it was considered unnecessary to match Mach number between engine and test-rig conditions as coolant remains in the incompressible range throughout the full testing range. A geometric scale factor of 11.4 was identified as an ideal aerodynamic case for which Re matching would be achieved between the engine and model for the same maximum engine pressure drop of 2.8%; however, a scale factor of this magnitude would correspond to a cooling tile measuring 1.77m by 1.44m. Construction, accommodation and operation of a rig this size is unfeasible and as such a smaller scale factor is required. In order to achieve Reynolds matching at a smaller scale factor, the jet velocity, and correspondingly the pressure drop across the tile would have to be increased. Although this would change conditions from the engine case it was considered acceptable due to the low jet Mach numbers meaning flow remained incompressible and associated effects would be negligible. Selecting a scale factor required constraints of the hardware to be considered, such that testing could be carried out over the entire representative Reynolds range without exceeding the capabilities of the equipment.

The maximum pressure rise and mass flow rate available to the fan were 11.88kPa and 1.79kg.s\(^{-1}\) respectively. For a scale factor of 11.4, approximately 2 kg.s\(^{-1}\) of air would pass through the rig; greater than can be handled by the fan. An additional consideration is the power available to the mesh heater, with a maximum output of 50kW. Assuming a correctly sized mesh and no losses in the supply, a maximum DT of 25.5°C is available. Reducing the scale factor would reduce \(\dot{m}\) and as a result increase \(\Delta T_{\text{max}}\); however the pressure drop load on the fan would increase. An additional concern associated with using a smaller scale factor is the impact on the heat transfer coefficient. This is inversely proportional to scale and for constant \(Nu\), \(h\) increases as scale factor drops. An equation developed by Goldstein et al. [55] correlating \(Nu\) to \(Re\) was used to predict stagnation \(Nu\) for the upper Reynolds limit.

\[
\frac{Nu}{Re^{0.76}} = \frac{(24 - |L/D - 7.75|)}{533 + 44(R/D)^{1.285}}
\]  

(5.4)

The value of \(Nu\) obtained from Goldstein's equation was used to generate peak \(h\) values over a range of scale factors, shown in Figure 5.13. The heat transfer coefficient can be seen to rapidly increase for smaller scale factors. For large values of \(h\), errors may be introduced to heat transfer measurements due to rapid crystal transition impacting upon measurement
accuracy. A solid line of $h = 500 \, W/(m^2 \, K)$ can be seen in Figure 5.13, this represents a value of $h$ above which transition speed can become problematic.

In order to determine the most appropriate scale for the model, five constraint factors were used. A maximum preferred tile area was set as $1m^2$. In order to account for additional losses within the rig a maximum allowable pressure drop of 8% across both skins was specified. A maximum mass flow rate was set as $1.5kg.s^{-1}$ to give a buffer from the fan maximum, and due to a reduction in available pressure drop for higher flow rates. A minimum available temperature rise of 35°C was specified based upon an assumption of 100% efficiency. A large value was desired to accommodate for potential losses between the supply and the heater due to the high current of the DC supply. The final constraint was a maximum desired $h$ value of 500 as illustrated in Figure 5.13. The variation of the specified constraint conditions have been non-dimensionalised and plotted against geometric scale factor in the constraint diagram, Figure 5.14. Values below 1 are within the specified constraints and it can be seen that no point achieves this for all parameters. A geometric scale factor of 7.5 was selected as the closest fit to match the compromise; the larger than desirable peak $h$ of ~550 and tile area of $1.09m^2$ were deemed as necessary compromises for the rig to be functional at all.

5.3.1 Normal Effusion Approximation

In the Tile-Rig, the film cooling effusion holes on the hot skin of the liner are modelled as normal to the surface, rather than using the design angle of 19° to the horizontal. As the full-Tile-Rig is only concerned with evaluating cold side heat transfer performance, exact modelling of the film cooling is not vital. It has been considered that altering the angle of the exhaust holes may impact upon flow migration within the cooling channel; however the change is considered necessary due to the negative impact of effusion holes on the ability to measure heat transfer performance.

The path of angled effusion holes through the plate will increase the surface temperature in the region above, due to heat transfer from the fluid within the hole, conducting through to the surface. This will corrupt the local heat transfer calculations, resulting in an over-calculation in this region. Additionally, angled holes present a larger footprint on the surface of the tile, with holes angled 19° to the normal, having a perceived open area ~3x larger than that of normal holes. The presence of the mesh heater requires that the cooling tile be viewed through the plate, meaning angled holes would obstruct the view of
liquid crystal above the entire hole length; an area approximately 10x that of a normally aligned hole.

5.3.2 Tile Thickness

Based upon the provided tile data, the hold and cold skins have thicknesses of 10.5\textit{mm} at the model geometric scale factor of 7.5. The findings of Lichtarowicz \textit{et al.} [35] show that \textit{L/D} has very little effect on \textit{C_D} for values higher than 1.5; this gives confidence that a small change from the exact thickness does not impact upon jet behaviour. In order to facilitate design, the model thicknesses have been changed to values of 12\textit{mm} for the cold skin and 25\textit{mm} for the hot skin. The large pressure drop across the impingement jet array, combined with the large open area of the cooling tiles results in significant downward force acting upon the cold skin. In order to increase its stiffness and reduce potential plate deflection, a thicker plate has been selected. An increase of 1.5\textit{mm} to a thickness of 12\textit{mm} increases the moment of inertia of the plate by \textasciitilde50\% reducing plate deflection by \textasciitilde33\%, whilst the connection of the hot and cold skins via the studs and ports was also considered to increase the overall stiffness to effectively damp deflection to insignificant levels. Modelling the effusion holes as normal, rather than angled holes effectively reduces the \textit{L/D} by a factor of 3 if the plate thickness is kept constant. Effusion holes angled at 19\textdegree{} to the surface would have a length of 32\textit{mm} at model scale. To compensate for the reduction in \textit{L/D} from the change of orientation, a plate of 25\textit{mm} is used for the hot skin.

5.4 Instrumentation

During an experiment, temperature and differential pressure data are recorded throughout the rig, to allow calculation of the surface heat transfer coefficient and for characterisation of the flow behaviour, including the impingement and effusion Reynolds numbers. Temperature measurements are taken using thermocouples as they provide a sufficiently fast response to capture the rapid change in fluid temperature resulting from mesh heater activation. Pressure measurements are taken across the rig using differential pressure transducers recording the pressure from various static pressure tappings positioned throughout the Tile-Rig. Sensors are connected to the lab PC using a selection of National Instruments hardware, detailed in Chapter 4 and measurements are recorded using Labview. The location of sensors throughout the test facility is detailed in this section.
5.4.1 Temperature Measurement

The large size of the Tile-Rig potentially creates some problems with spatial uniformity in the gas temperature; a local measurement of the flow temperature has therefore been made for each of the individual Measurement Regions described later in this chapter. Fine-wire exposed junction thermocouples have been used for this purpose (type K, wire diameter 0.05 mm). The gas thermocouples have been suspended across the inlet to impingement holes in the five regions shown in Figure 5.16. In this way, the spatially resolved heat transfer coefficients are based on a locally measured gas temperature history; with the thermocouple junctions positioned in appropriate locations to establish the inlet thermal boundary conditions. A typical distribution of gas-temperatures in the different regions is shown in Figure 5.16. An additional gas-thermocouple is positioned at the throat of the metering nozzle to allow accurate calculation of the fluid density passing through it. This is in order to ensure the mass flow rate through the metering nozzle is calculated accurately.

In addition to the gas-temperature measurements, surface temperature measurements are required for calculation of the surface heat transfer. Full area surface temperature is captured using liquid crystals as has been outlined in Chapter 4 using a precisely calibrated liquid crystal to provide indication of the time at which the surface reaches particular temperatures. For the method to be accurate, an in-situ calibration of the liquid crystal is required, as precise liquid crystal calibration depends on the crystal mixture and application. Due to this dependency on the application method, calibrations are required for the five different Measurement Regions shown in Figure 5.15 to provide a calibration for each. Additional temperature measurement instrumentation is required for the in-situ calibration to be performed, with thin foil thermocouples used for this purpose (type T, foil thickness 0.013 mm).

The liquid crystal calibration is performed as outlined within Chapter 4 using a slow progression of the heater to capture discrete surface temperature measurements and produce a calibration plot as shown in Figure 4.2. Calibration of the crystals on the Tile-Rig produced the precise green signal responses displayed in Table 5.6. Calibration of the crystals indicates a small influence of the different layers on precise calibration, with a peak variance between the different measurement regions of 0.3° C, 0.5° C and 0.1° C for the three crystals respectively.
Table 5.6: Precise thermocouple calibrations for the five different measurement regions shown in Figure 5.23.

### 5.4.2 Pressure Measurement

Differential pressure transducers are used to measure the stage pressure loss at various locations throughout the rig through measurement of the static pressure difference between various tappings, with locations shown in Figure 5.17. The arrangement of pressure tappings allows the pressure drop to be determined across the different stages. Ten pressure transducers are available for recording pressure in the full Tile-Rig: Three with a range of ±700mm H2O; six with a range of ±250mm H2O; and three with a range of ±50mm H2O. The numerical reference in Figure 5.17 refers to the different stages across which pressure drop may be measured and Table 5.7 indicates the data which may be obtained by using particular tappings and the range capability required by the transducers to measure the stage pressure difference.

The pressure at location 1 is measured with respect to atmosphere in order to evaluate the pressure drop across the mesh heater. It is also necessary in order that the fluid density may be correctly calculated for use in calculating jet Reynolds number. Similarly, the pressure at location 5 is measured with respect to atmosphere in order that the correct density at the nozzle is used for calculating rig mass flow rate, and with respect to tapping 6 in order to determine the flow velocity through the nozzle. These 3 pressure measurements are recorded during every test.

<table>
<thead>
<tr>
<th>Measurement Region</th>
<th>Crystal 1</th>
<th>Crystal 2</th>
<th>Crystal 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stud 1</td>
<td>32.10</td>
<td>36.76</td>
<td>39.91</td>
</tr>
<tr>
<td>Stud 2 / Array 2</td>
<td>32.27</td>
<td>36.55</td>
<td>40.01</td>
</tr>
<tr>
<td>Stud 3</td>
<td>32.11</td>
<td>36.80</td>
<td>39.98</td>
</tr>
<tr>
<td>Array 1</td>
<td>31.95</td>
<td>37.08</td>
<td>40.02</td>
</tr>
<tr>
<td>High pressure</td>
<td>Low pressure</td>
<td>Measurement</td>
<td>Required Δp range</td>
</tr>
<tr>
<td>---------------</td>
<td>--------------</td>
<td>------------------------------------------------------------------------------</td>
<td>-------------------</td>
</tr>
<tr>
<td>$P_0$</td>
<td>1</td>
<td>Pressure drop across the mesh heater.</td>
<td>±50mm</td>
</tr>
<tr>
<td>1</td>
<td>2</td>
<td>Pressure drop across the impingement tile.</td>
<td>±700mm</td>
</tr>
<tr>
<td>2</td>
<td>3, 4</td>
<td>Pressure drop across the effusion tile.</td>
<td>±250mm</td>
</tr>
<tr>
<td>3</td>
<td>4</td>
<td>Pressure drop across the restrictor plate</td>
<td>±250mm</td>
</tr>
<tr>
<td>$P_0$</td>
<td>5</td>
<td>Total pressure loss from inlet to Perspex plenum exit.</td>
<td>±700mm</td>
</tr>
<tr>
<td>5</td>
<td>6</td>
<td>Metering nozzle velocity for total rig mass flow rate.</td>
<td>±250mm</td>
</tr>
</tbody>
</table>

Table 5.7: Typical pressure measurements taken within the Tile-Rig and required transducer ranges.

Due to the use of plenum fed air to the impingement array, a single pressure drop measurement across the impingement or effusion plate is deemed representative of the stage, and may be used to calculate average jet velocity and Reynolds number. This approach may not always be used, as in certain tests restrictor plates are used to simulate pressure blockages across the hot skin as spanwise Pressure Zones (refer to Section 5.7). Several pressure tappings have been taken up and downstream of the impingement and effusion plates (locations 1-3) to allow local pressure measurements to be made. When flow restriction is presented, the pressure drop may vary in different streamwise locations and it is necessary to take pressure readings using tappings in all three Pressure Zones of the effusion plate. This allows the effect of the restriction on the effusion jet Reynolds in the different Pressure Zones to be determined. As flow restriction reduces local flow through the effusion plate, a pressure head may result that also affects impingement pressure drop distribution. As such it is necessary to use the local pressure tapping in the measurement region under investigation. Only a single tapping is required in the exit plenum (region 4) as the large volume results in negligible spatial variation of pressure.

### 5.5 Metering Plenum

The full tile facility features a metering plenum of length 3$m$ and height/width 1.2$m$, connecting the working section exit plenum to the fan. The walls are constructed from 1” thick plywood to ensure adequate resistance to the large pressure forces (~21$kN$ at maximum pressure drop. The plenum arrangement, shown in Figure 5.1, features two mesh screens.
either side of a convergent-divergent orifice of throat diameter \(160\text{mm}\) and inlet radius \(40\text{mm}\), used to measure the mass flow rate through the tile model. The layout of the plenum is arranged to minimise pressure losses through the nozzle, with the screens positioned in order to condition the flow prior to it entering the nozzle and the fan; and the diffuser section of the metering nozzle designed to provide pressure recovery. The metering nozzle may be used to calculate the rig mass flow rate using the known throat diameter with calculations of the flow density and velocity obtained by taking pressure and temperature measurements within the plenum and nozzle. Use of the radiused inlet \((R/D=0.25)\) has been shown to increase discharge coefficient [34] and mass flow rate is calculated for an assumed nozzle discharge coefficient of 1.

The metering nozzle design, shown in Figure 5.18, targets a restriction of its pressure losses. The \(160\text{mm}\) throat diameter is required due to the high mass flow rate passing through the test rig; such that the velocity and associated pressure drop is contained. The dimensions of the diffuser section of the nozzle were selected with reference to the work of McDonald [102] with the recovery contours shown in Figure 5.19 used to optimise the pressure recovery through judicious selection of diffuser length and angle. The diffuser length and expansion angle of \(320\text{mm}\) and \(2\phi=8^\circ\) correspond to a pressure recovery coefficient greater than 0.5. It can be seen from Figure 5.19 that pressure recovery increases with diffuser length-to-throat radius \((N/R_T)\); however, the large throat diameter placed a restriction on this value. The proposed design targeted a pressure loss from the metering nozzle of less than \(1\text{kPa}\) with the rig operating at the engine maximum Reynolds condition. Air density within the plenum will be sub-ambient due to pressure losses across the cooling tiles (~7%) and the elevated fluid temperature (~40°C in the plenum), with an estimated value of ~1.05 corresponding to these conditions. The \(160\text{mm}\) diameter was selected to give a maximum orifice velocity of \(~62\text{ms}^{-1}\) at \(m = 1.3\text{kg.s}^{-1}\), a pressure drop of \(~2\text{kPa}\). The recovery coefficient provided by the diffuser design brings the overall pressure loss to below the target limit.

5.6 Mesh Heater Commissioning

The mesh heater used in the Tile-Rig spans the entire inlet channel, and area of \(1.09\text{m}^2\). The heater is powered by a \(50\text{kW}\) (1000A, 50V) DC power supply unit (PSU), provided by DP Energy Services, with 4 pairs of 1” thick copper cables used to transport the current between the mesh heater and the PSU. The mesh is constructed of 40 micron stainless steel
wires, plain woven at 250 threads per inch. The current is supplied to the mesh heater through copper bus bars spanning the length of the heater. This arrangement is used to match the mesh resistance to the voltage-current ratio (0.05) of the PSU, giving the mesh heater a width of 922 mm and a length of 1170 mm, the reverse of the rig dimensions. Based upon these dimensions resistance of the mesh heater was calculated using equation (4.2) to a value of 0.045Ω. As a result the maximum power supply to the heater will be limited by the maximum current output of the PSU, reducing the available power to 45 kW. This is sufficient to satisfy the demands of the testing, corresponding to a maximum temperature increase of ΔT=35°C at a mass flow rate corresponding to the engine upper limit conditions.

The size of the mesh heater used in the Tile-Rig is significantly larger than any reported example of a similar such device. A review of mesh heaters used in transient experiments found no heaters larger than 0.135 m², 8x smaller than the Tile-Rig mesh heater. The large open area, led to concerns of non-uniform heating across the heater that could affect the accuracy of heat transfer calculations. To account for this, multiple gas-thermocouples were positioned around the tile at impingement jet exit. Figures 5.15 and 5.16 display the position of the gas thermocouples and typical simultaneous temperature measurements taken by each. The unsteadiness of the temperature measurements in Figure 5.16 is typical of thermocouple responses from all tests. The time averaged temperatures read by the different thermocouples after the temperature step has been imposed typically vary by up to 2-3°C between the different thermocouples; however, no definite trend is apparent with spatial location. Individual thermocouples were observed to exhibit a degree of consistency in relation to the others, typically measuring higher or lower than the combined average, although the degree of offset was seen to vary by up to 1-2°C. In order to account for the observed spatial variation, thermocouples positioned close to the particular measurement region were used for calculation of heat transfer coefficient. In addition to assessing temperature variations across the span of the mesh heater, an evaluation was carried out into the near wall temperature increase. Temperature traverses were carried out in the corner region of the rig and compared against a reference thermocouple in an open region in the same quadrant. Figure 5.20 shows the traverse arrangement used.

Measurements taken from the traverses (Figure 5.21 and Figure 5.22) show that the fluid temperature increases significantly in close proximity to the front wall of the rig, with a reduction in temperature occurring within 40 mm of the sidewall. It would appear from the traverse data that proximity to the corner studs impacts upon the mesh behaviour in this
region. An increase of temperature in the region near the walls is thought to be the result of reduced velocity through the mesh in this region; with the smaller localised mass flow rate resulting in a larger temperature increase, as shown by equation (4.2). The reduction in temperature from 60\textit{mm} in from the corners would be consistent with this theory if flow is accelerated in this region due to the blockage presented by the corner stud. The deviation of temperature in the near wall region appears to reduce rapidly with distance from the walls as can be seen in the behaviour of traverses A and D. The temperature rise exhibited by the two thermocouple traverse and the reference converge back towards one another within 40-50\textit{mm} from the wall in these cases (to within the levels seen in Figure 5.16).

5.7 Pressure Blockage Simulation

Lean burn combustion has been observed to experience unsteady combustion due to the low fuel-air ratio. This results in periods of burner flow impingement on the combustor liner surface, which is considered to raise the local static pressure, reducing the pressure differential across the effusion skin. A consequence of this is a reduction in the local effusion flow velocity, weakening the local film-cooling performance; and in the presence of a large rise in static pressure may result in flow reversal leading to hot gas ingestion. The variation in hot skin exhaust pressure distribution causes a redistribution of the flow and may lead to the presence of crossflow effects within the tile cavity that affect the impingement performance.

The full Tile-Rig has been designed to allow a degree of simulation of the combustion chamber pressure fluctuations, modelled as circumferential (spanwise) bands of blockage. Through testing it is sought to determine the effect of such variations on cold-side impingement cooling performance.

5.7.1 Pressure Zones

To simulate the spanwise bands of pressure blockage, the exit plenum downstream of the modelled hot skin has the capacity to be divided into three “zones”, shown in Figure 5.23 using porous baffle plates. The numbers of impingement and effusion holes contained within each of the Exit Static Pressure Zones is summarised in Table 5.8. The open area of these plates establishes a particular change in fractional pressure drop in that exit zone. Open area may be varied by blocking additional holes, and using plates in combination.
Table 5.8: Impingement and effusion hole numbers (and open area) in the three effusion exit zones.

In order to model the effect of locally elevated static pressure in the burner region the focus of the testing has been on manipulating the exit static pressure at the burner end and mid-axial sections of the tile (referred to as Zones 1 and 2 respectively).

The flow restrictor plates for Zones 1 and 2 each contain regular arrangements of 20mm diameter holes. The Zone 1 plate has 10 holes, presenting an open area equivalent to 33% that of the effusion array in that region. The Zone 2 plate has been configured to operate with either 8 or 12 holes, presenting an open area equivalent to 33% or 50% of the effusion array in that region. No blockage plate is used in Zone 3, at the rear end of the tile, as the distance from the burner region will reduce the local impact of pressure fluctuations from burner impingement. As Zones 1-3 all exit into the same chamber downstream of the baffle plates the total pressure drop in each zone will be roughly equal, excepting any large streamwise pressure gradient within the cooling channel; the presence of which should be minimised by the plenum fed arrangement of the rig. Based upon this assumption and assuming equal discharge coefficients for the effusion and blockage holes; the porosity of the baffle plates was designed to produce a reduction in pressure drop across the hot skin of ~90% in the case of an effusion area 3x the baffle area (Zone 1 and 8-hole configuration of Zone 2) when compared with the unblocked Zone(s). A reduction of ~80% will result from an effusion area twice the baffle area (12-hole configuration of Zone 2). Considering the system as a whole, the introduction of blockage will lead to an overall increase of fractional pressure drop between the cooling chamber and the exhaust plenum. For a defined overall tile pressure drop the introduction of blockage will reduce impingement Reynolds number, reducing cold-side cooling performance as a result.

5.7.2 Exit Static Pressure Cases

Using the Zone 1 and 2 blockage plates, four exit static pressure cases have been considered which are summarised in Table 5.9. Each of these cases is described below:
Exit Static Pressure Case B0: This case is of uniform exit static pressure, with an unobstructed exit plenum allowing uniform pressure drop occurring across the effusion-tile. This is considered the baseline case relative to which other arrangements are compared.

Exit Static Pressure Case B1: This case considers a reduction in overall tile pressure drop in the central (Zone 2) region of the tile. The open area of the Zone 2 blockage plate is equivalent to 33% of the respective effusion area. The local tile pressure drop in Zone 2 is reduced by 85% compared to Zones 1 and 3.

Exit Static Pressure Case B2: This case considers a reduction in overall tile pressure drop in both Zones 1 and 2. The open areas of the blockage plates in Zones 1 and 2 are equivalent to 33% and 50% of the respective effusion areas. The local tile pressure drop is reduced by 85% in Zone 1 and by 75% in Zone 2 when compared with the pressure drop across the effusion plate in the unblocked, Zone 3.

Exit Static Pressure Case B3: This case considers zero effusion flow in Zone 2, but equal tile pressure drops in Zones 1 and 3.

<table>
<thead>
<tr>
<th>Geometry Designation</th>
<th>Zone 1 (burner end)</th>
<th>Zone 2 (middle)</th>
<th>Zone 3 (exit end)</th>
</tr>
</thead>
<tbody>
<tr>
<td>B0</td>
<td>No blockage</td>
<td>No blockage</td>
<td>No blockage</td>
</tr>
<tr>
<td>B1</td>
<td>No blockage</td>
<td>0.00251 m²</td>
<td>No blockage</td>
</tr>
<tr>
<td>B2</td>
<td>0.00314 m²</td>
<td>0.00377 m²</td>
<td>No blockage</td>
</tr>
<tr>
<td>B3</td>
<td>No blockage</td>
<td>Fully blocked</td>
<td>No blockage</td>
</tr>
</tbody>
</table>

Table 5.9: Definition of Tile-Rig exhaust blockage designations including the open area of the blockage plates.

5.8 Heat Transfer Measurement Regions

The large area of the cooling geometry means that obtaining surface heat transfer maps for the entire surface is impractical. In order to represent the features of the full geometry, spatially resolved measurements of heat transfer coefficient have been obtained in five distinct Regions on the cold-side of the hot skin. These Regions have been selected as being representative of the various impingement regimes encountered across the tile architecture; with Regions in each Exit Static Pressure Zone and in close proximity to the ports and studs; as well as in open, unobstructed areas of the different density effusion arrays.

The five Measurement Regions have been designated with reference to the presence or absence of studs; with three stud regions and two array regions measured in detail. The locations of these Regions is shown schematically in Figure 5.23, in which each is shaded.
green and identified using the naming convention for the Measurement Regions. The following summarises the five Measurement Regions:

**Stud 1**: Burner End corner stud Region. This area is includes the stud and extends to the edge of the igniter port. The Stud 1 Region has an area of approximately $0.053m^2$ and features 67 effusion holes and 43 impingement jets, including 3 angled jets centred on the stud.

**Stud 2**: Central stud including a region of regular impingement jets between the two modelled ports. The Stud 2 Region has an area of approximately $0.057m^2$ and features 84 effusion holes and 51 impingement jets, including 3 angled jets centred on the stud.

**Stud 3**: Exit End corner stud Region. The Stud 3 Region has an area of approximately $0.048m^2$ and features 49 effusion holes and 32 impingement jets, including 3 angled jets centred on the stud.

**Array 1**: Burner End zone including areas close to the igniter port. Most of this area is a regular array of impingement jets. The Array 1 Region has an area of approximately $0.067m^2$ and features 75 effusion holes and 48 impingement jets.

**Array 2**: Exit End zone including areas close to the secondary port. Most of this area is a regular array of impingement jets. The Array 2 Region has an area of approximately $0.054m^2$ and features 70 effusion holes and 47 impingement jets.

The total area of the five measurement regions accounts for ~25% of the surface area of the cooling tile, including the two ports that have been modelled and essentially present dead space.

### 5.9 Closure

Design of the Full-Tile Test Facility has been considered in this chapter. This facility has been developed in order to investigate the cooling performance of a complete double skin combustor liner tile using an impingement-effusion cooling pattern, including the effects of structural blockages and variable exit static pressure conditions. A schematic is shown in Figure 5.1 with a photograph included in Figure 5.2.

The cooling geometry featured on this test facility has been designed by adapting a similar cooling tile from the ANTLE engine, but with porosity altered to match engine conditions supplied by Rolls-Royce PLC. In all other respects the liner tile that has been
modelled is representative of the EFE cooling tile illustrated in Figure 5.3, constructed to a geometrical scale factor of 7.5; this includes modelling of the fastening studs and spacers and the blockage presented by the ports present in the outer annulus. A means of simulating spatial pressure variations within the main combustion chamber that are a result of lean burn combustion has been designed, in which the variations are modelled as spanwise bands of pressure. In addition to a uniform exit static pressure condition, three other conditions have been modelled with various degrees of restriction to flow exiting through the first two zones of the target plate (as shown in Figure 5.23). The scale factor of 7.5 was selected as a compromise to various considerations. A large scale factor was desired to avoid high heat transfer coefficients that would impact upon experimental accuracy and improve resolution of results; however cost, manufacturing and accommodation requirements meant that a smaller scale factor was required. The value of 7.5 was selected as the largest that could be feasibly manufactured and housed.

Heat transfer measurements for this test facility are made using a transient liquid crystal thermography technique. Due to the large size of the target plate (1.09$m^2$) and the repeated cooling pattern measurements were not taken over the entire tile, but from five Measurement Regions. These regions, illustrated in Figure 5.15 the different blockage features. The experimental procedure and analysis technique for recording and reducing data captured using this test facility have been detailed in Chapter 4. The results and discussion are presented in Chapter 7, with a comparison of these results conducted against results from the Sector-Rig Test Facility in Chapter 9.
5.10 Figures

Figure 5.1: A schematic diagram of the Tile-Rig Test Facility

Figure 5.2: A photograph of the Tile-Rig working section
Figure 5.3: EFE initial outer igniter combustor hot skin casing design [6]

Figure 5.4: Schematic diagram of angled impingement jets used in the stud region shown in Figure 5.9.
Figure 5.5: ANTLE combustor liner cooling tiles. (a) Impingement jet array (b) Effusion jet array [6]

Figure 5.6: Impingement jet array for EFE combustor liner Tile-Rig.
Figure 5.7: Effusion jet array for EFE combustor liner Tile-Rig showing inlet and exit locations of jet holes.

Figure 5.8: Section 1A (Figure 5.6), impingement jet array pitches.

Figure 5.9: Section 1B (Figure 5.6), angled impingement jet arrangement surrounding studs.
Figure 5.10: Section 1C (Figure 5.6), additional impingement jet arrangement surrounding igniter port.

Figure 5.11: Section 2A (Figure 5.7), effusion jet array pitches.

Figure 5.12: Section 2B (Figure 5.7), additional effusion jet arrangement surrounding igniter port.
Figure 5.13: Variation of stagnation heat transfer coefficient with scale factor.

Figure 5.14: Constraint diagram for illustrating factors affecting determination of geometric scale factor
Figure 5.15: Spatial positioning of gas and surface thermocouples and illustration of liquid crystal Measurement Regions.

Figure 5.16: Gas temperature response at different spatial positions 1-5.
Figure 5.17: Location of static pressure tappings in the full file test facility.

Figure 5.18: Schematic of metering nozzle for measurement of rig mass flow rate
Figure 5.19: Pressure recovery contours for varying geometry conical diffusers [102]

Figure 5.20: Thermocouple traverse arrangement for mesh heater corner performance evaluation
Figure 5.21: Temperature variation with distance from side wall (mm)

Figure 5.22: Temperature variation with distance from front wall (mm)
Figure 5.23: A schematic diagram that illustrates the locations of heat transfer measurements (shown in green, including the naming convention) and axially arranged Exit Static Pressure Zones.
Chapter 6  Sector-Rig Test Facility

As has been detailed previously in Chapter 1, the typical method of cooling double-skin combustor liners features combinations of pedestals and slot injected film cooling, as illustrated in Figure 1.5 for cooling of, respectively, the cold and hot side of the hot combustor liner skin. Interest in assessing heat transfer performance in zero-effusion geometries has been inspired by experimental results indicating an increase in combustion efficiency and a reduction in NOx output in response to a reduction in the number of effusion holes [6]. Removing effusion holes from the cooling geometry represents a significant challenge to providing the necessary cooling performance. An absence of film-cooling on the hot-side of the inner-skin means that improved heat transfer performance is required on the cold-side to compensate. Additionally, the absence of effusion holes from the hot skin requires flow to exit from the end of the liner channel, with propagation of the spent jets towards the exit resulting in a significant crossflow between the two skins. As discussed in Chapter 2, the performance of impingement jet arrays has been widely reported to be hindered when issuing into channels with crossflow.

The Sector-Rig test facility has been developed to enable a study of impingement-based cooling systems in the absence of effusion holes in order to quantify both aerodynamic and cold-side heat transfer performance parameters. The test facility is used for investigation into cooling geometries that yield enhanced impingement performance, particularly in the presence of a strong crossflow within the cavity between the two skins. In order to utilise the unavoidable crossflow, cooling geometries will feature surface mounted roughness features to increase the surface area over which heat transfer can occur, and to increase $h$ through promoting increased turbulence. Investigations are focused on combinations of impingement jet and pedestal features combined in hybrid geometries.

Coolant feed to pedestal arrays in double-skin arrangements typically occurs from a single row of large impingement holes positioned upstream of the array; for this thesis it has been proposed that heat transfer performance could be improved by adopting a distributed feed arrangement through an array of small impingement holes. A parametric investigation will be performed of various pedestal arrangements positioned beneath a consistent array of impingement jets with parameters representative of those defined for use in the full-tile test facility. The Sector-Rig Test Facility has been developed for use in conducting measurements of the aerodynamic and heat transfer performance of various arrangements of
pedestals beneath an array of impingement holes representative of the dimensions tested on the Tile-Rig Test Facility. The primary objective of the investigation is to evaluate the potential for zero effusion cold side cooling and to propose an optimised design that could be compared against traditional cooling methods, considering heat transfer performance and the level of pressure losses. The ultimate objective of the parametric study is to evaluate whether the tested geometries could potentially be applied to a cooling tile similar to that tested on the Full Tile Test Facility and whether the results indicate that double skin combustor liners employing zero effusion cooling methods may be pursued as a viable option in lean burn gas-turbine engines.

6.1 Overview

The Sector-Rig is used to investigate the impact of pedestal height and density upon cooling performance. The cooling geometries investigate using the Sector-Rig are based upon dimensions from the EFE combustor liner, and the ANTLE inspired impingement array used within the Tile-Rig. A geometrical scale factor of 17 is used compared to engine scale geometry. The investigated cooling patterns use the same impingement jet array with eleven rows of impingement holes, each row containing five impingement holes of diameter 12 mm. This hole size is equivalent to an engine scale diameter of 0.7 mm, as used in the Tile-Rig. The space between the jets within the array (x/D=7.1, y/D=6.15) is also comparable to those used in the Tile-Rig, shown in Figure 5.8. The basic arrangement under investigation incorporates pedestals of various heights, arranged into equilateral triangular arrays beneath the impingement-jet array. Various pin-fin geometries have been considered experimentally and are described later.

A schematic diagram of the Sector-Rig Test Facility is shown in Figure 6.1 and a photograph of the working section is shown in Figure 6.2. Air is drawn from outside the test cell and blown through the sector model by a centrifugal fan. Prior to entering the working section of the tile, the flow is passed via a settling chamber to a metering orifice (throat diameter 74 mm) which is used to measure the rig mass flow rate. After passing through the orifice the flow is ducted to a contoured inlet passing through various flow conditioning screens in order to offer a uniform flow profile at entry to the working section. The flow passes through a mesh heater that is capable of heating the airflow by tens of degrees before entering the working section. The mesh heater (detailed further in Section 6.3) is important to the application of the transient liquid crystal method for measuring h. Downstream of the
mesh heater the flow is accelerated by a contraction into the feed-channel of the tile model, as shown in Figure 6.2. From this point the flow passes through the impingement holes and into the pin-fin channel. Heat transfer performance is measured using liquid crystal thermography. The working section and connecting duct of the rig are constructed from Perspex, as seen in Figure 6.2, so as to allow optical access of the crystal colourplay using a video camera.

Test conditions within the Sector-Rig are monitored using various pressure and temperature instrumentation that are similar to that used on the Tile-Rig. The pressure instrumentation consists of static pressure tappings that are connected to differential capacitance manometers. Temperature instrumentation consists of gas (miniature-bead) and wall temperature (surface mounted foil) thermocouples. The gas thermocouples are constructed from 0.05 mm K-type bare wire devices manufactured by Omega Engineering Ltd. The design of the gas temperature thermocouple installations is such that they can resolve the temporal variations in temperature observed during transient heat transfer measurements. The foil thermocouples are used in the calibration of the liquid crystal coating used in the spatially-resolved measurement of heat transfer. Data acquisition from these sensors is accomplished using National Instruments SCXI system signal conditioning and digitisation equipment.

6.2 Cooling Geometry Development

The Sector-Rig features cooling geometries based upon an EFE combustor liner tile. The same tile dimensions and aerodynamic conditions are used as those on which the Tile-Rig cooling geometry design was based. A selection of different pin arrays has been developed to featuring different combinations of pin-to-pin ($P/D$) spacing and height-to-diameter ($H/D$) ratios, in order to assess the effect of these parameters on cooling performance and aerodynamic behaviour. Additionally, an original design concept, dubbed ‘Shielded Impingement’ has been developed. A discussion of the design strategy for the different geometries follows.

6.2.1 Experiment Scaling

As with the Tile-Rig, this facility operates at near atmospheric conditions. In order for meaningful data to be obtained, the rig has been scaled based upon $Re$ and $Pr$ derived from the data received from Rolls-Royce regarding engine operating conditions. A geometric
scale factor of 17 has been determined for the Sector-Rig, with Table 6.1 displaying a summary of the range of engine conditions to be modelled and the corresponding Rig operating conditions. \( Re \) is based on the jet velocity \( (U = \sqrt{2 \Delta p / \rho}) \) and, for the engine conditions, an assumed impingement hole diameter of 0.7\textit{mm}. The values of \( Re_{jet} \) listed in Table 6.1 are based on assumptions of a pressure drop split between the impingement plate and the cooling channel equal to the impingement/effusion pressure split of the Tile-Rig. Changes to the pedestal array for the different cooling geometries will alter the pressure drop occurring across the entire tile due to additional blockage presented by the pedestals. As a result, the precise \( Re_{jet} \) for the design pressure drop will vary for the different cases; however, the scaled total pressure drop will remain constant at 1.65\% for the upper limit and 0.08\% for the lower limit.

<table>
<thead>
<tr>
<th>Engine upper limit</th>
<th>Engine lower limit</th>
<th>Sector-Rig</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P_o )</td>
<td>48</td>
<td>5</td>
<td>1</td>
</tr>
<tr>
<td>( T_o )</td>
<td>900</td>
<td>500</td>
<td>323</td>
</tr>
<tr>
<td>( \rho )</td>
<td>15.5</td>
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<td>( C_p )</td>
<td>1120</td>
<td>1025</td>
<td>1005</td>
</tr>
<tr>
<td>( \Delta p/P_o )</td>
<td>2.8</td>
<td>2.4</td>
<td>1.65</td>
</tr>
<tr>
<td>( U )</td>
<td>118</td>
<td>74</td>
<td>50</td>
</tr>
<tr>
<td>( Re_{jet} )</td>
<td>33,000</td>
<td>7,000</td>
<td>33,000</td>
</tr>
<tr>
<td>( Pr )</td>
<td>0.70</td>
<td>0.71</td>
<td>0.71</td>
</tr>
<tr>
<td>Geometric scale factor</td>
<td>1.00</td>
<td>17</td>
<td></td>
</tr>
</tbody>
</table>

Table 6.1: Engine and Sector-Rig operating conditions.

The scale factor of 17 used for the Sector-Rig is significantly larger than that used in the Tile-Rig facility. The reason for this is that, as has been discussed in Chapter 5, the scale factor of 7.5 used for the Tile-Rig facility is an artefact of the large size of the test plate. The scale factor used represents a necessary compromise between achieving a model that offers appropriate experimental accuracy whilst also not being excessively large. The Sector-Rig, being a smaller region of the full tile, can be modelled at larger scale and still retain a manageable working section for laboratory testing. Operating at a larger scale factor is preferable as it allows improved spatial resolution in the heat transfer measurements and avoids high \( h \) values that can add to experimental uncertainty due to excessively rapid crystal transition. The Sector-Rig cooling geometries were designed for use on a pre-existing rig
featuring a fan, plenum and mesh heater arrangement. The geometric scale factor of 17 was chosen with reference to the channel dimensions of 220x520mm, to which the test pieces for this setup must be attached and the maximum outputs of the fan and heater. The design of the working section is discussed later in Section 6.3.

6.2.2 Parametric Study

A matrix of different test geometries was developed around the use of three different pin-to-pin distances; \( P/D = 5, 2.5 \) and 1.67 and three different pin height case; \( H/D = 2.5, 1.25 \) and 0.63; equivalent to full, half and quarter channel height. The \( x/D \) and \( y/D \) values have been arranged to enable regular alignment of the triangular and rectangular arrays such that separation between the jets and pedestals remains constant in streamwise and spanwise directions.

**Geometry S1:** Features pins arranged into ten rows of alternately five and six pedestals, arranged in an equilateral triangular array with \( P/D=5 \). This is equivalent to the same streamwise and spanwise separation as the impingement array. This arrangement is shown in Figure 6.3.

**Geometry S2:** Features pins arranged into twenty-one rows of alternately ten and eleven pedestals, arranged in an equilateral triangular array with \( P/D=2.5 \). This is equivalent to a streamwise and spanwise separation half that of the impingement array. This arrangement is shown in Figure 6.4.

**Geometry S3:** Features pins arranged into thirty-one rows of alternately fourteen and fifteen pedestals, arranged in an equilateral triangular array with \( P/D=1.67 \). This is equivalent to a streamwise and spanwise separation one-third that of the impingement array. This arrangement is shown in Figure 6.5. Due to the high density of pedestals in this arrangement, flow from the jet holes impinges partially onto individual pins. In order to allow the jet to enter the channel unobstructed, in the full height case S3a the pedestals which overlap with the pins remain at quarter channel height \( (H/D=0.63) \).

**Geometry S4:** Features arrays of combinations of pins with \( H/D=0.63 \) and \( H/D=2.5 \) arranged to shield the impingement jets from the developing crossflow. These geometries are discussed in further detail in the following section.
The matrix of testing and the reference codes by which different geometries are referenced is included in Table 6.2, with all geometries featuring eleven streamwise rows of five impingement jets.

<table>
<thead>
<tr>
<th>Style</th>
<th>Test geometry code</th>
<th>Pin layout</th>
<th>Pin heights</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impingement into pedestals</td>
<td>S1a</td>
<td>P/D = 5.0</td>
<td>H/D = 2.5</td>
</tr>
<tr>
<td></td>
<td>S1b</td>
<td>P/D = 2.5</td>
<td>H/D = 1.25</td>
</tr>
<tr>
<td></td>
<td>S1c</td>
<td></td>
<td>H/D = 0.63</td>
</tr>
<tr>
<td></td>
<td>S2a</td>
<td>P/D = 2.5</td>
<td>H/D = 2.5</td>
</tr>
<tr>
<td></td>
<td>S2b</td>
<td>P/D = 2.5</td>
<td>H/D = 1.25</td>
</tr>
<tr>
<td></td>
<td>S2c</td>
<td></td>
<td>H/D = 0.63</td>
</tr>
<tr>
<td></td>
<td>S3a</td>
<td>P/D = 1.67</td>
<td>H/D = 2.5</td>
</tr>
<tr>
<td></td>
<td>S3b</td>
<td></td>
<td>H/D = 1.25</td>
</tr>
<tr>
<td></td>
<td>S3c</td>
<td></td>
<td>H/D = 0.63</td>
</tr>
<tr>
<td>Shielded Impingement</td>
<td>S4a</td>
<td>P/D = 5.0</td>
<td>H/D = 2.5 &amp;</td>
</tr>
<tr>
<td></td>
<td>S4b</td>
<td>P/D = 2.5</td>
<td>H/D = 0.63</td>
</tr>
<tr>
<td></td>
<td>S4c</td>
<td>P/D = 1.67</td>
<td></td>
</tr>
</tbody>
</table>

Table 6.2: Matrix of geometries proposed for testing in the Sector-Rig facility including test reference codes.

### 6.2.3 Shielded Impingement

In addition to the parametric investigation into the impact of P/D and H/D on heat transfer performance, an original geometry has been proposed combining full and quarter height pedestals (H/D=2.5 and 0.63 respectively). It has been theorised that, by positioning pedestals spanning the full height of the channel upstream of impingement holes, the issuing jets may be shielded from the developing crossflow, improving heat transfer performance in the downstream region of the channels by significantly reducing the jet deflection typically associated with large crossflow.

In addition to the full height ‘shielding’ pins, the proposed Shielded Impingement array design features pins of H/D=0.63 as illustrated in Figure 6.6. By offering a reduced restriction to the crossflow, it was predicted that the developing crossflow will be diverted.
into channels, between the full height shielding pins and the impingement jets. In these channels the short pedestals will act as a traditional short pin array, interacting with the developing crossflow to increase the local $h$. It was predicted that the relatively small blockage presented by the short pedestals of $H/D=0.63$ would result in an increase in pressure drop that would be small compared with that from the impingement jet array and the shielding pins. During the design stage a similar arrangement was considered in which only the shielding pins would be used, with the quarter channel height pins omitted. The chosen geometry was selected as it was felt that any reduction in pressure losses would not be sufficient to offset the improved heat transfer performance resulting from the additional pedestals. This theory was supported by the findings of Hong et al. [60] in whose study it was observed that for impingement arrays over sparse pedestal geometries, the wake produced by the pin fins washed away the wall jet, reducing $Nu$ in the inter-jet region.

Three different Shielded Impingement pedestal arrays, included in Table 6.2 have been proposed, for each of the pin layout cases, $P/D=5$, 2.5 and 1.67. The S4b case is shown in Figure 6.6, with the dark circles representing the pedestals with $H/D=2.5$ and the white circles representing the pedestals with $H/D=0.63$. The S4a and S4c cases follow the same pattern, with only those pins positioned directly upstream of an impingement hole having $H/D=2.5$. For the S4a (sparse array) and S4c (dense array) cases, this results in only every second jet being shielded from the crossflow, as the alignment of the jets and pedestals is such that a full height pedestal would block the impingement hole exit for every other row for case S4C (as may be seen in Figure 6.5) and no pedestal is present for case S4a (as may be seen in Figure 6.3).

### 6.3 Rig Design

The Sector-Rig test facility required design of a working section that can be attached to a pre-existing test facility for testing the cooling geometries listed in Table 6.2. The existing setup featured a fan capable of delivering up to 1 kg/s at a gauge pressure of 7 kPa and a mesh heater powered by a 12kW (60V, 200A) DC power supply. The mesh heater used in the test rig spans a channel measuring 220x520mm to which the newly designed cooling geometries will be attached. As these dimensions would correspond to a resistance of 0.14Ω, which is significantly below the design value of 0.3Ω based upon the Voltage and Current characteristics of the PSU the mesh heater was manufactured from two meshes connected in series. This doubled the effective length of the heater, with the corresponding resistance
calculated using equation (4.2) to be a value of 0.28Ω. This meant that the maximum power supply will be limited by the maximum current output of the PSU, reducing the available power to $11kW$. This is sufficient to satisfy the demands of the testing, corresponding to a maximum temperature increase of $\Delta T = 40°C$ at a mass flow rate corresponding to the engine upper limit conditions.

Measurements of surface heat transfer performance are captured using the same transient liquid crystal thermography method used in the Tile-Rig. Additionally, $h$ for the pedestals may also be measured using thermocouples positioned within an aluminium pin that may replace any of the standard pedestals within the array. In order for surface heat transfer measurements to be accurate it is necessary that heat losses to the pins be accounted for. To reduce heat losses to the pins within the cooling channel, as well as to the side walls of the rig; a closed-cell rigid expanded plastic material, Rohacell, supplied by eMKay Plastics ltd. is used for various components. Rohacell is a low density structure ($\rho = 51.3 \text{ kg/m}^3$) with low thermal conductivity ($k = 0.029 \text{ W/mK}$) and specific heat capacity of $c = 1400 \text{ J/kg.K}$. As the thermal conductivity of Rohacell is roughly equal to that of air at rig operating conditions ($k_{\text{air}} = 0.028 \text{ W/m.K}$ at 50°C) it may be assumed that no heat is lost from the air to Rohacell components.

### 6.3.1 Feed Channel Design

All cooling geometries tested in the Sector-Rig facility use a common inlet channel configuration featuring a side-feed impingement array with eleven rows of five 12mm diameter impingement jets. The inlet channel has constant width, 520mm, but contracts from an initial height of 220mm at the mesh heater to a height of 50mm just upstream of the impingement array in order to maintain an adequate supply feed velocity, for the length of the inlet. A wider channel is required at the mesh heater to tailor its electrical resistance in order to obtain the optimum power from the DC supply. The slower flow in this region also reduces the pressure drop that occurs across an active mesh heater.

Contraction of the inlet channel starts 160mm downstream of the mesh heater, in order to ensure the change in flow pattern does not influence the flow through the mesh heater. Approximately the same distance is used downstream of the contraction before first row impingement also to ensure the flow contraction does not influence jet behaviour. In order to reduce heat losses from the air to the channel prior to the air reaching the impingement array, the inlet channel is lined with Rohacell prior to contraction of the channel.
Following the flow contraction, the feed velocity is accelerated by a factor of 5.6 and approaches the impingement array in a side feed arrangement. Previous studies into side feed impingement arrays have identified significantly reduced performance to the final row of jets that may not just be attributed to heat loss. It was considered that a factor in this reduction may be a reduction in final row jet velocity as a result of reduced velocity feed at the rear end of the channel. In order to reduce the downstream deceleration of feed channel velocity a restrictor plate was added to the end of the inlet channel to allow a degree of flow to bypass the cooling channel. A series of porous restrictor plates, each featuring twenty holes of different diameters (3, 6 and 9\,mm) were produced that would present different amounts of blockage at inlet channel exit. Discharge coefficient of these holes was determined experimentally to be ~0.75 and by measuring the pressure drop across the restrictor, the level of bypass flow may be determined.

6.3.2 Cooling Plate Design

The impingement plate features 55 impingement holes of 12\,mm diameter, arranged in eleven rows of five jets with jet-to-jet spacings of 85.2\,mm in the spanwise direction (y/D=6.15) and 73.8\,mm in the streamwise direction (x/D=7.1). The impingement y/D is equal to (x/D)cos30° in order to maintain consistent spacing between the impingement jets and the pedestals, as shown in Figure 6.3 to Figure 6.5. The impingement tile is manufactured from Perspex and to match the engine scale L/D for the impingement of 0.5, a plate of thickness 25\,mm is used (L/D_{model}=0.48). All pedestal geometries are modelled using two plates and a collection of removable Rohacell cylinders, measuring 66.7\,mm long, with diameter: 17\,mm. The plates are manufactured from 12\,mm thick Perspex, with length: 852\,mm and width: 520\,mm. Each plate features an array of holes through which the Rohacell cylinders may be push fit to model the pedestals. The two plates may be used to model the three different pin densities listed in Table 6.2. One plate features 220 holes and can model the S1, S2 and S4a-b geometries. The second plate features 449 holes for modelling the S3 and S4c geometries. For the S1 geometries to be modelled, half of the holes need to be blocked off, flush to the target surface. The Rohacell pins are manufactured to a length of 66.5\,mm to allow the full range of pin heights to be modelled. To ensure uniform pin height, a backing plate was used, against which the pedestals were pressed as illustrated in Figure 6.7.
The nominal width of the cooling channel, 520 mm, is oversized for the cooling geometries under investigation as additional space beyond the outermost jets presents a reduced pressure region around which flow can bypass the main cooling geometry. Rohacell spacers have been produced to tailor the channel width to the particular geometries, with the spacers fitting tightly to the pedestals outside the impingement array as shown in Figure 6.3 to Figure 6.5. This gives a channel width of 443 mm for the S1, S2 and S4a-b geometries and a channel width of 415 mm for the S3 and S4c geometries.

6.4 Equipment for Acquisition of Experimental Data

Experimental temperature and differential pressure data are recorded throughout the rig, to allow calculation of the surface $h$ and for characterisation of the flow behaviour, including the Reynolds numbers and loss coefficients of flow through the impingement jet array and friction factors within the pedestal cooling channel. Temperature measurements are taken using thermocouples as they provide a sufficiently fast response to capture the rapid change in fluid temperature resulting from mesh heater activation. Pressure measurements are taken across the rig using differential pressure transducers recording the pressure from static pressure tappings positioned along the inlet and pedestal channels of the Sector-Rig. Sensors are connected to the lab PC using a selection of National Instruments hardware, detailed in Chapter 4 and measurements are recorded using Labview. The location of sensors within the Sector-Rig facility is detailed in this section.

6.4.1 Temperature Measurement Instrumentation

The side inlet feed arrangement used in the Sector-Rig results in a gradual decline of fluid mean temperature with distance from the mesh as heat is lost to the walls of the rig. In order to evaluate local jet temperatures, measurements are made of the flow temperature issuing from each successive row of jets. Fine-wire exposed junction thermocouples have been used for this purpose (type K, wire diameter 0.05 mm). The gas thermocouples have been suspended across the exit of the second impingement hole of each jet row, such that the thermocouple junctions are positioned in appropriate locations to correctly measure the temperature issuing from the jets. In order to have a baseline temperature against which the measured jet temperatures may be evaluated, an additional gas thermocouple is positioned downstream of the mesh heater in order to establish the gas temperature at inlet. A typical distribution of gas-temperatures in the different location is shown in Figure 6.8. A fine-wire
thermocouple is also positioned in the metering plenum to allow accurate calculation of the fluid density passing through the nozzle. This is in order to ensure the mass flow rate through the rig is accurately calculated.

In addition to the gas-temperature measurements, surface temperature measurements are required for in-situ calibration of the liquid crystal. Thin foil thermocouples are used for this purpose (type T, foil thickness 0.013 mm). As precise liquid crystal calibration depends on the crystal mixture and application, surface thermocouples are required to be positioned on each pedestal tile for in-situ calibration. The wall transient from the surface foil-thermocouple is shown in Figure 6.9.

6.4.2 Static Pressure Measurement Instrumentation and Locations

Static pressure tappings located at strategic locations within the Sector-Rig are used to measure stage pressure losses; the difference in static pressure between two points was measured by differential pressure transducers. Six pressure transducers are available for recording pressure in the Sector-Rig facility, each with a range of ±250 mm H₂O. The locations at which tappings have been taken are shown in Figure 6.10 and have been positioned to enable measurement of the pressure drop across the impingement plate and along the length of the pedestal and feed channels. The three tappings taken within the pedestal and feed channels allow evaluation of the pressure loss from channel inlet to exit. A tapping taken at the midpoint of the pedestal channel provides improved resolution of the pressure losses. Taking measurements between tappings across the impingement tile, at the same streamwise location, allows local impingement velocity and therefore Re to be calculated for the first and final jet stages. Measurement of the pressure loss between feed channel inlet and cooling channel exit allows the total pressure loss associated with a particular geometry to be measured. This is a useful measure for comparison of different test cases and the degree of blockage presented. In addition to differential pressures within the rig, several pressure measurements are made with respect to atmosphere; at locations 1 and 5 to allow correct calculation of fluid density within the region, in order to define the flow correctly. Referencing the pressure at tapping 6 to atmosphere allows the bypass velocity, and accordingly, mass flow rate to be measured.
6.5 Heat Transfer Measurement Techniques

In the Sector-Rig Test Facility transient liquid thermography is used to take end-wall heat transfer measurements. As the cooling performance of a pedestal array is augmented by heat transfer to the pins as well as to the liner skin, discrete measurements are also taken evaluating the heat transfer performance of the pedestals.

6.5.1 Measurement of Surface Heat Transfer Coefficient

In order to evaluate heat transfer performance of the cooling array with streamwise location, measurements on the Sector-Rig are taken over the entire length of the channel. Surface heat transfer measurements are taken using a transient liquid crystal thermography approach common to both this and the full-tile test facilities. Due to the technique being common to both test rigs, full details of the method by which measurements are taken and data analysed has been included in Chapter 4. As the Sector-Rig utilises a side-feed arrangement, an additional process has been incorporated into the analysis method to ensure that the Nusselt numbers produced are representative of the local fluid temperature such that reported data may be scaled for comparison under different conditions and remain valid. The mixed-bulk analysis method, also outlined in Chapter 4, determines \( h \) based upon the local mixed bulk gas temperature. This is achieved using an energy balance principle for a spatially isothermal wall and a surface heat transfer distribution calculated based on a measured inlet temperature.

The nature of the cooling geometries being tested on the Sector-Rig facility means that complex variations will exist in the gas temperature due to the gradual injection of air to the cooling channel through eleven discrete rows of impingement jets. Heat losses in the feed channel much smaller than those in the cooling channel will result in jet temperatures greater than the surrounding mixed bulk temperature for all rows after the first. The complex flow patterns render the local temperature transients impractically difficult to resolve and as such it was felt that the adopted analysis method represented the most appropriate solution.

6.5.2 Measurement of Pedestal Heat Transfer Coefficient

The presence of pedestals in the Sector-Rig cooling geometries needs to be carefully considered as any heat transferred from the air to the pedestals during a transient test will alter the experimental results. The selection of Rohacell as the material for the pins, as has been detailed earlier in this chapter, allows heat transfer to the pins to be disregarded for
analysis purposes. In engine applications the heat transfer performance of the pedestals within the array is of large significance when considering the overall performance of the cooling geometry as the additional surface area of the pedestals offered improved cooling performance. In order to supplement the surface heat transfer data, additional capability has been built into the Sector-Rig test facility to measure pedestal heat transfer performance.

For measurement of $h$ of all tested pedestals, single examples of pins with $H/D=2.5; 1.25$ and $0.63$ have been manufactured from aluminium, each featuring a k-type thermocouple sealed at its centre, in order to record the variation of the pin temperature in response to the mesh heater activation. A schematic of the aluminium pedestal setup is illustrated in Figure 6.11. Thermocouples positioned on the central axis at three vertical positions (base, mid-height and tip) indicated uniform temperature distribution throughout the pedestal). The thermocouple readings may be used alongside the known material characteristics of the aluminium pedestal ($\rho = 2800\text{kg/m}^3$, $c_p,\text{pin} = 900\text{J/kgK}$) to calculate the transient heat flow rate to the pedestal using equation (6.1). This may be used to solve for $h$ using equation (6.2).

$$\dot{q} = m_{\text{pin}}c_{p,\text{pin}} \frac{dT_{\text{pin}}}{dt} \quad (6.1)$$

$$h = \frac{\dot{q}}{A_{\text{pin}}(T_g - T_{\text{pin}})} \quad (6.2)$$

Pedestal heat transfer measurement is performed within the same experimental setup used for measuring the surface heat transfer performance. As such, the temperature rise of the pedestal is triggered in response to a step change in fluid temperature imparted by the mesh heater. Consequently, a gradual reduction in the rate of change of temperature of the pedestal occurs as the temperature difference between the pedestal and fluid reduces. This leads to a gradual reduction in heat flux illustrated in Figure 6.12. From equation (6.2) it can be seen that the reduction in $\Delta T$ between the fluid and the pedestal opposes this, with $h$ inversely proportional to $\Delta T$ for a constant heat flux. In practice, $h$ is variable with $\Delta T$; however examination of the variation with time of the pedestal $h$ shows the value to be constant within a value of $\pm 3\%$ over the course of a test. For the purpose of calculating Nusselt numbers, the mean value of $h$ measured over the course of the test is used. The

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1 Heat loss through the base of the aluminium pedestal has been considered negligible compared with the heat transferred in. Over the course of a typical test (~40-60 second duration) heat loss through conduction to the Perspex base and free convection to the air in the cavity has been estimated to be ~1-2\% of heat flow into the pedestal.
design of the rig, incorporating removable pedestals allows the aluminium pins to be placed into the cooling channel in place of any of the Rohacell pedestals. This enables an evaluation of pedestal heat transfer coefficient in regards to position relative to impingement jets and also any streamwise changes to heat transfer performance.

6.6 Summary

In this chapter the development of an experimental facility for performing parametric testing of a range of zero effusion cooling geometries has been detailed. A matrix of twelve different cooling geometries has been produced; each featuring a fixed array of eleven rows of five impingement jets discharging into various pedestal roughened passages.

The test matrix incorporates a total of nine pedestal geometries resulting from combining two variable parameters; the height-to-diameter \((H/D)\) ratio of the different pedestals and the separation between the equilaterally spaced pedestals within the array (presented as \(P/D\)). In addition to these baseline geometries, additional geometries have been proposed incorporating Shielded Impingement arrangements. Shielded Impingement geometries feature an array of predominantly short pedestals, but with pedestals at full channel height positioned at select positions upstream of impingement jets, with the goal of shielding the jet stagnation region from the developing crossflow.

Details have been included of measurement instrumentation used for acquisition of heat transfer and aerodynamic data in this test facility. Pedestal heat transfer measurements are taken using k-type thermocouples encased within aluminium pedestals as detailed within this chapter. A transient liquid crystal thermography approach is used for measurement of surface heat transfer. Due to the side-feed arrangement of the tested geometries an energy balance analogy is used in order to base reported Nusselt numbers on the local mixed bulk temperature such that they may be scaled for comparison with previously reported data. As this measurement technique is common to both this and the full-tile test facilities, precise details of the methodology used for both have been described in Chapter 4.
6.7 Figures

Figure 6.1: A schematic diagram of the Sector-Rig Test Facility

Figure 6.2: A photograph of the Sector-Rig working section
Figure 6.3: Sector-Rig pedestal layout; P/D = 5 (geometries S1a, S1b, S1c).

Figure 6.4: Sector-Rig pedestal layout; P/D = 2.5 (geometries S2a, S2b, S2c).
Figure 6.5: Sector-Rig pedestal layout; P/D=1.67 (geometries S3a, S3b, S3c).

Figure 6.6: Sector-Rig Shielded Impingement layout (geometry S4b).
Figure 6.7: Variable pedestal height setup.

Figure 6.8: Typical gas temperature histories recorded in the impingement holes of the Sector-Rig during a transient heat transfer test.
Figure 6.9: Fluid temperature transients at selected streamwise locations, and corresponding surface thermocouple transient.

Figure 6.10: Location of static pressure tappings in the Sector-Rig facility.
Figure 6.11: Diagram of aluminium pedestal used for measurement of pedestal heat transfer coefficient.

Figure 6.12: Variation with time of heat flux measured for aluminium pedestals in the Sector-Rig.
Chapter 7    Full Tile Experimental Results

Experimental results captured on the full-Tile-Rig are included within this chapter. Aerodynamic data are displayed in terms of the hot and cold skin loss coefficients and fractional pressure-drop. Testing has been carried out over a nominal impingement jet Reynolds number range of \( \text{Re} = 7 \times 10^3 \rightarrow 33 \times 10^3 \). In order for systematic testing to be carried out in the presence of variable pressure conditions, ten different test points are determined by fan setting, with the fan being operated in the range from 700 up to 2900rpm. Testing has been carried out in five different measurement regions on the test plate under four different exit static pressure conditions. Additionally, in order to assess the performance of the angled jets providing direct stud cooling, further data have been gathered in which the camera field of view is focused on a close-up region of studs situated in the central and corner regions of the tile.

Within this chapter, heat transfer data are presented as maps of surface Nusselt number \( (\text{Nu}) \) in the different measurement regions. Maps are displayed for a selection of the different Reynolds conditions at which the facility was operated and under the different exit static pressure (ESP) cases. In addition to displaying contour maps, heat transfer data have been reduced to area averaged values in order to compare the performance across the different measurement regions. Discussion of the data presented within this chapter is restricted to a comparison of the different regions and ESP conditions tested on the Tile-Rig facility. A more detailed discussion follows in Chapter 9 in which the reduced data for both rigs are evaluated against each other and previously published data.

7.1 Pressure Drop and Loss Coefficient Measurements

The pressure-drop characteristics of the impingement-effusion tile model have been measured using the Tile-Rig facility. These data have been acquired for a range of flow rates (Reynolds numbers) spanning the limits specified by the minimum and maximum engine conditions. In the reporting of the experimental results both loss coefficient\(^2\) and fractional pressure drop are used. Pressure drop measurements are taken across the cold skin \( (\Delta p_{\text{imp}}) \), hot skin \( (\Delta p_{\text{eff}}) \) and tile \( (\Delta p_{\text{T}}) \). The reported parameters are defined in Table 7.1.

\(^2\) The loss coefficients defined in Table 7.1 use the friction coefficient equation adopted by Metzger et al. [105] for flow through pedestal arrays. The same format has been used for calculating loss coefficients and friction factor on the Sector Rig (Table 8.1, page 174).
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cold skin loss coefficient:</td>
<td>( f_{\text{imp}} = \frac{\Delta p_{\text{imp}}}{2 \rho U_{\text{imp}}^2} )</td>
</tr>
<tr>
<td>Cold skin fractional pressure drop:</td>
<td>( \frac{\Delta p_{\text{imp}}}{\Delta p_{TN}} )</td>
</tr>
<tr>
<td>Hot skin loss coefficient:</td>
<td>( f_{\text{eff}} = \frac{\Delta p_{\text{eff}}}{2 \rho U_{\text{eff}}^2} )</td>
</tr>
<tr>
<td>Hot skin fractional pressure drop:</td>
<td>( \frac{\Delta p_{\text{eff}}}{\Delta p_{TN}} )</td>
</tr>
<tr>
<td>Nominal tile fractional pressure drop:</td>
<td>( \frac{\Delta p_T}{p_{\text{in}}} = \frac{\Delta p_{\text{imp}} + \Delta p_{\text{eff}}}{p_{\text{in}}} )</td>
</tr>
</tbody>
</table>

Table 7.1: Pressure drop characteristic parameter definitions.

### 7.2 Exit Blockage Conditions

The full Tile-Rig has been designed to allow a degree of simulation of variations in hot skin exit pressure distribution that result from pressure fluctuations within the combustion chamber and lead to localised reductions in film-cooling velocity. The pressure variations, detailed in Chapter 5, are modelled as circumferential (spanwise) bands of blockage. Four different Blockage Cases (B0-B3) have been defined in Chapter 5 (Table 5.9) by separating the tile into three different pressure zones (Z1-Z3) and using blockage plates of different open areas. Fractional pressure drop and loss coefficient data have been acquired for each of the static pressure exit conditions considered.

#### 7.2.1 Uniform Exit Static Pressure Case (B0)

Exit static pressure case B0 represents an unobstructed hot skin with uniform exit static pressure. This case is considered the baseline scenario, relative to which the other arrangements are compared. A general view of the static pressure drops across both the hot and cold skins can be seen in Figure 7.1, which shows the data as a function of total mass flow rate through the tile model. The ratio of hot and cold skin pressure drops is plotted in Figure 7.2. Over the range of flow rates considered, the ratio of hot to cold skin pressure drops is constant at ~3.1. The hot and cold skins were designed with a nominal hot/cold skin pressure split of 3.9/1 based on open area of the two arrays, assuming equal discharge.
coefficient for the jets in each tile. The smaller than predicted value is indicative of a lower $C_D$ for the holes in the effusion plate, as would be expected due to the highly turbulent flow occurring beneath the impingement jets. Also, as $(L/D)_{\text{eff}} \approx 2(L/D)_{\text{imp}}$ this is consistent with the findings of Lichtarowicz [35] regarding the gradual reduction of $C_D$ with increasing $L/D$.

The hot skin pressure drop behaviour is shown in the form of a loss coefficient in Figure 7.3 as a function of the average effusion $Re$. The equivalent data for the cold skin is shown in Figure 7.4. The loss coefficients across the hot and cold skins are similarly affected by $Re$ with $f$ decreasing as $Re$ increases. The rates of decrease can be observed to start levelling off for $Re > 2 \times 10^4$ as the jet $C_D$ stabilises. The higher loss coefficient across the hot skin is representative of the lower effusion $C_D$ experienced by the effusion holes due to the highly turbulent flow between the skins.

7.2.2 Modified Exit Static Pressure Cases B1, B2 and B3

For the modified Exit Static Pressure cases the overall tile pressure drop ($\Delta p_T$) is different in each of the three axially arranged zones (shown in Figure 5.23). The tile pressure drop data in each of these zones and for each ESP case is shown in Figure 7.5. Note that the nominal tile pressure drop ($\Delta p_{TN}$) is the measured pressure drop in Zone 3, the exit end of the tile for which exit flow is always unobstructed. The data show that, depending on the ESP case, the local tile pressure drop in the middle and burner end zones is up to 40% lower than the nominal value. The hot skin pressure drop ($\Delta p_{\text{eff}}/\Delta p_{TN}$) in the three Exit Static Pressure zones is shown in Figure 7.6 for the various ESP cases. Although the exit flow from Zone 3 (exit end of the tile) is not restricted in any of the ESP cases, the hot skin pressure drop does vary in this region due to changes to the static pressure distribution in Zones 1 and 2. This is seen in Figure 7.6 where reducing levels of hot skin pressure drop ($\Delta p_T$) in either or both of the first two zones cause increases to the hot skin pressure drop in the unobstructed regions. The solid line in Figure 7.6 indicates the baseline hot skin pressure loss associated with ESP case B0. This provides a good indication of the effect of pressure blockages on flow migration in the channel between the hot and cold tiles. For cases B1 and B3 in which only the central region experiences blockage, migration can be seen to occur in both streamwise directions, to the front and rear of the tile. For case B2, in which the first two zones are both blocked, migration occurs only from front to rear, indicating that the greatest crossflow development is associated with this case.
Figure 7.7 illustrates the influence of the flow blockage on cold skin pressure drop in the different zones. It can be seen that when compared with the baseline case B0 the overall impingement pressure loss reduces as a greater blockage is presented across the hot skin. The greatest reduction is associated with case B2. The different ESP conditions can also be seen to affect the distribution of flow through the cold skin in the different zones, with the most pronounced individual reduction in Zone 2 for case B2. This may well be attributed to the presence of crossflow within the first two zones presenting an increasingly large pressure head between the two skins. This head is removed at Zone 3 in the absence of hot skin exit blockage with the largest impingement velocity occurring in this region.

### 7.3 Heat Transfer Measurements

Heat transfer measurements taken from the Tile-Rig are expressed in terms of $Nu$, defined previously in Chapter 1. The measurement of $h$ is based on the analysis process defined in Chapter 4 using the gas temperature entering the impingement hole. $Nu$ have been obtained over an impingement $Re$ range of ~7,000 to 33,000 in five distinct Measurement Regions across the tile (shown in Figure 5.23). For convenience, the surface $Nu$ maps featured in this chapter are plotted alongside a reference image illustrating the region from which they are taken. For each of these Regions, measurements have been taken for each of the 4 Exit Static Pressure cases described in Chapter 5.

The heat transfer data included within this chapter are all calculated based on measurements of the first crystal reaction time, for crystal response at $T_w \approx 32^\circ C$. It has been necessary to take measurements from this crystal exclusively due to the low level of heat transfer coefficients occurring at mid-point between jet centres and at low $Re$. These low heat transfer coefficients meant that for most test cases the only complete reaction that was seen was that of the first crystal in all regions away from jet centres, with noise from a very slow second crystal response impacting on the ability to process the data in many locations. It was attempted to promote more rapid crystal response away from the jets by increasing gas temperature; but the small rig scale factor results in high heat transfer coefficients for a given $Nu$. This resulted in values of $h$ beneath the jets of up to 900 $W/m^2.K$, at the highest Reynolds cases. This value was much higher than anticipated and for $T_{aw}$ higher than 50°C excessively rapid transition times were seen for all crystals in the below-jet region. A solution to counter this was to use a $\Delta T_g$ varying from ~30°C to ~25°C, reducing the temperature step for higher Reynolds cases, and to calculate $h$ from the first crystal only. The
decision to use only the first crystal was for consistency of results; however, a rapid crystal response time below the jets impacts upon the uncertainty of \( h \) values in that region.

Heat transfer data obtained from the Tile-Rig are captured for investigation of the spatial distributions of \( Nu \) and of the influence of various Exit Static Pressure distributions. The overall spatial distributions of \( Nu \) are investigated over the full range of \( Re \) for evaluation of heat transfer performance in the different regions of the tile and in the presence of blockage features. Experiments are carried out to investigate performance of the normal jet array, and also of the angled jets surrounding the studs. For this purpose additional tests have been carried out focusing on the studs in measurement regions Stud 1 (corner stud) and Stud 2 (central stud). The impact of the various Exit Static Pressure distributions is investigated by considering the effect of the different Blockage cases on \( Nu \) in the different regions when compared against the baseline case, B0.

In addition to providing a qualitative analysis of the heat transfer performance, the obtained surface \( Nu \) maps allow a visual assessment of any spatial variation in cooling resulting from blockages within the channel or occurring due to the ESP distribution. The impact of the various Exit Static Pressure distributions is investigated by analysing the effect of the different Blockage cases on \( Nu \) in the different regions when compared against the baseline case, B0. For evaluation of heat transfer performance across the entire tile, average Nusselt numbers \( \overline{Nu} \), have been taken at various locations across the tile surface. Averaged values are compared at different \( Re \) and under different exit static pressure conditions to evaluate the effect of each.

### 7.3.1 Nusselt Number Distribution for Uniform ESP Case B0

The basic distribution of \( Nu \) in each of the five Measurement Regions (shown in Figure 5.23) for the uniform exit static pressure case, B0 is discussed in this section. Surface \( Nu \) maps are displayed for each of the five measurement regions at impingement \( Re_{jet} \) of \( \sim 10,000 \) and 20,000.

Distributions of \( Nu \) for measurement region Stud 1 are shown in Figure 7.8 (Re\( \sim 11000 \)) and Figure 7.9 (Re\( \sim 21000 \)). Behaviour appears typical of that usually encountered within an impingement array. A regular pattern of high \( Nu \) directly beneath the jets can be seen to reduce in magnitude with radial distance from jet centre. At locations where impingement holes are missing from the otherwise regular array (in the vicinity of the stud and spacers)
significant $Nu$ values are displayed. It can also be seen that the presence of effusion holes does not appear to have a strong effect on the local value of $Nu$, although some small influence is apparent near some of these holes. For effusion holes located close to impingement jets, contours of high heat transfer can be seen to skew towards the holes, with a minor increase in local $Nu$ exhibited between the impingement and effusion jets. For effusion holes with no impingement jet between them, there is an apparent reduction in $Nu$ in the space between. The effect of the effusion holes on local $Nu$ is likely a result of the effusion holes attracting and swallowing the wall jet. Heat transfer is augmented for flow accelerating towards the holes but an associated reduction occurs in areas which the wall jet can less easily penetrate as flow is diverted. Very similar behaviour is exhibited at measurement region Stud 3 (the opposite corner stud), shown in Figure 7.12 ($Re$~10000) and Figure 7.13 ($Re$~20000).

Examination of the $Nu$ contours beneath the impingement jets in regions Stud 1 and Stud 3 shows evidence of non-uniformity in the local jet performance. During commissioning of the mesh heater, it was found that the fluid temperature at the edges and corners of the rig varied from the bulk temperature, although the major effects appeared to occur within ~50mm of the walls. The observed effects showed an increased temperature in this region which it has been surmised is due to reduced local mass flow rate. This premise is supported by the $Nu$ maps, as the presence of a local $T_{gas}$ higher than that used to calculate $Nu$ would result in an exaggerated result. As $Nu$ reduces significantly in the corner region, this must indicate a significantly retarded jet velocity, and reduced local $Re_{jet}$. As such, heat transfer data taken from the edges of the rig must be discounted from quantitative analysis.

The typical measured $Nu$ distribution for case Stud 2 is shown in Figure 7.10 ($Re_{jet}$~11000) and Figure 7.11 ($Re_{jet}$~21000). The stud and spacer are clearly visible in the central region of the data, while the ports can be seen in the upper-left and lower-right corners of the image. It may be concluded from Figure 7.10 and Figure 7.11 that sensibly uniform heating is maintained across the entire region, except for the locations around the central stud where jets from the normal array have been omitted due to the obstruction it presents. The alignment of the cooling array with the stud in this region results in a significantly larger area in which jets are absent. This highlights the small area of influence of individual jets in this geometry as a very acute reduction of $Nu$ is apparent in this region. Due to this large area of low heat transfer, the influence of the effusion holes on the passage of the wall jets is more apparent around the central stud than in regions Stud 1 and Stud 3.
is most apparent in Figure 7.11 that the $Nu$ contours emanating from the jet touchdown locations spread towards the effusion holes surrounding the central stud, but no similar contours occur between effusion holes. The shape of the contours remains generally uniform and circular over the entire region, suggesting that any crossflow that may be present is of insufficient strength to affect the jets. This is consistent with observations of regions Stud 1 and Stud 3, with no apparent change imparted by the close proximity of the local ports. For all three of the stud measurement regions, the performance of the angled stud cooling jets are partially resolved in the data, appearing to show relatively weak performance compared to the surrounding normal jets. Poor resolution due to the overall size of the measurement region limits the ability to satisfactorily assess heat transfer performance on the stud fillet. To allow for further assessment, testing has been repeated with the camera focussing more closely on the studs in regions Stud 1 and Stud 2. This is illustrated in Figure 7.20 and Figure 7.21 and discussed in Section 7.3.1.

Typical $Nu$ for region Array 1 are shown in Figure 7.14 ($Re_{jet} \sim 11000$) and Figure 7.15 ($Re_{jet} \sim 21000$) and for region Array 2 in Figure 7.16 ($Re_{jet} \sim 10000$) and Figure 7.17 ($Re_{jet} \sim 21000$). Both regions contain arrays of jets uninterrupted by the presence of studs and spacers; however each is in the vicinity of one of the larger blockage features. Region Array 1 borders the igniter port, with a small portion evident in the lower-left corner of the data. Region Array 2 borders the secondary port, which is clearly visible in the upper-right corner of the data. The lower right section for Array 2 contains no data as that is an area of the plate to which liquid crystal was not applied.

As with the stud regions both of the array regions display largely uniform impingement performance, although a small effect caused by the relative position of the impingement and effusion holes is evident for Array 1. This is seen as a gradual change in shape of the high heat transfer impingement regions with streamwise position. This further demonstrates the attraction of the impingement jet towards the effusion holes, with the troughs of $Nu$ again occurring between effusion holes. The influence of the array density ($X/D$) in the different regions is apparent when comparing Array 1 (Figure 7.14 and 7.15) and Array 2 (Figure 7.16 and 7.17), as a much greater independence of individual jets is apparent for the larger $X/D$ in Array 2. This reduced interaction between jets results in the individual impingement points in Array 2 being characterised by circular areas of high heat transfer.
*Nu* have been averaged across thirteen different areas spanning the five measurement regions. Areas *Nu*_ave1 and *Nu*_ave3 from region Stud 1 (Figure 7.9); *Nu*_ave4 and *Nu*_ave5 from region Stud 2 (Figure 7.11) and *Nu*_ave7 and *Nu*_ave8 from region Stud 3 (Figure 7.13) feature averaged Nusselt numbers from the uninterrupted jet array and from the region surrounding the studs. Averaged Nusselt numbers from the uninterrupted jet arrangement are also taken from regions Array 1 (*Nu*_ave9 and *Nu*_ave10, Figure 7.15) and Array 2 (*Nu*_ave11 and *Nu*_ave12, Figure 7.17). Additional averaged Nusselt numbers are taken about areas bordering the upstream (*Nu*_ave6) and downstream (*Nu*_ave13) igniter ports, with data also averaged about a spacer in the Stud 1 region (*Nu*_ave2). For consistency, the unobstructed averaged Nusselt numbers have been taken about areas corresponding to 2x2 jet pitches about a single central jet. The averaged Nusselt numbers surrounding the different studs have been selected to best capture the local performance based upon the alignment of jets to the different studs, covering an area of 3x2 jet pitches. Averaged Nusselt numbers beneath the uninterrupted jet array are compared across the full Reynolds testing range for the five different Measurement Regions in Figure 7.18, with local variations of similar representative areas within regions Array 1 and Array 2 compared in Figure 7.19. It can be seen from Figure 7.18 that whilst *Nu* follow a sensible Reynolds trend, spatial performance is somewhat variable. For the average values that have been captured, broadly similar *Nu* values are displayed across the full Reynolds range. The captured Nusselt numbers display a degree of variation, with differences between the minimum and maximum *Nu* of order 20-30% when comparing individual areas across a range of test cases, although on average the different measurement regions exhibit close performance, with average variance between regions of ~10%. The weakest performance appears to occur with region Stud 3, for area *Nu*_ave7 and the highest *Nu* can be typically seen for the *Nu*_ave4 area, in the Stud 2 measurement region. It can be seen from Figure 7.18 that there is not an obvious, consistent trend in terms of ranking the relative heat transfer performance of the different regions as the averaged values plotted vary somewhat from a consistent trend line. Typically, marginally lower values of *Nu* may be seen in the measurement regions situated in the sparser cooling array (areas *Nu*_ave7 and *Nu*_ave11) towards the rear of the plate, with higher averaged Nusselt numbers associated with the denser jet array (areas *Nu*_ave5, *Nu*_ave4 and *Nu*_ave9) as would be expected due to the greater jet interaction. The variations in *Nu* apparent in Figure 7.18 can be seen to not only occur across measurement regions, with variations of a similar magnitude seen to occur in Figure 7.19 within the Array 1 region. *Nu* for the two averaged areas within this
measurement region can be seen to differ by up to 20%. This is the result of a spatial variation in stagnation region $Nu$ that may be clearly observed in Figure 7.14 and 7.15, with $Nu$ increasing with streamwise and spanwise position from top right to bottom left of the figures. The difference in performance appears from Figure 7.19 to become more pronounced for $Re_{jet}$ greater than 20000. The same behaviour is not apparent in the Array 2 region; the averaged Nusselt numbers in this region remain much closer across the entire Reynolds range. The large size of the cooling tile and the mesh heater meant that variations in the performance between measurement regions were anticipated, and the degree of spatial variation in Nusselt number appears to be of a sensible magnitude.

### 7.3.2 Nusselt Number Distribution for Direct Stud Cooling Angled Jets

All studs on the Tile-Rig are cooled using three angled jets arranged as shown in Figure 5.4. Holes drilled at two different angles are used, with two angled at 50° to the horizontal and one at 30°. In order to evaluate the performance of the angled jets, heat transfer experiments have been carried out with the camera field of view focussed directly on the stud region. This is shown in Figure 7.20 and 7.21 for regions Stud 1 and Stud 2 respectively at $Re_{jet} \approx 27000$.

The different angles of the stud jets results in different touchdown locations as can be seen in Figure 5.4. The 30° hole, visible below the stud in Figure 7.20 and 7.21, makes contact approximately halfway up the stud shaft, whilst the 50° holes, visible to the upper right and upper left of the studs in Figure 7.20 and Figure 7.21, touch down on the fillet. The angle of the different jets results in only a small portion of the 30° jets being visible; however, the centres of the 50° jets can be seen in Figure 7.20 and Figure 7.21 impinging on the fillet, indicated by the dashed lines. The cooling provided by the stud jets is particularly clear in Figure 7.21 and it can be seen that all of the angled jets provide significantly less heat transfer augmentation to the normally aligned equivalents. The stagnation Nusselt numbers of the 50° stud jet and the surrounding standard array jets are compared in Figure 7.22. The 50° angled jets are shown to provide cooling at ~55% the level of the normal impingement jets. The 30° jet cooling, although not shown in Figure 7.21 is weaker still; however as the jet centre is not visible on the target plate this is to be expected. In order to assess the performance of the cooling jets in the region of the fastening studs, Figure 7.23 has been produced displaying $\overline{Nu}$ of the immediate stud area in the three Stud based measurement regions relative to the local $\overline{Nu}$ for the uninterrupted array. From Figure 7.23 it can be
observed that typical values of $\overline{Nu}$ are of order 30-40% lower in the stud region than the surrounding unobstructed area. Of the three examined stud regions, the weakest performance is associated with the corner studs; however the apparent reduction in jet strength at the corners of the rig, mentioned previously in this chapter, may result in the greater reduction seen in these regions.

Data in Figure 7.23 have been plotted for Reynolds number greater than 20000, as the low Nusselt numbers in the stud regions mean that in certain cases, areas around the stud do not experience a full crystal reaction during the time frame of the experiment. This produces a drop out of data that affects $\overline{Nu}$, as only the regions of higher heat transfer coefficient are captured.

### 7.3.3 Effect of Exit Static Pressure Condition on Nusselt Distribution

Heat transfer tests on the Tile-Rig have been performed under four different exit static pressure conditions, detailed in Chapter 5, in order to simulate elevated pressure from within the combustion chamber impinging on the liner wall. Heat transfer experiments have been carried out under the different ESP cases and the effects of the different pressure conditions on surface Nusselt number have been investigated.

The different exit static pressure distributions have been implemented through the use of restrictor plates positioned downstream of the hot skin. The restrictor plates are manufactured from clear Perspex plates featuring varying numbers of 20mm diameter holes. In order to hold the restrictor plates in position and divide the hot skin into the three different pressure zones shown in Figure 5.23, vertical plywood mounts have been fitted into the exit plenum. The restrictor plates and mounts present obstructions to the camera field of view, both directly and as a result of reflections introducing glare to the image. Efforts were made during experiments to arrange the camera such that interference and obstruction was minimised; however, for certain arrangements reflections were unavoidable. This is most evident in the contour plots from region Stud 2 under all pressure conditions except B0, with reflection off the plywood mount resulting in a horizontal strip of unprocessed data along the centre of the image. This can be seen in Figure 7.24 in which gaps in the data caused by obstructions from the edges of holes in the restrictor plate may also be seen. In all other measurement regions it was possible to eliminate reflections by appropriate positioning of the camera; however, a small level of data is also lost in region Array 1 under pressure condition B2, due to obstructions from the edges of holes in the restrictor plate.
The variations in exit static pressure for the different blockage cases result in the central region of the plate experiencing the greatest variety of exit pressure blockage, with contour maps of surface $Nu$ included for each of the four.

Contour maps of surface $Nu$ are presented for the Stud 2 measurement region under Pressure conditions B1, in which flow through the hot skin in zone 2 is restricted (shown in Figure 7.24); under Pressure condition B2, in which flow through the hot skin is restricted in both zones 1 and 2 (shown in Figure 7.25) and under Pressure condition B3, in which flow through the hot skin in zone 2 is completely blocked (shown in Figure 7.26). The three blockage case contour maps are plotted for $Re_{jet} \sim 20000$. The baseline pressure case, B0 has been plotted previously at comparable $Re_{jet}$ in Figure 7.11.

Similar cooling patterns can be observed for pressure conditions B1 and B3 as were encountered in B0, with each experiencing very similar baseline performance. There is evidence of local pressure blockage appearing to result in higher jet centre $Nu$, although this effect is less pronounced for the full central blockage case, B3. A significantly different contour map can be seen in Figure 7.25 for pressure condition B2 than for the other cases. In addition to significantly lower $Nu$ over the entire region, the presence of exit blockage in both zones 1 and 2 can be seen to have resulted in significant crossflow build up, as flow unable to exit through the hot skin in the first two pressure zones is diverted towards the rear of the tile, to the unobstructed zone 3. The level of crossflow can be seen to increase largely across Figure 7.25 in the direction of Zone 3, with the effects most noticeable in the regions between the two ports, near the central stud. The presence of more defined jet contours at the upper right and lower left corners of Figure 7.25 seem to indicate that the open area between the two ports acts as a nozzle through which the crossflow accelerates. The influence of the crossflow appears to increase as it approaches the effective throat of the channel between the ports, with the centre stud and spacer also obstructing the flow in this region. Stagnation $Nu$ can be seen to decrease up to this point with jet distortion becoming significantly more pronounced when compared to the circular contours seen in Figure 7.11 for the B0 case. Beyond this point the crossflow appears to decrease as the channel opens wider, with stagnation $Nu$ increasing. Despite the reduction in crossflow, it still appears to have a significant effect on jet performance at the most downstream location in the Stud 2 region which borders the end of zone 2 and the start of zone 3. As the crossflow that has been observed in zone 2 has been attributed to pressure blockages in zones 1 and 2 forcing the
flow towards the rear of the tile, the impact that this crossflow has in the neighbouring zones has also been investigated.

The extent to which the crossflow has developed within zone 1 and the degree to which it continues into zone 3 has been investigated by observing the Nusselt contour plots from regions Array 1 and Array 2 under ESP case B2. Measurement region Array 1, as shown in Figure 7.27, is situated in zone 1 and the rear of the region (at the bottom of Figure 7.27) aligns with the border of zones 1 and 2. The influence of crossflow within region Array 1 appears to be minimal over the first few jet rows; however, evidence of minor crossflow is apparent, with its appearing to increase with proximity to the border of zones 1 and 2. The contours of $Nu$ beneath the jets appear to change shape, becoming more elliptical in the direction of the flow. From Figure 7.27 it appears that the crossflow in zone 1 has developed to a sufficient extent to influence jet performance, but to a level significantly below that seen around the central stud. As was seen in Figure 7.25 for the central region, the crossflow appears to have the greatest influence on the jets adjacent to the igniter port, suggesting a degree of crossflow is accelerated around the obstacle.

It can be seen in Figure 7.28, illustrating the Nusselt distribution of region Array 2 under ESP case B2 that a significant degree of the crossflow that has developed over the first two zones continues into zone 3. The jets immediately downstream of zone 2 can be seen to be largely affected by the crossflow with the shape and strength of the jet Nusselt contours clearly altered from those shown in Figure 7.17 for baseline case, B0. The influence of the crossflow can be seen to decrease rapidly as it continues into zone 3, with an identifiable reduction in the effect on each subsequent jet row as streamwise distance from zone 2 increases. Beyond the secondary port, the impact of the crossflow has reduced to such an extent that the Nusselt contours resulting from individual jets return to a circular profile very similar to that seen in Figure 7.17 for the baseline case, B0. The visible impact of crossflow in the contour plot (Figure 7.28) dissipates completely for the final two jet rows, at the rear end of the Array 2 region. The impact of the crossflow passing from zone 2 to zone 3 appears to be at its greatest on the jets adjacent to the local port, this behaviour is consistent with observations from zones 1 and 2 under pressure condition B2.

Whilst the large crossflow that develops under exit static pressure condition B2 appears to have the greatest impact on the $Nu$ contour plots in region Stud 2, the presence of crossflow in region Stud 2 is also apparent, to a greatly reduced degree, for ESP condition B3. For this pressure condition, only pressure zone 2 is blocked, but no flow may exit
through the hot skin in this region and a degree of crossflow must develop as flow diverts towards zones 1 and 3. Crossflow is only apparent in this region to the lower-right of the image, where it appears that flow is accelerated between the port and the central stud, with the profile of the contours exaggerated in the direction of the crossflow towards zone 3. The low level of crossflow in the Stud 2 region is expected as any crossflow developing under ESP condition B3 starts within zone 2, developing towards the front and rear of the tile. Considering this, the effects should be most clear at the borders to the zone, and the extent to which the crossflow impacts upon heat transfer performance in zones 1 and 3 is investigated by investigating Nusselt contours for the Array 1 and Array 2 regions. $Nu$ contours for region Array 1 under blockage condition B3 are shown in Figure 7.29, with very little impact of crossflow apparent. The visible effects appear to be limited to a small degree of distortion of the jets immediately adjacent to zone 2 (at the bottom of Figure 7.29) with the Nusselt contours extending further in the direction of the crossflow. Elsewhere in the region the contours appear unaffected, with jet behaviour appearing similar to that shown in Figure 7.15 for the baseline case, B0.

The impact of crossflow is much more apparent in Figure 7.30 for region Array 2, than for the Stud 2 or Array 1 regions. The crossflow can be seen to extend into zone 3 in a similar fashion to that observed in Figure 7.28 under condition B2, with the effects on the jet contours reducing rapidly with streamwise distance into zone 3. Beyond the secondary port the contours return to the circular profiles observed in Figure 7.15 for the baseline pressure condition, B0. The level of crossflow within the Array 2 region appears much lower than was observed for the B3 pressure condition, with a significantly lower degree of distortion seen in the jet $Nu$ contours. As was observed in Figure 7.28 the crossflow appears to have the greatest impact on the performance of jets in the vicinity of the port.

The effect of the different exit static pressure conditions on the Nusselt contours in the corner stud regions is much less pronounced than has been observed for the Stud 2 and Array regions. The Nusselt contours for the Stud 1 region are shown in Figure 7.31, and it may be seen by comparing the performance to the uniform static pressure distribution shown in Figure 7.9 that over much of the region, no significant changes are evident in the distributions or magnitudes of $Nu$. The contour magnitudes beneath impingement jets approaching the injector appear marginally reduced when compared with the baseline B0 case. This may be the result of minor crossflow development due to the exit blockage in zone 1; however, the effects are minimal. For all other ESP cases in this region no apparent variation in the
contours of surface $Nu$ could be observed; similarly in the Stud 3 measurement region, changes to exit pressure condition did not appear to alter heat transfer performance.

**7.3.4 Effect of Exit Static Pressure Condition on Averaged Nusselt Numbers**

The effect of the exit static pressure distributions on the central regions of the plate has been examined by investigating visible effects on the distributions of $Nu$ contours. The discussion in the previous section has been primarily focused on the central region as very little effect was apparent on the contour plots in the corner regions. In addition to changes to the distribution of $Nu$, altering the exit static pressure condition impacted upon the displayed magnitudes of $Nu$ due to the redistribution of the flow through the tile when compared to the uniform exit pressure case. In order to quantify the effects, area average Nusselt number $\overline{Nu}$, has been plotted for the different ESP cases.

Values of $\overline{Nu}$ from measurement region Stud 2 are shown in Figure 7.32 for the area $Nu_{ave4}$ (refer to Figure 7.11). As expected, averaged Nusselt numbers are significantly reduced in this region for the B2 ESP case, with values of $\overline{Nu} \sim 67\%$ of those encountered under uniform ESP condition, B0. Under the other ESP conditions (B0, B1 and B3) $\overline{Nu}$ appears equal for $Re_{jet}$ up to $\sim 20000$. For $Re_{jet}$ greater than this the performance under the B3 condition appears to fall below the B0 and B1 cases. This behaviour has been attributed to the higher Reynolds number resulting in more significant crossflow under the B3 condition, in which no flow can exit through the tile in zone 2, in which measurement region Stud 2 is situated. The behaviour of $\overline{Nu}$ in Figure 7.32 is reflected in the values taken from area $Nu_{ave6}$, adjacent to the local port with similar magnitudes also present. Obstructions from the restrictor plate and reflections from internal rig reflections impede the ability to fully resolve the average Nusselt numbers surrounding the central stud under the different ESP cases; however a similar trend can be identified.

Examination of the Nusselt contour plots of the Array 2 region under ESP conditions B2 and B3 indicated a large spatial variation in performance. Crossflow developing from earlier blocked measurement zones appeared to have a large effect on the jet performance just downstream of zone 2. This effect appeared to dissipate as streamwise distance into the pressure zone increased. The averaged Nusselt numbers close to the zone 2 exit are plotted in Figure 7.33, for area $Nu_{ave12}$; the averaged Nusselt numbers for a position further from the
exit of zone 2 can be seen in Figure 7.34 for area $\text{Nu}_{\text{ave}11}$. Both these areas are shown in Figure 7.17, as well as area $\text{Nu}_{\text{ave}13}$ showing average Nusselt numbers around the secondary port.

Analysis of Figure 7.33 shows the averaged Nusselt numbers from the $\text{Nu}_{\text{ave}12}$ area are affected by the B2 pressure conditions in a similar manner to the central measurement region. The reduction in $\overline{\text{Nu}}$ for area $\text{Nu}_{\text{ave}12}$ under ESP case B2 is comparable to that reported for the $\text{Nu}_{\text{ave}4}$ area, with values of $\overline{\text{Nu}} \sim 67\%$ of the level experienced for uniform exit static pressure condition B0. Figure 7.33 also shows the degree to which the crossflow effects that were observed in Figure 7.30 impact upon the Nusselt numbers at area $\text{Nu}_{\text{ave}12}$. The effects of crossflow are less pronounced under condition B3 than for B2; however, a significant reduction in $\overline{\text{Nu}}$ is still apparent, with values $\sim 85\%$ that of the level experienced under uniform exit static pressure condition B0. Observation of the Nusselt contours under the different ESP conditions B2 (Figure 7.28) and B3 (Figure 7.30) appeared to indicate a more significant impact of cross flow in the region of the port than in the open area; however, the effect of the pressure conditions on the heat transfer performance in area $\text{Nu}_{\text{ave}13}$ were also compared, with Nusselt numbers in this area affected to a similar degree as those for area $\text{Nu}_{\text{ave}12}$.

Examination of Figure 7.34 allows an assessment to be made of the degree to which the crossflow effects dissipate over measurement region Array 2. Area $\text{Nu}_{\text{ave}11}$ is separated from area $\text{Nu}_{\text{ave}12}$ by five impingement jets (nine effusion holes) and this is sufficient for a marked reduction in crossflow. Comparing the averaged Nusselt numbers shown in Figure 7.34 to those that were encountered in Figure 7.33 highlights the large difference in performance, with $\overline{\text{Nu}}$ for area $\text{Nu}_{\text{ave}12}$ under ESP conditions B2 and B3 $\sim 90-95\%$ of the level encountered under the uniform condition, B0. Beyond the Array 2 region, a slightly smaller degree of reduction in $\overline{\text{Nu}}$ has been observed for area $\text{Nu}_{\text{ave}8}$, an unobstructed section of region Stud 3 (see Figure 7.13) situated further to the rear of the tile.

The $\text{Nu}$ contours and averaged Nusselt numbers in the Array 2 region highlight that the largest impact of the pressure blockages in terms of cold-side heat transfer performance is on the introduction of crossflow to the system. This measurement region is situated in zone 3, in which no obstruction is placed on exit flow; however crossflow produced due to the exit conditions in zones 1 and 2 has a clear effect on performance. Additionally, from comparison of $\text{Nu}$ in areas $\text{Nu}_{\text{ave}11}$ and $\text{Nu}_{\text{ave}12}$ it can be seen that effusion holes are highly
effective at removing crossflow from the channel due to the short distance over which this is achieved.

Values of $\overline{Nu}$ within the measurement regions contained in zone 1 are less susceptible to changes to the exit static pressure condition that the Stud 2 and Array 2 regions. Plots of the variation of $\overline{Nu}$ under the different exit pressure conditions are shown in Figure 7.35 for area $\overline{Nu}_{ave10}$ taken from region Array 1 (see Figure 7.15) and in Figure 7.36 for area $\overline{Nu}_{ave3}$ taken from region Stud 1 (see Figure 7.9). These two areas experience similar changes under the different pressure conditions, with very similar performance exhibited under all pressure conditions except for B2, in which flow through the hot skin in zones 1 and 2 is obstructed. Under ESP B2 a reduction in $\overline{Nu}$ is apparent in both areas. This reduction is most likely a result of the crossflow that has been observed to occur under condition B2. The level of reduction in $\overline{Nu}$ compared with the B0 case in the two areas is consistent with this, as area $\overline{Nu}_{ave10}$, which is situated further from the front of the rig appears more greatly affected than area $\overline{Nu}_{ave3}$. Very similar behaviour is exhibited in area $\overline{Nu}_{ave5}$ to that shown in these areas, with a level of reduction similar in this area comparable to that shown in Figure 7.36 for area $\overline{Nu}_{ave3}$. The effect of the different pressure conditions in area $\overline{Nu}_{ave1}$ (see Figure 7.9) around the front stud is plotted in Figure 7.37. It can be seen in this region that there is no discernible effect of changing the pressure condition on the heat transfer performance in this region. The Nusselt numbers in area $\overline{Nu}_{ave7}$, surrounding the rear stud are similarly unaffected by the different exit pressure conditions as has been observed in the front stud region.

### 7.4 Uncertainty Analysis

The uncertainty of the Nusselt numbers within this chapter have been calculated using the Kline-McClintock method discussed previously in Chapter 2. The uncertainty is calculated using equation (8.3) [94].

$$u_R = \sqrt{\left(\frac{\partial R}{\partial x_1}u_1\right)^2 + \left(\frac{\partial R}{\partial x_2}u_2\right)^2 + \left(\frac{\partial R}{\partial x_3}u_3\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n}u_n\right)^2}$$  \hspace{1cm} (7.1)

The solution equation used to measure the surface heat transfer coefficient, equation (4.5) ((page 73), introduces three sources of error into the measured value of $h$. These are:
Surface and fluid temperatures ($T_o$, $T_w$ and $T_aw$)

These temperatures are connected in the solution equation, with $(T_aw - T_i)$ and $(T_w - T_i)$ both used as terms within the equation. The fluid temperatures are measured using fine-wire thermocouples that have been calibrated against a platinum resistance thermometer to remove bias error. The wall temperature is determined by the liquid crystal applied to the surface exhibiting a defined colour response at a specific, calibrated reaction temperature. Calibration of the liquid crystal provides a high level of certainty in $T_w$ at the reaction temperature; however, the initial wall temperature must be determined based on the initial gas temperature measurement as the experiment starts at equilibrium and at $t=0, T_w = T_i$. As such, the uncertainties for both the temperature terms have a level of error based on the accuracy of the thermocouples, with standard errors for both equal at $u(T_aw - T_i) = u(T_w - T_i) = \pm 0.5^\circ C$.

Thermophysical properties of the heat transfer surface, $\kappa = \sqrt{\rho c k}$

The heat transfer surface is manufactured from Perspex, for which density, specific heat capacity and thermal conductivity can vary based upon the mode of manufacture and the atmospheric conditions. The values used within the solution equation are $\rho = 1190 \text{ kg/m}^3$; $c = 1500 \text{ J/kg.K}$ and $k = 0.18 \text{ W/m.K}$ [103-104] for a value of $\kappa = 567 \text{ W.s}^{0.5}/\text{K.m}^2$. Although these properties will not be subject to random uncertainty there will be a degree of bias introduced by temperature changes and from differences between the stated and actual values of the constituent components of $\kappa$. In order to account for this a standard error of $u_\kappa = \pm 28.3 \text{ W.s}^{0.5}/\text{K.m}^2$ is used for the uncertainty analysis.

Time, $t$ at which the crystal reaction is observed

The uncertainty in the time of the crystal reaction is reduced through the use of the LED to synchronise the start of the experiment. This essentially reduces the error in $t$ to the frame rate of the camera, $u_t = \pm 0.04 \text{ s}$.

Applying these sources of uncertainty to equation (8.3) allows equation (8.4) to be derived. This may be used to calculate the uncertainty in the measured value of $h$ based upon the accuracies of the individual measurements.

$$u_h = \sqrt{\left(\frac{\partial h}{\partial t} u_t\right)^2 + \left(\frac{\partial h}{\partial \kappa} u_\kappa\right)^2 + \left(\frac{\partial h}{\partial (T_aw - T_i)} u(T_aw - T_i)\right)^2 + \left(\frac{\partial h}{\partial (T_w - T_i)} u(T_w - T_i)\right)^2} \quad (7.2)$$
Equation (8.4) has been derived based on the step-based solution of Fourier’s one-dimensional conduction equation that is commonly used to calculate $h$ based on transient liquid crystal measurements. As a summation (step-series) method has been used, the calculated value of $h$ is based upon a series of infinitesimal temperature steps (of duration 0.04 s). For the purpose of calculating the uncertainty, the value of $\partial (T_{aw} - T)$ is based on the mean fluid temperature exhibited up to reaction time, $t$. Figure 7.38 illustrates the variation of $T_{aw}$ with time for a typical gas temperature transient.

Due to the significance of different sources of uncertainty being dependent on the duration of the experiment, equation (8.4) is solved for a range of $h$ corresponding to the full range observed in the experiment; the resulting percentage uncertainty is shown in Figure 7.39. It can be seen from Figure 7.39(a) that the level of uncertainty decreases as the time for the crystal response increases. The percentage uncertainty in $h$ decreases from a maximum of 10.0% for a crystal reaction occurring after 1 second, decreasing rapidly and levelling off towards a minimum uncertainty of 6.2% for a crystal reaction occurring at 22 seconds or greater. The maximum uncertainty is associated with very high heat transfer coefficient, $h \sim 900 \, W/m^2.K$ ($Nu \sim 165$) that only occurs beneath jet centres at high Reynolds numbers. The level of uncertainty can be seen in Figure 7.39(a) to decrease rapidly towards a steady value, with uncertainty dropping below 8% for crystal responses occurring only 0.3 seconds later (corresponding to $h \sim 525 \, W/m^2.K$ and $Nu \sim 100$). The variation of uncertainty with Nusselt number may be seen in Figure 7.39(b), with the strong influence of $Nu$ easily apparent. In practice, the Nusselt distribution over the surface is such that the uncertainty at most measurement points is below 7% and typical distributions of uncertainty are shown for the Array 1 measurement region in Figure 7.40(a) at the maximum Reynolds case ($Re \approx 32000$) and in Figure 7.40(b) at $Re \approx 24000$ in order to illustrate this.

<table>
<thead>
<tr>
<th></th>
<th>$Re \approx 32000$</th>
<th>$Re \approx 24000$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Proportion of region under 9% Uncertainty</td>
<td>97.5%</td>
<td>99.9%</td>
</tr>
<tr>
<td>Proportion of region under 8% Uncertainty</td>
<td>89.5%</td>
<td>95.8%</td>
</tr>
<tr>
<td>Proportion of region under 7% Uncertainty</td>
<td>69.4%</td>
<td>74.5%</td>
</tr>
<tr>
<td>Proportion of region under 6.5% Uncertainty</td>
<td>43.7%</td>
<td>46.4%</td>
</tr>
<tr>
<td><strong>Averaged % Uncertainty</strong></td>
<td><strong>6.87%</strong></td>
<td><strong>6.76%</strong></td>
</tr>
</tbody>
</table>

Table 7.2: Uncertainty statistics of calculated heat transfer coefficient in measurement region Array 1

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The contour maps consist of $2 - 4 \times 10^5$ individual measurement points and from tabulating data from Figure 7.40 into Table 7.2 it can be seen that even at the highest Reynolds case, averaged uncertainty is $\sim 6.9\%$, with this figure dropping for the lower Reynolds number test as would be expected from Figure 7.39(b).

### 7.5 Closure

The experimental results from the full-tile test rig demonstrate the ability to model a full combustor liner tile successfully for the purpose of cold-side heat transfer measurements. The hot and cold skin jet arrays have been successfully sized, with the ratio of pressure drops across the two liner skins observed to be at a sensible level, close to design. The pressure drop across the hot skin is slightly higher than anticipated due to increased turbulence between the skins. This results in a higher loss coefficient through the effusion jets than for the impingement array that can be seen in Figure 7.3 and Figure 8.7.

Nusselt number data has been successfully gathered across five different measurement regions over a range of jet Reynolds numbers from $7 - 33 \times 10^3$, and over a range of four different exit static pressure conditions. Uncertainties of the measured heat transfer coefficients have been shown to be dependent on Nusselt number, with averaged uncertainty evaluated as $6.8\%$. The experimental results have revealed that Nusselt numbers at a given $Re$ are relatively consistent across the entire tile, with average values of similar regions experiencing comparable values. A variance of up to $20\%$ has been observed in $\bar{Nu}$ values of the different measurement regions under similar jet Reynolds numbers. This difference in Nusselt numbers observed across the entire tile is also seen in local variations within one particular measurement region (Array 1). This would appear to indicate that the difference in performance is related to local flow behaviour rather than as an artefact of uneven flow distribution entering the rig.

Changes in exit static pressure have been shown to have a strong influence on the development of crossflow within the rig, particularly in zone 2 and the start of zone 3 under pressure condition B2. The strongest crossflow is associated with this ESP condition due to exit pressure restrictions in the first two zones. Less pronounced effects of crossflow are also observed under pressure condition B3, as spent air is forced away from the central zone due to the total exit blockage in that region. Under this condition the impact is most pronounced in region Array 2, immediately downstream of zone 2. The heat transfer performance of
measurement regions located within areas of large crossflow has been shown to be significantly impaired, with $\overline{Nu}$ dropping by more than 30% from values exhibited under uniform static pressure conditions in the worst affected areas. In the absence of crossflow only a very limited impact of ESP distribution is apparent. This suggests that the presence of large pressure fluctuations within the combustion chamber will have its most significant effect if exit flow is consistently obstructed over a significant portion of the tile, although restrictions on flow exit will impact on film cooling coverage of the hot side of the tile.

The performance of the angled jets positioned around the fastening studs has been investigated, with Nusselt numbers shown to be significantly weakened compared to the normal effusion jets. Stagnation jet values for the 50° jets were recorded as being ~55% the strength of the normal equivalents, with the 30° jets performance weaker still (exit stagnation values could not be quoted as impingement does not occur on the tile surface). Evaluation of the $\overline{Nu}$ in the stud regions indicated Nusselt number levels are 60-70% that of the uninterrupted array within the same region. The failure of the adopted cooling geometry to provide levels of cooling consistent with those typically observed indicates that further consideration needs to be given to the direct cooling of such components.

A more detailed discussion of the results from the Tile-Rig, published within this chapter is included in Chapter 9, in which heat transfer performance is compared against the results from the Sector-Rig and against previously reported correlations of impingement heat transfer performance.
7.6 Figures

Figure 7.1: The cold skin and hot skin pressure drops as a function of total mass flow rate (Exit Static Pressure Case B0).

Figure 7.2: The ratio of cold skin to hot skin pressure drop as a function of Tile-Rig mass flow rate (Exit Static Pressure Case B0).
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Figure 7.38: Comparison of fluid temperature transient and averaged fluid temperature up to time, $t$. 
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Figure 7.40: Distribution of uncertainty in region Array 1 at a) Jet Reynolds number = 32000 and b) Jet Reynolds number = 24000
Chapter 8  Sector-Rig Results

Aerodynamic and heat transfer measurements from the parametric study performed on the Sector-Rig facility are detailed within this chapter. The reported data identify that altering the $H/D$ and $P/D$ of pedestals positioned in an array beneath a distributed impingement feed has significant effects upon both the porosity of the geometry and the cooling performance. Testing has been carried out over a range of aerodynamic conditions, with experiments operated over an impingement jet Reynolds number range of $Re_{jet} = 7 \times 10^3 \rightarrow 3.3 \times 10^4$, representative of the engine conditions provided by Rolls-Royce plc and detailed in Chapter 6. Additional testing has been carried out for certain geometries at values beyond this range up to $Re_{jet} \approx 4 \times 10^4$. The parametric study has been conducted for a selection of the different geometries proposed in Chapter 6, with data captured and presented for geometries S2a-c, S3a-c and S4b-c; whilst testing on the $P/D=1.25$ geometries, S1a-c and S4a, that were deemed of lower priority were omitted due to time constraints.

Aerodynamic measurements have been carried out in order to evaluate the influence of changes to the pedestal geometry on the effective porosity of the different cooling patterns. Captured aerodynamic data are displayed in terms of the fractional pressure-drop across the impingement array and along the channel length and the friction-factor based upon the total system pressure loss. A porosity coefficient is also presented as a ratio of the effective open area of the impingement array, based upon the total pressure drop across the geometry, to the actual measured open area. Measurements have been taken of surface and pedestal heat transfer coefficient, with data presented in terms of non-dimensional Nusselt numbers ($Nu$). Surface heat transfer data are presented in three primary formats within the chapter: contour maps; span-averaged plots and area averaged values, with measurements identifying superior heat transfer performance for the Shielded Impingement geometries. Heat transfer data for pedestals are presented to supplement the surface heat transfer measurements; however obtained results have been limited to only two geometries but suggest little variation of $Nu$ with $H/D$ beyond the effect on velocity due to a reduction in open area.

Discussion of the data presented within this chapter is restricted to a comparison of the different geometries tested on the Sector-Rig facility. In addition to the discussion conducted within this chapter a comparison of these results and those taken from the Full Tile Test Facility (reported in Chapter 7) is conducted in Chapter 9, in which the reduced data for both rigs are evaluated against each other and previously published data.
8.1 Aerodynamic Measurements

Aerodynamic measurements have been taken at a range of flow rates extending beyond the limits specified by the minimum and maximum engine conditions, with tests carried out over an impingement jet Reynolds range of $Re = 7 \times 10^3 \rightarrow 4.4 \times 10^4$. In order to characterise the porosity and loss coefficients of the various impingement-pedestal cooling geometries for which testing has been carried out using the Sector-Rig facility fractional pressure drop measurements have been taken across the cold skin impingement jet array ($\Delta p_{\text{imp}}$), over the length of the channel from inlet to exit ($\Delta p_{\text{chan}}$) and over the entire cooling geometry from feed channel inlet to cooling channel exit ($\Delta p_{\text{tot}}$). These pressure drop measurements have been used for calculation of loss coefficient across the cold skin ($f_{\text{imp}}$) and friction factor through the pedestal array ($f_{\text{chan}}$), defined in Table 8.1. In addition to the stage loss coefficients that have been defined, the total tile loss coefficient is characterised in terms of the total pressure drop, measured across the tile from feed channel inlet to cooling channel exit and the mean impingement jet velocity, based upon the mass flow rate measured through the tile.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cold skin loss coefficient</td>
<td>$f_{\text{imp}} = \frac{\Delta p_{\text{imp}}}{2\rho U_{\text{imp}}^2}$</td>
</tr>
<tr>
<td>Pedestal channel friction factor: $f_{\text{chan}} = \frac{\Delta p_{\text{chan}}}{2\rho U_{\text{chan}}^2}$</td>
<td></td>
</tr>
<tr>
<td>Tile loss coefficient:</td>
<td>$f_{\text{tile}} = \frac{\Delta p_{\text{tot}}}{2\rho U_{\text{imp}}^2}$</td>
</tr>
</tbody>
</table>

Table 8.1: Pressure drop characteristic parameter definitions.

A portion of fluid entering the rig is allowed to bypass the cooling geometry as illustrated in Figure 8.1, in order to encourage feed flow to continue through the inlet channel. The mass flow rates quoted within this chapter account for this bypass mass flow rate with the mass flow rate passing through the rig defined by subtracting the measured bypass value from the level passing through the metering nozzle.
8.1.1 Pressure Drop Variation across Different Geometries

Pressure drop data has been captured across the Sector-Rig at the front and rear of the feed and pedestal channels, with an additional tapping at the centre of the pedestal channel as shown in Figure 6.10. Analysis of the pressure data indicated that the presence of the pedestals in the cooling channel beneath the impingement array produces an uneven distribution of suction through the jet holes. Pressure losses within the cooling channel result in impingement jet velocity increasing through each row of holes in the direction of the channel exit as illustrated in Figure 8.1. The different pedestal arrangements were observed to have a significant effect on the degree of variation in the impingement jet velocities. This is illustrated in Figure 8.2 showing the ratio of the pressure drop across the cold skin at the first and eleventh (last) jet rows. From Figure 8.2 it is immediately noticeable that there is a significantly larger streamwise variation in jet velocities for the S3a geometry than for all other geometries, with the average ratio approximately twice that of the next highest case for mass flow rates up to 0.25 kg/s. For mass flow rates greater than this a marked reduction in the difference in pressure drop at the positions is apparent; however, the variation remains significantly larger than for all other cases. It appears that the variation in jet velocity is primarily influenced by the average level of blockage present within the cooling channel with the cases of highest blockage exhibiting the greatest increase in jet velocity to the rear of the channel. This suggests that increasing amounts of flow bypass the cooling channel as the level of blockage increases. The significant increase observed for case S2a would appear to be a result of the significantly greater blockage presented by this cooling geometry (~2x that of the S2a case). Similar behaviour was reported by Andrews et al. [5] for rectangular pedestals.

In order to calculate $Re_{jet}$, a mean jet velocity was determined from pressure measurements taking at the three different points along the channel. The variation of mean impingement pressure drop with mass flow rate is shown in Figure 8.3 and it can be seen that for the majority of cases, streamwise variation in jet velocity has very little impact on the mean pressure drop across the impingement plate. The exception to this is geometry S3a, which demonstrates a change in behaviour for elevated mass flow rates ($\dot{m} > 0.25 \text{ kg/s}$). This is consistent with observations from Figure 8.2, where the rate of impingement pressure drop was seen to decrease dramatically. This indicates an increase in the effective open area of the impingement array, and thus an increase in the discharge coefficient of the impingement holes under high mass flow rate conditions for this geometry. The mean jet
discharge coefficients for the different geometries are displayed in Figure 8.4, from which it can be seen that typical $C_D$ of ~0.81 was observed for all different geometries, except geometry S3a for which a $C_D$ of ~0.75 was identified as typical. As may be expected from the observations of Figure 8.3 the discharge coefficient for the jets within the S3a geometry increase for the elevated mass flow rate tests, with values rising to levels similar to those displayed for the remaining geometries ($C_D = 0.83$).

Variation of pressure drop with mass flow rate through the pedestal channel is shown in Figure 8.5 with the ratio of channel/impingement pressure loss for the different geometries illustrated in Figure 8.6. From these figures it can be seen that changes in pedestal geometry have a significant effect on the amount of pressure lost in the cooling channel. Very similar losses are experienced by the low blockage cases, S2c and S3c featuring pedestals at quarter channel height and Shielded Impingement case S4c; with the pressure drop in the channel approximately 10% of the mean value measured across the impingement plate. Slightly higher losses are associated with the half channel height cases S2b and S3b and the densely packed Shielded Impingement case, geometry S4c. Pressure losses within the pedestal channel for these cooling geometries are ~15-20% of those occurring across the impingement array. The geometries featuring full height pedestals exhibit the greatest channel pressure losses, with $dP_{chan}/dP_{imp}$ for case S2b approximately 33%. The channel losses occurring under the S3a geometry are ~60% that of the impingement loss. This value is shown in Figure 8.6 to increase for mass flow rate greater than 0.25 kg/s as the pressure channel pressure losses continue to rise while the impingement losses appear to stagnate.

8.1.2 Cold skin and Channel Loss Coefficient Measurements

The loss coefficients across the cold skin and within the pedestal channels for the different geometries have been measured. The variation of $f_{imp}$ with mass flow rate is shown in Figure 8.7 and the corresponding values of $f_{chan}$ are shown in Figure 8.8. As reflected by the pressure drops shown in Figure 8.3, the cold skin loss coefficient is consistent for most geometries, measuring ~0.4 for all cases. The exception is geometry S3a, for which a value of ~0.45 is observed. The wider variation in losses shown in Figure 8.5 is matched by a wider spread in the channel friction factors, with values ranging from 0.3 – 0.6. From initial low mass flow rates through the channel the friction factors are seen to increase towards a steady value for all geometries not featuring pedestals of $H/D=2.5$. For these cooling geometries (S2a and S3a) $f_{chan}$ is shown to gradually decrease over the entire range
of testing. Another observation that can be made from Figure 8.8 is that consistently higher channel friction factors are observed for the $P/D=2.5$ cases than for the equivalent pedestal heights at $P/D=1.67$.

### 8.1.3 Consideration of Tile Loss Coefficient on Effective Porosity

Observed differences in the pressure drop through the channel for the different cooling geometries indicate a variable obstruction being presented to the flow by the different pedestal arrays. Considering that a consistent impingement array has been used across the different geometries, changes to the pressure loss within the channel at equivalent mass flow rate will equate to a variation in the effective porosities being presented by the different cooling geometries. When referring to effective porosity this is considered in terms of the pressure loss incurred for a particular mass flow rate through the geometry. These are the parameters on which sizing of the impingement array, designed for the Full Tile Test Facility was based, referencing EFE cooling flow conditions supplied by Rolls-Royce [6]. The impingement array tested on the Sector-Rig is representative of jet arrangements used for the Full Tile Test Facility but has been altered slightly to accommodate the pedestal array (as detailed in Chapter 6). As such it has been designed to match engine conditions close to a fractional pressure split of 80% across the cold skin with the remaining 20% occurring within the channel. As the rig porosity has been designed to match engine specified conditions, changes in the relationship between mass flow rate and total tile pressure drop will have a negative impact on cooling tile performance as both parameters are restricted by the amount of available coolant and the pressure drop by the fuel injector requirements.

Changes in porosity may be characterised using the tile loss coefficient $f_{tile}$ derived in Table 8.1 (page 173). This can also be expressed in terms of mass flow rate through the tile:

$$f_{tile} = \frac{\Delta p_{tot}}{2 \rho U_{imp}^2} \left( \frac{\rho A^2}{\rho A^2} \right) \equiv \left[ \frac{\rho A^2}{2 m^2} \right] \frac{\Delta p_{tot}}{2 m^2}$$

(8.1)

Measured values of $f_{tile}$ are displayed for all geometries in Figure 8.9 with the target value ($f_{tile} = 0.465$), based on the engine conditions also included for comparison. From Figure 8.9 it can be seen that $f_{tile}$ for geometry S3b matches the target value, corresponding to a mass flow rate and pressure drop matching the ratio from engine specified conditions. As such it is considered to have an ‘effective porosity’ equal to the target value. For the full height pedestal geometries $f_{tile}$ is shown to exceed the target value, corresponding to pressure
blockages larger than the design value and a reduced effective porosity. In line with previous observations of channel pressure drop, $f_{tile}$ for geometry S3a is significantly larger than for all other geometries. The remaining geometries display values of $f_{tile}$ below the target value. Geometries for which the displayed values of $f_{tile}$ exceed the target value indicates pressure losses in the channel are above those required to match engine conditions. In order for the total pressure drop to be reduced in line with target porosity, the number of pedestal rows must be reduced or the open area of the impingement array increased. This latter method would result in a slower velocity through the jets reducing the stage pressure loss, whilst mass flow rate through the channel would remain unaltered, pressure drop through the channel unaffected. Conversely, to increase pressure drop to match the engine settings additional pedestal rows could be added, or the open area could be reduced to produce jets of higher velocity and greater pressure drop for equivalent mass flow rate.

It is interesting to note that, in the absence of exclusively full height pedestals, increasing the overall blockage presented by the pedestal array appears to have a muted impact on the effective porosity of the tile. This is apparent in the marginal change in porosity coefficient from the lowest blockage case S2c, when either doubling the height (geometry S2b), or the number (S3c) of the pedestals. It is not until both parameters are increased that a significant increase in pressure blockage is observed, resulting in a reduced effective porosity as displayed by the larger value of $f_{tile}$ for geometry S3b. Similarly, for the Shielded Impingement geometries in which only select pedestals are at full channel height, only a marginal increase is apparent in the S4b case, for which one in four pedestals has no tip clearance. The larger $f_{tile}$ values displayed by geometry S4b when compared with the more numerous pedestals in geometry S4c is also most likely related to this. Although arrangement S4c features more than twice as many pedestals as geometry S4b, it features only half the number of full height pedestals. This is consistent with the hypothesis that the number of full height pedestals has a more significant impact than the overall number of pedestals with tip clearance. From observations on the effect of pedestal arrangement on tile effective porosity it would appear that the height of the short pedestals in the Shielded Impingement arrays could be increased without resulting in a significant reduction in porosity.
8.2 Surface Heat Transfer Measurement

Measured surface heat transfer data are presented within this section, with the captured data presented in the form of surface contour maps displaying area resolved heat transfer performance as well as plots of span and area averaged $\overline{Nu}$ for the different geometries. As with experiments performed on the full Tile-Rig, heat transfer measurements taken from the Sector-Rig are expressed in terms of $\overline{Nu}$, defined previously in Chapter 1. The different nature of the cooling geometries tested with the Sector-Rig from the more traditional impingement-effusion arrangements previously explored requires a slightly altered $\overline{Nu}$ definition, shown in equation (8.2), than that used for the Tile-Rig. The presence of the pedestal array means that the pin diameter, rather than the jet diameter, is used as the characteristic dimension to determine $\overline{Nu}$ from the measured heat transfer coefficient. Measurements from the Sector-Rig have also been made of pedestal $h$ and for consistency the same definition of $\overline{Nu}$ must be used for both.

$$Nu = \frac{h D_{pin}}{k} \tag{8.2}$$

Heat transfer data have been obtained over an impingement Reynolds number range of ~20,000 to 40,000 for all test cases, S2a-c, illustrated in Figure 6.4; S3-a-c illustrated in Figure 6.5 and S4b-c illustrated in Figure 6.6. A greater range of test points has been investigated over particular geometries in order to assess relationships between $Re$ and $\overline{Nu}$. Heat transfer measurements of geometries S2a-c and S3a-c conducted for the parametric study are compared within this section. Comparisons of surface $\overline{Nu}$ contour maps are conducted in Section 8.2.2. Comparisons of spanwise averaged $\overline{Nu}$ are conducted in Section 8.2.3. Separate comparisons are conducted for heat transfer measurements from the investigation into Shielded Impingement cooling. Comparisons of surface $\overline{Nu}$ contour maps are conducted in Section 8.2.4. Comparisons of spanwise averaged $\overline{Nu}$ are conducted in 8.2.5.

Similar to the heat transfer data presented in Chapter 7, all surface heat transfer data shown in this chapter are calculated from measurements of the first crystal reaction time, for crystal response at $T_w \approx 32^\circ$C. The large scale factor for this rig resulted in lower heat transfer coefficients than were observed on the full-Tile-Rig meaning the reaction of the 37$^\circ$C crystal was limited to near jet regions in all but the highest $Re$ cases. In order to promote reaction of the crystal across the full range of $Re$, the temperature change was varied to
increase gas temperature for lower Re cases, with $T_g$ ranging from 45-78°C across the different cases. This temperature range was restricted by the output of the mesh heater. The maximum achievable temperature increase was limited by the power supply, with more power required for greater amounts of air passing through the heater.

### 8.2.1 Mixed Bulk Temperature Correction

Due to the side feed arrangement of the Sector-Rig Test Facility additional consideration needed to be given to heat lost to the rig if measurements of $h$ at all locations are to be based on the local temperature. An analytical approach has been employed to use the local mixed bulk temperature in the calculation of $h$. This method, developed by Metzger and Larson [89] and subsequently adopted as common practice [1] [90] is detailed in Chapter 3. The analysis procedure which has been developed to calculate $h$ based on the local mixed bulk temperature ($h_{mb}$) from measurements of $h$ based on the inlet thermocouple reading ($h_e$) is detailed in Chapter 4.

The variation for the resultant $h_{mb}/h_e$ across the tested geometries is illustrated in Figure 8.10 for $Re_{jet} \approx 21000$. It can be seen that the magnitude of the mixed bulk correction varies between the different geometries by up to 12%. Figure 8.10 however, shows that the variation with streamwise position follows a similar profile for all geometries, with the value at channel exit falling between 1.29-1.35 for all but two test cases, a spread of ~5%. This variation of $h_{mb}/h_e$ between the different geometries appears largely consistent with the previously discussed parameters, with $h_{mb}/h_e$ generally increasing for the larger exposed areas. An additional factor in the magnitude of $h_{mb}/h_e$ is the measured value of $h_e$ based on inlet temperature, with more heat loss associated with greater heat transfer coefficients. This parameter can be considered accountable for the higher ratios seen for the S4 cases over similar geometries with larger values of $\bar{y}$. The variation with $Re_{jet}$ of $h_{mb}/h_e$ is illustrated in Figure 8.11 for geometry S2c, from which a clear reduction in the mixed bulk heat transfer coefficient is apparent for increasing Reynolds numbers (and accordingly mass flow rate). This trend is consistent with the theory and previously reported observations [1].

### 8.2.2 Parametric Study: Comparison of Nusselt Number Contour Maps

The distribution of $Nu$ in each of the standard cooling geometries that have been tested is discussed in this section. Maps of surface $Nu$ are displayed in Figure 8.12 and Figure 8.13 for each of the S2 and S3 test cases at $Re_{jet}$ of $\sim 20,000$ and $30,000$ respectively. Due to
slight variations in mass flow through the geometry resulting from a variable amount of flow bypassing the geometry through the restrictor plate, $Re_{jet}$ is quoted for each contour map within Figure 8.12 and Figure 8.13. Additionally, the contour map shown in Figure 8.13(e) for geometry S3b has gaps in the profile as measurements were only made across portions of the test surface for the presented $Re_{jet}$. In spite of the missing regions, sufficient data has been collected to perform useful comparisons. $Nu$ contour maps are presented for the entire length of the channel, over a region spanning a single jet pitch centred on the central jet row. Due to variations in the pedestal geometry of the different test cases and for ease of comparison, reference dimensions are given in terms of the impingement hole locations, which remains consistent for all tests.

The contour plots illustrate the influence of pedestal density on $Nu$ beneath the impingement jets. For the S2 geometries with pedestals at $P/D=2.5$, the contour distributions resemble those seen for impingement arrays over smooth passages, with largely circular $Nu$ contours observed to occur beneath the impingement jets. The smaller separation of the pedestals in the S3 geometries with pedestals at $P/D=1.67$ appears to break up the jet flow in the stagnation region, giving less defined peaks but a broader distribution. Figure 8.12 and Figure 8.13 also illustrate the impact of the developing crossflow on $Nu$, with a strong influence identified for all test cases. The contours of high $Nu$ visible beneath the first several rows of impingement holes become increasingly less defined for increasing streamwise position. Peak $Nu$ can be seen to decrease progressively, with the jet touchdown location (indicated by peak $Nu$) moving progressively further downstream of the hole location, as indicated by the $x$-axis.

The geometries for which the strongest and weakest influence of crossflow is apparent are the two full height pedestal cases S2a and S3a. For geometry S2a, the full height pedestals appear to protect the impingement jets from the crossflow as the jet contours maintain a more consistent pattern with streamwise position than all other geometries. For the S3a geometry a very different distribution is apparent, demonstrating the least defined jet contours of all tested geometries; additionally, the magnitudes of $Nu$ increase with streamwise position although a minor reduction in peak $Nu$ is apparent. This is likely a result of the increasing impingement jet velocity with jet location (previously discussed in Section 8.1.1, page 174) combining with the increasing crossflow streamwise position, increasing the channel $Re$. Although geometries S2a and S3a exhibit more consistent streamwise $Nu$ than with the other geometries, this can be seen to be offset by lower $Nu$ in
the inlet region of the channel. Magnitudes of $Nu$ for geometry S2a towards the exit of the cooling channel appear larger than for all other geometries, but the reduced magnitude displayed in the inlet region appears to indicate a weaker performance than for the equivalent density arrays S2b and S2c. For geometry S3a similarly low $Nu$ values are displayed at channel inlet, but no obvious improvement can be seen towards channel exit. The highest magnitude $Nu$ values are displayed by geometries S2c and S3c.

In addition to the observations regarding the differences in the impingement cooling performance, increased $Nu$ can be seen to occur due to interaction of the jets and crossflow with the pedestals. One area that such interaction may be identified is in lines of elevated $Nu$ in the wake immediately downstream of the pedestals aligned midway between jets (at $y/D \pm 0.5$) and may be most easily observed in Figure 8.13(d) for the S3a geometry. The magnitude of this elevated pedestal cooling appears to increase with crossflow. The increase in $Nu$ observed in the wake from the pedestal is further seen to increase with streamwise position. These pin jets can be seen to some extent in the contour maps for all geometries.

### 8.2.3 Parametric Study: Comparison of Span Averaged Nusselt Numbers

For quantitative comparison of heat transfer performance between geometries, spanwise-averaged Nusselt numbers ($\overline{Nu}$) have been taken between $y/D \pm 0.5$ across the entire tile length. $\overline{Nu}$ are compared for similar $Re_{jet}$, consistent with those of the $Nu$ contour maps displayed in Figure 8.12 and Figure 8.13. Comparisons of $\overline{Nu}$ for geometries S2a-c are presented at $Re_{jet} \sim 22000$ in Figure 8.14 and $Re_{jet} \sim 33000$ in Figure 8.15; comparisons of for geometries S3a-c are presented at $Re_{jet} \sim 22000$ in Figure 8.16 and $Re \sim 33000$ in Figure 8.17.

Figure 8.14 and Figure 8.15 illustrating $\overline{Nu}$ for the S2 geometries, indicate the strongest heat transfer performance is displayed by geometry S2c, with the weakest apparent for geometry S3a. Similar distributions of $\overline{Nu}$ are displayed by geometry S2b and S2c, featuring pedestals of $H/D = 1.25$ and 0.63 respectively, but with $\overline{Nu}$ typically higher for geometry S2c. For all geometries illustrated in Figure 8.14 and Figure 8.15, $\overline{Nu}$ the minima and maxima displayed by the span-averaged profiles can be seen to initially increase over a number of jet rows that varies with geometry (jet row four for geometries S2a and S2b, jet row five for geometry S2c) before a gradual decline in the maxima of $\overline{Nu}$ occurs from jet row five-six onwards. $\overline{Nu}$ drop to similar levels, with geometry S2c maintaining slightly larger magnitudes than S2b. A more gradual streamwise reduction of $\overline{Nu}$ for Geometry S2a case
can be seen, resulting in magnitudes of $\overline{Nu}$ from jet row ten onwards, exceeding those for geometries S2b and S2c cases. This is in contrast to significantly lower $\overline{Nu}$ values displayed from channel inlet to midpoint. While the maxima of $\overline{Nu}$ have been shown to decline with increasing streamwise position, typically from jet row five onwards; streamwise variation of the minima of $\overline{Nu}$ does not follow this pattern. It can be seen in Figure 8.14 and Figure 8.15 that for geometries S2b and S2c, the minima of $\overline{Nu}$ start to increase from jet row eight onwards. This is most likely a result of a stronger influence of the pedestals as crossflow increases. For geometry S2a, minima of $\overline{Nu}$ can be seen to increase along the full length of the channel. As with the observations of peak $\overline{Nu}$, the minima rise from values lower than for geometries S2b and S2c, but exceeds the values for these geometries from jet row nine onwards. Considering the strengthened performance at channel exit displayed by geometry S2a alongside the typically smaller peak $\overline{Nu}$ values, results in a significantly more uniform, but weaker streamwise distribution of $\overline{Nu}$ than for cases S2b and S2c. Of these geometries, S2c consistently exhibits the strongest heat transfer performance. All three geometries display the same general trend in streamwise variation of $\overline{Nu}$, characterised by an increase in $\overline{Nu}$ from channel inlet to typically jet row four with a subsequent reduction in $\overline{Nu}$ to channel outlet.

Span averaged Nusselt numbers for the S3 geometries are shown at Re~22000 in Figure 8.16 and Re~33000 in Figure 8.17. These figures illustrate an increased influence of the pedestals on heat transfer performance. Streamwise variation of $\overline{Nu}$ for geometry S3c can be seen to closely resemble the profiles displayed for geometries S2b and S2c shown in Figure 8.15. Once again the maxima of $\overline{Nu}$ can be seen to initially increase over the first four jet rows before gradually decreasing towards channel outlet. The minima of $\overline{Nu}$ between peaks exhibit slightly different streamwise variations in $\overline{Nu}$, initially increasing before reducing after the fourth jet row, after which the minima of $\overline{Nu}$ can be seen to increase steadily in the direction of channel outlet. A large drop in $\overline{Nu}$ can be seen in Figure 8.16 and Figure 8.17 following the fourth jet row for all S3 geometries. The contour plots in Figure 8.12 and Figure 8.13 suggest that this is the result of pedestals positioned directly between jets in the fourth and fifth rows. These appear to obstruct the wall jet from entering this region and reduce the local heat transfer performance accordingly. Although they can only be evaluated for the positions where heat transfer data were gathered, streamwise variation of $\overline{Nu}$ for geometry S3b appears similar to that displayed by S3c, but with smaller typical magnitudes. The relative rates of change of the maxima and minima of $\overline{Nu}$ with
streamwise position for geometries indicates a general trend in streamwise variation of \( \overline{Nu} \), consistent with those displayed in Figure 8.16 and Figure 8.17 for the S2 geometries. This is characterised by an increase in \( \overline{Nu} \) from channel inlet to jet row four, with a subsequent reduction in \( \overline{Nu} \) to channel outlet. Although similar streamwise variation in \( \overline{Nu} \) is exhibited by geometry S3b over the majority of the channel, \( \overline{Nu} \) beneath the fifth jet row significantly lower than the equivalent row for geometry S3c. Figure 8.12(e) suggests that this is likely due to deflection of the jet by the crossflow causing it to foul upon the downstream pedestal.

Cooling geometry S3a exhibits a streamwise variation of \( \overline{Nu} \). (shown at \( Re \sim 22000 \) in Figure 8.16 and \( Re \sim 33000 \) in Figure 8.17) that is largely inconsistent with those seen for the other tested geometries. The effects of crossflow appear to affect heat transfer performance from earlier within the channel, with a decrease in peak \( \overline{Nu} \) seen after the third jet row, as opposed to the fourth or fifth row seen with the other geometries. Despite this, overall \( \overline{Nu} \) can be seen to increase towards the channel outlet, with the minima of \( \overline{Nu} \) observed to increase across the entire channel length, rather than demonstrating the oscillation seen for the other geometries. In addition to this, the maxima of \( \overline{Nu} \) increase following the sixth jet row towards channel outlet. As shown in Figure 8.16 and Figure 8.17, the maxima of \( \overline{Nu} \) for geometry S3a at channel outlet are approaching the magnitudes seen at the second and third jet rows. Although the peak values are smaller (~90% of maximum), the increased minima of \( \overline{Nu} \) indicate a net improvement in \( \overline{Nu} \) towards channel outlet. Given the observations from Figure 8.2 regarding the large increase in jet velocity through jet row eleven for the S3a geometry, an increasing trend in \( \overline{Nu} \) is unsurprising and consistent with the observations of Andrews et al [5]. The presence of shallower, but broader peaks in the span-averaged plots would suggest that the improved performance is only partially the result of increased local \( Re_{jet} \). This is supported by the contour plots for the S3a geometry shown in Figure 8.12(d) and Figure 8.13(d). It is indicated that although the contours beneath the jets approaching channel outlet exhibit lower \( Nu \) values than at jet rows two and three, their influence extends further, with magnitudes of \( Nu \) away from the jet much higher than seen at channel inlet. This indicates that the increasing \( \overline{Nu} \) with streamwise position results from increased interaction between the impingement jets and the pedestals.
8.2.4 Shielded Impingement: Comparison of Nusselt Number Contour Maps

Nu contour distributions for the two Shielded Impingement geometries investigated (S4b: P/D = 2.5 and S4c: P/D = 1.67) are shown in Figure 8.18 at $Re_{jet}$ of ~20,000 and ~30,000. The presented contour distributions appear to indicate differences in the behaviour of the two investigated Shielded Impingement geometries, with Nu bearing closer resemblance to those seen for the baseline arrays of equal pedestal density, previously shown in Figure 8.12 and Figure 8.13. It can be seen however, that a common trait between the contour distributions for the Shielded Impingement geometries exists. High Nu magnitudes are maintained along the entire length of the channel, showing less decrease in heat transfer performance than for the unshielded geometries and consistently higher magnitudes of Nu than for the full height geometries.

The distribution of Nu for the S4b geometry appears to be the least affected by crossflow, although the negative influence of crossflow on the jets may still be detected through the deformation of the circular distribution and the reduction in peak Nu values towards channel outlet. It can be seen in Figure 8.18 however, that this occurs to a much smaller degree than for all other test cases, including the full height geometries (S2a and S3a). Unlike the contour distributions for the S2a and S3a geometries however, the good downstream performance does not occur at the expense of reduced cooling towards the channel inlet.

The distribution of Nu for the S4c geometry appears to offer similar inlet cooling performance to that seen in the S4b geometry, with comparable magnitudes of Nu. As observed for the other $P/D = 1.67$ pedestal geometries (S3a-c), the increased density of pins can be seen to alter the contour profiles as the jet flow must be accommodated between the pedestals, resulting in a breakdown of the defined contours exhibited for geometries S2a-c and S4b. The crossflow appears to have an increased impact on Nu magnitude at channel exit for geometry S4c than for S4b, although the effect appears to be smaller than that observed for the S3 geometries. This is likely a result of the alignment of the jets and pedestals resulting in only every second jet being shielded. It should again be noted, that despite this, Nu appears to remain higher than for the comparable S3 geometries, but not to the same degree as is shown for the S4b geometry over the comparable S2 geometries.
8.2.5 Shielded Impingement: Comparison of Span Averaged Nusselt Numbers

In order to evaluate the heat transfer performance of the Shielded Impingement arrays and compare performance with the baseline S2 and S3 arrays discussed in the previous section, Span-average Nusselt numbers are compared for the S4 geometries. $\overline{Nu}$ for the S4b and S4c geometries are compared at $Re_{jet} \sim 22000$ in Figure 8.19 and at $Re_{jet} \sim 33000$ in Figure 8.20. For comparison of heat transfer performance of the Shielded Impingement geometries against the baseline configurations, $\overline{Nu}$ for geometry S4b is compared against pedestal geometries S2a and S2c in Figure 8.21 and $\overline{Nu}$ for geometry S4c is compared against the equivalent density pedestal geometries, S3a and S3c in Figure 8.22. The comparison is made against the ‘a’ and ‘c’ geometries as these feature pedestals of heights consistent with the Shielded Impingement arrays.

The span comparisons of $\overline{Nu}$ illustrated in Figure 8.19 and Figure 8.20 illustrate very similar values of $\overline{Nu}$ for the two cases over the first half of the channel. Larger values of $\overline{Nu}$ can be typically seen for the S4b case however this may be attributed in part to the higher $Re_{jet}$ seen for this geometry in both presented figures. Due to the increased shielding, the S4b geometry displays increasingly superior cooling performance towards channel outlet, this may be expected because the presence of higher $Nu$ for the S4b geometry in the downstream region appear to be most pronounced beneath the odd numbered impingement jet rows for which the shielding pedestals are absent in the S4c geometry. Although to a lesser degree, $Nu$ beneath the even numbered downstream jets are also seen to be larger for the S4b geometry. This suggests that the consistent presence of shielding pins in consecutive rows is beneficial in protecting the jets from the developing crossflow.

Span averaged heat transfer performance of the Shielded Impingement geometries have been compared against the equivalent full and quarter height pedestal arrays. The relative performance of the different cooling tiles are illustrated at $Re_{jet} \sim 33000$ for geometry S4b, alongside geometries S2a and S2c in Figure 8.21 and at $Re_{jet} \sim 33000$ for geometry S4c, alongside geometries S3a and S3c in Figure 8.22. Examination of these streamwise profiles of $\overline{Nu}$ provides confirmation of the improved heat transfer performance for the S4b and S4c geometries over the equivalent S2 and S3 configurations. This is consistent with preliminary conclusions from the contour maps in Figure 8.12; Figure 8.13 and Figure 8.18. The poorer heat transfer performance of the arrays exclusively featuring full height pedestals has been
previously discussed and both the S4b and S4c geometries have been shown to display
greater magnitudes of $\overline{Nu}$ than the corresponding full height pedestal geometries over the
entire channel length. When comparing performance against the short pedestals, S2c and S3c
however, the improvement only becomes apparent in the latter half of the channel. It is
suggested that this occurs when the crossflow has increased to a level that disturbs the
unshielded jets in the short pedestal arrays.

As has been previously noted, $\overline{Nu}$ for the S2c and S3c geometries decrease significantly
with streamwise position over the latter half of the channel and it is in this region that the
benefits of the shielding pedestals can be identified. Examination of $\overline{Nu}$ profiles for the
Shielded Impingement geometries indicates that this downstream reduction in cooling
performance occurs to a reduced degree. This is illustrated in Figure 8.21, where $\overline{Nu}$ for the
S4b geometry are consistently higher than for geometry S2c from jet row five onwards,
following the steep drop in magnitude of $\overline{Nu}$ displayed by geometry S2c. This behaviour is
consistent for different $Re_{jet}$ cases; however, for increasing $Re_{jet}$ the reduction in $\overline{Nu}$ for the
S2c case was seen to occur further downstream, following jet row six, whilst little variation
was apparent for geometry S4b. The improvement in heat transfer performance of the S4c
Shielded Impingement geometry over the equivalent short pedestal array has been observed
to be less pronounced than that seen for the S4b case; behaviour that has been primarily
attributed to the omitted shielding pedestals. This conclusion has been drawn as the largest
apparent increases in $Nu$ occur beneath the jet rows in the second half of the channel that are
preceded by shielding pins (rows six, eight and ten).

8.2.6 Comparisons of Surface Heat Transfer Performance

In order to provide an assessment of the relative surface heat transfer performance of
the eight different geometries that have been tested, area averaged Nusselt numbers ($\overline{Nu}$)
have been recorded over the full range of tests. $\overline{Nu}$ is plotted against $Re_{jet}$ in Figure 8.23 and
also against the total tile pressure drop in Figure 8.24. As was discussed earlier in this
chapter and previously in Chapter 6, $Re_{jet}$ required to match engine conditions varies for the
different geometries due to changes to the effective porosity of the cooling tile. As such, the
different methods of comparing area averaged heat transfer data have been used to enable a
comparison of data at equivalent mass flow rate and at equivalent pressure drop.
Examination of the relative performance of the cooling geometries featuring pedestal arrays of equivalent $P/D$ spacing indicates that that the surface heat transfer performance of the different geometries may be ranked according to the height of the pedestals featured within the cooling array. For both the $P/D = 1.67$ and $P/D = 2.5$ cases, the area averaged Nusselt numbers for the different geometries increase with total pressure drop, as pedestal height decreases for the uniform $H/D$ cases. A further increase in $\overline{Nu}$ is seen for the equivalent S4 geometries. The Shielded Impingement geometries exhibit the largest averaged Nusselt numbers; with superior performance apparent for the S4b case over that of the S4c geometry. The effect of the pedestal height on heat transfer performance can also be seen to be more pronounced for the more densely spaced pedestal arrays, with a much wider range of $\overline{Nu}$ apparent for the different geometries in the $P/D = 1.67$ case than for $P/D = 2.5$. As has been identified for the Shielded Impingement configurations, the dense pedestal array geometries can be seen to provide weaker cooling performance than the wider spaced pedestal geometries, with $\overline{Nu}$ for the S2 cases observed to be consistently higher than for the equivalent S3 geometries. The degree of variation in performance between the different array densities appears to be more pronounced for taller pedestals. The same behaviour is also apparent in Figure 8.23 when comparing performance based on $Re_{jet}$; however the larger pressure losses associated with the taller pedestals can be seen to reduce the differences in performance between the geometries.

Observations of all the surface heat transfer data presented within this chapter appear to identify that the most significant heat transfer augmentation results from the impingement jet array, with the pedestals complementing rather than dictating performance. This is particularly evident in the weakened heat transfer performance of the full height pedestal arrays and with the general reduction in $\overline{Nu}$ for increasing pedestal height and density illustrated in Figure 8.24. An increase in the height or density of the pedestal array has been shown in Figure 8.5 to increase the channel pressure drop and accordingly reduce the jet Reynolds number for a given total tile pressure loss. For the geometries with high levels of pedestal blockage the increased channel pressure losses result in a reduced $Re_{jet}$. This trend would suggest that any associated increase in surface heat transfer performance resulting from the increased channel blockage is insufficient to account for the reduction in jet velocity. The improved heat transfer performance of the S4b Shielded Impingement geometry provides exception to this, with superior heat transfer performance being displayed despite exhibiting proportional channel pressure losses of equivalent or greater value to all
but the full height pedestal geometries. The larger magnitude of $\overline{\text{Nu}}$ alongside observations of the surface contour plots for the S4 arrays indicate that the shielding pedestals succeed in protecting the impingement jets such that downstream jet performance is significantly improved over the other tested geometries. The degree to which this is achieved is highlighted particularly clearly in Figure 8.21.

These combined observations would suggest that while interaction of the pedestals and the developing crossflow can improve heat transfer performance, a net improvement in heat transfer performance requires that this not be at the expense of a reduction in jet strength. By preserving the impingement jet performance in the downstream region the Shielded Impingement geometries appear to have achieved a successful interaction of impingement and pedestal cooling.

8.3 Pedestal Heat Transfer Measurement

The overall cooling performance of the different tiles is a function of heat transfer between the coolant and the exposed surface area of the tile. In addition to promoting increased turbulence within the cooling channel, pedestals also act as heat exchangers with the potential to significantly improve heat transfer performance through the increased surface area they present to the coolant. As a result, the heat transfer performance of the pedestal arrays is a function of both the surface (liner tile) heat transfer coefficient and that of the pedestals themselves. The pedestal arrays used for the different cooling geometries present widely different exposed surface areas to the coolant flow. Across the range of tile concepts investigated the total wetted area, accounting for endwalls and pedestals, ranges from below $0.5m^2$ for geometry S2c, to a value approaching $1.2m^2$ for geometry S3a, 80% of which is accounted for by the pedestals. This highlights the importance of accounting for the contribution of pedestal heat transfer in assessing the overall cooling potential of the different tile concepts. During the course of testing, efforts have been made to evaluate the heat transfer performance of the pedestals within the arrays in order to supplement the reported surface data. Pedestal Nusselt numbers have been captured by positioning an aluminium pin containing k-type thermocouples at its centre. The process by which the heat transfer coefficient is calculated has been discussed in Chapter 6.
8.3.1 Pedestal Nusselt Numbers

Heat transfer data have been measured for pedestals over a jet Reynolds range of $Re = 10 - 45 \times 10^3$ for a variety of streamwise and spanwise locations within the S3a and S3b cooling geometries. The spanwise locations at which measurements have been taken, illustrated in Figure 8.25, are centred about the second of five jets in the row, with the number of pedestal locations at which data was gathered alternating between four (when pedestals are staggered to the jet array) and five (when pedestals are in line with the jet array). Pedestal heat transfer measurements have been taken over a range of streamwise locations (illustrated in Figure 8.25) grouped into sets of three rows, with each group of three starting immediately downstream of an impingement jet. The measured pedestal $Nu$ are illustrated in Figure 8.26 to Figure 8.31. $Nu$ for rows 1-3 of geometry S3a are shown in Figure 8.26; $Nu$ for rows 13-15 of geometry S3a are shown in Figure 8.27 and $Nu$ for rows 25-27 of geometry S3a are shown in Figure 8.28. Similar illustrations of $Nu$ are shown for rows 7-9 of geometry S3b in Figure 8.29; rows 16-18 of geometry S3b in Figure 8.30 and rows 25-27 of geometry S3b in Figure 8.31. The figures displaying $Nu$ illustrate that the location of a pedestal has a marked effect on its heat transfer performance.

Sensible spanwise variations of $Nu$ are displayed for pedestals within a particular row considering the proximity of the individual pedestals to the impingement jet. As might be expected, the pedestals situated closest to the jets typically display $Nu$ larger than those situated further from the hole; however, the degree to which this behaviour occurs varies significantly across the range of pedestal rows for which data have been gathered. For the inline pedestal rows significantly larger values of $Nu$ are displayed for the central pedestal. The remaining four data points within these rows exhibit broadly equal $Nu$ which may be expected as each of these pedestals are equidistant from jet holes. Similar behaviour is apparent for the offset pedestal rows, with $Nu$ of the pedestals at $y/D \pm 0.17$ typically greater than for the pedestals midway between jets at $y/d \pm 0.5$.

The level of influence that the spanwise distance of the pedestals from the impingement jets has on $Nu$ can be seen to be much more pronounced for the first row downstream of the jet, with much smaller variations of $Nu$ displayed for the different pedestals within the third row downstream of the jets, as can be seen in Figure 8.26 to Figure 8.31 by the consistently flatter spanwise profiles that are displayed in all tested cases for the third (blue) row. Note that by referring to the ‘third’ row, this relates to downstream location from the previous...
impingement jet and refers to every third row display (rows 3, 9, 15, 18, 27) with the same inference for the ‘first’ and ‘second’ rows. The observed difference in the spanwise profile of the first and third downstream rows indicates that pedestal heat transfer performance is more significantly affected by the upstream jets than those downstream. The $x/D$ spacing of the pedestal array is such that the third row downstream of each impingement jet is situated just upstream of the following one. As $Nu$ for the pedestals within the third row are broadly uniform, it can be inferred that the crossflow within the channel results in the impingement jets affecting primarily the pedestals in the direction of the channel exit. Also by comparing the data displayed for the different geometries, it can be observed that the level of deviation between $Nu$ for the different pedestals are typically more pronounced for geometry S3b than for geometry S3a. An additional observation that can be made from the illustrated pedestal data is that $Nu$ for the negative $y/D$ positions are consistently lower than for the equivalent positive positions. This would appear to indicate that marginally weaker heat transfer performance occurs towards the edge of the Sector-Rig tile than at the centre.

From Figure 8.26 to Figure 8.31 a changing level of influence of the local impingement jets on the pedestal $Nu$ can be identified through the variable profile displayed for the second row downstream of the jets. For both the S3a and S3b geometries, the spanwise variation of $Nu$ for the different pedestals can be seen to be greatly reduced, or absent in the data captured over the central pedestal rows (13-18). This can be seen in Figure 8.27 for geometry S3a and in Figure 8.30 for geometry S3b, with the typical spanwise variation seen to return for the further downstream pedestal location of rows 25-27 (Figure 8.28 for geometry S3a and Figure 8.31 for geometry S3b). This is likely a result of the diversion of the impingement jet by the developing crossflow and as the impingement location changes with streamwise location so too does the nature of the interaction between the jets and the pedestals.

8.3.2 Row-Averaged Pedestal Nusselt Variations

The Nusselt numbers displayed for the different pedestals in Figure 8.26 to Figure 8.31 indicate a streamwise variation in the magnitude of pedestal $Nu$ that seems particularly pronounced for the S3a geometry. In order to assess the streamwise variation of pedestal heat transfer performance, row averaged pedestal Nusselt numbers ($\overline{Nu}$) for geometries S3a and S3b are displayed in Figure 8.32. From this figure $\overline{Nu}$ can be seen to increase significantly with streamwise position for the S3a geometry, with $\overline{Nu}$ for pedestal rows 25-27 of order twice the magnitude of those in row 1. A much smaller streamwise variation of $Nu$ is
apparent for geometry S3b, with only a minor increase for certain pedestal rows within geometry S3b appearing to indicate a much smaller upward trend.

This variation in $\langle Nu \rangle$ from the front to the rear of the cooling channel is likely a result of the uneven jet velocity distribution throughout the array that results from the pressure losses within the pedestal channel. The pressure drop across the cold skin has been shown in Figure 8.2 to be $\sim 3.5$ greater at the rear of the channel than at the inlet, corresponding to a jet Reynolds number for the final row impingement $\sim 1.9$ that of the first row. Correspondingly, the channel pressure drop for geometry S3b is much smaller, with the final row cold skin pressure drop $\sim 1.5x$ that of the first row. This level of pressure drop variation, which is much more representative of the typical levels exhibited by the remaining pedestal arrays, corresponds to the jet Reynolds number increasing by a factor of $\sim 1.2$ from the first to the final jet row. These levels of variation appear to follow with the trends of streamwise $\langle Nu \rangle$ variation displayed for the two geometries in Figure 8.32 were the data continued to the end of the channel. The similarity in the level of channel pressure drop for the short pedestal ($H/D \leq 1.25$) and Shielded Impingement arrays mean that the variations in pedestal $\langle Nu \rangle$ with streamwise position will likely be similar for these cases. It was considered that the different streamwise profiles of $\langle Nu \rangle$ could be attributed to the gradual increase of crossflow from channel inlet to channel outlet; however, this would not explain the initial lower magnitudes of $\langle Nu \rangle$ displayed by the full-height pedestals, or the relatively stable streamwise values of $\langle Nu \rangle$ observed for the S3b array. As such the pedestal heat transfer performance is thought to be primarily affected by the wall jet developing from the local impingement jet, with the larger $\langle Nu \rangle$ displayed for pedestals in the S3a array resulting from the increased flow velocity past the pins due to the smaller open area.

In order to investigate the Reynolds sensitivity of pedestal cooling performance, the row averaged pedestal Nusselt numbers are plotted against jet Reynolds number for geometry S3a in Figure 8.33 and for geometry S3b in Figure 8.34, with the heat transfer levels for the individual pedestal rows highlighted within these figures. These data are repeated in Figure 8.35 with $\langle Nu \rangle$ for both tested geometries plotted on the same figure to allow comparison of performance. In line with the comparisons made for the surface heat transfer performance, pedestal $\langle Nu \rangle$ are plotted in Figure 8.36 against total tile pressure drop in order to allow comparisons of heat transfer performance at equivalent mass flow rate and pressure drop. Lines of $\langle Nu \rangle$ for the different geometries plotted alongside the data points on Figure 8.35 indicate that when considered against jet Reynolds number, the area averaged
pedestal heat transfer coefficient is of order 10-20% larger for the full height pedestals than for the half height equivalents. When plotted against a constraint of total pressure loss across the whole geometry, as shown in Figure 8.36, very similar heat transfer characteristics may be identified, with marginally improved performance displayed by the half height pedestals. This marginal improvement may be an artefact of the pedestals tested. The full height pedestal data have been gathered from earlier rows than for the half height case, and due to the gradual increase in $\overline{Nu}$ with streamwise position displayed, to different extents, by both geometries this would provide an exaggerated assessment of the performance of the S3b pedestals.

The displayed data would appear to indicate that for a given mass flow rate of coolant, superior heat transfer coefficient results from the full height pedestals, due to the increased velocity between the pedestals. The resulting additional pressure loss is such that in the case of half channel-height pedestals, a greater mass flow rate may be passed through the jet array for an equivalent pressure-loss. Under these conditions, Nusselt numbers of equivalent magnitude result for both geometries, but superior cooling would still result from the full height pedestals due to the surface area of this array being close to twice that of the half height pedestal array. Heat transfer data have not been captured for pedestals in the S2 geometries; however, the apparent influence of the wall jet on pedestal heat transfer performance would suggest that this relationship will vary for the different $P/D$ cases. Further testing would be beneficial to further characterise the heat transfer performance of the pedestals in different geometries.

### 8.3.3 Comparison against Surface Heat Transfer Performance

For comparison of pedestal and surface heat transfer performance streamwise variations of pedestal $\overline{Nu}_p$ and wall $\overline{Nu}_w$ have been compared in Figure 8.37. Variation of $\overline{Nu}$ with $Re$ was compared for the same geometries in Figure 8.38. Variations in streamwise heat transfer performance for geometry S3a indicate magnitudes of $\overline{Nu}_p$ and $\overline{Nu}_w$ that are almost equal over the length of the channel. The discrete values of $\overline{Nu}_p$ appear to track $\overline{Nu}_w$ despite the steep gradients displayed by the continuous profile. From Figure 8.37 it would appear that magnitudes of $\overline{Nu}_p$ and $\overline{Nu}_w$ are strongly connected. For geometry S3b, it would appear that although $\overline{Nu}_p$ are of smaller magnitude than $\overline{Nu}_w$, similar tracking of the surface heat transfer performance is indicated. For both measured geometries, a streamwise increase of $\overline{Nu}_p$ is apparent in comparison to $\overline{Nu}_w$. The variation of $\overline{Nu}$ with $Re$ is compared in
Figure 8.38. \(\overline{Nu}\) values for geometry S3a are almost equal for \(Re_{jet} < 2.5 \times 10^4\). Deviation is seen as pressure characteristics for geometry S3a change at high mass flow rate as illustrated in Figure 8.6. For \(Re_{jet} > 3 \times 10^4\) increased proportional pedestal channel pressure drop and velocity appear to be the reason for increased pedestal \(\overline{Nu}\), but with decrease in wall \(\overline{Nu}\). For geometry S3b, wall values of \(\overline{Nu}\) are higher than for the pedestals situated within that cooling tile, and also \(\overline{Nu}\) for the walls and pedestals of geometry S3a.

Measurements of relative heat transfer performance indicate a change in characteristics from typical pedestal array behaviour. Reported data has consistently indicated higher \(Nu\) for pedestals than for the endwall [16-19]. For the literature surveyed the level of increase varied, but the presence of larger \(Nu\) for the pedestals was consistent. Due to change in feed arrangement it would that distributed impingement is characterised by a relative increase in surface \(Nu\) compared with that for the pedestals. Whether this is a benefit or of detriment to overall cooling performance needs further information regarding the performance of the different arrays relative to traditional impingement cooling. The effect of pressure drop is also important as it will determine the number of rows that may be used. While comparisons have been conducted for equal jet Reynolds numbers, the pressured drop across the entire tile for geometry S3a is significantly larger than for S3b, as characterised by \(f_{tile}\) (Figure 8.9).

### 8.4 Uncertainty Analysis in Heat Transfer Measurements

The uncertainty of the Nusselt numbers within this chapter have been calculated using the Kline-McClintock method discussed previously in Chapter 3. The uncertainties in reported values for both the surface and pedestal Nusselt numbers are calculated using equation (8.3) [94].

\[
\begin{align*}
{u_R} = & \sqrt{\left(\frac{\partial R}{\partial x_1} u_1\right)^2 + \left(\frac{\partial R}{\partial x_2} u_2\right)^2 + \left(\frac{\partial R}{\partial x_3} u_3\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} u_n\right)^2} \\
& (8.3)
\end{align*}
\]

#### 8.4.1 Uncertainty in Surface Heat Transfer Coefficient

The solution equation used to measure the surface heat transfer coefficient, equation (4.5) (page 73), introduces three sources of error into the measured values of \(h\). These sources of uncertainty have been previously defined in Chapter 7 as the measurements of surface and fluid temperature (\(T_o\), \(T_w\) and \(T_{aw}\)); the thermophysical properties of the heat transfer surface, \(\kappa = \sqrt{\frac{\rho c k}{k}}\) and the time, \(t\) at which the crystal reaction is observed.
The same data acquisition equipment was used for heat transfer measurements performed on the Sector-Rig as were used for the Tile-Rig experiments. This involved the same calibration procedures being followed for the thermocouples and the liquid crystals as were previously outlined for the Tile-Rig. As such, uncertainties for the temperature terms used to measure $h$ are equal to those quoted in Chapter 7 for the Tile-Rig, with $u_{(T_{aw}-T_i)} = u_{(T_w-T_i)} = \pm 0.5^\circ C$. To allow optical access of the liquid crystal colourplay, the heat transfer surface of the rig has once again been constructed of Perspex. The thermophysical properties for the Perspex remain unchanged, with the same reported values of density ($\rho = 1190 \text{ kg/m}^3$); specific heat capacity ($c = 1500 \text{ J/kg.K}$) and thermal conductivity ($k = 0.18 \text{ W/m.K}$) \[103-104\] giving a value of $\kappa = 567 \text{ W.s}^{0.5}/\text{K.m}^2$. Uncertainties in the precise values of the constituents of $\kappa$ result in the same standard error as previously quoted of $u_{\kappa} = \pm 28.3 \text{ W.s}^{0.5}/\text{K.m}^2$. The same camera has been used to record the liquid crystal colour transitions for the Sector-Rig tests as was employed in the full-Tile-Rig experiments, with an LED once again positioned in the camera view as a trigger mechanism to allow $t=0$ to be pinpointed. As such the standard error in $t$ is equivalent to the frame rate of the camera, giving $u_t = \pm 0.04s$.

Applying these sources of uncertainty to equation (8.3) allows equation (8.4) to be derived. This may be used to calculate the uncertainty in the measured value of $h$ based upon the accuracies of the individual measurements.

\[
u_h = \sqrt{\left(\frac{\partial h}{\partial t} u_t\right)^2 + \left(\frac{\partial h}{\partial \kappa} u_\kappa\right)^2 + \left(\frac{\partial h}{\partial (T_{aw}-T_i)} u_{(T_{aw}-T_i)}\right)^2 + \left(\frac{\partial h}{\partial (T_w-T_i)} u_{(T_w-T_i)}\right)^2} \tag{8.4}
\]

Using Equation (8.4) the measured heat transfer coefficients have been shown to have uncertainty of $u_h \approx 5.7\%$. Despite the measurements being taken using the same technique with equipment calibrated to equal precision, the larger scale factor of the Sector-Rig (x17, as opposed to x7.5) results in maximum heat transfer coefficients significantly lower than those for the Tile-Rig. Due to typically longer crystal reactions, the relative uncertainty in the time-step and the temperature measurements is significantly smaller resulting in the reported lower level of uncertainty. As was previously reported, there is still a variation of $u_h$ with $h$ (shown in Figure 8.39); however, the variation is minimal over the practical range of heat transfer coefficients encountered.
8.4.2 Uncertainty in Pedestal Heat Transfer Coefficient

In addition to calculation of uncertainty in the quoted surface heat transfer coefficients, the uncertainty of the pedestal heat transfer coefficients are also calculated using the Kline McClintock method. The different calculation method used to determine pedestal heat transfer results in different parameters contributing to the uncertainty in the reported values of \( h \). Referring back to the solution equation through which the measured data were used to calculate \( h \) the sources of uncertainty for use in equation (8.3) may be determined.

\[
\dot{q} = m_{pin}c_{p, pin} \frac{dT_{pin}}{dt} \tag{8.5}
\]

\[
h = \frac{\dot{q}}{A_{pin}(T_g - T_{pin})} \tag{8.6}
\]

Sources of uncertainty that have been identified from equations (8.5) and (8.6) are the pedestal and gas temperature measurements; the material properties and the time step. The area and volume of each of the aluminium pedestals are set and are considered to have negligible uncertainty.

- **pedestal and fluid temperatures \((T_g, T_{pin})\)**

  Temperature of the pedestal and of the fluid are both measured using k-type thermocouples that have been calibrated against a platinum resistance thermometer to remove bias error with uncertainties equivalent to those identified for the surface heat transfer coefficient. As such, the uncertainties for both the temperature terms have a level of error based on the accuracy of the thermocouples, with standard errors for both equal at \( u(T_g - T_{pin}) = u_{dT_{pin}} = \pm 0.5°C \).

- **Thermophysical properties of the heat transfer surface, \( c_p \) and \( \rho \)**

  The pedestals are manufactured from aluminium, which has better defined physical characteristics than Perspex, due to the variation in Perspex properties based upon its manufacture and its high level of thermal expansion. As has been stated for the surface heat transfer uncertainty, the material properties are not subject to random uncertainty; however, due to potential inaccuracies in the quoted values there will be a degree of bias introduced by temperature changes and from differences between the stated and actual values. In order to account for this, standard errors of \( u_\rho = \pm 50 \text{ kg/m}^3 \) and \( u_{c_p} = \pm 20/\text{kg.K} \) are used in the uncertainty analysis.
• Time step, $dt$

The lack of a visual signal for the measurement of the pedestal heat transfer reduces the sources of uncertainty in the time step to the precision of the measurement and the determination of the point at which the mesh heater is activated. Use of the LED provides a signal trigger indicating to the data acquisition equipment the time at which the heater is activated. This essentially reduces the error in $dt$ to the frequency of the acquisition, giving $u_c = \pm 0.01s$. Applying these sources of uncertainty to equation (8.3) allows equation (8.7) to be derived. This may be used to calculate the uncertainty in the measured value of $h$ based upon the accuracies of the individual measurements.

$$u_h = \sqrt{\left(\frac{\partial h}{\partial c_p} u_{c_p}\right)^2 + \left(\frac{\partial h}{\partial \rho} u_{\rho}\right)^2 + \left(\frac{\partial h}{\partial (T_g - T_i)} u_{(T_g - T_i)}\right)^2 + \left(\frac{\partial h}{\partial (T_{pin} - T_i)} u_{(T_{pin} - T_i)}\right)^2}$$  \hspace{1cm} (8.7)

As with the uncertainty in the surface heat transfer measurements, the uncertainty associated with the reported pedestal heat transfer coefficients is variable with their magnitude. This variation of $u_h$ with $h$ is illustrated in Figure 8.40. It can be seen that, contrary to the trend seen for uncertainty in the surface heat transfer coefficient, the uncertainty in the pedestal Nusselt numbers is related to the inverse of $h$. The higher uncertainty associated with low measured values of $h$ can be attributed to the greater proportional error in the temperature differential between the pedestal and the fluid. From Figure 8.40 it can be seen that over the range of tests that have been conducted, $U_h$ decreases rapidly from a value approaching 7% for $h < 30$, dropping below 5% uncertainty for $h > 35$ before levelling off to a steady value of $\sim 3.5\%$ from $h = 80.0$

8.5 Closure

Aerodynamic and heat transfer data recorded from the parametric study into zero effusion cooling geometries have been reported within this chapter. The investigation, carried out using the Sector-Rig facility, focussed on eight different geometries featuring an engine representative impingement array feeding into a roughened channel of pedestals. The investigated geometries feature pedestals at two different pin-to-pin spacings: $P/D = 1.67$ and $P/D = 2.5$ and at three different height-to-diameter ratios: $H/D = 2.5$; $H/D = 1.25$ and $H/D = 0.63$. Additional testing has been carried out on ‘Shielded Impingement’ geometries featuring pedestals of $H/D = 2.5$ (full channel height) upstream of impingement jet locations, amongst an array of predominantly short, $H/D = 0.63$, pedestals. The levels of uncertainty in
the reported surface Nusselt numbers have been calculated at 5.7%, with uncertainty in the pedestal heat transfer data typically ~3.5%.

Aerodynamic data captured from the rig has revealed that for geometries in which a gap exists between the pedestal tip and the cold skin, altering the number of pedestals within the channel has only a minor impact on the resultant pressure losses. Similarly, changes to the pedestal height have little apparent effect provided the tip clearance remains, although these observations have only been made based on pedestals up to half-channel height. The use of exclusively full height pedestals within an array has been shown to have a marked negative impact on the level of pressure loss resulting from the additional channel blockage; however, full height pedestals within the Shielded Impingement array are sufficiently sparsely arranged as to avoid increasing the channel pressure loss. In addition to the negative impact on pressure losses, surface $Nu$ for the full height pedestal geometries are lower than for all other tested geometries when considered against jet Reynolds number (flow-rate) and against total tile pressure drop. The greatest average surface Nusselt numbers have been identified as occurring for the Shielded Impingement geometries, although similar heat transfer performance as for the Shielded Impingement arrangements is displayed by the quarter and half height pedestal geometries over the first half of the channel. A rapid decline in $Nu$ with increasing streamwise position observed to occur for the short pedestal geometries is shown to be reduced through the addition of the shielding pins.

In addition to surface heat transfer data, measurements of pedestal Nusselt number have been recorded for the S3a and S3b geometries. Nusselt numbers are shown to vary for individual pins with streamwise and spanwise position based upon the velocity of the local impingement jet and the proximity of the jet to the pedestal. Nusselt numbers for the full height pedestals are larger than for the half height when compared at equivalent Reynolds number (mass flow rate) but for equivalent pressure drop, area averaged Nusselt numbers are broadly equal. When compared at equal pressure drop, the pedestals within the S3a geometry providing an effective cooling performance approximately twice that of the S3b pedestals due to the additional surface area of the pins. When compared at equal jet Reynolds numbers, the relative heat transfer performance of the pedestals within the S3a geometry is further increased over the S3b geometry.

Considering the heat transfer performance displayed for the different geometries and the associated pressure losses, the best cooling performance was provided by Shielded Impingement geometry S4b, followed closely by the second Shielded Impingement geometry.
S4c, for which heat transfer performance appeared to be reduced relative to geometry S4b, due to the reduced number of shielding pedestals. Improved heat transfer performance was achieved with less than 20% of pressure loss occurring within the channel and for only marginally higher values than for the shortest lowest blockage pedestal case. Results indicated that ‘Shielded Impingement’ cooling strategy offers potential for improved cooling performance over geometries of uniform height. Through optimisation of design heat transfer performance could be further improved.

Of the remaining geometries the heat transfer performance appears to reduce with increasing height, with geometries S2c and S3c ($H/D = 0.63$) exhibiting similar levels of average heat transfer performance for low channel pressure losses. The weakest cooling performance was displayed by geometry S3a. This was attributed to the large blockage presented by the pedestals forcing the coolant to bypass the pedestal array resulting in the poorest heat transfer performance for the highest pressure loss. An extension to the discussion of results from the Sector-Rig, reported within this chapter is included in Chapter 9 in which heat transfer performance is compared against the results from the Full Tile Test Facility and against previously reported correlations of impingement and pedestal heat transfer performance.
8.6 Figures

Figure 8.1: Profile of impingement jet velocity variation with streamwise position

Figure 8.2: Impingement pressure drop for the tested geometries as a function of Sector-Rig mass flow rate.
Figure 8.3: Mean impingement pressure drop of tested geometries as a function of mass flow rate.

Figure 8.4: Mean jet discharge coefficient of tested geometries as a function of mass flow rate.
Figure 8.5: Pedestal channel pressure drop for the tested geometries as a function of Sector-Rig mass flow rate.

Figure 8.6: Ratio of impingement jet to pedestal channel pressure drop as a function of Sector-Rig mass flow rate.
Figure 8.7: Cold skin loss coefficient ($f_{imp}$) as a function of Sector-Rig mass flow rate.

Figure 8.8: The pedestal channel friction factor ($f_{chan}$) as a function of Sector-Rig mass flow rate.
Figure 8.9: The total tile loss coefficient ($f_{tile}$) as a function of Sector-Rig mass flow rate.

Figure 8.10: Comparison of heat transfer coefficient based on inlet gas temperature vs. heat transfer coefficient based on local mixed bulk gas temperature for different geometries at Rejet=21000
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Figure 8.12: Surface Nusselt contour maps for tested cooling geometries at Jet Reynolds number ~22000.
Figure 8.13: Surface Nusselt contour maps for tested cooling geometries at Jet Reynolds number ~33000.
Figure 8.14: Span averaged surface Nusselt Number for S2 cooling Geometries, Rejet=22000

Figure 8.15: Span averaged surface Nusselt Number for S2 cooling Geometries, Rejet=33000
Figure 8.16: Span averaged surface Nusselt Number for S3 cooling Geometries, Rejet=22000

Figure 8.17: Span averaged surface Nusselt Number for S3 cooling Geometries, Rejet=33000
a) Geometry S4b (Re=23000)

b) Geometry S4c (Re=21000)

c) Geometry S4b (Re=34000)

d) Geometry S4c (Re=32000)

Impingement Jet Location

Figure 8.18: Surface Nusselt contour maps for Shielded Impingement geometries S4b and S4c.
Figure 8.19: Span averaged surface Nusselt Number for S4 cooling Geometries, Rejet=22000

Figure 8.20: Span averaged surface Nusselt Number for S4 cooling Geometries, Rejet=33000
Figure 8.21: Comparison of span averaged surface Nusselt Number for Shielded Impingement geometry S4b against equivalent full (S2a) and quarter (S2c) height pedestal arrays at Re$_{jet}$≈33000

Figure 8.22: Comparison of span averaged surface Nusselt Number for Shielded Impingement geometry S4c against equivalent full (S3a) and quarter (S3c) height pedestal arrays at Re$_{jet}$≈33000
Figure 8.23: Area averaged surface Nusselt number for all tested geometries vs. Jet Reynolds number.

Figure 8.24: Area averaged surface Nusselt number for all tested geometries vs. total tile pressure drop.
Figure 8.25: Pedestal heat transfer measurement locations for a) geometry S3a and b) geometry S3b.
Figure 8.26: Nusselt numbers for pedestals in rows 1-3 of geometry S3a.

Figure 8.27: Nusselt numbers for pedestals in rows 13-15 of geometry S3a.
Figure 8.28: Nusselt numbers for pedestals in rows 25-27 of geometry S3a.

Figure 8.29: Nusselt numbers for pedestals in rows 7-9 of geometry S3b.
Figure 8.30: Nusselt numbers for pedestals in rows 16-18 of geometry S3b.

Figure 8.31: Nusselt numbers for pedestals in rows 25-27 of geometry S3b.
Figure 8.32: Variation of pedestal Nusselt number with streamwise position for geometries S3a and S3b.

Figure 8.33: Variation of pedestal Nusselt number with jet Reynolds number for cooling geometry S3a.
Figure 8.34: Variation of pedestal Nusselt number with jet Reynolds number for cooling geometry S3b.

Figure 8.35: Comparison of pedestal Nusselt number variation with jet Reynolds number for geometries S3a and S3b.
Figure 8.36: Comparison of pedestal Nusselt number variation with total (impingement plus channel) pressure drop for geometries S3a and S3b.

Figure 8.37: Comparison of pedestal and wall Nusselt number variation with streamwise position for geometries S3a and S3b at Rejet~26000.
Figure 8.38: Comparison of pedestal Nusselt number variation with total (impingement plus channel) pressure drop for geometries S3a and S3b.

Figure 8.39: Effect of variation of surface heat transfer coefficient on the uncertainty of the measured $h$ based upon green response of the first crystal.
Figure 8.40: Effect of variation of pedestal heat transfer coefficient on the uncertainty of the measured $h$. 
Chapter 9   Comparisons of Heat Transfer Performance

Investigations that have been performed for this thesis have been conducted on two independent test facilities for which the experiments and results have, up to this stage, been presented in isolation. The investigations have comprised an evaluation of the cooling performance of an entire combustor liner outer tile utilising a traditional impingement-effusion array representative of that currently employed on Rolls-Royce engines. The large area presented by the full-tile model required a dedicated rig on which to conduct the investigation, dubbed the ‘Full-Tile Test Facility’. In order to identify means by which the heat transfer performance may be improved, a parametric study has been conducted into novel methods of cooling in the absence of effusion holes. The parametric investigation was carried out on a representative sector of the full-tile geometry. As such a much smaller cooling array was used for the experiments, with testing performed on a separate, Sector-Rig Facility. The design procedure for each of these test facilities has been reported in Chapter 5 for the Full Tile Test Facility and Chapter 6 for the Sector-Rig Test Facility. Aerodynamic and heat transfer measurements gathered from the Full-Tile Test Facility were presented in Chapter 7 and results from the parametric study on the Sector-Rig Test Facility in Chapter 8. As the reported results have been presented in isolation, they do not alone provide sufficient information for an assessment of the performance of the different investigated geometries. In order to analyse the findings in a wider context and allow a meaningful assessment of the significance of the research, a comparison of the results from the different investigations is conducted in this chapter.

9.1  Summary of results and objectives

In Chapter 7 and Chapter 8, reported results from the different test facilities have been discussed in which analyses have been conducted in isolation, with observations and discussions restricted to identification of trends and relative performance as well as explanations for the observed behaviour. In order to provide a complete picture of the results for the conducted experiments, the analysis must be expanded and the results placed in a wider context, considered against the objectives of the thesis. As such a summary of the objectives are included here, followed by a brief review of the previously reported results from both test facilities.
9.1.1 Review of Investigation Objectives

The experiments that have been detailed throughout this thesis were designed to satisfy several objectives based around the central aim of proposing, identifying and evaluating improved methods for cold-side cooling of combustor liners in lean burn gas-turbine applications. Modelling a complete combustor liner tile for the Full-Tile Test Facility was carried out in order to evaluate the heat transfer performance of an impingement jet array used within an impingement-effusion cooling arrangement in an engine representative setting. This was aimed in part at assessing the impact of localised structural blockages on the cooling performance of the tile in the affected regions and also investigating the impact on cold-side cooling performance of pressure blockages to flow exiting through the hot skin. An evaluation into the heat transfer of the unobstructed array was also conducted for comparison against idealised benchmark tests from representative sectors.

The Sector-Rig was developed to measure the heat transfer performance of various arrangements of pedestals beneath an array of impingement holes. The primary objective was to establish an optimised design that could be compared against traditional cooling methods such that the potential of combining cooling techniques could be assessed, considering heat transfer performance and level of pressure losses. The ultimate objective of the parametric study is to evaluate whether the tested geometries could potentially be applied to a cooling tile similar to that tested on the full-tile test facility and whether the results indicate that double skin combustor liners employing zero effusion cooling methods may be pursued as a viable option in lean burn gas-turbine engines.

9.1.2 Summary of Previously Reported Results

A range of experiments measuring cold-side heat transfer performance of cooling geometries designed for use in double-skin combustor liner tiles have been performed on two independent test facilities. The Full-Tile Test Facility features a model of a complete liner tile featuring arrays of impingement and effusion jet holes, arranged for full coverage cooling. The Sector-Rig Test Facility was designed for a parametric study into zero-effusion cooling methods. Measurements were taken characterising heat transfer performance of an impingement jet array in which the jets feed into pedestal roughened passages. Heat transfer measurements have been displayed in three primary formats over the course of the results chapters:
1. Area resolved contour plots of surface Nusselt number ($Nu$) have been presented in order to identify patterns and spatial variations of heat transfer performance.

2. Averaged Nusselt numbers, with $Nu$ averaged over a representative sector of the cooling tile ($\overline{Nu}$).

3. Span-averaged Nusselt numbers specific to the Sector-Rig, illustrating streamwise variation of $Nu$.

**Full Tile Test Facility**

- A sensible degree of consistency in heat transfer performance across the different measurement regions was observed when under uniform exit static pressure condition (Figure 7.18).

- Regional reductions in heat transfer performance were displayed in the vicinity of structural blockage features, due to a reduced presence of impingement jets.

- Angled jets positioned in these regions to provide direct stud cooling were observed to provide worsened heat transfer performance, with stagnation $Nu$ reduced to ~60% of typically observed values (Figure 7.22).

- For uniform exit static pressure distribution no influence of crossflow could be identified.

- Strong crossflow was evident between blockage features (two modelled ports) under uneven exit static pressure distribution (Figure 7.25). Localised reductions in surface $Nu$ were observed when crossflow was evident.

- Crossflow was most evident for case B2 when pressure blockage restricted flow through hot skin in Zones 1 and 2. Crossflow was seen to rapidly disperse [105] upon reaching the final, unobstructed zone (Figure 7.28).

**Sector-Rig Test Facility**

- Largest surface $Nu$ were displayed by the newly proposed Shielded Impingement geometries S4b, with geometry S4c displaying the second largest $Nu$. Full height pedestal geometries S2a and S3a displayed typically smaller $Nu$ than all other geometries. Lowest $Nu$ values were displayed by geometry S3a (Figures 8.24-8.25).

- Geometries S2c and S3c, featuring pedestals of quarter channel height ($H/D = 0.63$) displayed $Nu$ comparable with Geometries S4b and S4c in regions of low crossflow.
Towards channel outlet a large decrease in $Nu$ was observed for geometries S2c and S3c due to increasing crossflow (Figures 8.24-8.25).

- Pressure losses due to pedestal blockage within the channel were largest for geometry S3a, followed by geometry S2a. The lowest pressure loss was displayed by geometry S3c; very similar values displayed by geometries S2b, S2c and S4c (Figure 8.5).

- Measurements were taken of pedestal $Nu$ for geometries S3a and S3b to supplement surface heat transfer measurements and indicated significantly different streamwise distributions of $Nu$. An increase of $Nu$ with streamwise position was displayed by both geometries, but with much larger variation apparent for geometry S3a than S3b (Figure 8.32).

- $\overline{Nu}$ for geometry S3a was higher than for geometry S3b when compared against jet Reynolds number (Figure 8.35); however, when compared against total tile pressure drop geometry S3b displayed larger $\overline{Nu}$ than for geometry S3a (Figure 8.36), with values closer than in the Reynolds number based comparison.

### 9.2 Correlations of Heat Transfer Performance

Correlations of heat transfer performance have been derived in terms of a power relationship between area averaged Nusselt number $\overline{Nu}$, and jet Reynolds number $Re_{jet}$ in keeping with similar correlations for impingement [55-56] and pedestal [105] cooling arrangements. Using this relationship between the two parameters, correlations between $Re$ and $\overline{Nu}$ of the format shown in equation (9.1) have been defined for surface heat transfer measurements from the Full Tile and Sector-Rig Test Facilities and of pedestal measurements from the Sector-Rig. The coefficient, $A$ and exponent, $B$ governing the relationship were obtained by plotting natural logarithms of $\overline{Nu}$ against $Re_{jet}$, resulting in a linear representation of the data points.

$$\overline{Nu} = ARe_{jet}^B$$  \hspace{1cm} (9.1)

Due to the volume of data that has been produced from the range of heat transfer measurements, correlations of averaged $Nu-Re$ relationship have been derived to assist in the comparison of heat transfer measurements. Correlations for $Nu$ from various different geometries, measured on two independent test facilities are derived separately within this
section. Correlations for surface heat transfer measurements from the impingement-pedestal geometries tested on the Sector-Rig are presented in Section 9.2.1. Correlations for surface heat transfer measurements from the Full Tile Test Facility are presented in Section 9.2.2. Correlations for pedestal heat transfer performance are presented in Section 9.2.3. The different effective porosities of the various impingement-pedestal geometries have not been account for, with correlations essentially derived at equal mass flow rates. A method by which the porosity of the tiles may be normalised is discussed in Section 9.3

9.2.1 Surface Nu-Re Correlations for Impingement-Pedestal Geometries

Correlations relating $\overline{Nu}$ and $Re_{jet}$ that have been derived from the experimental data reported in Chapter 8 are displayed in Figure 9.1 for the S3 and S4c geometries (corresponding to equal $P/D = 1.67$) and in Figure 9.2 for the S2 and S4b geometries (corresponding to equal $P/D = 2.5$). In deriving $Nu-Re$ correlations for the impingement-pedestal cooling geometries displayed in Figure 9.1 inconsistencies in the $Nu-Re$ relationship for tests conducted at the lowest mass flow rate (corresponding to $Re_{jet} \approx 7 \times 10^3$) were observed. $Nu$ at this lowest flow rate appear larger for all tested cases than would be expected from the trend displayed by the data gathered at higher mass flow rates. Because of this, the first datum point for each of these geometries has been disregarded from the linear fits shown in Figure 9.1. The fit lines have been illustrated continuing beyond the first datum points in order to highlight the disparity between the correlation and the experimental data; however it is clear that there exists a lower limit on the range of $Re_{jet}$ over which the correlations may be confidently applied set by the extents of the data used to derive the correlations. The coefficients A and exponents B from the derived correlations are presented in Table 9.1.
## 9.2.2 Nu-Re Correlations for Full Tile Impingement-Effusion Geometry

Averaged Nusselt numbers from the Tile-Rig, specifically measurements taken from unobstructed regions removed from the blockage features, have been collated to evaluate trends in Nusselt variation with Reynolds from the different measurement regions. The collated heat transfer data, illustrated in Figure 9.3 indicate that under similar flow conditions quite a broad spread of \( \bar{Nu} \) values is displayed across the different measurement regions. Analysis of the data allowed a \( \bar{Nu} - Re \) trend to be determined, although the resulting correlation indicates a much stronger \( Re \) dependency than anticipated with an exponent of 0.904 observed. A review of published \( Nu - Re \) correlations has found no other exponents for \( Re \) of this order, with typical values varying between 0.7 and 0.8 [11] [56] [61] [106]. In order to verify that the value of 0.904 was not a product of combining measurements from different regions, trends from individual areas were plotted revealing exponents of a similar order (0.88-0.95) for all cases. As the same analysis process has been used for the Sector-Rig for which more typical \( Nu - Re \) trends (0.68-0.73) have been identified and the experimental procedure has been checked for error, the correlation \( \bar{Nu} = 0.0055Re^{0.904} \) must be considered a genuine result. Due to the absence of similar \( Nu - Re \) trends in published literature, the unusually strong \( Re \) dependency of heat transfer performance on the Tile-Rig warrants further investigation in order to verify the observed trends.

### Table 9.1: Coefficients and exponents for use in equation (9.1) correlating pedestal \( Nu \) vs. measured \( Re \) for the investigated impingement-pedestal geometries.

<table>
<thead>
<tr>
<th>Cooling Geometry</th>
<th>( A )</th>
<th>( B )</th>
</tr>
</thead>
<tbody>
<tr>
<td>S2a</td>
<td>0.0630</td>
<td>0.685</td>
</tr>
<tr>
<td>S2b</td>
<td>0.0679</td>
<td>0.681</td>
</tr>
<tr>
<td>S2c</td>
<td>0.0578</td>
<td>0.701</td>
</tr>
<tr>
<td>S3a</td>
<td>0.0469</td>
<td>0.705</td>
</tr>
<tr>
<td>S3b</td>
<td>0.0409</td>
<td>0.727</td>
</tr>
<tr>
<td>S3c</td>
<td>0.0579</td>
<td>0.701</td>
</tr>
<tr>
<td>S4b</td>
<td>0.0697</td>
<td>0.692</td>
</tr>
<tr>
<td>S4c</td>
<td>0.0460</td>
<td>0.728</td>
</tr>
</tbody>
</table>
9.2.3 Pedestal $Nu$-$Re$ Correlations for Impingement-Pedestal Cooling Tiles

Pedestal heat transfer data were captured for two of the impingement-pedestal cooling tiles for which surface heat transfer data have been presented, with measurements taken for geometries S3a and S3b. Correlations of $\overline{Nu}$-$Re$ have been derived from the mean Nusselt numbers for the pedestals illustrated in Figure 8.35. The resulting correlations are plotted against the relevant experimental data points in Figure 9.4. The coefficients for the correlation are displayed in Table 9.2.

<table>
<thead>
<tr>
<th>Cooling Geometry</th>
<th>$A$</th>
<th>$B$</th>
</tr>
</thead>
<tbody>
<tr>
<td>S3a</td>
<td>0.0116</td>
<td>0.846</td>
</tr>
<tr>
<td>S3b</td>
<td>0.0102</td>
<td>0.845</td>
</tr>
</tbody>
</table>

Table 9.2: Coefficients and exponents for use in equation (9.1) correlating pedestal $Nu$ vs. measured $Re$.

9.3 Impact of fractional pressure losses on tile effective porosity

For comparisons between different cooling geometries to be valid for application to engine scenarios, investigations should target matching mass flow rate and pressure drop characteristics of tested models against engine specified conditions; however, the cooling arrangements employed on the Sector-Rig Facility were not all able to match the parameters. Aerodynamic data measured from the Sector-Rig indicated that the total pressure drop occurring across the different cooling geometries is a function of losses across the cold skin and across the pedestal array. The common impingement array used for all tile geometries mean that pressure drop across the cold skin was generally consistent for all geometries. Changes enforced on the pedestal array for the different cooling geometries however, were found to affect the pressure losses within the cooling channel due to the variable levels of obstruction presented by the pedestals. Matching mass flow rate and pressured drop characteristics is important because both characteristics are determined by engine conditions. Pressure drop is set by fuel injector requirements and the mass flow rate is determined by the available coolant. Due to the high increased proportion of compressor air entering directly into the combustion chamber in lean burn engines, a strict limit is placed on the available mass flow rate available for component cooling. As pressure drop and mass flow rate conditions were not matched in all cases, heat transfer measurements from the Sector-Rig were compared at equal mass flow rate (characterised by $Re_{jet}$) and equal tile pressure drop.
respectively. Neither of these fully satisfy the engine conditions as only one parameter is matched. It was reported in Chapter 8 that the relative porosity of different cooling geometries could be compared through the use of the total tile loss coefficient, \( f_{tile} \), with the distributions illustrated in Figure 8.9 indicating \( f_{tile} \) with reference to the engine specified value of \( f_{tile} = 0.47 \).

A means of unifying the heat transfer data from the different geometries has been derived by the author; using this heat transfer performance is corrected for equivalent porosity, by applying the porosity coefficient \( C_{por} \), derived in Section 9.3.1 (page 229). In addition to comparing performance of the heat transfer performance, the fractional pressure drop across the pedestal array (\( f_{chan} \)) has been used to predict the pressure loss within the channel for increased numbers of pedestals. This has been conducted to evaluate the possibility of applying pedestals over tiles matching the EFE geometry modelled on the Full Tile Test Facility.

### 9.3.1 Porosity Coefficient for Modelling Equalised Tile Porosity

In order to quantify the overall effect of the blockage presented by a given geometry a porosity coefficient has been derived by the author based on the tile loss coefficient \( f_{tile} \), that was defined in Chapter 8 (Table 8.1, page 173). \( f_{tile} \) relates the mass flow entering the cooling geometry through the impingement array to the total pressure drop incurred across the entire tile. Thus, to match engine design conditions of mass flow rate and total pressure drop \( f_{tile} \) must be matched. The definition of the porosity coefficient \( C_{por} \), given by equation (9.2) is the ratio of the loss coefficient \( f_{SNx} \) (with the subscript referring to the geometry in question), for a particular tile geometry over the reference tile loss coefficient \( f_{ref} = 0.465 \) based on the defined engine conditions provided by Rolls-Royce PLC [6].

\[
C_{por} = \frac{f_{SNx}}{f_{ref}} \quad (9.2)
\]
Where:

$$A_{equiv} = A_{open} \frac{\sqrt{\Delta P_{SNx}}}{\sqrt{\Delta P_{ref}}} \equiv A_{open} C_{por}$$  \hspace{1cm} (9.4)$$

Considering that the mass flow rates for the two geometries must be equal for a matching condition and assuming equal density, the porosity coefficient may be expressed in the form shown in equation (9.3). As all geometries use a common impingement geometry that has been sized based on the dimensions derived from EFE data regarding mass flow rate and pressure drop, the open impingement area is equal for both cases. This reduces the porosity coefficient to being equivalent to the ratio of the total tile pressure drops. Using this relationship, it can be demonstrated that for an adjusted open area as shown in equation (9.3), the ratio of pressure losses may be equalised. This provides a definition shown in equation (9.4), for the equivalent area of impingement array required to match the porosity characteristics of the target tile. As such, a porosity coefficient greater than 1 indicates a cooling geometry that must be increased in size to match engine pressure drop characteristics at the defined mass flow rate. For a value smaller than 1 the reverse would be true. This change in effective porosity of the tile is applied in order to reduce the stage pressure loss across the impingement array such that total tile pressure loss is equalised with the reference value. This results in an altered jet Reynolds number, affected by the increase in hole diameter (by a factor of $\sqrt{C_{por}}$) and the reduction in jet velocity (by a factor of $1/C_{por}$). The combined influence of these changes leads to the equivalent Reynolds number $Re^*$, as defined in equation (9.5).

$$\therefore Re^* = Re_{ref} / \sqrt{C_{por}}$$  \hspace{1cm} (9.5)$$

$Re^*$ represents the Reynolds number at which the altered impingement geometry would be operating at the same mass flow rate and pressure drop as the reference geometry $Re_{ref}$. Using the Nusselt-Reynolds correlations that have been derived in Section 9.2 for the different geometries from the experimental results, comparison of the heat transfer performance of the different arrays may be made at equal mass flow rate and pressure loss by correcting the correlations for the equivalent porosity Reynolds number, $Re^*$. 

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9.3.2 Fractional Pressure Loss Variation with Number of Pedestal Rows

As the porosity coefficient is a function of $f_{tile}$ its value will change in line with any alteration in the number of pedestal rows assuming the impingement array parameters (jet velocity, hole diameter, jet-to-jet and jet-to-target spacing) remain constant. For constant density flow, the pressure drop across an impingement array is dependent primarily on the velocity through the jet holes. As such, regardless of the number of additional jet rows added, the impingement pressure drop will remain constant. For a pedestal array in which all flow migrates in one direction, the pressure drop will increase for every additional row. In a traditional pedestal array, in which all mass enters the channel upstream of the first pedestal row; the pressure loss, characterised by friction factor per row, increases at a steady rate as the mass flow rate is constant. For the distributed impingement arrangement however, the pressure drop per row increases as the spent jet air from each subsequent jet row adds mass to the coolant within the channel. This increases the crossflow velocity and subsequently the pressure drop across downstream pedestal rows.

For a consistent geometry over the channel it can be assumed that the friction factor per row remains constant and as such, the total friction factor within the channel may be evaluated for an altered number of pedestal rows. Modelling pressure drop in terms of friction factor per row is common practice for traditional pedestal arrays [29] [105]. Measurements of channel friction factor ($f_{chan}$) taken from the Sector-Rig Test Facility (displayed in Figure 8.8.) indicated that, with the exception of geometries S2a and S3a for which a trend of decreasing $f_{chan}$ with mass flow rate was displayed, typical values were largely invariant with increasing mass flow rate for values above 0.15 kg/s. These observations would suggest that additional rows could be added without significantly altering the friction factor per row. As such, $f_{chan}$ may be evaluated, as in equation (9.6) for a defined number of pedestal rows, $N$. The value, $N_{SN}$ is the number present in the tested geometry: 21 for S2a-c and S4b; 31 for S3a-c and S4c.

$f_{chan}$ may thus be used to calculate the pressure drop after $N$ rows using equation (9.7) where $\bar{U}_{max}$, as previously defined in Chapter 8 is the mean, maximum velocity through the channel. This is modelled on a mean mass flow rate of $\frac{1}{2} \dot{m}$ in order to account for the gradual addition of fluid to the channel. Constant pressure loss across the cold skin corresponds to a mean mass flow rate through the impingement holes that is invariant with increasing numbers of jets. As such, the total mass flow rate within the channel increases
with the addition of impingement rows. The subscript $N$ refers to the row number for which the values of $\dot{m}$ and $\bar{U}_{\text{max}}$ are applicable, with $\dot{m}_N$ equal to the mass flow rate per row multiplied by the number of rows.

$$f_N = \frac{f_{\text{chan}N}}{N_{SNx}}$$  \hspace{1cm} (9.6)

Considering:

$$\dot{m} = 2\rho A_{\min} \bar{U}_{\text{max},N}$$

$$dP_{\text{chan},N} = f_N 2\rho \bar{U}_{\text{max},N}^2 \equiv \frac{f_N \dot{m}_N^2}{2\rho A_{\min}^2}$$  \hspace{1cm} (9.7)

The increasing level of pressure drop occurring within the channel, calculated using equation (9.7) is illustrated in Figure 9.5, expressed as a fraction of the total permissible pressure drop. This value has been taken from the ratio $\dot{m}/\sqrt{dP}$ from the Tile-Rig definition. Data have been plotted in relation to the impingement hole row number in order to account for the different $P/D$ values of pedestal arrays considered. The data are plotted up to 30 impingement rows as that is the number present in the modelled full liner tile geometry. From Figure 9.5 it can be seen that the pressure drop occurring across each pedestal row is initially very small, increasing progressively as the mass flow rate and therefore crossflow velocity within the channel increases. This results in the channel pressure drop for all pedestal arrangements exceeding the design total allowable pressure drop being over the tile length illustrated in Figure 9.5. The number of pedestal rows across which this occurs can be seen to vary widely for the different design concepts and the value of $N$ for which channel fractional pressure losses of 0.2; 0.5 and 1 (as a proportion of the allowable value) are listed in Table 9.3. A fractional pressure drop of 0.2 (indicated by the dashed line on Figure 9.5) corresponds to the design channel pressure loss, based on the impingement open area.

<table>
<thead>
<tr>
<th>Fractional pressure loss $(dP_{\text{chan}}/dP_{\text{total}})$</th>
<th>Allowable number of pedestal rows (from channel start)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>S2a</td>
</tr>
<tr>
<td>0.2</td>
<td>19</td>
</tr>
<tr>
<td>0.5</td>
<td>26</td>
</tr>
<tr>
<td>1.0</td>
<td>33</td>
</tr>
</tbody>
</table>

Table 9.3: Number of pedestal rows corresponding to particular channel pressure losses, expressed as a fraction of the maximum allowable value.
The increasing level of pressure drop as mass is added to the channel restricts the use of pedestals over a complete liner tile. The large pressure losses for the full height pedestal geometries render their use impractical within a lengthy channel; however, the significantly smaller pressure drops associated with the arrays of pedestals in which tip clearance is present would still require selective positioning, although the application of pedestals to select sections of the array is a possibility. Positioning pedestals at the leading edge of the channel would result in the smallest pressure drop increase due to a lower crossflow velocity in this region, but as a consequence of the low flow rate, their effectiveness would be compromised. Additionally, as was observed to occur to an extent over the geometries tested on the Sector-Rig, the placement of pedestals within the cooling channel promotes a redistribution of flow, with more of the coolant passing through the rear impingement holes to avoid the channel obstruction. This was shown by the increase in jet velocity from the first to the final jet row, illustrated in Figure 8.2, with particularly pronounced effects apparent for the dense, full height geometry S3a in which the large blockage resulted in final row jet velocities of order 3-4x that of the first row jet. This problem could be negated by applying pedestals later within the channel; however, the increasing mass flow rate with streamwise position would result in a higher pressure drop for each row, reducing the number that may be employed from those listed in Table 9.3.

9.4 Application of Porosity Correction to Heat Transfer Measurements

The definition of porosity coefficient derived in Section 9.3 has been used to derive a method by which pressure drop characteristics of the different geometries may be brought into unison, through use of the porosity coefficient as a correction factor. Application of this method require it to be assumed that small changes to the diameter of holes in the impingement array would have negligible effects on the $\overline{Nu}$-$Re$ correlations derived in Section 9.2, beyond the change in Reynolds number required to maintain equal mass flow rate distribution. Due to the lack of reported data to support this assumption, the reasoning has been qualified in Section 9.4.1 (page 236).

9.4.1 Porosity Corrected Surface Heat Transfer Performance

As the coefficients listed in Table 9.1 (page 227) were obtained for the variation of surface Nusselt number with the Reynolds number of the flow issuing from the impingement
jets, using the listed values of $A$ and $B$ in equation (9.1) gives a comparison of the performance of the different cooling configurations at roughly equivalent mass flow rate (excepting any variations in the discharge coefficient through the impingement array for the different tile geometries). In such a comparison, no account is taken of the varying amounts of total pressure drop that have been identified for the different cooling geometries; however, adopting the method proposed in Section 9.3 for matching effective porosity allows a comparison at equal pressure drop and mass flow rate, using the porosity coefficient as a correction factor. By adjusting the open area of the impingement array, the jet-velocity required for a given mass flow rate may be increased or decreased; with the pressure-drop across the impingement tile changing accordingly. By considering the porosity coefficient for a particular array against the reference porosity of 0.733, the size of holes required within the impingement array for correct matching of the mass flow rate and pressure drop characteristics of the reference tile (based upon engine data provided by Rolls-Royce PLC [6] may be determined.

As correcting the Reynolds number for equal porosity across the different geometries requires a change in the impingement array to be modelled, it must be assumed that altering the diameters of the holes in the impingement array has a negligible effect on the $Nu-Re$ correlations if the correction factor is to be considered valid. This has been justified due to the minor alteration of tile geometry being considered by the model. As the proposed change in hole diameter being modelled is considered at constant mass flow rate, the distribution of flow entering the cooling channel and accordingly, the mass flow throughout remain unaffected. Coupling this with the fact that no change to the pedestal array is being modelled, the only parameters that would be altered in addition to the jet Reynolds number required for a particular mass flow rate are the jet-to-jet and jet-to-target spacings. A brief discussion into the possible impact on the $Nu-Re$ correlations of altering these parameters $(X/D, Y/D, Z/D)$ follows the description of the correction model.

In order to compare $Nu$ at equal mass flow rate and pressure drop for the different geometries, a method has been derived to correct the $Nu-Re$ correlations evaluated from jet Reynolds numbers indicating uniform mass flow rate but variable porosity, to correlations of $Nu-Re^*$ for which both parameters are matched across geometries. As the correction to $Re^*$ involves altering the feed array, its value for a given mass flow rate varies across geometries. This variable parameter can however be related back to the reference Reynolds number (corresponding to the impingement array on the Sector-Rig Facility) using equation (9.5)
from which it was derived. Substituting equation (9.5) into equation (9.1) enables \( \overline{Nu} - Re^* \) correlations for the different geometries to be calculated by inputting \( Re_{ref} \) and the previously measured coefficients \( A \) and \( B \) listed in Table 9.1 into equation (9.8). Using this equation, an adjusted coefficient, \( A^* \) may be derived using the respective porosity coefficients of the different arrays, as defined in equation (9.9). The resulting adjusted coefficients are listed in Table 9.4.

\[
\overline{Nu} = A \left( Re_{ref} / \sqrt{C_{por}} \right)^B \equiv A \left( \sqrt{C_{por}} \right)^{-B} Re_{ref}^B \quad (9.8)
\]

\[
A^* = A \left( \sqrt{C_{por}} \right)^{-B} \quad (9.9)
\]

Replacing coefficient \( A \) with \( A^* \), enables the correlations of area averaged Nusselt number, corrected for equalised porosity to be plotted. Porosity corrected correlations are illustrated in Figure 9.6, with \( Nu \) plotted against \( Re_{ref} \), with a given x-position corresponding to the same mass flow and pressure drop conditions for all geometries. The constant porosity correction imposes a fixed percentage change \( \overline{Nu} \) when compared with the baseline results illustrated in Figure 9.1 and Figure 9.2. To give an indication of the predicted effect on Nusselt number of adjusting for equivalent porosity the percentage change in \( \overline{Nu} \) resulting from the correction factor is listed in Table 9.5 for the different geometries.

The changes in surface heat transfer performance modelled using the equivalent porosity correction factor appear in line with observations of relative performance when plotted against total tile pressure drop in Figure 8.24. Compared with the \( \overline{Nu} \) variation versus jet Reynolds number illustrated in Figure 8.23, a much wider spread was apparent, with the relative heat transfer performance of the full height pedestal geometries (S3a and S3b) appearing noticeable reduced, with less pronounced improvements apparent for the predominantly short pedestal geometries, particularly S3c and S4c. The effect on the \( \overline{Nu} - Re \) trend of correcting to \( \overline{Nu} - Re^* \) appears less pronounced than those seen previously, but modelling heat transfer performance at a reduced pressure drop but equal mass flow rate would be expect to result in Nusselt numbers between those measured for the two previously plotted examples, as is predicted by the correction factor.
Table 9.4: Coefficients and exponents for use in equation (9.1) correlating pedestal $Nu$ vs. equivalent porosity $Re^*$ for the investigated impingement-pedestal geometries.

<table>
<thead>
<tr>
<th>Cooling Geometry</th>
<th>$A^*$</th>
<th>$B$</th>
</tr>
</thead>
<tbody>
<tr>
<td>S2a</td>
<td>0.0624</td>
<td>0.685</td>
</tr>
<tr>
<td>S2b</td>
<td>0.0691</td>
<td>0.681</td>
</tr>
<tr>
<td>S2c</td>
<td>0.0589</td>
<td>0.701</td>
</tr>
<tr>
<td>S3a</td>
<td>0.0436</td>
<td>0.705</td>
</tr>
<tr>
<td>S3b</td>
<td>0.0409</td>
<td>0.727</td>
</tr>
<tr>
<td>S3c</td>
<td>0.0592</td>
<td>0.701</td>
</tr>
<tr>
<td>S4b</td>
<td>0.0704</td>
<td>0.692</td>
</tr>
<tr>
<td>S4c</td>
<td>0.0471</td>
<td>0.728</td>
</tr>
</tbody>
</table>

Table 9.5: Predicted percentage change in $\overline{Nu}$ for cooling tile adjusted to reference porosity coefficient.

```
<table>
<thead>
<tr>
<th>Percentage change</th>
<th>S2a</th>
<th>S2b</th>
<th>S2c</th>
<th>S3a</th>
<th>S3b</th>
<th>S3c</th>
<th>S4b</th>
<th>S4c</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>-1.02</td>
<td>1.69</td>
<td>1.54</td>
<td>-6.79</td>
<td>0.10</td>
<td>2.08</td>
<td>0.90</td>
<td>2.24</td>
</tr>
</tbody>
</table>
```

Due to the primary assumption of this comparison that any impact on the relationship between $Re$ and $Nu$ will be sufficiently small for a negligible impact to be seen in the correlations, it must be noted that such a comparison provides only a guideline of performance at equivalent conditions. Nevertheless, a qualification of the stated assumption is included here. The changes required to the diameter of holes within the impingement array and the resultant jet velocity for matching mass flow rate alter more than just the jet Reynolds numbers. Maintaining the same tile geometry but with altered jet diameter will affect the $X/D$, $Y/D$ and $Z/D$ of the holes within the impingement array, as well as altering the relative magnitudes of jet and crossflow velocities. An increase in the diameter of the impingement holes requires a reduction in jet velocity in order for a constant mass flow rate to be maintained, thereby increasing the relative strength of the crossflow to the jet-flow due to unchanged conditions within the channel, with the reverse being true for a reduction in hole size. Changing the hole diameter will also alter the jet-to-target spacing, which has been shown in various studies to influence the impact of the crossflow on impingement performance. Although studies into the performance of arrays of impingement jets above
smooth passages have shown jet performance to be more affected by crossflow as $Z/D$ reduces [61], in scenarios where the ratio of impingement to crossflow velocities remain constant the opposite behaviour has generally been observed [56] [107]. As the changes modelled by use of the correction factor are considered at constant mass flow rate, the crossflow would remain unaltered regardless of the change in jet diameter. As such, the level of influence of the crossflow on impingement performance would be expected to rise and fall with $Z/D$. The level of influence exerted by the crossflow on impingement performance was also shown by Florschuetz et al. [56] to be sensitive to changes in $X/D$ and $Y/D$ but with the effects of changing each of these shown to be opposed. As such, the described correction factor should be considered as a guideline of performance that would result if altering the geometries for equal porosity, with further testing required if the effect of changing the impingement over fixed pedestal geometries is to be determined.

### 9.4.2 Porosity Corrected Pedestal Heat Transfer Performance

<table>
<thead>
<tr>
<th>Cooling Geometry</th>
<th>$A$</th>
<th>$A^*$</th>
<th>$B$</th>
</tr>
</thead>
<tbody>
<tr>
<td>S3a</td>
<td>0.0116</td>
<td>0.0106</td>
<td>0.846</td>
</tr>
<tr>
<td>S3b</td>
<td>0.0102</td>
<td>0.0102</td>
<td>0.845</td>
</tr>
</tbody>
</table>

Table 9.6: Coefficients and exponents for use in equation (9.1) correlating pedestal $Nu$ vs. measured $Re$ and equivalent porosity $Re'$ for geometries S3a and S3b.

Additionally, the method detailed in the previous to correct the correlations for heat transfer performance at equivalent porosity has been used to derive the coefficients, $A^*$ also listed in Table 9.2. The mean pedestal Nusselt numbers are plotted at equivalent Reynolds number in Figure 9.7, using the correction factor to adjust the experimental data points and to produce the lines illustrating the $\bar{Nu} - Re^*$ correlations for equivalent porosity. From Figure 9.4 and Figure 9.7 it can be seen that for comparison at equal mass flow rate and pressure loss, the mean Nusselt numbers are very similar for the two geometries, with $\bar{Nu}$ for geometry S3a 4.5% higher than for geometry S3b over the range of Reynolds numbers considered. This is reduced from a difference of ~14% for the $\bar{Nu}-Re$ correlation based on experimental measurements. As with the changes in surface heat transfer performance discussed in the previous section, the comparative reduction in $\bar{Nu}$ for the pedestals in the S3a geometry appears in line with observations of the relative performance when plotted against total tile pressure drop in Figure 8.36. For comparisons against total tile pressure
loss, mean pedestal Nusselt numbers for geometry S3b were higher than those for geometry S3a, which is reversed for comparisons against $Re$ and $Re^*$. As modelling heat transfer performance at a reduced pressure drop but equal mass flow rate would be expected to result in Nusselt numbers between those measured for the two previously plotted examples, the observed performance appears sensible.

9.5 Comparison of Impingement-Pedestal Tiles versus Baseline Geometries

The investigation carried out on the Sector-Rig into combined impingement-pedestal geometries was aimed at developing improved methods of cold side cooling. For an evaluation of whether the investigated approaches indeed offer improved performance over traditional methods the results that have been reported must be compared against previously published heat transfer data for relevant, equivalent geometries. For this purpose, surface Nusselt measurements and correlations are compared against traditional impingement and pedestal arrays as well as Nusselt data captured from the full-tile test facility. Comparisons of the heat transfer and aerodynamic performance of the different tiles studied using the Sector-Rig facility have thus far indicated superior surface heat transfer performance results from the Shielded Impingement geometries with the weakest surface Nusselt numbers occurring under geometry S3a, the most densely packed array with pedestals at $H/D = 2.5$ and $P/D = 1.67$. Rather than repeating the data for all geometries, comparisons have largely been restricted to these select best and worst case geometries.

9.5.1 Comparison of Surface Nusselt Numbers versus Impingement Array

In order to evaluate the heat transfer performance of the combined impingement-pedestal cooling geometries, comparisons are drawn against a similar array of impingement jets issuing into a smooth channel as well as to the heat transfer data captured from the full-tile test facility. For comparison of the averaged Nusselt numbers from the two rigs, adjustments must be made to the $Nu$ values that have previously been reported for the combined cooling geometries. As was mentioned in Chapter 8, for pedestal and surface Nusselt numbers to be evaluated together and for comparisons against traditional pedestal geometries, Sector-Rig Nusselt number were calculated taking the pedestal diameter ($1\text{mm at engine scale}$) as the characteristic length. The Nusselt numbers against which the Sector-Rig data are compared in this section are all calculated based on jet hole diameter ($0.7\text{mm at}$
engine scale) and as such, in order to bring the values in line, a factor of 0.7 has been applied to the Sector-Rig Nusselt numbers. Comparison of cold-side heat transfer performance from the different rigs indicates that while Nusselt numbers from the full tile model exceed those displayed by geometry S3a over the entire Reynolds range, lower averaged Nusselt numbers are typically seen for the impingement-effusion cooling system than for the Shielded Impingement geometries. This behaviour holds true for jet Reynolds numbers below $Re_{jet} = 3 \times 10^4$; however, the steeper increase of $Nu$ with Reynolds number means that beyond this value the impingement-effusion arrangement exhibits higher Nusselt numbers than all tested combined geometries. The steep variation of $\bar{Nu}$ with $Re$ displayed for the Tile-Rig correlation has been discussed in 9.2.2.

In addition to drawing comparisons between the two different cooling strategies investigated for this thesis, experimental heat transfer measurements are also compared against correlations modelling the performance of a simple impingement array. As can be clearly identified from Figure 9.8, the published correlations indicate that the modelled Nusselt numbers exceed the experimental measurements from both test rigs, with $\bar{Nu}$ typically of order 15% larger than for the Shielded Impingement geometries. This is accounting only for surface heat transfer performance however, with the contribution of the pedestals not considered in the data illustrated in Figure 9.8. The selected correlation for modelling the performance of impingement jets, against which experimental data have been compared, was derived by Florschuetz et al [56] details included in Chapter 2. This correlation was derived over a range of parameters which are matched by the geometries tested using the Sector-Rig and Tile-Rig Test Facilities, including $Re_{jet}$, $X/D$; $Y/D$; channel aspect ratio and ratio of crossflow to jet velocity. The only parameter to fall outside the range of the correlation was $Z/D = 3.54$ for the tested impingement array. This exceeds the maximum value employed by Florschuetz et al. ($Z/D = 3$) but it was deemed that conditions were sufficiently close to the accepted range for the correlation to be applied.

For a cooling geometry such as considered here, the propagation of flow towards channel exit results in a gradual increase of the ratio of crossflow to jet-velocity. Using the correlation, $Nu$ may be modelled as an area averaged value, but the predicted variation of $Nu$ for the modelled impingement array for increasing crossflow build up may also be considered for comparison against combined impingement-pedestal geometries. To expand on the information gleaned from the comparison of area averaged heat transfer performance plotted in Figure 9.8, point-by-point streamwise variation in Nusselt number, as evaluated by the
correlations, is plotted in Figure 9.9. This is compared against equivalent experimental data from selected geometries tested on the Sector-Rig. Viewing the surface Nusselt number in terms of variation with streamwise position clearly illustrates the superior heat transfer performance for the correlated, smooth impingement array. The streamwise profile for the simple arrangement is characterised by initial Nusselt numbers far in excess of those displayed for the combined geometries, but with a much more pronounced reduction with streamwise position as the crossflow increases. As can be seen from Figure 9.9 the rate of reduction is such, that for the rearmost jets taken over the length of the experimental section, Nusselt numbers drop to a level comparable with those of the combined arrays and lower than for the Shielded Impingement geometry, S4b. The rate of reduction in heat transfer performance with increasing crossflow that is predicted by Florschuetz et al.’s correlation is such that despite the superior initial Nusselt numbers, the addition of any further jet rows would appear likely to see subsequent values drop below those of the remaining combined geometries. This difference in streamwise variation highlights the regions where the addition of pedestals to the channel can offer improved surface heat transfer, but also where their presence is potentially obtrusive. Additionally, it should be noted, that despite the larger surface Nusselt numbers displayed, when taking into consideration the additional heat transfer through the pedestals (even in the upstream regions of low crossflow), heat transfer performance for the combined geometries would be elevated above those of the smooth channel.

9.5.2 Comparison of Surface Nusselt Numbers versus Pin-Fin Arrays

In order to conduct a comparison of the tested distributed impingement-pedestal arrays against traditional feed pin-fin arrays correlations developed for Nusselt number and friction factor, derived by Chang et al. [108] have been used. Note that throughout this section the term ‘pin-fin’ is reserved for the traditional pin-fin arrangement to avoid confusion when comparing against the combined impingement-pedestal geometries investigated for this thesis. Dimensions of the comparison array are broadly consistent with geometries from the Sector-Rig. Both channels accommodate pedestals up to \( H/D = 2.5 \). \( P/D = 2 \) for pin-fin array, falling between the pedestal densities used for the S2 (and S4b) and S3 (and S4c) geometries. Chang et al. also evaluated the effect of tip clearance on heat transfer and pressure drop. Friction factor was seen to drop significantly for increasing amounts of tip clearance, with \( f/row \) dropping from 0.138 for \( H/D=2.5 \) to \( f = 0.052 \); for \( H/D=1.75 \)
applying the definition of friction factor used throughout this thesis. Correlations estimated \( f = 0.027 \) for \( H/D=1.25 \) and even lower were it extended to \( H/D = 0.063 \). Using these values and applying a mass flow rate and maximum pressure drop defined by the EFE conditions maximum numbers of pedestal rows before the allowable pressure drop was exceeded were calculated for the different arrays. These corresponded to 5; 22 and 61 rows for \( H/D=2.5, 1.75 \) and 1.25 respectively, for arrays featuring an inlet feed of a single row of 5x51mm impingement holes (3mm at engine scale, as indicated in Figure 1.5 for typical pedestal geometries). Comparison with the data in Table 9.3 indicates that fewer pedestal rows may be employed in an inlet feed arrangement than with distributed impingement, for \( H/D=2.5 \) and 1.75; however for \( H/D = 1.25 \), the friction factor has dropped sufficiently for a greater number of rows to be used.

Using the correlations for heat transfer performance, also derived by Chang et al. [108] average Nusselt number for the modelled pin-fin arrays have been calculated. These are shown in Figure 9.10 alongside porosity corrected Nusselt numbers for geometries S3a, S4b and S4c. It can be seen from Figure 9.10 that there is a large decrease in \( Nu \) with decreasing pin-fin height. This is in opposition to the behaviour observed for the geometries tested on the Sector-Rig. It can also be seen that the full height pin-fin array displays Nusselt numbers significantly larger than for all other geometries, including Shielded Impingement geometries, S4b and S4c. Conversely, correlated \( Nu \) for the pedestal arrays with clearance display lower \( Nu \) than for the Shielded Impingement geometries, but with larger \( Nu \) than for the porosity corrected values for geometry S3a.

From this comparison it would appear that the Shielded Impingement geometries proposed and tested for this thesis exhibit superior cooling performance over all of the traditional pin-fin arrays against which they have been compared. Although the largest \( Nu \) was observed for the full height traditional arrangement, the pressure drop for this case was too large to maintain cooling beyond five pin-fin rows, resulting in poor area cooling. As regards the pin fins with clearance, both Nusselt number and pressure drop are less favourable than for geometries S4b and S4c for pin-fins with \( H/D = 1.75 \). For pedestals at \( H/D = 1.25 \), the pressure drop per row has reduced such that a greater surface area could be cooled; however, the associated \( Nu \) for this geometry shown in Figure 9.10, would appear to be too small to recover. It should be considered however, that this comparison only accounts for endwall heat transfer performance and, as has been previously discussed in Section 8.3.3 (page 192), in traditional pedestal arrays, \( Nu_{pin} > Nu_{wall} \). For pedestals in the distributed
impingement arrangement however, the reverse has been observed, with $N_u_{pin} \leq N_u_{wall}$ for the tested geometries.

9.6 Closure

Within this chapter, comparisons have been made of the performance of geometries measured from the different Test Facilities used for this thesis. A comparison of heat transfer performance for the Tile-Rig and Sector-Rig geometries, illustrated in Figure 9.8, most obviously indicates a very different $\overline{N_u}$-$Re$ relationship for the Tile-Rig than for all geometries tested on the Sector-Rig and for published correlations for heat transfer performance beneath similar impingement jet arrays. The correlations for the Tile-Rig indicated a much stronger $Re$ dependency of $\overline{N_u}$ than for all other observed cases, with $Nu \propto Re^{0.904}$ much higher than encountered in the literature. The reason for this high Reynolds dependency has not been confirmed; however, examination of reported Tile-Rig data indicates that similar variations of $\overline{N_u}$ with $Re$ appear to be displayed across the full range of measurement regions and pressure conditions implying a genuine result. As a result it may be characteristic of some aspect of the specific cooling geometry and is worthy of further investigation in order to confirm or refute the indicated trend. When compared against the performance of geometries from the Sector-Rig, it could be seen that the full tile $N_u$ were higher than the values for S3a over the full Reynolds range. Typically smaller $N_u$ were seen for the Tile-Rig than for geometries S4b and S4c; however, the large dependency on $Re$ would see the Tile-Rig $\overline{N_u}$ increase higher than for the Shielded Impingement geometries at $Re > 3 \times 10^4$.

When comparing the measured geometries against a widely regarded correlation of impingement performance [56] in Figure 9.8, the correlated values displayed significantly larger $\overline{N_u}$ than for any of the tested geometries. Despite the larger Nusselt numbers displayed by the impingement correlation, there are several good arguments that the impingement-pedestal geometries may offer superior heat transfer performance. The variation of streamwise Nusselt number for the different geometries in Figure 9.9 can be seen to decrease significantly with streamwise position as crossflow increases. All combined impingement-pedestal geometries display greater resistance to the crossflow indicating that for any increase in tile length, their relative heat transfer performance would improve. A second observation from Figure 9.9 is that a significant difference is apparent in $\overline{N_u}$ for the impingement jet location. All geometries tested on the Tile-Rig displayed lower values of
$Nu$ for the first jet row before subsequently increasing. This is not modelled by the correlation and appears to result an exaggerated value for jet row one. A final consideration is that for Geometries S3a and S3b, pedestal $Nu$ were of similar order to the surface values. If similar trends are apparent for the remaining geometries then accounting for pedestal heat transfer would result in superior heat transfer performance for the Shielded Impingement geometries. The best comparison would be to perform a zero pedestal test on the same geometry for comparison against experimental data.

A further comparison that has been made in this chapter is the comparison of geometries S3a, S4b and S4c against pin-fin arrays with similar geometric properties. Comparison of data indicated that although Nusselt numbers for the full height pin-fin case were higher than for the combined cooling geometries, the reported friction factor was such that pressure drop would exceed the maximum allowable value for the EFE tile after 5 pin rows, making it impractical for use. Comparisons were also made against similar pin-fin arrays with clearance; however, although more pedestal rows could be used due to the reduction of friction factor for reducing pin-fin height, Nusselt numbers also reduced and the combined geometries exhibited superior heat transfer and lower pressure losses.
9.7 Figures

Figure 9.1: Experimental data points and correlated lines of Surface Nusselt number variation with $Re_{jet}$ for geometries with $P/D = 1.67$.

Figure 9.2: Experimental data points and correlated lines of Surface Nusselt number variation with $Re_{jet}$ for geometries with $P/D = 2.5$. 
Figure 9.3: \( Nu - Re \) correlations derived from surface heat transfer measurements from unobstructed regions of the full liner tile model.

Figure 9.4: Experimental data points and correlated lines of Pedestal Nusselt number variation with \( Re_{jet} \) for geometries S3a and S3b.
Figure 9.5: Variation with increasing numbers of pedestal rows of channel pressure drop expressed as a fraction of the maximum allowable value.

Figure 9.6: Correlated lines of Surface Nusselt number variation with reference Reynolds number $Re_{ref}$ corrected for equivalent porosity Reynolds number, $Re^*$ using coefficient $A^*$. 
Figure 9.7: Experimental data points and correlated lines of Pedestal Nusselt number variation with reference Reynolds number $Re_{ref}$ corrected for equivalent porosity Reynolds number, $Re^*$ using coefficient $A^*$.

Figure 9.8: $Nu - Re^*$ correlations derived from data captured from the full liner tile model and combined cooling sector geometries, compared against published data for simple impingement jet arrays [56]
Figure 9.9: Comparison of streamwise variation of averaged Nusselt number for selected combined cooling geometries against published data for impingement jet arrays over smooth channels [56].

Figure 9.10: Comparison of average surface Nusselt number for combined geometries and of traditionally fed pedestal arrays with variable tip clearance [108] versus mean channel Reynolds number.
Chapter 10  Conclusions

Driven by a desire to increase performance and reduce operational costs through increased propulsive efficiency, power output and lifespan, continual developments have been made to the design of gas-turbine engines throughout their history. This search for constant improvement has promoted a gradual rise in turbine entry temperatures (TET) over the years, to achieve increased specific thrust. An undesirable consequence of high TET is an increased formulation of pollutants, specifically oxides of Nitrogen (NOx). Increasingly strict regulations on levels of NOx and the other pollutants have prompted a significant push in the consideration of reducing emissions. One means of reducing NOx emissions being explored is to use lean burn combustion, operating at an increased air/fuel ratio at which NOx emissions are reduced. In addition to increasing NOx output, high TETs require constant cooling of combustor liner and turbine components. Lean burn combustion presents complications to cooling due to the increased proportion of air entering into the fuel injector reducing the amount available for cooling purposes.

For double skin combustor liners, the inner liner acts as a heat shield protecting the outer structural shell from the high combustor temperatures. These are typically cooled using arrays of pedestals between the skins in combination with film cooling slots, for hot side cooling. The smaller available mass of available air in lean burn combustion requires a change from current practice, to geometries of reduced porosity. Traditional arrangements have been replaced by arrays of impingement and effusion holes for cold side forced convection cooling and hot side film cooling respectively. Complications of an impingement-effusion geometry to lean burn, combustor liner cooling include difficulty in cooling fastening studs and spacers and concerns over the adverse effect of film cooling on the production of NOx and CO emissions.

Whilst research has been conducted into the heat transfer performance of the various cooling geometries adopted for lean burn combustion applications, little research has been conducted into the effect of structural blockages on performance. This thesis has presented an investigation into the cold-side heat transfer performance of an impingement-effusion cooled model of a full lean burn combustor liner tile, targeting evaluation of the effects of structural blockages on localised cooling performance. A separate branch of investigation has been conducted into zero effusion cooling geometries for application to the same
geometry. For this investigation a parametric study has been conducted for geometries partly based on a representative sector of the full tile model.

10.1 Full-Tile Model Impingement-Effusion Cooling

The full tile test facility features a laboratory-based model of a complete combustor tile built to a geometric scale-factor of 7.5. The modelled tile, based on a representative outer annulus cooling tile from early EFE (Environmentally Friendly Engine) combustor designs, measures approximately 1.2×1 m in plan form. In order to simplify manufacture, the tile model is flat and therefore does not simulate the curvature present in the annular engine geometry; this however, is considered to represent a minor effect on the coolant flow. An impingement-effusion cooling system was adapted for the tile inspired from a previous pattern used in ANTLE as no confirmed EFE cooling pattern existed at the time of rig design. The investigation was targeted at identifying the impact of localised structural blockages on the cooling performance of the tile in the affected regions and also investigating the impact on cold-side cooling performance of pressure blockages to flow exiting through the hot-skin. This second area of investigation was in response to observed static pressure fluctuations occurring within the combustion chamber as a result of lean burn combustion. Structural blockages were presented in terms of fastening studs and spacers, with additional large blockage presented by the presence of two ports. The blockage presented does not just consist of protrusion into the cavity between the hot and cold skins, but also to the obstruction presented to the placement of impingement holes. Investigations into the effect of blockages on local cooling performance also incorporated an analysis of the performance of dedicated angled jets for the purpose of stud cooling. Pressure blockage was modelled in spanwise bands across zones of the cooling tile, simulated through the use of restrictor plates downstream of the hot skin.

The investigation into the effect of structural blockages on cooling performance in a full tile setting identified regions in which significantly reduced cooling performance was displayed. The blockage presented by the fastening studs reduced the presence of cooling holes in the region, with \( Nu \sim 0.6 \times \) values typically measured for the unobstructed array. Heat transfer performance of angled impingement holes, deployed for direct cooling of studs also demonstrated worse performance than the normally aligned jet holes. Comparison of maximum \( Nu \) at jet impingement location indicated that for angled jets \( Nu \sim 0.55 \times \) typical. These findings indicate that further consideration needs to be given to methods of cooling the
stud region. Little consistent evidence was found of any similar reduction of heat transfer performance due to the presence of the two ports under a uniform exit static pressure condition.

Effects of pressure blockage simulation highlighted minor increases in cold side heat transfer performance under pressure condition B1. This was attributed primarily to reduced passage of flow through the target plate, increasing interaction of jet air with surface. Negative impact of exit static pressure blockage was much more pronounced, but only in select regions where large crossflow was seen to develop. The greatest impact was seen for exit static pressure condition B2, for which flow through both Zones 1 and 2 was obstructed. The largest crossflow was observed in measurement region Stud 2, due to accumulation of spent air from the first two zones. Crossflow was observed to rapidly accelerate over the course of a single measurement region due to channelling of the flow between the two modelled ports. The impact of this large crossflow in region Stud 2 significantly impaired heat transfer performance such that $Nu$ under pressure condition B2 was $\sim 0.7 \times Nu$ that under ESP B0. Observations of heat transfer measurements in Zone 3 indicated a rapid dissipation of crossflow following removal of the obstruction, with magnitude and profile of $Nu$ contours returning to uniform conditions over the course of measurement region Array 2. This suggests that the presence of large pressure fluctuations within the combustion chamber will have its most significant effect if exit flow is consistently obstructed over a significant portion of the tile, although restrictions on flow exit will impact on film cooling coverage of the hot side of the tile.

In addition to investigating the implications of structural and pressure blockages on heat transfer performance, an evaluation into the general, unobstructed array was of interest. As many $Nu-Re$ correlations of heat transfer performance are calculated from benchmark tests of regular arrays; an evaluation of the performance measured from a representative tile could establish any deviations from such predictions derived within a more controlled environment. It was identified that results from the Tile-Rig displayed a much higher $Re$ dependency than values encountered in the literature with measurements indicating $Nu \propto Re^{0.9}$. Due to the deviation of the $Nu-Re$ relationship from typically accepted values it is considered worthy of further investigation to verify the behaviour and to identify the source of the increased $Re$ dependency when compared with similar geometries.
10.2 Sector-Rig Model Impingement-Pedestal Cooling

Interest in assessing heat transfer performance in zero-effusion geometries has been inspired by experimental results indicating an increase in combustive efficiency and a reduction in NOx output in response to a reduced number of effusion holes. Removing effusion holes from the cooling geometry represents a significant challenge to providing the necessary cooling performance. The Sector-Rig Test Facility was developed to study impingement-based cooling systems in the absence of effusion holes in order to quantify both aerodynamic and cold-side heat transfer performance parameters. The test facility was used for investigation into cooling geometries that yield enhanced impingement performance, particularly in the presence of this strong cross-flow within the cavity between the two skins. In order to utilise the unavoidable cross-flow investigations are focused on combinations of impingement jet and pedestal features combined in hybrid geometries.

The investigation into hybrid geometries consisted of a parametric investigation into the impact of pedestal height \((H/D)\) and density \((X/D)\) upon cooling performance. The cooling geometries investigated using the Sector-Rig are based on dimensions from the EFE combustor liner, and the ANTLE inspired impingement array used within the Tile-Rig. A geometrical scale factor of 17 is used compared to engine scale geometry. The investigated cooling patterns use the same impingement jet array with eleven rows of impingement holes, each row containing five impingement holes of diameter 12\(\text{mm}\). This hole size is equivalent to an engine scale diameter of 0.7\(\text{mm}\), as used in the Tile-Rig. The space between the jets within the array \((x/D=7.1, y/D=6.15)\) is also comparable to those used in the Tile-Rig. The basic arrangement under investigation incorporates pedestals of various heights, arranged into equilateral triangular arrays beneath the impingement-jet array.

In addition to the parametric investigation into the impact of \(X/D\) and \(H/D\) on heat transfer performance, an original geometry has been proposed combining full and quarter height pedestals \((H/D=2.5\) and 0.63 respectively\) in the same array. It was theorised, that by positioning pedestals spanning the full height of the channel upstream of impingement holes, the issuing jets may be shielded from the developing cross-flow and improving heat transfer performance in the downstream region of the channel. By offering a reduced restriction to the cross-flow, it was predicted that the developing cross-flow will be diverted into channels, between the full height shielding pins and the impingement jets. In these channels the short pedestals will interact with the developing cross-flow to increase the local \(h\). It was predicted
that the relatively small blockage presented by the short pedestals of $H/D=0.63$ would result in an increase in pressure drop that would be small compared with that from the impingement jet array and the shielding pins.

The ultimate objective of the parametric study is to evaluate whether the tested geometries could potentially be applied to a cooling tile similar to that tested on the full-tile test facility and whether the results indicate that double skin combustor liners employing zero effusion cooling methods may be pursued as a viable option in lean burn gas-turbine engines.

Investigations into heat transfer performance revealed promising performance for the newly proposed ‘Shielded Impingement’ strategy of impingement-pedestal cooling. The parametric study indicated the largest surface $Nu$ were displayed by Shielded Impingement geometry S4b, followed by Shielded Impingement geometry S4c. Improvement in $\overline{Nu}$ was primarily in regions of high crossflow, with performance channel inlet comparable to that for short pedestal geometries S2c and S3c. Improved heat transfer performance was achieved with less than 20% of pressure loss occurring within the channel and for only marginally higher values than for the shortest lowest blockage pedestal case. Results indicated that ‘Shielded Impingement’ cooling strategy offers potential for improved cooling performance over geometries of uniform height. Through optimisation of design heat transfer performance could be further improved.

Heat transfer performance for the remaining geometries could be ranked according to increasing height and density of pedestal spacing indicating a decrease in heat transfer performance. The decreasing heat transfer performance of the full height arrays was combined with greater pressure losses due to the increased channel blockage. For the remaining geometries similar levels of channel pressure loss were displayed, with pressure losses across the impingement plate the largest contribution to total geometry pressure loss.

Measurements of pedestal $Nu$ for select geometries indicated a reduced proportional contribution of pedestals to heat transfer performance from the typical reported behaviour, for which $Nu_{ped}$ has been identified as greater than $Nu_{wall}$. For the geometries tested it was observed that typically, $Nu_{ped} \leq Nu_{wall}$ although it is unclear to what extent this is due to reduced pedestal heat transfer, or to increased surface heat transfer.

Comparisons of the heat transfer performance for ‘Shielded Impingement’ geometries were conducted against traditional impingement and pedestal geometries. In relation to pedestal arrays, improved performance was apparent due to the significantly lower pressure.
loss incurred by the Shielded Impingement arrangement allowing its application over a significantly increased area than for pedestal geometries. For comparisons with traditional impingement cooling, a correlation was used to model a representative impingement array. The correlated Nusselt numbers were larger than those for the Shielded Impingement geometry. Considering the streamwise variation of heat transfer performance indicated a large reduction in $Nu$ with increasing streamwise position for the impingement array, as crossflow increases. Predicted performance saw surface heat transfer for the Shielded Impingement geometry exceeding that for the smooth-impingement arrangement over the final two jet rows. Considering this and accounting for additional heat transfer provided by the extra surface area presented by the pedestal array it is considered that the application of Shielded Impingement to cold side cooling of combustor liners will offer enhanced performance.
References


