An Experimental Study Of The Unsteady Heat Transfer Process
In A Film Cooled Fully Scaled Transonic Turbine Stage

by

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Abstract

In the present study, the physical phenomena that influence the external heat transfer to the rotor blades of a high pressure axial turbine is experimentally investigated. The experimental apparatus used was the MIT BlowDown turbine transient facility. This facility is a fully scaled rotating turbine stage with all the non-dimensional quantities representing the engine environment. Among the analysis tools, a two-dimensional, time-accurate viscous computational code (UNSFLO) was extensively used. Time-accurate comparisons of experimental and numerical surface heat transfer results are used to explain their unsteady nature with reference to the external flow conditions (shocks, wakes, etc.). An approach for the scaling of the rotor blade heat transfer with respect to the external flow Reynolds number is suggested in which different parts of the blade (leading edge, suction and pressure surfaces) are scaled separately. An empirical correlation is suggested which seemed to collapse the pressure surface heat transfer. This correlation is compared against other turbine blade measurements reported in the literature.

The experimental facility was modified to allow for the addition of coolant fluid at the engine matched conditions. The turbine stage was initially tested with the Nozzle Guide Vane trailing edge coolant ejection. The results showed that the pressure surface heat transfer rose when compared to the no injection results.

The fully cooled turbine stage (with rotor blade film cooling and nozzle guide vane trailing edge coolant injection) was tested at the engine representative conditions. It is the first time that the unsteady film cooled rotor heat transfer has been measured. The film cooled surface heat transfer measurements at 3 span-wise position provide a database for future comparisons. The film cooled data were compared against the uncooled results at the mid and tip span-wise positions. On the suction surface, the coolant film provided good protection of the surface and high film effectiveness measurements were observed. It was shown that the rotor time-averaged suction surface heat transfer was considerably lower than the steady state cascade measurements. On the pressure surface, the film cooling effectiveness was typically low and film lift off from the surface was occasionally observed. On the suction surface, the form of the time-resolved film cooled heat transfer seemed to be different from the uncooled data. It was shown that the interaction between the rotor and stator resulted in an unsteady blowing from the coolant holes, which resulted in an unsteady coolant film effectiveness. Using a simple model of the coolant flow through the hole combined with a flat plate film cooling correlation, the unsteady nature of the film cooled heat transfer was captured.

Thesis Supervisor: Professor A.H. Epstein
To my parents and my dear wife whose love and constant support were always with me.
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<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>coolant injection angle relative to normal</td>
</tr>
<tr>
<td>$A(\omega)$</td>
<td>Fourier coefficient at frequency ($\omega$)</td>
</tr>
<tr>
<td>$a, b, n$</td>
<td>constants</td>
</tr>
<tr>
<td>$a_1, x_g$</td>
<td>experimentally determined correlation constants</td>
</tr>
<tr>
<td>$ac$</td>
<td>speed of sound in coolant</td>
</tr>
<tr>
<td>$C$</td>
<td>axial chord</td>
</tr>
<tr>
<td>$c$ from $\sqrt{\rho c k}$</td>
<td>thermal capacity of kapton poly-amide sheet</td>
</tr>
<tr>
<td>$C_p$</td>
<td>pressure coefficient</td>
</tr>
<tr>
<td>$D$</td>
<td>cylinder diameter</td>
</tr>
<tr>
<td>$\Delta T_E$</td>
<td>indicated temperature difference between upper and lower sensors during post-run calibration</td>
</tr>
<tr>
<td>$\delta$</td>
<td>boundary layer thickness</td>
</tr>
<tr>
<td>$d$ from $k/d$</td>
<td>thickness of kapton poly-amide sheet</td>
</tr>
<tr>
<td>$\delta^*$</td>
<td>displacement thickness</td>
</tr>
<tr>
<td>$\epsilon$</td>
<td>mean convection rate of coolant film as a fraction of freestream velocity</td>
</tr>
<tr>
<td>$\phi$</td>
<td>perturbation potential</td>
</tr>
<tr>
<td>$f(x)$</td>
<td>function of length</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>specific heat ratio</td>
</tr>
<tr>
<td>$\Gamma$</td>
<td>Goertler number</td>
</tr>
<tr>
<td>$G(\omega)$</td>
<td>damping coefficient</td>
</tr>
<tr>
<td>$\eta$</td>
<td>effectiveness</td>
</tr>
<tr>
<td>$h$</td>
<td>heat transfer coefficient</td>
</tr>
<tr>
<td>$k$ from $\sqrt{\rho c k}$</td>
<td>thermal conductivity of kapton poly-amide sheet</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>modified reduced frequency</td>
</tr>
<tr>
<td>$L$</td>
<td>length of coolant slot</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>wave length</td>
</tr>
<tr>
<td>$M$</td>
<td>Mach number</td>
</tr>
<tr>
<td>$\mu$</td>
<td>viscosity</td>
</tr>
<tr>
<td>$\nu$</td>
<td>kinematic viscosity</td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number based on axial chord, difference between the inlet total and wall temperatures and main gas conductivity at the wall</td>
</tr>
<tr>
<td>$P$</td>
<td>pressure</td>
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<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
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<tr>
<td>$Q$</td>
<td>heat flux</td>
</tr>
<tr>
<td>$\theta$</td>
<td>momentum thickness</td>
</tr>
<tr>
<td>$R$</td>
<td>radius of curvature</td>
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<tr>
<td>$\rho$ from $\sqrt{\rho c k}$</td>
<td>density of kapton poly-amide sheet</td>
</tr>
<tr>
<td>$\rho$</td>
<td>gas density</td>
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<tr>
<td>$Re$</td>
<td>Reynolds number</td>
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<tr>
<td>$Re_c$</td>
<td>$Re$ based on axial chord isentropic exit Mach number and properties at exit condition</td>
</tr>
<tr>
<td>$Re_D$</td>
<td>$Re$ based on cylinder diameter</td>
</tr>
</tbody>
</table>
$S$  
**surface length**

$s$  
**fractional wetted surface**

$\sigma^2$  
**variance**

$S_0$  
**reference position at the exit of slot**

$S_u$  
**upper sensor scale factor**

$(t)$  
**time**

$T$  
**temperature**

$\tau$  
**ratio of boundary layer diffusion time scale to external periodic oscillation time scale**

$U$  
**velocity**

$u$  
**perturbation velocity**

$V$  
**voltage**

$\Omega$  
**reduced frequency**

$\omega$  
**oscillation frequency**

$W_g$  
**surface length of sensor/ non-dimensionalized by cylinder radius**

$X$  
**projected length scale**

$\xi$, $\beta$  
**non-dimensional blowing parameter**

$x$  
**length scale**

$Z$  
**a reference temperature**

**Subscripts:**

0  
**mean conditions**

ad  
**adiabatic**

aw  
**adiabatic wall**

c  
**coolant**

cal  
**from bath calibration**

des  
**at the design conditions**

f  
**film cooling gas**

fc  
**film cooled**

l  
**lower surface**

m, g  
**main gas**

r  
**recovery**

t  
**total condition**

$T_{rel}$  
**total relative**

u  
**upper surface**

w  
**wall, blade surface**

$\infty$  
**for upstream conditions**

**Superscripts:**

$'()$  
**perturbation quantity**

$()$  
**time averaged condition**
Chapter 1

1. Introduction and Literature Survey

1.1 Motivation

Advances in modern aircraft gas turbine technology over the past four decades have led to an increase in overall engine pressure ratio and turbine entry temperature, and this has lead to large gains in the specific thrust of engines. A measure of these gains is the maximum specific thrust which for an ideal cycle turbojet is given [1] by:

\[ \text{Max Specific Thrust} = M_0 \left( 1 + \left( \frac{T.E.T.}{T_0} \right)^{1/2} \cdot \left( \frac{\gamma - 1}{2} \cdot M_0^2 \right) \right)^{1/2} - 1 \]  

Eq (1.1.1)

where \( M_0 \) is the flight Mach number, \( T_0 \) is the ambient temperature, \( \gamma \) is the specific heat ratio of the gas and T.E.T. is the turbine entry temperature. It is observed that raising the turbine entry temperature results in an increase in the specific thrust of the engine. A complete analysis of the relation between performance and turbine inlet temperature has been discussed by Bagby[2]. The trend in the operational turbine entry temperature [3] is shown in Figure (1.1). A key feature of Figure (1.1) is that current turbine entry temperature (1900 K) is considerably higher than state of the art of maximum metal temperature (1300 K). Thus, even though the improvements in the materials technology have resulted in the use of higher temperature super-alloys [4], the major increase in the turbine entry temperature can be attributed to the development of cooling technology and improved aerodynamic design.

The need to cool the blading and endwalls of the turbine has resulted in considerable research and development including many different cooling schemes [5], such as internal convection
cooling, internal impingement cooling, transpiration cooling and film cooling. In stationary gas turbines, liquid internal convection cooling has been shown to provide the method with the highest cooling levels \(^6\), but for aircraft propulsion gas turbines, the principal means of cooling is the use of air bleed from the later stages of the high pressure compressor.

The requirement of maintaining high aerodynamic cycle efficiency for the turbine has resulted in the need to minimize the fraction of the cooling air for the maximum allowable metal temperature and thermal stress. For the turbine blades, control of the thermal stress limits the metal temperature gradient in the airfoil. Detailed knowledge of the heat transfer processes to the metal surface (both external and internal for cooled blades) is required to allow the computation of the temperature distribution within the blade. Temperature gradients within the metal result in thermal stress and reduces the operational life of the blade. The requirement of lower temperature gradient, and hence lower thermal stress, is more severe in turbine rotor blades than in the stationary vanes.

### 1.2 Similarity in Heat Transfer

The convective heat transfer to the turbine blades is controlled by the dynamics of the flow around the profile. In order to study the heat transfer around the blade, the physical parameters that influence the transport of mass, momentum and energy of fluid flow are introduced. Fluid flow is described by the thermal equation of state and the conservation of mass, momentum and energy. The description of the flow in Cartesian indicial notation is given \(^7\) by

\[
\frac{\partial \rho}{\partial t} + \sum_{j=1}^{n} \frac{\partial \rho u_j}{\partial x_j} = 0
\]

\text{Eq (1.2.1)}
Momentum
\[
\frac{\partial (\rho u_j)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ik}}{\partial x_k} + \rho f_i \tag{1.2.2}
\]

Energy
\[
\frac{\partial (\rho H)}{\partial t} + \frac{\partial (\rho u_i H)}{\partial x_j} = \frac{\partial p}{\partial t} + \frac{\partial (u_j \tau_{ik} - q_k)}{\partial x_k} + \rho f_i u_i \tag{1.2.3}
\]

State
\[g(p, T, \rho) = 0 \tag{1.2.4}\]

For an isotropic fluid in which stress is a linear function of the rate of strain and heat transfer is a linear function of the temperature gradient,

\[
\tau_{ij} = \frac{1}{3} (\kappa - 2 \mu) \delta_{ij} \frac{\partial u_i}{\partial x_1} + \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \tag{1.2.5}
\]

where
\[q_j = -k \frac{\partial T}{\partial x_j} \tag{1.2.6}\]

\[H = c_p T + u^2/2 \tag{1.2.7}\]

Here, \(t\) = time, \(\rho\) = density, \(p\) = pressure, \(T\) = temperature, \(u_i\) = the \(i\)th component of velocity, \(x\) = spatial coordinate, \(\mu\) = viscosity, \(\kappa\) = bulk viscosity, \(k\) = conductivity, \(c_p\) = specific heat capacity at constant pressure, \(q\) = heat flux, \(f\) = body force per unit mass, and \(\delta_{ij} = 1\) if \(i=j\), \(\delta_{ij} = 0\) if \(i \neq j\). The specific heat capacity at constant pressure \((c_p)\) is nearly constant [7].

The conservation equalities of Equations (1.2.1), (1.2.2) and (1.2.3) describe the exact motion of the fluid. In the particular cases where the bulk of the inertial forces are greatly larger than viscous shear forces, certain simplifying approximations can be used. The fluid may be considered inviscid (where the inertial terms dominate) except in a thin layer close to the solid surface, as suggested by Panton [8]. In this thin layer, the diffusion terms balance the inertial terms. This thin shear layer is called the boundary layer. By comparing the relative order of
magnitude, it can be shown [9] that for a shear layer with a thickness much smaller than the streamwise length scale of the solid surface, the gradient of the some of the terms (e.g. shear stress in the streamwise direction) is much smaller than those across the layer and as such can be ignored. With the thin shear layer assumptions for a two-dimensional flow in absence of body forces, the boundary layer equations in Cartesian coordinates are written as

Continuity
\[
\frac{\partial p}{\partial t} + \frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} = 0
\]
Eq (1.2.8)

x- momentum
\[
\rho \frac{\partial u}{\partial t} + \rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial y} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial y}\left(\mu \frac{\partial u}{\partial y}\right)
\]
Eq (1.2.9)

Energy
\[
\rho c_p \frac{\partial T}{\partial t} + \rho c_p u \frac{\partial T}{\partial x} + \rho c_p v \frac{\partial T}{\partial y} = \frac{\partial p}{\partial t} + u \frac{\partial p}{\partial x} + \frac{\partial}{\partial y}\left(\frac{k}{\partial y} \frac{\partial T}{\partial y}\right) + \mu \left(\frac{\partial u}{\partial y}\right)^2
\]
Eq(1.2.10)

x and y correspond to the \(x_i\) and \(x_j\), \(u\) and \(v\) correspond to \(u_i\) and \(u_j\), \(x\) is in the streamwise direction and \(y\) is the normal to the plane of the shear layer. The momentum and the energy equations are coupled through the pressure and fluid property terms.

Dimensional consideration of the Equations (1.2.8 through 1.2.10) suggests that the temperature profile across the boundary layer is dependent upon the geometry, time, gas properties, fluid velocity and pressure. Assuming that the Prandtl number (\(Pr = \mu c_p / k\)) is constant, for a body of surface length \(L\), local wall temperature \(T_w\), external fluid velocity \(U\) and gas conductivity \(k\), the surface heat flux is written as

\[
\dot{Q} = \left(\frac{k}{\partial y}\right)_{\text{at wall}} = f^\alpha \left[\frac{x}{L}, \frac{U}{L}, Pr, \frac{\rho UL}{\mu}, \frac{\mu(T)}{\mu(T_w)}, \frac{L}{\rho U^2 \partial x}\right]
\]
Eq(1.2.11)
In general, the dependence of the surface heat flux to the right hand side terms of Equation (1.2.11) is non linear. To determine the surface heat flux (numerically or experimentally) every term on the right hand side of the Equation (1.2.11) must be correctly simulated. Using the following assumptions, the form of Equation (1.2.11) can be simplified. For a steady constant pressure boundary layer, it can be shown [10] that by transforming the variables \((x, y)\) to \((x, \psi)\) where \(\psi\) is given by Equation (1.2.12), the momentum and energy equations are written as

\[
\frac{\partial \psi}{\partial x} = - \rho v, \quad \frac{\partial \psi}{\partial y} = \rho u \quad \text{Eq (1.2.12)}
\]

Momentum (from Equation (1.2.9))

\[
\frac{\partial u}{\partial x} = \frac{\partial}{\partial \psi} \left( \mu \rho \frac{\partial u}{\partial \psi} \right) \quad \text{Eq (1.2.13)}
\]

Energy (from Equation (1.2.10));

\[
\frac{\partial T}{\partial x} = \frac{1}{Pr} \frac{\partial}{\partial \psi} \left( \mu \rho \frac{\partial T}{\partial \psi} \right) + \frac{\mu \rho u}{cp} \left( \frac{\partial u}{\partial \psi} \right)^2 \quad \text{Eq (1.2.14)}
\]

The influence of the property variation and compressibility on the transfer of heat and momentum is confined to a single term, namely the product of density and viscosity \((\rho \mu)\). The gas density at constant pressure is inversely proportional to the temperature. The viscosity of air is approximated by Sutherland’s law:

\[
\frac{\mu}{1.45 \times 10^{-6}} = \frac{T^{3/2}}{T + 110} \quad \text{Eq (1.2.15)}
\]
\( \mu \) has a unit of Kilogram per meter per second and \( T \) has the unit of Kelvin. An approximate power law to Sutherland's law may be found at any mean temperature. Viscosity of air is approximately proportional to temperature to the power of 0.76 at a temperature of 300 K and to the power of 0.5 at 1300 K (typical turbine blade wall temperature).

For the particular case of small temperature variation (when compared to the absolute level) the property variation is negligible and Equation (1.2.15) is linear in temperature. Furthermore, for an isothermal surface (ignoring the dissipation term in the energy equation), the velocity and temperature profiles become similar. In this case, the form of the temperature profile is independent of the absolute value of the wall temperature, and heat flux at the surface is given by the Rate Equation (1.2.16):

\[
\text{Heat Flux per unit area} = \dot{Q} = h (T_m - T_w) \quad \text{Eq (1.2.16)}
\]

For the case of variable wall temperature, the heat transfer is a function of the upstream history of the boundary layer \([11]\) and the local definition of the heat transfer coefficient becomes inappropriate. In the present context, however, the turbine blades are typically designed to operate at or near isothermal condition and as such the use of the Rate Equation is a reasonable approximation.

### 1.3 Turbine Rotor Heat Transfer

Convective heat transfer accounts for most of the heat load to a turbine rotor blade. With reference to Equation (1.2.11), the convective heat transfer to a turbine blade surface is a function of geometry, unsteadiness, Reynolds number and the flow acceleration. Daniels \([12]\) provides a comprehensive review of the convective heat transfer in a turbine environment. The
influence of flow Reynolds number and the unsteadiness are of particular interest in the present study and are discussed in this section.

1.3.1 Unsteady Processes

The flow in a turbomachine is unsteady [1], with many sources generating different forms of unsteadiness in the flow [13]. The flow that passes through the nozzle guide vanes into the rotor of the first stage of a high pressure turbine experiences many sources of unsteadiness. These unsteady variations are caused by any stator exit temporal or spatial variations (seen as temporal variation by a rotor blade). Some of these sources and the corresponding length and time relative to a typical turbine blade geometry (C = chord) and throughflow time (C/U, where U = throughflow velocity) scales are

<table>
<thead>
<tr>
<th>Type of Unsteady Flow</th>
<th>Typical Length Scale</th>
<th>Typical Time Scale</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wakes from Upstream</td>
<td>C</td>
<td>C/U</td>
</tr>
<tr>
<td>Potential Field Interaction</td>
<td>C</td>
<td>C/U</td>
</tr>
<tr>
<td>Circumferential Non-Uniformity</td>
<td>Circumference</td>
<td>Circumference/Rot.Speed</td>
</tr>
<tr>
<td>Turbulence</td>
<td>Less than C</td>
<td>Less than C/U</td>
</tr>
</tbody>
</table>

Circumferential non-uniformities are typically caused by combustor instabilities and geometric non-uniformities of the combustor and the nozzle guide vanes upstream. The freestream turbulence is generated by vigorous mixing of the fluid in the combustor chamber.

For a uniform inflow turbine rotor blade, unsteadiness is of two forms [14]: periodic fluctuations (generated by the relative motion of the rotor and stator blade rows) and random fluctuations (turbulence). For a turbine rotor blade, the periodic unsteadiness is divided into
two parts: potential field influence and wake interaction. The random fluctuations are generated by many sources; among them, boundary layer instability and freestream turbulence.

The important physical parameter that characterizes the influence of periodic unsteadiness is the time scale of unsteadiness relative to the convection and diffusion time scales. The ratio of the temporal variation to the convection term is approximately given by Equation (1.3.1). This ratio is called the reduced frequency.

\[
\frac{\frac{\partial u}{\partial t}}{\frac{\partial u}{\partial x}} \sim \frac{(\text{Length Scale})(\text{Frequency of Unsteadiness})}{\text{Throughflow velocity}} = \frac{C \omega}{U} = \Omega
\]

Eq (1.3.1)

The blade chord and the throughflow velocity are used for a typical length and velocity scale. Reduced frequency for a turbine rotor/stator interaction flow (based on blade passing frequency) is typically of the order of unity or higher; i.e. periodic unsteadiness is important.

In this section, the previous studies of the influence of unsteadiness on turbine heat transfer are presented and divided into three parts: transition and turbulence, potential field interaction, and wake interaction.

**Transition and Turbulence:**

Even when the inviscid part of the flow is steady, the boundary layer can undergo a viscous shear instability, resulting in a turbulent state within the layer. This process is called transition, and has been extensively studied. Here, the criterion for the stability of the boundary layer is
introduced and the influence of other flow features (such as the freestream turbulence) on the transition process are reviewed.

The stability of a boundary layer is studied using a small perturbation analysis of the unsteady Navier-Stokes equations. By applying a small perturbation potential of the form given in Equation (1.3.2), the equations of motion for a two-dimensional incompressible boundary layer can be [9] written as in Equation (1.3.3).

\[
\phi = \phi(y) \exp[i(\alpha x - \beta t)] \quad \text{Eq (1.3.2)}
\]

\[
\beta = \beta_R + \beta_I, \quad \text{where } \beta_R \text{ and } \beta_I \text{ are the real and imaginary parts of the complex wave speed } \beta.
\]

\[
(u - \beta/\alpha) \left[ \frac{d^2\phi}{dy^2} - \alpha^2 \phi \right] - \phi \frac{d^2u}{dy^2} = \frac{1}{i\alpha \text{Re}} \left[ \frac{d^2}{dy^2} - \alpha^2 \right] \phi^2 \quad \text{Eq (1.3.3)}
\]

\(\bar{u}\) is the mean velocity profile and \(\text{Re}\) is the Reynolds number based on the boundary layer thickness. Equation (1.3.3) is known as the Orr-Sommerfield equation with the no slip boundary conditions at the wall \((y=0)\). This equation is linear and for a given mean velocity profile \(\bar{u}(y)\), the eigenfunction \(\phi(y)\) and the eigenvalue \(\beta\) can be evaluated. For a positive value of \(\beta_I\), the perturbation is amplified and the boundary layer is unstable. The value of \(\beta_I\) is a function of the Reynolds number and disturbance wavelength, and the marginal stability region is given by the curve in which \(\beta_I\) is equal to zero. The marginal stability region for the Blasius boundary layer profile [9] and a boundary layer with an adverse pressure gradient is shown in Figure (1.2). Within the loop, the perturbation waves (known as Tollmein- Schlichting waves) grow and soon become three-dimensional. For a Blasius profile, at approximately a value of 1480 for the critical Reynolds number (based on the boundary layer thickness), the boundary layer first becomes unstable.
The modelling of the boundary layer transition has been known to be essential to the heat transfer prediction. An important factor that influences the onset and the extent of the transition is the level of the free-stream turbulence level. In the gas turbine environment, free-stream turbulence intensities (defined as the RMS of the fluctuations divided by the mean of the velocity) of 17 percent have been observed \cite{15} at the exit of the combustor. Acceleration of the flow through the nozzle guide vanes reduces the magnitude of the turbulence intensities to a lower level (approximately 3 to 5\%) at the inlet to the rotor. Rued and Wittig \cite{16} demonstrated that for the case of a flat plate at constant pressure, an increase in the free-stream turbulence intensities from 1\% to 5\% results in a lowering of the transition onset Reynolds number by a factor of two and a shortening of the length of the transition by a factor of three. They concluded that for a high level of freestream turbulence (above 5\%), onset of transition is very close to the leading edge in a typical turbine blade.

**Potential Field Interaction:** The potential field associated with the flow around a blade profile extends both upstream and downstream, decaying exponentially upstream for subsonic flow with a length scale of the order of the chord. In the case of a rotor and stator moving relative to each other, the pressure field from the stator blade influences the flow around the rotor and vice versa. A measure of the importance of this interaction is the ratio of the axial gap between the rotor and stator to the chord (typically 0.3 for a high pressure turbine). For an axial gap which is much smaller than the chord, the potential field interaction can be important. The rotor/ stator potential interaction results in a fluctuating pressure and velocity field in the inviscid region of the flow around the blade.

The influence of a periodic external velocity (superimposed on a mean flow) on the boundary layer behavior for simple geometries has been extensively studied. For an oscillating laminar boundary layer, the balance of the temporal and diffusion terms (of energy and momentum), is characterized \cite{17} by the length scale \( \sqrt{\nu/ \omega} \), known as the Stokes length. Here, \( \nu \) is the
kinematic viscosity and \(\omega\) is the frequency of oscillation. The relative size of the Stokes length to the steady state boundary layer thickness, given in Equation (1.3.4), is a measure of unsteadiness of the boundary layer. For a laminar boundary layer,

\[
\sqrt{\frac{\omega^2}{v}} \sim \sqrt{\frac{\omega x}{U}} \quad \text{Eq (1.3.4)}
\]

\(x\) is the streamwise length scale and \(U\) is the mean freestream velocity. The parameter \((\omega x/U)\) is a form of the reduced frequency. For a value of \((\omega x/U)\), which is much less than unity, the boundary layer is quasi-steady; while for \((\omega x/U)\) of the order of unity, the unsteadiness is important. For the periodic flow associated with the rotor/ stator potential interaction, the range of this reduced frequency is approximately \(0 < (\omega x/U) < 2\).

By considering a convected unsteady disturbance on a constant pressure flat plate, Patel [18, 19] showed that for the range of \(1.6 < (\omega x/U) < 6.7\), the time averaged development of a laminar and a turbulent boundary layer is unchanged from steady state. In another study [20], in the absence of transition or separation, the oscillation of the freestream velocity did not influence the mean boundary layer development for a typical turbine rotor. The influence of the periodic external flow on a flat plate boundary layer transition was experimentally studied by Obremski and Fejer [21]. They showed that the onset of the transition is primarily a function of a Reynolds number based on the amplitude and the frequency of the oscillation.

In a transonic turbine, shock waves are generated from the upstream nozzle guide vanes, which are unsteady in the rotor frame of reference. The strength of these shocks depends on the nozzle guide vane exit Mach number. Most transonic turbines operate at the guide vane exit Mach number of less than 1.3. Therefore, these shock waves are weak and generally do not result in the separation of the rotor boundary layer. The interaction of shock waves with the
downstream blade is also characterized by the reflection of compression and expansion waves. A thorough review of the previous studies of steady shock interaction with turbulent boundary layers is given by Delery [22].

The unsteady impingement of shock waves on the downstream rotor blades have been the subject of many investigations. Johnson et al [23] presented a model based on a linear unsteady perturbation analysis of the boundary layer. This model accounted for the transient isentropic heating and cooling of the boundary layer by the passing shock and rarefaction waves. This shock passing resulted in high temperature gradients near the surface, and hence large conductive heat transfer in the laminar sublayer of the boundary layer. Johnson et al tested the validity of this model in a linear transonic cascade with simulated shock and wake passing from a rotating bar upstream. By using the measured fluctuating wall pressure around the profile, they were able to predict the unsteady heat transfer rate and found it in good agreement with the measurements.

The effect of the unsteady shock structure on a rotor blade profile was experimentally investigated by Doorly and Oldfield [24], Johnson et al [25] and Ashworth et al [26]. These studies show that the stator shock structure interacts with the rotor downstream resulting in an almost instantaneous (compared to the throughflow time scale) increase in the surface heat flux by many times (typically three to five times) the mean level.

**Wake Interaction:** The relative motion of the blade rows results in an unsteady interaction of a blade row with the wake from the upstream blade row. In contrast to the potential field interaction, where the extent of the influence was limited to a chord length, wakes are convected downstream with a much smaller rate of decay. The wake is characterized by a highly turbulent flow at a lower velocity than the main flow.
The low velocity fluid in the wake tends to lower the incidence angle for the downstream blade row. The rotor blade velocity triangle for the main flow, as well as the wake flow, are shown in Figure (1.3). The 'wake slip' velocity shown in Figure (1.3), results in a migration of wake fluid from the pressure surface on to the suction surface. By tracing the wake fluid (using tracer gas) within a compressor stator passage downstream of a rotor, Kerrebrock and Mikolajczak [27] showed that the wake fluid migrates from the suction to the pressure surface (from pressure to the suction surface for a turbine). By using an inviscid time marching numerical scheme, Hodson [28] predicted the wake generated unsteady flow in a turbine and obtained good agreement with measurement. Hodson's result suggests that away from the boundary layers, the phenomena associated with the rotor/stator wake 'chopping' interactions are dominated by the inviscid rather than viscous effects and an inviscid calculation scheme is sufficient to predict the trajectory of wake fluid.

The high level of freestream turbulence within the wake structure also influences the boundary layer transition and heat transfer processes. Pfeil et al [29] investigated the boundary layer transition on a flat plate being disturbed by the periodic wake structure from an upstream rotating bar. They concluded that the combined influence of the stochastic fluctuations (turbulence level) in the wake and periodic external flow resulted in an early boundary layer transition. Using hot film measurements on the surface of a rotor blade in a rotating low speed turbine facility, Addison and Hodson [30] showed that the boundary layer transition is unsteady and is dominated by the upstream stator wake turbulence.

Doorly [31] studied the wake interaction process in a linear turbine cascade with rotating bars upstream. The bars were scaled to simulate the correct wake velocity deficit and reduced frequency (based on the wake passing period). Doorly's measurements [31] of heat transfer with wake passing at a position near the leading edge on the suction surface for a turbulent (tripped) boundary layer and a laminar boundary layer are shown in Figure (1.4). With the
wake impingement, the laminar boundary layer undergoes a rapid transition and a turbulent patch is formed. This rapid transition, which is coupled to the wake interaction process, results in a high level of heat transfer in regions where the boundary layer might otherwise (in the absence of the wakes) have been laminar. Doorly observed that the effect of the wake impingement on a turbulent boundary layer was comparatively (to the laminar case) small. The same trends were observed by Dong and Cumpsty \([32]\) in a compressor cascade experiment with bar passing.

Wittig et al \([33]\) showed that with a periodic wake passing, the dominant parameters affecting the heat transfer are the ratio of the transient time associated with the wake passage to the time of the blade passing, as well as the difference between the turbulence intensities within the wake and the free-stream flow. Sharma et al \([34,35]\) have suggested a model in which the periodic wake interaction increases the intermittency factor (defined here as the fraction of period the boundary layer is turbulent) of the natural modes of transition. They suggest that this increase in the intermittency factor is a linear function of the fraction of time during which the wake directly impinges on the boundary layer. The comparison of the their model and the measured blade heat transfer in a rotating low speed turbine facility shows good agreement.

1.3.2 Influence of Reynolds Number

Reynolds number effects have an important influence on turbine heat transfer. Generally, increasing the Reynolds number raises the convection of momentum and hence heat transfer (by Reynolds analogy). The heat transfer (non-dimensionalized by the length x, temperature difference between the wall and the main flow, and conductivity of the gas at the wall) for an incompressible constant pressure boundary layer on an isothermal flat plate is given \([9]\) by Equations (1.3.5) and (1.3.6).
For a laminar boundary layer,

\[
\frac{\dot{Q}_x}{k (T_\infty - T_w)} = 0.339 \Pr^{1/3} \Re_{(x)}^{1/2} \quad \text{Eq (1.3.5)}
\]

For a laminar boundary layer,

\[
\frac{\dot{Q}_x}{k (T_\infty - T_w)} = 0.0296 \Pr^{0.4} \Re_{(x)}^{0.8} \quad \text{Eq (1.3.6)}
\]

Pr is the Prandtl number and Re\(_{(x)}\) is the Reynolds number based on the freestream velocity and length from the start of the boundary layer \((x)\). It is seen that in this simple case, depending on the state of the boundary layer, the heat transfer scales with the Reynolds number to the power of 0.5 to 0.8.

The influence of the Reynolds number on turbine heat transfer has been extensively studied \([36,37,38,39]\). It has been shown that the Reynolds number is the most important parameter influencing the position of the transition point and the level of heat transfer rate. The need to account for this strong dependence has prompted researchers to model the influence of the Reynolds number and develop correlations that are used by designers. The influence of the Reynolds number is discussed in detail in Chapter 5.

1.4 Film Cooling Studies

The need to protect surfaces exposed to the high turbine entry temperature has resulted in the development of many different cooling methods. One method introduces a secondary fluid into the boundary layer of the surface to be protected. The introduction of a secondary fluid at discrete locations (series of holes or slots, see Figure (1.5)) along a surface exposed to a high
temperature environment is called film cooling. In a turbine, some fraction of the flow is bled off from the high pressure compressor stage and is used as the secondary fluid. The temperature of the coolant air is well below the metal temperature of the turbine blades, and thus can be used for the internal cooling before being ejected as film cooling. The introduction of the coolant increases the fluid mass within the boundary layer and can be considered to be equivalent to thickening the insulating effect of the boundary layer between the wall and the hot gas and resulting in a lower mean temperature within the boundary layer.

The conventional method of describing film cooling, following that given by Goldstein [40], is discussed in this section. For a constant property flow on an isothermal wall, the Equation(1.2.16) can be used to determine the heat flux. In the case of a film cooled wall, the heat flux is normally described in terms of a local variation in the gas to wall temperature downstream of the holes and a heat transfer coefficient in the presence of injection.

\[
\dot{Q}_{fc} = h_{fc} (T_{ad} - T_w) \tag{1.4.1}
\]

Subscript fc refers to the film cooled condition and \(T_{ad}\) is the adiabatic wall temperature in the presence of the injection. In this scheme, the injection affects both the driving temperature difference (\(T_{ad} - T_w\)) and the heat transfer coefficient. Using this method, detailed knowledge of the two variables (\(T_{ad}\) and \(h_{fc}\)) are required, which are measured independently. \(T_{ad}\) is measured on an adiabatic surface downstream of the injection and is typically specified in terms of an 'adiabatic wall effectiveness'; defined as

\[
\eta_{ad} = (T_{\infty} - T_{ad}) / (T_{\infty} - T_c) \tag{1.4.2}
\]

Subscripts \(\infty\) and \(c\) refer to the main and coolant flow condition. \(\eta_{ad}\) is a function of length scale downstream of the injection, geometry of injection and freestream conditions.
An important parameter of interest is the reduction in the heat transfer in the presence of film cooling. Combining Equations (1.2.16) for the uncooled wall and (1.4.1) for the film cooled case, the ratio of the film cooled to uncooled heat flux is written as

\[ \frac{Q_{fc}}{Q_0} = \frac{h_{fc}}{h_0} \left( \frac{T_{ad} - T_w}{T_{\infty} - T_w} \right) = \frac{h_{fc}}{h_0} \left( 1 - \Theta \eta_{ad} \right) \]

where

\[ \Theta = \frac{T_{\infty} - T_c}{T_{\infty} - T_w} \]

Here, subscript 0 refers to the uncooled condition. The fractional reduction in the heat flux for an isothermal wall is called the 'isothermal film effectiveness'. Experimental measurements of the ratio of the heat transfer coefficient (for Equation (1.4.3)) are obtained from the case when \( \Theta \) is equal to zero. At any wall temperature, coolant gas temperature and main gas temperature, the reduction in the surface heat flux is obtained from the knowledge of two parameters: (1) Ratio of film cooled to uncooled heat transfer coefficients, (2) Adiabatic wall effectiveness.

On an isothermal flat wall under steady state zero pressure gradient conditions and a given injection geometry (pitch/ diameter of holes, inclination angle, etc.), dimensional analysis considerations suggest that adiabatic film cooling effectiveness and the ratio of film cooled to uncooled heat transfer coefficients are dependent upon coolant to main gas mass flux ratio, coolant to main gas momentum flux ratio, length scale downstream of the injection, Reynolds number and the state of the boundary layer upstream of the injection holes.

\[ \eta_{ad} = f_1 \left( I, B, \frac{x}{D}, \text{Re}_D, \frac{\delta^*}{D} \right) \]

\[ \frac{h_{fc}}{h_0} = f_2 \left( I, B, \frac{x}{D}, \text{Re}_D, \frac{\delta^*}{D} \right) \]
where
\[ B = \frac{(p \cdot U)_{\text{Coolant}}}{(p \cdot U)_{\text{Main}}} \quad , \quad I = \frac{(p \cdot U^2)_{\text{Coolant}}}{(p \cdot U^2)_{\text{Main}}} \]
Eq (1.4.7)

\( x \) is the length downstream of the coolant holes of diameter \( D \), \( \delta^* \) is the displacement thickness of the boundary layer upstream of the holes and \( \text{Re}_D \) is the Reynolds number based on the hole diameter and the freestream conditions.

There are several drawbacks in using the above scheme to determine the film cooling behavior in a turbine. This procedure assumes linearity of the energy equation, which is only applicable for cases when the velocity field remains unchanged for any variation in the temperature field. In a turbine application, the coolant and the main temperatures are typically at 900 and 1900 K respectively, resulting in the density of the injected fluid being twice that of the main flow. The assumption of constant properties is no longer valid and the energy and momentum equations are coupled. Another problem with this approach is that at the point of injection, the wall is at the coolant temperature, which is typically lower than the mean metal temperature (around 1300K), and hence the isothermal assumption is no longer valid. Despite these objections, this method of characterizing the film cooling process has been widely used.

1.4.1 Heat Transfer Coefficient

Injection through discreet holes results in a highly three-dimensional interaction of coolant and the freestream fluid. The aerodynamic interaction pattern of discrete coolant jets in crossflow results in the generation of counter-rotating vortices \([41,42]\) from the coolant holes. This addition of fluid through coolant holes results in an increase in the turbulent mixing (increasing \( h_{fc}/h_0 \)), and thickening of the boundary layer (decreasing \( h_{fc}/h_0 \)) when compared to the uncooled case.
The influence of the coolant injection on the heat transfer coefficient was investigated by Ammari et al [43] in a flat plate experiment, and by Horton et al [44] in a turbine linear cascade. Using a heat-mass transfer analogy, Ammari et al [43] measured the ratio of film cooled to uncooled heat transfer coefficients on an isothermal zero pressure gradient flat plate. Their experiments were performed for the range of $0.5 < \textit{M} < 2.0$ (coolant to main mass flux ratio), $0.16 < I < 4.0$ (coolant to main momentum flux ratio) and two injection angles (35 and 90 degrees to the streamwise direction). The laterally averaged ratio of the heat transfer coefficient with and without injection for the range of the tested parameters is shown in Figure (1.6a) for the 35 degree and Figure (1.6b) for the 90 degree (normal) injection angles. Negligible change (less than 5%) due to the variation of the momentum ratio is observed for the normal injection case of Figure (1.6b). In the 35 degree injection case shown in Figure (1.6a), a 66% increase in the momentum flux ratio resulted in an increase (approximately 10% close to the holes, decaying to zero further downstream) in the heat transfer coefficient, except at the highest blowing and momentum ratios ($M = 2.0$ & $I=4.0$). For this high blowing case, the authors report that the coolant film lifted off from the surface. In the range of $0.5 < M < 1.5$ and $0.16 < I < 2.25$, away from the injection region ($x/D > 10$), the ratio of film cooled to uncooled heat transfer coefficients is in the range of $1.0 < h_{fc}/h_0 < 1.2$.

By comparing their results with that of other researchers, Ammari et al [43] showed that for the range of $0.14 < \delta^*/D < 0.4$, $0.34E-4 < \textit{Re}_D < 2.2E-4$ and $x/D > 5$, the influence of $\delta^*/D$ and $\textit{Re}_D$ on the ratio of cooled to uncooled heat transfer coefficient is less than 10%. Ammari et al [43] also demonstrated that in the vicinity of the coolant holes (5 to 8 hole diameter downstream), the lateral distribution of the heat transfer coefficient is highly non-uniform. This non-uniformity was argued to be due to the discreteness of each injected coolant jet, and soon disappeared after the jets interact downstream.
1.4.2- Studies of Effectiveness

The lack of a clear description of the interaction dynamics between the boundary layer and the injected film from holes has been a severe constraint to the development of analytical and computational models of film cooling. The use of semi-analytical correlation formulas based on the ratios of the coolant to the external mass and momentum fluxes has been almost universal, see Goldstein and Haji-Sheik [45] amongst others. These two-dimensional correlations lack accuracy in the highly three dimensional flow in the vicinity of the holes.

Here, an approach for determining the influence of coolant injection on the adiabatic wall temperature based on an energy balance model [40] is presented. In this method, a control volume extending from the origin of the upstream boundary layer (x') to some distance downstream of the injection region (shown in Figure (1.7)) is considered, within which the total mass and enthalpy of the fluid (injected and entrained) are conserved. The flow is steady, two-dimensional, and the injected and the entrained fluid are considered completely mixed with constant properties. Balancing the total mass flow within the control volume

\[ \dot{m} = \dot{m}_1 + \dot{m}_c = \int_0^\delta (\rho u) \, dy \]

Eq (1.4.8)

where \( \dot{m}_1 \) is the entrained mass, \( \dot{m}_c \) is the coolant mass, \( \rho \) is density, \( c_p \) is the specific heat capacity at constant pressure and \( u \) is the velocity within the layer. By assuming a 1/7th power [9] turbulent velocity profile, the rate of entrainment of mass (\( \dot{m}_1 \)) is predicted and is given [40] by

\[ \dot{m}_1 = 0.329 \rho_\infty U_\infty x' Re_x^{-0.2} \]

Eq (1.4.9)
For the adiabatic condition, the balance of enthalpy at any position downstream of the injection point is used to determine a mean temperature ($\overline{T}$) within the boundary layer.

$$\frac{T_{\infty} - \overline{T}}{T_{\infty} - T_c} = \left(1 + \frac{\dot{m}_1 c_{p_{\infty}}}{\dot{m}_c c_{pc}}\right)^{-1}$$

Eq (1.4.10)

where the average temperature is defined by the following expression.

$$T_{\infty} - \overline{T} = \frac{\int_{0}^{\delta} [\rho u c_p (T_{\infty} - T)] dy}{\int_{0}^{\delta} (\rho u c_p) dy}$$

Eq (1.4.11)

Assuming a power law velocity profile and a similar temperature profile, mean temperature $\overline{T}$ is related to the adiabatic wall temperature [40] by Equation (1.4.12). Hence, by combining Equations (1.4.7), (1.4.9), (1.4.10) and (1.4.12), the adiabatic wall effectiveness is given by the Equation (1.4.13).

$$\frac{T_{\infty} - T_{ad}}{T_{\infty} - \overline{T}} = 1.9 \text{ Pr}^{2/3}$$

Eq (1.4.13)

$$\eta_{ad} = 1.9 \text{ Pr}^{2/3} \left(1 + 0.329 \left(\frac{1}{B} \frac{x'}{D}\right)^{0.8} \left(R_{ce} \frac{\mu_c}{\mu_{\infty}}\right)^{0.2} \frac{c_{p_{\infty}}}{c_{pc}}\right)^{-1}$$

Eq (1.4.13)

$B$ is the mass flux ratio and $R_{ce}$ is the Reynolds number based on the coolant fluid through the slot and the slot width. Distance $x'$ is the length from the origin of the boundary layer. Many different approaches of determining $x'$ have been used in the literature [40]. In the simplest form of these methods, boundary layer is assumed [46] to start from the injection point ($x = x'$).
This assumption neglects the mass of fluid within the boundary layer upstream of injection and is only valid at the limit of zero $\delta^*$. However, this assumption has been shown [40] to be a good approximation for the cases where the boundary layer thickness is smaller than the width of the injection slot. The slot angle has also been shown to affect the entrainment of the main fluid by the injection, which correlates [45] with the following expression:

$$\eta_{ad} = 1.9 \text{Pr}^{2/3} \left( 1 + \beta \right) 0.329 \zeta^{0.8} \left( \frac{Re_c \mu_c}{\mu_\infty} \right)^{-0.2} \left( \frac{c_{p*}}{c_{p}} \right)^{-1}$$  \hspace{1cm} \text{Eq (1.4.14)}

where

$$\zeta = \frac{1}{B} \frac{x}{D}$$  \hspace{1cm} \text{Eq (1.4.15)}

For an injection angle of $\alpha$, the parameter $\beta$ is given by

$$\beta = 1 + 1.5 \times 10^{-4} \left( \frac{Re_c \mu_c}{\mu_\infty} \right) \sin \alpha$$  \hspace{1cm} \text{Eq (1.4.16)}

For the case of film cooling through discreet holes (of diameter $D$), Goldstein et al [47] show that downstream of the coolant holes ($x/D > 6$), the film cooling effectiveness from discreet holes is equivalent to an equivalent slot ($S_e$), the width of which is given by the hole area divided by the pitch ($P$) between the holes. For cylindrical holes, this slot width is given by

$$\frac{S_e}{D} = \frac{\pi}{4} \frac{D}{P} \approx 0.8 \frac{D}{P}$$  \hspace{1cm} \text{Eq (1.4.17)}

The influence of the freestream pressure gradient and the Reynolds number on film cooling was investigated by Kutateladze et al [48]. The effect of surface roughness on film cooling performance was investigated by Goldstein et al [49], who showed that roughness reduces the film effectiveness for the low coolant mass blowing ratio and increases the film effectiveness
for high blowing ratio, when compared with the smooth wall results. The additive nature of multiple rows of cooling holes on the film effectiveness was shown experimentally by Muska [50], et al. Jurban and Brown [51].

In a completely different approach, Forth and Jones [52] showed that superposition can be used in characterizing film cooling even in the compressible, variable properties environment. They argued that the film cooling process can be divided into two broad flow regimes (depending on the mass and momentum ratios): the weak flow regime in which the coolant flow does not penetrate through the boundary layer, and the strong injection in which the coolant flow actually enters some distance into the mainstream.

### 1.4.3 Studies of Turbine Rotor Film Cooling

The research in the application of film cooling techniques to the gas turbine environment has been exclusively experimental. The blade wall curvature, three-dimensional external flow structure, free-stream turbulence, blade rotation and unsteadiness collectively contribute towards the interaction of the main fluid and the injected coolant film. This interaction not only affects the heat transfer conditions, but also increases the aerodynamic losses as reported by Grives [5] and Ito et al [53].

Ito et al [53] showed that the blade surface curvature influences the film cooling effectiveness particularly in the vicinity of the injection holes, with greater effectiveness on the convex surface and less on the concave surface, when compared to a flat plate case. On the concave surface, Schwarz and Goldstein [54] suggested that the unstable flow along the concave surface promotes lateral mixing of the film cooling jets, which in turn results in a two-dimensional behavior of the film effectiveness. Near the endwalls on the suction surface of the blade
profile, Goldstein and Chen [55] showed that the film cooling jets are swept away from the surface by the passage vortex, resulting in very low levels of film effectiveness.

On the concave surface, the film cooling was unaffected by the endwall effects. Wadia and Nealy [56] investigated and provided design guide lines on the influence of film cooling at the blade leading edge. Dring et al [57] experimentally investigated the effect of film cooling on a rotating turbine rotor blade. They showed that on the rotor midspan, the suction surface film cooling effectiveness is similar to existing flat plate measurements, while on the pressure surface, a much higher decay rate of effectiveness was measured. Dring et al observed a large radial displacement of the coolant jet on the pressure surface, which they concluded to be the main cause of the low level of effectiveness measured.

1.5 Scope of the Present Study

There are many uncertainties in the relative significance of different phenomena on the heat transfer in a transonic turbine rotor blade. While there is particular interest in determining the contribution to the mean heat load of the unsteady rotor/ stator interaction in a transonic turbine, the unsteady interaction process must first be understood. The addition of film cooling on the rotor blade and the coolant injection from the upstream nozzle guide vanes may also influence the mean and unsteady heat transfer and further complicate the unsteady interaction process. To clarify some of these uncertainties, the present study is concerned with answering the following questions:

1) Is the rotor blade heat transfer unsteady? If so, can the periodic unsteadiness be attributed to the details of the rotor/ stator shock and wake interaction?

2) For the case where the nozzle guide vanes are cooled, what is the impact of the guide vane coolant injection on the downstream rotor blade heat load?
3) In a fully cooled turbine stage, what is the mean rotor blade heat transfer in the presence of film cooling? What are the effects of three-dimensionality and unsteadiness on the film cooling process?

4) How does the rotor heat load vary with the flow Reynolds number?

To answer these questions, the present study provides an experimental examination of the heat transfer on the rotor of a fully cooled transonic turbine. The emphasis is on using detailed time-resolved rotor heat transfer data to study the influence of various unsteady physical phenomena in an engine-like rotating environment. This study also presents the first measurements of unsteady film cooled rotor blade heat transfer in a transonic turbine.

1.6 Review of Contents

In Chapter 2, the experimental facility and method of data acquisition are covered. The design and calibration of the coolant facility is introduced. The heat transfer gauge instrumentation of the blades, the calibration and the method of determining the confidence level in the data are also discussed.

In Chapter 3, the numerical scheme (UNSFLO) used to calculate the heat transfer distribution is introduced. This code is an unsteady time-marching Euler solver coupled to a Navier-Stokes solver around profile. The code was validated using steady state experimental data and its limitations discussed.

In Chapter 4, the experimental unsteady measurements of the heat transfer distribution around the rotor blade at mid-span are compared against the unsteady numerical solution. This comparison in conjunction with the results of the calculations of the external flow, clearly identifies the process of the shock and wake interaction on the rotor heat transfer.
In Chapter 5, from the experimental results, a correlation relationship for the heat transfer as a function of the Reynolds number is suggested. This correlation was further compared against the data from other rotor profiles available in the literature.

In Chapter 6, the influence of the upstream nozzle guide vane trailing edge coolant injection on the heat transfer at the mid-span of the downstream rotor is examined. The results of this chapter provide the base line heat transfer distribution for the film cooling studies.

In Chapter 7, the influence of film cooling on the rotor heat transfer is experimentally investigated. The experimental results were obtained at three span-wise positions. The influence of unsteadiness and three-dimensionality are discussed. A model is proposed that accounts for the quasi-steady influence of the potential field interaction on the film cooling effectiveness.

In Chapter 8, the points of conclusion for this study are covered and recommendations for future work are presented.
Chapter 2

2. Experimental Apparatus

The present experimental results were obtained in the MIT Blow-Down Turbine (BDT) rotating facility. In this short duration facility, a full turbine stage is tested at the scaled conditions representing the engine operating environment. In this section, a brief description of the main part of the BDT (without film cooling section) is given; for a full description see Epstein et al. [58]. In order to perform the film cooled part of the experiment, a new coolant supply section was designed and manufactured by the author. The design and final configuration of the coolant supply is described in Section 2.2. In Section 2.3, a description of the instrumentation and the data acquisition system is presented. The heat flux gauges are discussed in Section 2.4. The calibration, heat transfer data reduction scheme and error analysis are discussed in Section 2.5. A summary of all the test conditions and tables of the test parameters are given in Section 2.6.

The high pressure axial turbine stage used in the present study is called ACE (Advanced Core Engine) by Rolls-Royce Plc. This profile has been extensively tested by Rolls-Royce Plc. The rotor blade co-ordinates have previously been reported by Ashworth [39]. The stage geometry and profiles are shown in Figure (2.1a). The mid-span velocity triangles for the rotor blade at 100% corrected speed are shown in Figure (2.1b). There are 36 nozzle guide vanes and 61 rotor blades, in this turbine stage.

2.1 Blowdown Turbine Facility

The facility consists of a supply tank (volume ≈10 m³), a large diameter fast acting valve, the test section and a discharge dump tank, see Figure (2.2). The apparatus has been designed
such that flow parameters which influence the turbine fluid mechanics and heat transfer are correctly scaled to the engine conditions. These non-dimensionalized parameters are; Reynolds number, corrected speed, corrected mass flow, Prandtl number, ratio of the specific heats and gas to metal temperature ratios.

The gas temperature is adjusted by circulation of hot oil (up to a maximum of 500F) in the supply tank jacket. To match the ratio of specific heats to the engine combustor exit condition, the supply tank is filled with a mixture of argon/Freon-12 gases. The test section is initially at room temperature and the gas temperature scaled in order to preserve the correct gas-to-metal temperature ratio. The Reynolds number of the flow is varied by adjusting the supply tank pressure. The power generated by the turbine is absorbed in an eddy brake system. The brake controls the reduction in the rotational speed such that, combined with the temperature decay due to the isentropic expansion of the gas, a constant rotor corrected speed for the duration of the test period (0.3 second) is maintained. The nozzle guide vanes are choked during the test time which results in a constant corrected mass flow through the stage. The stage operating point can be varied by changing the area of the choke plate between the test section and the dump tank. A row of vanes is used behind the turbine stage to remove the flow swirl, which in turn increases the operating time before the choke plate unchokes. Table (2.1) summarizes the design conditions for the blowdown facility and compares them to the full scale turbine quantities.

Initially, the entire facility is evacuated and the supply tank and fast acting valve are heated by the circulating oil. Once the temperature of the supply section has stabilized at the required value, the valve is closed. With the test section still at vacuum, the supply tank is filled with the required gas mixture. After the gas temperature in the supply tank has equalized, the turbine rotor is bought up in speed, using a small electric motor, to slightly above the targeted operating speed. The power to the electric motor is then cut, and the rotor begins to slow
down due to the bearing drag. A comparator is triggered as soon as the rotational speed matches a pre-set value, which in turn simultaneously opens the main valve (and coolant valve for the cooled tests) and energizes the eddy brake. A start-up transient occurs during the first 250 ms, and is followed by a steady state period lasting 300 ms, over which, the corrected parameters are constant to better than one percent. The eddy brake keeps the rotor at constant corrected speed during the test time (250 ms - 550 ms). The thermal inertia of the metal in the test section maintains an isothermal wall condition through the test time. Conventional temperature, pressure, and shaft speed transducers are used to measure the steady state operating conditions of the turbine. For the cooled tests, the coolant valve was closed after one second after the start time.

2.2 Coolant Supply Facility

In a film cooled turbine, the other non-dimensionalized parameters of interest are the ratio of the coolant to main mass flux, the coolant to metal temperature ratio, coolant to main momentum ratio and the coolant specific heat ratio. The ability to independently adjust these parameters and maintain their values at a constant level during the test period is an important part of the facility design. In the main facility, the corrected mass flow is constant, but the actual mass flux is decaying in time. To achieve a constant coolant to main mass flux ratio, the coolant mass flow must also vary with the same time constant as the main flow. It was decided to design a second Blow-Down tank for the coolant supply and match its decay time-constant with that of the main supply tank.

With the test section at room temperature, the present temperature scaling results in a cryogenic coolant temperature of around 200° K. By mixing a monatomic (Argon) with a heavier gas (Freon 14) at a set coolant temperature, the ratio of the specific heats for the coolant gas can be independently adjusted. Freon-14 coolant gas is chosen because of its low condensation
temperature. For a given coolant gas temperature and properties, the maximum coolant supply pressure is the pressure at which the gas condensates. For any coolant temperature, coolant pressure and the gas properties, the coolant mass flux and its rate of decay are fully determined by the coolant supply tank volume and the orifice discharge hole size. The scaling of the coolant supply tank and the orifice plate and their calibration is presented in Appendix I.

The coolant supply system configuration is shown in Figure (2.3). A jacketed stainless steel tank is used as the coolant supply. The coolant gas is discharged through a choked orifice, designed to ASME specifications. A fast opening stainless steel 3" ball valve is located in between the tank and the orifice plate. The valve is fully open in 30ms. After the orifice plate, passing through a 3" pipe and a right angle bend, the coolant enters the plenum within the nozzle guide vane disc, see Figure (2.4). Each cooled nozzle guide vane blade has an internal coolant plenum which is fed from holes that are in between the endwall platform and the dovetail supports. For the cooled rotor blades, the coolant is fed through stationary pre-swirl holes and into the rotor disc plenum. Again, each cooled rotor blade is fed from holes below the endwall platforms. For the test series where the nozzle guide vanes are cooled and the rotor blades are uncooled, the passage of the coolant to the rotor disc are blocked. A labyrinth seal is used to reduce the coolant leakage through the endwalls.

The coolant supply tank is a jacketed stainless steel pressure vessel of 0.114 m$^3$ capacity. For coolant exit and instrumentation access, two 3" stainless steel pipes with flanged ends were welded into both ends of the vessel. The instrumentation flange carried 3 thermocouples (2 metal and 1 gas), 2 pressure transducers, the coolant gas feed pipe and a high performance mixing fan. A commercial refrigeration system was used to cool down the heat exchanger fluid (Freon 11). Using a sump type pump, the heat exchanger fluid was recirculated within the jacket and back to the cooling system. The coolant tank and all the connecting pipes were insulated in order to reduce natural convection heat transfer. Insulating washers and gaskets
were used in order to reduce conduction heat transfer between the cooled and the room temperature piping to the test section. All the piping and surfaces within the test section in the coolant path were coated with a 1/4 inch thick layer of insulating compound (SP-40) to avoid excessive heat transfer to the coolant gas. The length of the connecting pipe between the orifice plate and the test section was the minimum possible, given the space constraints. The minimum equilibrium temperature of the supply tank was the temperature at which the maximum output of the refrigeration system was equal to the heat picked up at that temperature. During the tests, the heated main supply tank substantially increased the room temperature, which further reduced the performance of the refrigeration system. In practice, the minimum supply tank temperature attainable with the present configuration was 210 K.

2.3 Instrumentation and Data Acquisition

The data acquisition system consisted of a digital clock, a digital counter, forty five high speed analogue to digital (A/D) channels, four multiplexer units with sixteen channels each, low and high speed amplifier banks, a 20 megabyte core memory unit and a controlling microprocessor. The microprocessor controls the sequencing and operation of each unit and is programmed using an interactive menu-driven software running on a Vax workstation. In Figure (2.5), the schematic of the acquisition system is shown. For a complete description of the data acquisition system refer to Guenette. [59] The digital clock can be programmed into four different sampling intervals for different times within the test and calibration periods. The maximum A/D sampling frequency is at 200 KHZ and distribution of the sampling intervals for the period before and after the test time is a function of the available memory space. In Table(2.2), the timing information for each high speed channel is shown. The test sequence is characterized by the simultaneous opening of the main valve, coolant valve and the energizing of the eddy current brake. The flow parameters stabilize after the first 250 ms. The period between the 250ms and 550ms is the test period during which time the maximum data
acquisition rates are used. After one to four seconds depending on the test conditions, the choke plate in between the test section and the dump tank unchokes. The rotor is slowed down and stops after one minute. In order to perform post-test calibration checks, a low rate of data acquisition is maintained for 10 minutes after the test time.

All the low frequency response instrumentation were multiplexed into the analogue to digital (A/D) converters, which reduced the sampling frequency of those channels by a factor of sixteen. Apart from the tachometers, the upper sensors of the heat flux gauges were the only instrumentation which were allocated directly to the A/D's and not through the multiplexers. A list of the facility instrumentation which were used to characterize the mean flow is given in Table (2.3). The thermocouples were commercially available J and K type thermocouples. All the pressure sensors were Kulite type differential transducers, except for the coolant total pressure sensor on the rotor, which was an absolute type transducer. The resistance temperature devices were both platinum and ceramic type sensors. The signals from the transducers on the rotor were transmitted through a 45 channel uncooled "contact brush" type slip ring and then to the amplifier boxes. The heat flux measurements were made using thin film gauges, installed around the rotor blade profile which will be discussed in Section 2.4.

2.4 Thin Film Gauges

An important part of the present study is the ability to accurately measure the time-resolved surface heat flux at the desired measurement points. The development of reliable, accurate and robust heat flux instrumentation was addressed in the report by Guenette [59]. Guenette developed a multi-layer thin film gauge, which was the exclusive method of heat transfer measurement in the present study. For details of the theory, calibration and the use of thin film gauges see Epstein et al [60] The present study covers the initial series of tests where thin film
were used on the rotor blades. The particular data reduction schemes used and a brief description of the sensors are given below.

Each multi-layer gauge is comprised of two thin film (≈10^{-7} m thickness) Nickel temperature sensors positioned on either side of a 25 μm thick polyamide insulator (Kapton), see Figure(2.6). The sensors are formed into serpentine patterns (to increase their length to width ratio) which cover an area of 1.3 mm^2. Initially, the Nickel was deposited using standard vacuum deposition technique. It was discovered that this particular mode of production resulted in excessive build-up of tensile residual stress within the Nickel, which resulted in cracking and non-repeatable calibration. A new production scheme was later developed by Guenette which resulted in a lower residual stress within the sensor metal, and provided reliable and repeatable calibration results. In this scheme, the gauge pattern is etched through a layer of optical photo-resist using ultra violet radiation. Then, Nickel is deposited over this etched pattern using an electroless deposition technique. The connecting leads to the gauge are made up of thin layers of electroplated Copper. The gauges were deposited in batches of twelve on a single sheet. This production scheme was used by the author to produce all the thin film gauges required for the cooled tests. The gauges were adhesively bonded to the blade surface using strain gauge bonding techniques.

At low frequencies, the gauge is essentially a thermal shunt. The temperature drop across the insulator, as measured by the two sensors, becomes a direct measure of the heat flux through the gauge. Above a roll off frequency of 20 Hz, the thermal waves are dampened through the insulator. The Kapton layer appears as an infinitely thick insulator to the thermal waves of frequencies higher than 1 kHz. A one-dimensional unsteady heat conduction model is used to infer heat flux from the temperature history of the two sensor. The response of the gauge through the entire frequency domain is numerically reconstructed. The high end of the frequency range is scaled by the time associated with a fluid particle to pass over the sensor. A
fluid velocity of 300\(\text{ms}^{-1}\) corresponds to the typical upper frequency of about 300 KHZ for the
gauge. The Nyquist frequency associated with the data acquisition is at 100 KHZ, which
actually limits the upper frequency range for the sampled data. This frequency limit
corresponds to over 25 times the nozzle guide vane passing frequency, as measured in the rotor
frame of reference.

For the uncooled rotor blade, the Kapton sheet (with the gauges) was wrapped around the
blade, fully covering the blade surface. The gauges had to be fitted in between the coolant
holes for the film cooled blades, covering from the tip to the hub of the blade, see
Figure(2.7a). The gauges were laid out such that the copper leads were directed towards the
hub of the blade. At the end-wall of the blade, thin (G#40) external wire leads were soldered
onto the copper leads. To increase the mechanical strength, the soldered joints and the wire
leads were encapsulated in epoxy. The wire leads were fed through small holes in the end-wall
platform and were directed over a supporting shim plate and soldered into a connector. The
shim plate and its connector were designed to provide a more modular disc and blade assembly
in which the disc wiring was independent of the blade instrumentation. Also, they provided
support for the leads during the assembly and the test period. The connector from the shim
plate was firmly fitted into a supporting "junction box" on the disc, see Figure (2.7b). A
ribbon cable, with a similar connector in the junction box, provided continuity to the slip ring.
The stationary connections after the slip ring were the Wheatstone bridge, the amplifiers and
the A/D's.

The position of the gauges for the uncooled tests are listed in Table (2.4a). For the film cooled
tests, 3 blades were instrumented with thin film gauges, see Figures (2.8). The position of the
gauges for the film cooled tests are listed in Table (2.4b). The unwrapped surface of the blade
and the position of the heat flux sensors are shown in Figures (2.9). In Figures (2.9a),
representing the tip-section instrumented blade cT, (2.9b) representing the mid-section
instrumented blade cM, (2.9c) representing the hub-section instrumented blade cH and their sensor identification numbering, are shown. In this scheme, each sensor is identified by the blade name followed by the sensor position number and the qualifier. For example, sensor cH25B, represents the film cooled hub-section instrumented blade (cH), leading edge (gauge 25), bottom sensor (B). The exact position of the heat flux sensors for each instrumented blade is shown. The scale of the drawing is 1:1 and the position of the sensors are accurate to within a sensor length. The coolant hole geometry of the instrumented blades were different from the remaining rotor blades. To provide symmetry in the flow pattern, two other blades (un-instrumented) with similar film cooled hole patterns, were mounted on the disc on either side of each instrumented blade.

Time-averaged data is measured by averaging the heat transfer from 300 to 500ms and is presented with respect to the fractional wetted surface. The wetted surface is 1.38 and 1.84 times the axial chord for the pressure and the suction surfaces respectively. The heat transfer measurements are non-dimensionalized in the form of the Nusselt number distributions. The Nusselt numbers are based on the difference between the total inlet relative (from streamline curvature calculation) and the blade surface (upper sensor indicated) temperatures, axial chord and the thermal conductivity of the main gas evaluated at the blade surface temperature. This presentation form was chosen such that the non-dimensionalization is in accordance with the form of the reported results of Oxford University cascade experiment.

The unsteady heat transfer data had a bandwidth of 100 KHZ for the uncooled rotor experiments. In the film cooling experiment, all the high speed channels picked up a noise source (believed to be the digital counter) at exactly 360 times the rotational speed. For the nominal design incidence, the rotational speed was at about 100 rounds per second, which corresponded to a pick-up frequency of 36 KHZ (and higher harmonic). All the high speed channels were digitally filtered at 32 KHZ for the film cooled experiment. For the unsteady
heat transfer data, this upper frequency bandwidth corresponded to above 8 times the nozzle guide vane passing frequency for the 100% and above 7 times the nozzle guide vane passing frequency for the 120% nominal corrected speed. The rotor time-resolved data, was ensemble averaged (with respect to the nozzle guide vane passing) for 10 revolutions of test time.

2.5 Calibration

The calibration and the accuracy of the low frequency response tunnel instrumentation were discussed by Guenette [59]. In this section, the calibration, data reduction and the analysis of the signal to noise ratio of the thin film gauges will be covered. All of the calibrations were performed with the entire data acquisition chain in line (except for internal rig wiring).

2.5.1 Static Calibration

The two sensors of a thin film gauge are resistance thermometers providing measurement points at the external surface and a depth of $d$. The knowledge of the properties of Kapton, the value of $d$ and the temperature coefficient resistivity of the two sensors, fully characterizes the operation of the thin film gauge. Apart from the temperature coefficient resistance sensitivity of the sensors, the unknowns that determine the response of the gauge are the $\sqrt{\rho c k}$ and $k/d$, where $\rho$ is the density, $k$ is the conductivity and $c$ is the specific capacity of Kapton.

The temperature coefficient resistivity calibration of the sensors were performed in a conventional heated immersion bath of varying temperature. All the thin film gauges in the present study were temperature coefficient resistance calibrated before and after (those that survived) the test series by the author. Guenette [59] presented a scheme in which the $\sqrt{\rho c k}$ and the $k/d$ are determined by the use of a laser pulse step radiant heating. To reduce the risk of damaging the thin film gauges, the laser calibration is only performed after a series of
tests have been completed. The thin film gauges that survived the test series were then calibrated and the results are listed in Table (2.5). The calibration results showed that the $\sqrt{\rho c k}$ of Kapton remains constant at a value of 581 (± 5) (W/m$^2$)(°C/K) for the calibrated thin film gauges which was later confirmed by Haldeman (mean of 584). This result suggests that the material properties of Kapton is unaffected by the gauge manufacturing procedure.

The calibration for the $k/d$ values were performed after the completion of the test series. Because some of the sensors were destroyed during the test series, not all the sensors were calibrated. However, a large enough number of the thin film gauges survived to allow at least one sensor from each manufactured sheet to be calibrated. It is assumed that because each sheet undergoes the same etching and deposition process, the thickness of Kapton ($d$) and hence the calibration values will be similar. This assumption was verified by calibrating two thin film gauges from the same sheet, which resulted in $k/d$ values within the experimental error. The measured $k/d$'s for the calibrated thin film gauges are also listed in Table (2.5).

### 2.5.2 Run Time Calibration

The run time calibration is used to verify the accuracy of the temperature coefficient resistance sensitivity and offset values (scale and zero) for each sensor. By matching the initial 30 ms (prior to the valve opening) and the last 30 sec (after 10 min the blade is at equilibrium) of a sensor output to the resistance temperature device indicated temperature, the scale and zero are re-evaluated. It was found that the sensor scale remains unaltered from test to test, but the zero did vary. The shift in the zero was attributed to drift of the amplifier, power sources and a small change in the resistance of the sensors, due to erosion from test to test. In the following data reduction schemes, the zeros of the sensors were determined by forcing their indicated output to match the resistance temperature device indicated blade mean temperature. The scale values determined from the static temperature coefficient resistance calibration were unaffected.
For the film cooled tests, the calibration of the thin film gauge signals from the rotor was complicated by the existence of random "jumps" in the output of a few sensors during the test time, see Figure (2.10). These jumps resulted in a shift in the run-time calibration of the sensors which had to be accounted for. The majority of these jumps were observed on the pressure surface top sensors, and all the jumps occurred after the valve opening, and prior to the rotor coming to a stop. It is postulated that these jumps could be due two possible effects; (1) small particles loosened by the start-up transient in the tunnel, impacting on the rotor surface and changing the resistance of the sensors, and (2) failure of ribbon cable from the slip ring to the junction box. These two effects result in an abrupt increase in the line resistance, hence an apparent positive jump in the measured temperature, which is consistent with the experimental observation. During the film cooled tests series, two major tunnel failures occurred which are believed to account for the small loose particles in the test section. In one occasion, a few gallons of heat exchanger fluid was accidentally discharged (O ring failure) into the section during a post run period, and in another occasion, the O ring sealing the main valve, was dislodged and ingested into the test section. The measurements taken during the second tunnel failure are suspect and not presented.

The pressure surface thin film gauges had the highest occurrence of jumps. Even though the top sensors were the most affected by the appearance of the jumps, some bottom sensors were also affected which is due to the ribbon failure. The magnitude of the jumps typically corresponded to a change in the resistance of a few Ohms (a sensor resistance is around 3000 Ohms). The existence of the jumps did not alter the scale value of a sensor, however, it changed the value of zero. Every sensor was carefully checked for a sharp change in the mean value at any one point for the entirety of the data acquisition time. After all the jumps were identified, the measured voltages were adjusted for the value of the jump at the point where they occurred, see example in Figure (2.10). The error associated with the evaluation of the
zeros was determined by the offset between the resistance temperature device and the sensor indicated temperatures at the end of the data acquisition, ten minutes from the start. The accuracy of this jump removal scheme was checked by comparing the results of the a test where no jumps were observed with a similar test with jumps.

2.5.3 Uncertainty Analysis

An uncertainty analysis of the time-averaged heat flux measurements is performed in Appendix II. Every step required to calibrate a sensor, contributes to the overall measurement uncertainty, and should be accounted for. In Appendix II, individual measurement uncertainties are suitably combined to yield an overall error band. The major assumption used in determining the error bands is that the uncertainty contributions are unrelated. The error bands (95% confidence level) of the thin film gauge measurements are calculated for every test and are listed with the presented data. The test to test repeatability of the measurements is another indication of the relative precision of the experiment. In Figure (2.11), the measured Nusselt number distribution at the nominal hub section for two identical fully cooled stage test conditions (T71 and T75) is shown. The relative gauge uncertainty is typically around 5%. The absolute calibration uncertainty is about 10%.

2.6 Summary of Test Conditions

There are three different test series presented in the present study; (1) Uncooled stage, (2) Cooled nozzle guide vanes (with trailing edge coolant injection) and uncooled rotor, and (3) Fully cooled stage (cooled nozzle guide vanes with trailing edge coolant injection and full coverage film cooled rotor blades).
In the uncooled stage test series, two rotor blades were used. The test condition parameters are listed in Table (2.6a). The tests numbers are; T47, T50, T51, T52, T53 for the mid-span instrumented blade and T112 for the tip section instrumented blade. In Table (2.6b), the measured time-averaged Nusselt numbers for all the thin film gauges (and their associated error bands) in this test series are given.

In the cooled nozzle guide vanes and uncooled rotor blades tests, only the mid-span instrumented uncooled rotor blade was used. The test numbers for this series are; T55, T56, T57, T60 and T61. The test parameters are listed in Table (2.7a) and the measured time-averaged Nusselt numbers are given in Table (2.7b).

In the fully cooled stage test series, three instrumented rotor blades were used (nominal Tip, Mid and hub sections). The test numbers for this series are T63, T64, T65, T66, T67, T70, T71, T72, T73, T74 and T75. The test parameters are listed in Table (2.8a) and the measured time-averaged Nusselt numbers are given in Table (2.8b).

The two uncooled rotor blades (mid-span and tip sections) were instrumented by Guenette who performed the uncooled stage test series. The calibration, data reduction and the analysis were performed by the author. Mr. C. Haldeman performed the temperature coefficient resistance calibration of the tip section blade. The nozzle guide vane heat transfer experiments were performed by Guenette. The second test series (nozzle guide vanes cooled, rotor blades uncooled) were performed by the author using the previously instrumented mid-span blade. The fully cooled stage series were performed by the author using the three instrumented uncooled blades.
Chapter 3

3. Numerical Procedure

In experiments performed at the MIT's short duration blowdown facility, the unsteady heat transfer to the surface of the turbine has been measured. These surface measurements do not provide sufficient information to couple the unsteady external flow features with the unsteady heat transfer processes. A Computational Fluid Dynamics (CFD) code was used to furnish numerical solutions for comparison with the measured quantities. In the following chapters, it will be shown that the combination of the experimental and computational results establishes a powerful tool for the quantitative analysis of the heat transfer processes, and its coupling to the external flow. The primary purpose of this chapter is to introduce the CFD procedure used and discuss its strengths and limitations. In Section 3.1, the details of the CFD code (named UNSFLO) that was used, is introduced. In Section 3.2, the steady state comparisons between the rotor linear cascade and the steady UNSFLO solution are used to show the validity and the accuracy of the code, and the discrepancies between the measurement and the calculations for both the suction and pressure surfaces are explained. The unsteady comparisons of the present experiment and UNSFLO results are presented in Chapter 4.

3.1 UNSFLO Description

The numerical tool used for this study is a quasi-three dimensional, Reynolds-averaged, multi-blade row, unsteady, viscous computational fluid mechanics code known as UNSFLO, developed by Giles \[^{61,62}\]. UNSFLO uses a hybrid Euler/Navier-stokes algorithm. The Euler equations are solved in an outer inviscid flow region (where viscous effects are neglected) using a Ni-type Lax-Wendroff algorithm \[^63\] . Around the profiles on an O-type grid structure, the thin shear layer Navier-Stokes equations are solved using an alternating direction implicit
method. UNSFLO is one of the first Navier-Stokes program able to analyze stator / rotor interaction with arbitrary pitch ratios. A time-inclined computation plane facilitates the calculation of stages with unequal rotor-stator numbers, greatly reducing the computation time required [61]. In the present study, the computation for a stage was done for one nozzle guide vane and two rotor blades. The time-tilting then serves to adjust the calculation to the desired blade row pitch ratio. UNSFLO can also be used in the steady state mode in which the pitch-wise average of the stator exit flux components are matched (through iteration) to the inflow fluxes for the rotor domain. The input to this mode of the calculation is the stage pressure ratio. This feature is useful for matching the present experimental conditions where no pressure measurements in between the rotor blades and the stators is available.

Within the boundary layer region, a Baldwin-Lomax [64] algebraic turbulence model is used with typically eighteen grid points across each layer. The steady state position of the transition point on each surface is an input to the code. The transition point is defined here by the position on the blade at which the turbulence closure model is turned on.

UNSFLO has a feature which allows for variation of the streamtube height (in the third dimension) to be used in the calculation. This input was based on streamline curvature solutions at the test conditions. This approximate correction to the streamtube height was found to be essential in order to obtain the correct inflow/ outflow Mach numbers for this transonic stage, and represents a quasi 3-D solution of a 3-D flow field. The influence of the streamtube variation on the numerical solution will be discussed in section 3.3.

A typical grid structure used in the calculations is shown in Figure (3.1), for which approximately 16,000 grid points were used. A typical unsteady solution took approximately 20 hours of CPU time on a Stellar super-mini computer. The converged unsteady solution were visualized using color contour plotting of the calculated variables (Mach number,
pressure, enthalpy, etc.) on the Stellar. This interactive visualization allows the user to follow the unsteady interaction process and interpret the variations in the surface quantities with the observed features of the flow. These observations allowed a detailed examination of the propagation of the nozzle guide vane trailing edge shock structure which will be discussed in Chapter 4.

3.2 Steady State Comparison with Oxford Cascade Measurements

In order to check its validity and accuracy, the numerical code was used in the steady state mode to calculate the flow around the ACE rotor profile. The results were compared against the steady state (with no bar-passing) cascade measurements of Ashworth et al. [65] at a design incidence, corresponding to that of test (T47). The streamtube height variation used in the calculation was arranged to match the geometric divergent angle of the end-walls (10°) in the experiment.

In the numerical calculations, the flow around the blade was calculated for both the fully laminar and fully turbulent boundary layers. This was achieved by setting the transition point at the trailing edge (laminar case) or at the leading edge (turbulent case). The distribution of the Nusselt number around the blade calculated by UNSFLO and measured by Ashworth, is shown in Figure (3.2a). The two different symbols in Figure (3.2a), represent the measurements reported by Ashworth with and without a turbulence grid. Calculated wall static pressure and the measurement are shown in Figure (3.2b).

It is observed that on the suction surface the calculated laminar heat flux matches both measurements up to 20% of the wetted surface, after which point, the measurement follows the turbulent calculation. At around 65% of the wetted surface on the suction side, the shock from the pressure surface of the adjacent rotor blade interacts with the boundary layer, see
Figure (3.3). In the schlieren photograph taken by Bryanston-Cross [66] for the same geometry at the same test condition, this shock interaction is clearly seen, see Figure (3.4). Detailed observations of the interaction region, show a rapid thickening of the boundary layer downstream of the region, as well as an apparent large increase in the level of turbulence. The thickening of the boundary layer is expected to reduce the level of heat transfer, while the increase in the turbulent transport within the boundary layer should increase the heat transfer. The overall influence of the shock interaction on the heat transfer is a balance between these two counter-acting phenomena. The calculation does underestimate the heat load at the back of the suction surface, which could be attributed to the inadequacy of the present turbulence model. The present boundary layer turbulence model has been shown [67] to underestimate the skin friction coefficient downstream of a shock/ boundary layer interaction by as much as 100%. This difference between the measurements and the numerical solution at the back of the suction surface is observed in all the test cases.

On the pressure surface, the calculation of the turbulent case compares well with the cascade measurements with the turbulence grid. The no grid measurements seem to follow the laminar solution for the first 35% of the wetted surface, presumably going through transition, and then match the turbulent solution from the 50% wetted surface on. These results seem to agree with the observations made by other workers that the increase in freestream turbulence moves forward the transition point of the boundary layer.

3.3 Effect of Streamtube Height Variation

The influence of the variation in the streamtube height on the UNSFLO predictions was numerically tested. Using three different streamtube height values, the flow around the ACE rotor for the Ashworth's test case was predicted. In these tests, the streamtube height was arranged to correspond to a constant divergent angle from the inlet to the outlet, see
Figure (3.5). The tested streamtube height variation correspond to divergent angles of to 0°, 10° and 15°. All other test condition are similar to the test case of section 3.2. The calculated surface pressure distribution around the blade for the three cases are shown in Figure (3.6a). In Figure (3.6b), the corresponding calculated Nusselt number distribution are shown. It is observed that changing the value of the streamtube height variation has a direct influence on the prediction. In Figure (3.6a), the pressure distribution shows that the main difference is near where the flow Mach number is close to one. Increasing the streamtube height in a supersonic flow should result in a decrease in the surface pressure, which is consistent with the result of Figure (3.6a). The influence of the streamtube height on the heat transfer, also shows some change on the front part of the suction surface.

3.4 Summary

In this chapter, the numerical code (UNSFLO) was introduced and its validity and accuracy was checked against data from literature. This code would be used as a tool in the next chapter to predict the unsteady flow for the rotor/stator test case. The steady state fully laminar and fully turbulent solutions from UNSFLO were evaluated by comparing against Ashworth's cascade data. The input streamtube height variation was shown to have a direct influence on the profile pressure distribution and heat transfer.
Chapter 4.

4 Comparison of Uncooled Rotor Heat Flux with Calculation at Mid-Span

Introduction: The detailed studies of fluid mechanics of the heat transfer to the high pressure turbine rotor blades have until recently focused on the steady state analysis of the flow. In the last decade, a number of experimentalists have simulated the interaction of the wake and shock structures from the nozzle guide vanes on the downstream rotor, Doorly [31], and Ashworth[39]. This method which was originally devised by Doorly [31], provided the incoming unsteady flow by means of a stationary rotor cascade and a moving bar upstream, simulating the passage of the vanes. In conjunction with schlieren photography and high frequency data acquisition techniques, the influence of the wake and the shock structures on the instantaneous surface heat flux were analyzed.

Doorly [31] demonstrated that for a laminar boundary layer, the wake interaction (resulting in an unsteady transition of the boundary layer) can increase the blade heat transfer by as much as 100%, see Figure (4.1). The unsteady experimental results showed that the relative duration of laminar or turbulent boundary layer at any given blade position depended on the width of the corresponding wake. Doorly [31] also showed that the turbulent patch resulting from the wake impingement propagates at a speed lower than the freestream velocity, which culminates in a time lag for the boundary layer to relax to the laminar state. This time lag results in history effect, especially on the pressure surface. The wake interaction resulted in a smaller increase in heat transfer for a fully turbulent boundary layer, which was shown to be due to the higher levels of freestream turbulence.

In the bar passing experiments performed by Ashworth [39] on the same blade profile as the present study, it was shown that the simulated nozzle guide vane wake and trailing edge shock
had an important effect on the rotor heat transfer, see Figure (4.2). Ashworth [39] also measured the unsteady surface pressure and compared to the measured unsteady heat transfer. This comparison showed that the unsteady rapid rise in the surface pressure associated with the shock impingement results in an abrupt rise in the instantaneous surface heat flux. The influence of this shock interaction was mainly to increase the heat transfer in the front part of the blade (before the crown) especially on the suction surface. In a follow on to the Ashworth's experimental results, Rigby et al [68] used a simple one dimensional model and the measured unsteady surface pressure to predict the surface heat flux with unsteady shock impingement. Rigby et al assumed that the unsteady shock enhancement in heat transfer could be modelled by the compression of the inner sub-layer by the pressure perturbation and the isentropic heating associated with this compression. The theory was experimentally validated by applying it to Ashworth's [31] experimental results.

The main objective of this chapter is to identify the unsteady physical phenomena that influence the heat flux to an uncooled rotor blade. The time-resolved ensemble averaged heat transfer measured around the rotor profile at mid-span, at 2 different test conditions (nominal design incidence (T47) & -10° incidence (T52) ) are reported and compared with their respective viscous numerical prediction (using UNSFLO).

The time-averaged experimental measurements (pressure, temperature, rotational speed and etc.) are used to define the input flow boundary condition in the code. The streamtube height variation through the stage is determined from the Streamline Curvature solution obtained from Norton [69]. Steady state solutions for both fully laminar and fully turbulent rotor boundary layers were obtained. For the unsteady solutions, the rotor blade was specified to be fully turbulent. Attempts to obtain a fully laminar unsteady solution did not succeed due to the occurrence of large scale flow separation in the prediction. The transition point on the nozzle guide vanes affects the velocity and thermal boundary layer growth on the nozzle guide vane,
which influences the blockage. This effect is minor when compared to the uncertainties in the streamtube height variation. The transition point of the nozzle guide vane boundary layer was determined from the previously available experimental data obtained and are shown in Figure(4.3). The numerical grid has approximately 16,000 grid points. There are eighteen points through the viscous grid with twelve of these points in the inner layer.

4.1 Time-Averaged Rotor Heat Transfer

In this section, the steady state and the time-averaged (average of a vane passing period) Nusselt number distribution around the rotor profile obtained from UNSFLO are compared against the experimental time-averaged results (averaged from 300 to 500 ms, roughly 20 rotor revolutions).

4.1.1 Comparison with Steady Calculation

There is a common observation based on low speed data that the level of heat transfer to the rotor is somewhere between laminar and turbulent depending on the local boundary layer state, see Sharma[34] and Mayle [70]. The unsteadiness can be accounted for by increasing the intermittency factor account for this deterministic phenomena. Intermittency factor is defined here as the amount of time the heat transfer is above a threshold level (typically half way between the turbulent and laminar values) divided by the total time. This view has been established through studies of low speed cascade and low speed rotating experiments. In contrast to this view, the cascade with bar passing results of Ashworth, in particular, showed that in a transonic turbine, the level of heat transfer could be above the turbulent levels. In this section, the time-averaged experimental Nusselt number distribution around the profile (shown in Figure (4.4)) is compared to the steady state laminar (transition point set at the trailing edge) and turbulent (transition set ahead of the leading edge) solutions from UNSFLO.
In Figure (4.5), the comparison between the measurement and steady state calculation at the design incidence (test T47) is shown. The low level of measured Nusselt number compared to the calculation at the leading edge is due to the spatial and temporal averaging by the sensor, which will be discussed in Chapter 5. It is observed that the measured Nusselt number is higher than even the calculated turbulent level for the first 50% of the fractional wetted surface for both the suction and pressure surfaces. This is the region where the maximum influence of the unsteady interaction of the nozzle guide vane shock structure with the rotor is expected.

In Figure (4.6), the same comparison between the measurement and steady state calculation is made for the -10° incidence case (test T52). It is observed that the difference between the experimental results and the turbulent calculation is much less than in the design incidence. The discrepancy between the measurement and calculation at the back of the suction surface (approx. 70% wetted surface) is in the region where the influence of the shock from the pressure surface of the adjacent rotor blade is important.

4.1.2 Comparison with Time-Averaged Unsteady Calculation

UNSFLO time averages of the unsteady solutions are compared against the experimental data for design incidence (test T47) and the -10° incidence (test T52) test conditions. For the sake of comparison, the steady state solutions previously shown in Section 4.1.1 are also included.

For the design case test T47, shown in Figure (4.7), it is observed that the time-averaged and the steady state turbulent solutions are very close, and essentially the only difference seems to be at around 30% wetted surface on the suction surface and the first 20% on the pressure surface. For the -10° incidence test (T52), the comparisons are shown in Figure (4.8) and the trends were similar to the (T47) test case. The only notable difference seemed to occur at the
first 50% of the pressure surface, where the time-averaged unsteady calculation seem to be in between the steady state turbulent and laminar solutions, eventually rising towards the turbulent value. Note that at the leading edge, the time-averaged unsteady prediction is lower than the steady state results, which is primarily due to the movement of the stagnation point and the subsequent temporal averaging as will be shown. Qualitatively, the time-averaged unsteady calculations show good agreement (within 20%) with the rotor measurements (with a 10% absolute uncertainty in data). Discrepancies between the measurements and the calculations will be discussed in the next section when the detailed unsteady comparisons are made.

4.2- Time-Resolved Rotor Heat Transfer at Midspan

4.2.1 Observations from the Unsteady Numerical Solution

The unsteady numerical solution could be graphically viewed using visualization software packages developed at Computational Fluid Dynamics Lab (CFD) at MIT.

The typical UNSFLO numerical solution has the correct relative timing information, but the point of reference to the experiment is arbitrary. To compare the unsteady solution to the experimental results, a timing reference point had to be determined. In the absence of any other, the time associated with the large peak in heat transfer measured by gauge #7 was matched between the numerical solution and the measurement, see Figures (4.11b) and (4.12b). This value of the shift in the time was applied to the remaining files. This method of matching of the timing information ensured that the details of the time-resolved measurement could be directly compared with the external flow features such as shocks and wakes.

Shock Interaction: The unsteady shock motion predicted by UNSFLO for one blade passing (nozzle guide vane) period is shown in Figures (4.9). In this animation, the rotor
blades are kept stationary while the nozzle guide vanes move down towards the bottom of the page. This allows the flow features to be observed from the rotor reference frame. These figures are produced by viewing an instantaneous picture of the flow at eight different times within the period and outlining the spatial pressure gradient associated with the shocks. Due to the nature of the shock motion, an identifying scheme is adopted by which the shocks and their reflections are tracked. The nozzle guide vanes are numbered (N2, N3, etc.). The first number of the shock identifier corresponds to the vane where that shock originated, for example, shock 2a corresponds to a shock originated from vane N2. In the next section, the term "blade" corresponds to the reference rotor blade (shaded blade) and only the shocks that interact with this blade are discussed. The thickness of the lines representing the shocks have no relations to the strength of the shocks. The strength of these shocks varies, such as shock 2a in Figure (4.9a) has a normal Mach number of about 1.2, while some others are barely stronger than a sonic wave.

The shock from R3 rotor blade is reflected from the back of the suction surface on the reference blade, see Figure (4.9a). This shock is stationary in the relative frame and therefore it is not numbered. In Figure (4.9a), the shock (2a) from the trailing edge of the N2 vane is just about to interact with the blade. In Figure (4.9b), it is seen that this shock reflects from the crown of the blade and is moving towards the leading edge (Figure (4.9c)). It is also seen that the reflection of this shock (2b) interacts with the pressure surface of the adjacent R3 rotor blade (terminating in sonic lines) and is gradually bending towards upstream. Once the shock (2a) has passed over the leading edge, its reflection moves towards the upstream vane row, see Figure (4.9d). In the Figure (4.9e), it is seen that the 2a shock is moving down the pressure surface of the blade while its original reflection (2b) is moving upstream and just about to interact with the N3 vane. The 2a shock rapidly moves down the pressure surface and interacts with the R1 rotor blade, see Figure (4.9f), while the original reflection (2b) has reflected from the back of the suction surface of the N3 vane and its reflection is moving
downstream. At the same time, it is observed that the shock from N2 vane (2d) and reflected from the back of the N3 vane (2e), interacts with the crown of the blade, see Figure (4.9f). In Figure (4.9g), it is seen that the 2a shock is completely moved down the pressure surface and its reflection (2b') from the R1 rotor blade, now interacts with the blade at the back of the pressure surface. In the Figure (4.9g), it is also seen that the 2e shock has moved down the suction surface towards the leading edge. The reflection of the 2b shock from the back of the N3 vane (2c) now is reflected from the pressure surface of the R3 rotor blade and is moving towards the reference blade. In Figure (4.9h), the shock 2c interacts with the crown of the blade and is reflected while the 2e shock is just about at the leading edge. The same shock structure repeats itself for the next period. It is seen that in any period, the shock structure from 2 adjacent nozzle guide vanes interact with the blade.

**Wake Interaction:** The nozzle guide vane wake trajectory, at 3 different times within one period, as it interacts with the downstream rotor blade rows are shown in Figures (4.10). Entropy is used to trace the wake as it originates from the nozzle guide vane trailing edge through the rotor blade row. On the suction surface where the wakes from different vanes appear to merge, a certain amount of judgement was involved in determining the relative position of the individual wakes. The nozzle guide vane numbering scheme is consistent with the same vanes as described in the previous paragraph. The gauge positions on the reference blade are also shown for later reference to the surface measurements.

On the suction surface, it is seen that the wake from the N0 vane (not shown) is being immediately replaced by the wake from N1 vane, see Figure (4.10a). In the next 2 Figures (4.10b&c), it is seen that the wake from the N1 vane completely wraps around the suction, being later replaced by the wake from the N2 vane. These figures suggest that on the suction surface, the blade boundary layer interacts with the turbulent wake from the nozzle guide vanes at all times. On the pressure surface, however, the wakes are being "chopped" by the blade
and the pressure surface portion of the wake is convected downstream. This results in an intermittent wake/boundary layer interaction on the pressure surface.

4.2.2- Surface Heat Flux Comparison: Suction Surface & Leading Edge

In this section, the detailed time-resolved comparison of the heat transfer distribution on the suction surface of the blade as measured and predicted are presented. These comparisons are presented in Figures (4.11) for test (T47) and in Figures (4.12) for test (T52).

Design Incidence (T47): The region between the leading edge and the crown of the blade is represented by Figures (4.11b) and (4.12c). In this portion of the blade, it is seen that the peak to peak level of the unsteadiness in the Nusselt is of the same order or larger than the mean level for both the measurements and calculations. This result is consistent with the observations of Section 4.2.1, in which the trajectory of the shock interaction with the suction surface was shown to be mainly between the leading edge and the crown. The features of the unsteadiness are well predicted by the code, but the level of the peaks are lower than in the measurements. Past the crown of the blade in Figures (4.11d,e&f), the unsteady components in the measured and predicted Nusselt numbers are still evident which is due to the convection of the high turbulent patch from the upstream shock interaction, see Ashworth. In Figures (4.11d,e&f), the prediction captures the nature of unsteady component of the heat transfer. In Figure (4.11e), just downstream of the position of the shock (from the R3 blade, see Figures (4.9)) interaction with the back of the suction surface, the code over-predicts the mean level of the Nusselt number.

-10° Incidence (T52): In Figure (4.12b), the comparison between the measurement and prediction is much better than in the (T47) test case, but the code still under-predicts the peak value of the measured Nusselt number. Near the crown in Figure (4.12c), the levels of the
peak to peak unsteadiness seems to be correctly predicted but an apparent time lag is also observed.

**Leading Edge:** When visualizing the results of the calculations, it was observed that the instantaneous stagnation point could move by as much as 3-5% of the wetted surface on the suction surface and 8-10 percent on the pressure surface. The presented calculated values of the Nusselt number are spatially averaged over the same area as the heat flux sensor covers for direct comparison. In this averaging, it is assumed that the sensor linearly averages the spatial variation of heat flux over its surface. In Figures (4.11a) and (4.12a), the unsteady calculation at the leading edge exhibits a large unsteady peak which is not observed in the measurements. The leading edge measurement did not exhibit such a large level of unsteadiness, especially in the (T52) (Figure (4.12a)) test case where very little unsteady component of heat transfer were observed. The reason for this difference between the calculations and measurements is unknown.

4.2.2.1 Shock Interaction

The predicted Nusselt number at positions corresponding to gauges 6 (leading edge), 7 & 9 is shown in Figure (4.13) for the (T47) test case. The arrows pointing to the peaks in the heat transfer correspond to the compressional heating effect due to the shock impingement as predicted by the code. It is seen that the large peak in the Nusselt number is moving in time towards the leading edge sensor. This was previously observed when the shock trajectories were analyzed in Section 4.2.1.

To show that the peaks in the calculations are due to the shock impingement, in Figure (4.14b) and (4.14c) the time-resolved Nusselt number and static wall pressure (non-dimensionalized by the time-averaged relative stagnation pressure) at the gauge 7 position are plotted respectively,
(note that Figures (4.14b) and (4.11b) are identical). By comparing these 2 figures, it is seen that the large peak in the calculated heat transfer occurs exactly at the same time as the rapid rise in the pressure which characterizes the passage of the shock.

The discrepancy between the measured and predicted unsteady heat transfer with the passage of shocks is due to the inaccuracies in either the inviscid flow calculation (and shock strength) or in the viscous calculation. In order to separate the two effects, the Rigby theory (see the introduction to this chapter) is used, in which the pressure output signal of the Figure (4.14c) is used as an input to the model. Rigby suggests that the rise in the heat transfer due to the passage of the shock can be determined by the compression (bracketed term on the right side of Equation (4.1) ) and the isentropic heating of the gas (second term on the right side of Equation (4.1) ) in the sub-layer. From Rigby, the modulation of heat transfer is given as;

\[ \frac{\text{Nu}(t) - \text{Nu}_0}{\text{Nu}_0} = \left( \frac{p(t)}{p_0} - 1 \right) + \frac{\sqrt{p c_p/k}}{\text{Nu}_0} \frac{C g(t)}{T_g - 1} \]

Eq (4.1)

Where,

\[ g(s) = L (g(t)) = \gamma s L \left( \left( \frac{p(t)}{p_0} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right) \]

Eq (4.2)

The symbol \( s \) is the Laplacian operator and the subscript 0 corresponds to the steady state values. The unsteady pressure signal calculated by UNSFLO is decomposed in the frequency domain and the resulting Fourier components are used to numerically evaluate the terms of the Equation (4.1) & (4.2). The mean value heat transfer is obtained from the steady state solution at that position. The result of this evaluation for gauge 07 is plotted against the measurement and UNSFLO prediction in Figure (4.14a), which shows that the use of Equation (4.1) results in the same unsteady heat transfer as predicted by UNSFLO. Therefore, the discrepancy between the measurement and the calculation is due to the prediction of the
inviscid flow, and in particular the shock strength, and not in the prediction of the viscous layer.

The lack of sufficient grid resolution through the boundary layer, especially at the wall, could contribute to this peak under-prediction. The new version of the numerical grid generation for UNSFLO (which has become available recently) allows a geometric stretching of the grids within the boundary layer (compared to the previous linearly spaced version). This feature results in a much higher resolution within the inner layer at the wall. Two numerical solutions of the stator/rotor interaction case were obtained for the same input conditions and number of grid points. The only difference was that in one the grid was linearly spaced through the boundary layer while on the other the spacing between the grid points was geometrically decreased. In Figure (4.15), the predicted surface heat flux for these 2 test cases at gauge 07 positions are shown. It is seen that the higher resolution at the wall has resulted in a small increase in the peak calculated Nusselt number values. The influence of the grid resolution on the time-resolved predicted heat transfer is less than 2%.

In Figure (4.16), the calculated Nusselt number at position 9 and its associated animation of the shock impingement as a function of the time in the period is shown. The shock identifier numbers are consistent with the previous Figures of (4.9). The arrows over the Nusselt number plot point to the passage of the shock and its subsequent compressional heating effect. The interaction of the 2a shock with blade at the gauge 9 position is identified to be the cause of the peak shown by the arrow in Figure (4.16a). In Figures (4.16b &c), the 2e and 2c shocks are identified to cause the sharp rise in the calculated Nusselt number respectively. The existence of the 2c shock is independently verified by analyzing the heat transfer on the suction surface of the nozzle guide vane. The time-resolved Nusselt number measurements at the back of the nozzle guide vane suction surface are shown in Figure (4.17). The peak identified in Figure (4.17a) is consistent with an interaction of the 2c shock structure. The measurement
further upstream (just prior to the throat) shows no sign of coherent unsteadiness in Figure (4.17b).

4.2.3 Surface Heat Flux Comparison: Pressure Surface.

In this section, the detailed time-resolved comparison of the Nusselt distribution on the pressure surface of the blade as measured and predicted are presented. These comparisons are presented in Figures (4.18) for design incidence (test T47) and Figures (4.19) for -10° incidence (test T52) cases. As opposed to the suction surface where the strong shocks cause high levels of compressional heating, on the pressure surface, the shock structures and their reflection tend to be much weaker. On the pressure surface, however, it was previously shown that the wakes are intermittently chopped by the rotor blades and in the rotor relative frame an unsteady wake/boundary layer interaction is established.

Design Incidence (T47): Near the leading edge on the pressure surface (Figure (4.18a)), the calculation under-predicts the mean level of heat transfer. The time-resolved comparison, however, seem to suggest that the calculation is correctly modelling the unsteadiness. Further down the pressure surface, the calculations show a distorted sinusoidal distribution (Figure(4.18b) ). The peaks in the calculation do match the corresponding peaks in the measurement, but the measurements do not exhibit the same troughs as in the calculation. The comparisons shown in Figures (4.18c &d), still exhibit slightly different unsteady component between the measurement and calculation. Near the trailing edge, the comparison between the calculation and the measurement show a good agreement both in the mean levels and in the nature of unsteadiness.

-10° Incidence (T52): For the -10° Incidence test case in Figures (4.19), similar to the suction surface comparisons, the unsteady component of the calculation seem to be somewhat
different, especially near the leading edge. The first half of the pressure surface still exhibited the troughs that were not observed in the data. The last two sensors on the pressure surface, in Figures (4.19 d &e), do seem to exhibit reasonable agreement between calculation and data.

4.2.3.1 Limitations of Algebraic Turbulence Modelling

A possible reason for the discrepancy between the measurement and the solution from UNSFLO could be due the way wakes are treated in the code. UNSFLO has neither any mechanism of associating turbulence content with the nozzle guide vane wake (limitation of the Navier Stokes/Euler hybrid method) nor transferring that turbulent energy from the freestream to the boundary layer (limitation of the current turbulence modelling).

The comparison between the data and the calculation in Figure (4.18b) is again shown in Figure (4.20b). The plot in Figure (4.20a) corresponds to the instantaneous Mach number profile through the boundary layer as calculated by UNSFLO at time equal to 0.75 for a sensor position on the pressure surface. The arrow in Figure (4.20b) points to the exact time in the period when this comparison is made. At this particular time within the period, both the data and the prediction exhibit high values of the Nusselt number. The external Mach number is at a relatively high value of about 0.3. At the same time, the calculated turbulent viscosity (non-dimensionalized by the laminar viscosity) within the boundary layer is plotted against normal height from the surface (non-dimensionalized by blade axial chord) in Figure (4.20c). This figure shows that the calculated turbulent viscosity is an order of magnitude greater than the laminar viscosity, hence the boundary layer at this position is predicted to be turbulent.

The same figures are plotted at a different time within the period and are presented in Figures(4.21). At a time equal to 1.1, the instantaneous Mach number profile through the boundary layer as calculated by UNSFLO is shown in Figure (4.21a). It is observed that, the
freestream Mach number is reduced to a value of about 0.1. The trough in the calculated Nusselt number is observed to correspond to this time within the period. In Figure (4.21c), the calculated turbulent viscosity within the boundary layer is shown for this particular window in time. It is observed that the calculated turbulent viscosity is now predicted to be of the same order as the laminar viscosity. At this time, the boundary layer in the code is not generating enough turbulence, which is consistence with the problems of this particular algebraic turbulence [64] model.

The algebraic turbulence model used in the code has no history effects within it and responds instantaneously to any change in the external flow, while in the physical flow, the turbulence has a physical time constant. The other physical phenomenon not correctly modelled is the freestream turbulence within the wake. In a wake/ boundary layer interaction process, the high freestream turbulent energy within the wake is believed to increase the turbulent mixing within the boundary layer.

4.3 Summary and Conclusion

In this chapter, through detailed time-resolved comparison between the measurements and the prediction, the deterministic unsteady phenomena that influence the heat transfer to the rotor were studied. The comparisons were performed at two different test conditions representing the nominal design and \(-10^\circ\) incidence test conditions.

1. The time-averaged heat transfer measurements were at or above the steady state turbulent predictions. In the front 50% of the wetted surface, the greatest difference between the time-averaged data and the steady state solutions were observed for both suction and pressure surfaces. Apart from the leading edge, the time-averaged unsteady prediction followed the calculated steady state turbulent levels.
2. The trajectory of the shock structures from the nozzle guide vanes as they propagated and were reflected were graphically displayed as a function of the time. The numerical solution predicted an unsteady shock reflection process with the shocks from more than one nozzle guide vane interacting with a rotor blade within a blade passing period. This detailed tracking of the shocks was later verified by comparing the surface peak heat flux with the prediction. The presented data from suction from the upstream nozzle guide vanes were also used to confirm the shock interaction process. Therefore, it is concluded that the numerical code is correctly capturing the nature of the unsteady shock interaction process.

3. The calculation predicted that wake/ boundary layer interaction on the suction surface occurs at all times within a blade passing period as the wakes from subsequent nozzle guide vanes tend to merge with each other. This suggests that in the rotor relative frame of reference, the wake/ boundary layer interaction on the suction surface can be considered as a steady phenomenon. On the pressure surface, however, the nozzle guide vane wakes are chopped by the rotor blade and convected downstream, which suggests the possibility of an intermittent wake/ boundary layer interaction. This intermittent interaction could not be clearly identified from the time-resolved measurements.

4. The comparisons of time-resolved heat transfer on the suction surface showed large levels of unsteadiness due to the passage of the shocks. The timing between the numerical calculation and the measurement was accurately predicted. This allowed the origin and the position of the individual shocks, as they interacted with the rotor blade surface to be identified. The sharp peaks in heat transfer were shown to be due to the shock interaction. Comparison of the numerical results, present data and the Rigby analysis, suggested that the inviscid phenomena are mainly responsible for the large peaks in the heat transfer. Under-prediction by UNSFLO of the level of the peaks in heat transfer when compared to the data, were shown to be due to
the inaccuracies of the inviscid part of the numerical solution. It is believed that the inaccuracies were due to incorrect specification of the streamtube height variation rather than a numerical problem. By demonstrating the sensitivity of the rotor heat transfer to the nozzle guide passage geometry, the results of Appendix (III) further support this conclusion. At the leading edge, the code predicted large unsteady heat transfer levels due to the passage of the 2a shock. The measurement, however, showed very little unsteady components. The cause of this discrepancy is not known.

5. By studying the details of the calculated levels of turbulent energy within the boundary layer, it was shown that periodically on the pressure surface, the predicted boundary layer had a low level of turbulence which resulted in a reduction of the predicted heat transfer. This reduction in the turbulence content of the boundary layer was attributed to the particular turbulence modeling used, and a one or two equation turbulence model might resolve this problem.
Reynolds number has an important influence on turbine heat transfer. It is often necessary to scale heat transfer measurements to Reynolds numbers other than those at which the data was taken for reasons of comparison among experiments, verification of computations, turbine design and evaluation of influence of operating flight conditions. The Reynolds number dependence of heat transfer through laminar and turbulent boundary layers on flat plates has been established, see Schlichting, but these techniques do not work reliably in turbines. The objective of the work described in this chapter is to develop relations for the scaling turbine heat transfer with Reynolds number.

Here, we develop the models, fit them to the data from the present turbine, and then apply them to several turbines extracted from the literature. Heat transfer from different sections of the blade will be compared against correlations for simple geometries. In particular, we will model the leading edge as a cylinder in cross flow with circulation and resolve the influence of the unsteadiness. The suction surface boundary layer will be treated as a fully turbulent flat plate, while the pressure surface boundary layer will be treated as a fully turbulent flat plate with enhanced mixing. The rational behind these choices is mainly empirical. Simple geometry models (flat plate and cylinder in cross flow) have been shown to provide a reasonable comparison measurements, see L.C. Daniels [71].

Researchers have studied the effect of Reynolds number on the mean heat transfer rates to the rotor of a gas turbine. Nicholson [72] studied two rotor blade profiles in a linear cascade and he showed that with the increase in the Reynolds number, the mean Nusselt number in both the laminar and turbulent regions is increased. He also showed that the transition point (when it
could be located) was moved towards the leading edge with the increase in the Reynolds number. Nicholson's results indicated that the level of freestream turbulence influenced the position of the transition point. Ashworth (1985) reported that the heat transfer to the ACE profile in a linear cascade was increased with the increase in the Reynolds number all around the profile. Blair's (1988) investigations on the rotor heat transfer of a rotating turbine stage revealed the same level of increase in the mean heat transfer as was previously reported in cascade experiments.

The external heat transfer to the rotor blade is commonly compared against the isothermal flat plate correlations of the form

\[ \text{Nu} = C \text{ Re}^n \]

Where \( c \) is a function of the Prandtl number and the gas to wall temperature ratios.

\[ n = 0.8 \] for turbulent boundary layers
\[ n = 0.5 \] for laminar boundary layers

In this section, the dependence of the measured heat transfer to the Reynolds number is investigated in detail. The profile will be separated into three regions; Leading Edge, Pressure Surface and Suction Surface. The experimental results obtained for each different region will be discussed and compared with the results of other workers.

5.1 Leading Edge Heat Transfer

The research on the leading edge heat transfer of a blade has been focused on using heat transfer data form cylinder in cross flow experiments, see L.C. Daniels. Many correlations have been suggested that relate the leading edge heat transfer to the Reynolds number and
freestream turbulence, eg. Lowery and Vachon [73]. The heat transfer at the leading of the rotor is dependent on the inflow Reynolds number. In this section, this dependence is investigated and comparisons with cylinder in crossflow is made. A particular attention has been made to the influence of unsteadiness on the presented measurements.

There are certain aspects of the present experimental setup that have to be taken account of before a direct comparison between the test data and cylinder in crossflow can be made.

1. The heat flux gauge covers a large portion of the leading edge circle area.

2. The unsteady potential field interaction between the stator and rotor in the experiment results in the movement of the stagnation point which was clearly observed in the unsteady numerical calculations performed, see Section 4.2.2. This movement changes the relative position of the instantaneous stagnation point with respect to the center of the measuring sensor, and hence results in a temporal averaging of the output of the heat flux sensor.

3. The heat flux gauge is positioned on the rotor blade by approximating the correct average location of the stagnation point; gauge output would be sensitive to any error in this position.

We will now examine the influence of each of the the above considerations, starting with the influence of the spatial averaging of the heat flux gauges mounted offset from the stagnation point of a cylinder in crossflow.

Cylinder in crossflow has been investigated extensively by Frossling [74] amongst others. Frossling showed that for air (Pr= 0.7), the heat transfer for $\theta= 0^\circ$ to $55^\circ$ onto a cylinder is given by:
\[
\frac{\text{Nu}_D}{\sqrt{\text{Re}_D}} = 0.945 - 0.510 X^2 - 0.596 X^4
\]

Eq (5.1.1)

\text{Re}_D is Reynolds number based on the cylinder diameter, \( \theta \) is the angle from the stagnation point and \( X = 0.5 \sin \theta \). Equation (5.1.1) can be written as

\[
\frac{\text{Nu}_D}{\sqrt{\text{Re}_D}} = 0.945 - \frac{0.510}{4} \sin^2(\theta) - \frac{0.596}{16} \sin^4(\theta)
\]

Eq (5.1.2)

Or in terms of multiples of the angle \( \theta \),

\[
\frac{\text{Nu}_D}{\sqrt{\text{Re}_D}} = 0.8673 + 0.0824 \cos(2 \theta) - 0.0047 \cos(4 \theta)
\]

Eq (5.1.3)

This expression slightly under-estimate the heat transfer (by 5%) at the stagnation point but is satisfactory elsewhere within the specified bounds.

In this section, the output of a heat flux gauge mounted in front of a cylinder which is 'slowly' oscillating in crossflow shown in Figure (5.1) is simulated and the output compared with the experiments. Assuming that the response of the sensor to varying heat load over its surface is to linearly average the heat flux, Eq (5.1.3) could be integrated to include the effect of spatial averaging. The oscillations, of frequency \( \omega \), are considered quasi-steady which implies that the diffusion time through the boundary layer is much smaller than the oscillation time, ie.

\[
\frac{\delta^2}{\nu} \ll \frac{2 \pi}{\omega}
\]

Eq (5.1.4)

Near the stagnation point, from Schlichting [9],
\[ \delta = 2.4 \sqrt{\frac{\nu D}{4 U_\infty}} \quad \text{Eq (5.1.5)} \]

Therefore Equation (5.1.4) and (5.1.5) are reduced to the form

\[ \frac{\omega D}{U_\infty} \ll 1 \quad \text{Eq (5.1.6)} \]

For the test cases relating to the turbine condition, Equation (5.1.6) is satisfied. It is assumed that the center of the heat flux gauge is following oscillatory motion of the form given in Equation (5.1.7), where \( \theta_0 \) is the steady state offset angle of the gauge center from the stagnation point and \( \theta' \) represents the amplitude of the oscillatory angle. Equation (5.1.3) is integrated both in time (for one period of oscillation) and in space for one gauge width (an angle of \( \theta_g \)) and the results compared with the leading edge heat transfer of the rotor.

\[ \theta_2 = \theta_0 + \frac{\theta_g}{2} + \theta' \sin(\omega t), \quad \text{and} \quad \theta_1 = \theta_0 - \frac{\theta_g}{2} + \theta' \sin(\omega t) \quad \text{Eq (5.1.7)} \]

Where \( \theta_1 \) and \( \theta_2 \) give the range of the spatial and temporal integration. Integrating Equation (5.1.3) with respect to \( \theta \), the spatially averaged Nusselt number is given by

\[ \left( \frac{NuD}{\sqrt{ReD}} \right)_{\text{Spatially Averaged}} = \frac{1}{\theta_g} \int_{\theta_1}^{\theta_2} F \, d\theta = F_0 - F_1 \sin(\omega t) - F_2 \sin^2(\omega t) \quad \text{Eq (5.1.8)} \]

And

\[ F_0 = 0.8673 + 0.0824 \left( \frac{\sin(\theta_g)}{\theta_g} \right) \cos(2\theta_0) - 0.0047 \left( \frac{\sin(2\theta_g)}{2\theta_g} \right) \cos(4\theta_0) \quad \text{Eq (5.1.9)} \]

\[ F_1 = 0.0824 \left( \frac{\sin(\theta_g)}{\theta_g} \right) (2\theta') \sin(2\theta_0) - 0.0047 \left( \frac{\sin(2\theta_g)}{2\theta_g} \right) (4\theta') \sin(4\theta_0) \quad \text{Eq (5.1.10)} \]
\[ F_2 = 0.0824 \left( \frac{\sin(\theta_g)}{\theta_g} \right) (2\theta^2) \cos(2\theta_0) - 0.0047 \left( \frac{\sin(2\theta_g)}{2\theta_g} \right) (8\theta^2) \cos(4\theta_0) \] Eq(5.1.11)

\( F_0 \) represents the spatial averaging of the sensor, while \( F_1 \) and \( F_2 \) represent the oscillatory change in the measured Nusselt number.

In the unsteady calculations of the stator rotor interaction, the position of the leading edge stagnation point was observed to move by approximately 5% of the chord around the mean which corresponds to \( \theta' = 0.99 \) radian, and the angle corresponding to the sensor length is given by \( \theta_g = 1.00 \) radian. For the case where the center of the sensor is exactly at stagnation point \( (\theta_0 = 0.0) \),

\[ \frac{\text{Nu}_{D}}{\sqrt{\text{Re}_D}} \text{ (spatially averaged, } t) = 0.9346 - 0.120 \sin^{2}(\omega t) \] Eq (5.1.12)

Integrating with respect to time for one period, the average Nusselt number is given by

\[ \left( \frac{\text{Nu}_{D}}{\sqrt{\text{Re}_D}} \right)_{\text{mean}} = 0.875 \] Eq (5.1.13)

The expression in Equation (5.1.13) predicts the mean output of a heat flux gauge mounted on an oscillating cylinder in crossflow. The spatial and temporal averaging result in a 7% lower measured Nusselt number compared to the leading edge value. In order to estimate the sensitivity of the measured heat flux to the exact positioning of the sensor, the integral in Equation (5.1.8) is evaluated with the value of \( \theta_0 \) corresponding to one gauge width. Equation (5.1.8) is integrated in time for one period and the average Nusselt number is given by

\[ \text{For } \theta_0 = 1.0, \quad \left( \frac{\text{Nu}_{D}}{\sqrt{\text{Re}_D}} \right)_{\text{mean}} = 0.819 \] Eq (5.1.14)
The inputs to this model are the extent of the movement of the stagnation point, the width of the sensor and the diameter of the cylinder. It is seen that the time averaged measurement of Nusselt number could be lowered than the leading edge value by as much as 13%. \( \text{Nu}_D/\sqrt{\text{Re}_D} \) vs time from Equation (5.1.12), and the measured heat flux from the leading edge sensor for the design test case is plotted in Figure (5.2) which gives a reasonable comparison between the measurement and the model. The relative timing relation is obtained from the results of the unsteady calculation of Chapter 4. The time-averaged measured (from the turbine tests) Nusselt number vs \( \sqrt{\text{Re}_D} \) is plotted in Figure (5.3) which clearly shows the linear dependence. The expressions in Equations (5.1.13) and (5.1.14) are also plotted in Figure (5.3). Figure(5.3) also may indicate the sensitivity of the measurements to the positioning of the leading edge sensor. In particular, Equation (5.1.14) is shown to match the data with the bounds of the experimental error.

### 5.2 Pressure Surface

On a curved surface, as the flow follows the curvature, a normal pressure gradient to the wall is induced, which is positive (away from the surface) in a convex wall and negative in a concave wall. In the boundary layer flow, the normal pressure gradient combined with the no slip condition has a stabilizing effect for a convex curvature and a destabilizing effect for a concave curvature. The instability due to the concave curvature can result in longitudinal vortex structures within the boundary layer \(^{75}\). Schlichting reports experimental results which suggest that the instability occurs for Goertler numbers (defined in Equation (5.2.1)) exceeding a value of 7, i.e.

\[
\text{Gö} = \frac{\theta U_{\infty}}{v} \sqrt{\frac{\theta}{R}} > 7
\]

Eq. (5.2.1)
The distribution of the Göertler number on the pressure surface of the rotor profile, calculated at the design Reynolds number, is shown in Figure (5.4). This calculation was performed using the steady state predicted (by UNSFLO) momentum thickness distribution around the profile. The blade boundary layer was specified to be fully turbulent. It is seen that the instability limit is exceeded near the leading edge, at around 10% wetted surface, and before the position of the first heat flux sensor. Ashworth performed a detailed analysis of the transition process on this profile, using the measured intermittency factors, in which he suggested that on the pressure surface, the natural boundary layer transition is superseded by the Göertler instability-induced transition. Using the present measurements, however, there is no method of determining the existence of Göertler vortices.

5.2.1 Pressure Surface Heat Transfer Correlation

Examination of the distribution of pressure side heat transfer revealed a common shape to the curves such that the Nusselt number distribution on the pressure surface collapsed to a single curve for varying flow Reynolds number. This form is given by;

\[
\text{Nu} = \text{Const. } \text{Re}^{0.8} \left[ 1 + a_1 \text{Re}^{n_1} \left( \frac{x}{s} \right)^{0.8} \right] \tag{5.2.1.1}
\]

where \(a_1\) and \(n_1\) are experimentally determined constants. The spatial term to the power of 0.8 corresponds to the growth of a turbulent boundary layer thickness on a flat plate. This expression is similar to a flat plate correlation with an enhancement term in the larger bracket. This equation was evaluated empirically for the 3 tested Reynolds numbers and the resulting fit is given below.

\[
\text{Nu} = \text{Const. } \text{Re}^{0.8} \left( 1 + 1.95 \times 10^{-8} \text{Re}^{1.25} \left( \frac{x}{s} \right)^{0.8} \right) \tag{5.2.1.2}
\]
This fitted equation was then compared with the available data for other turbine rotor geometries and test cases. It was found that there seems to be an effective origin where the enhancement term seems to apply. The final form of the correlation is shown in Equation (5.2.1.3). This expression is compared against the data from Nicholson cascade and Dring low speed rotor and the comparison is reported below. The Reynolds number is calculated based on rotor relative exit Mach number, axial chord and conditions for all the presented data.

\[
\text{Nu} = \text{Const. } \text{Re}^{0.8} \left( 1 + a_1 \text{Re}^{1.25} \left( \frac{x - x_g}{S} \right)^{0.8} \right)
\]

or

\[
\frac{\text{Nu}}{\left[ \text{Re}^{0.8} \left( 1 + a_1 \text{Re}^{1.25} \left( \frac{x - x_g}{S} \right)^{0.8} \right) \right]} = \text{Constant}
\]

\(x_g\) represents the effective origin of the correction term. The two parameters \(a_1\) and \(x_g\) are obtained experimentally for each geometry. The second term in the first bracket of the Equation (5.2.1.2) will be referred to as the correction term in the subsequent discussions.

The ACE profile, tested at MIT rotor facility and at Oxford cascade facility, represents the high pressure ratio military turbines. The measured Nusselt No. distribution for the ACE profile divided by the Reynolds No. to the power of 0.8 is plotted in Figure (5.5a) and the left hand side of the Equation (5.2.1.4) is plotted in Figure (5.5b) for \(x_g = 0\). It is observed that the 3 cascade results collapse into a single curve for a previously evaluated (from the rotor measurements) value of \(a_1 = 1.95\times10^{-8}\). Of course, the 3 sets of rotor data that were used to determine the terms of the Equation (5.2.1.2) do not validate the expression and are only included for comparison.
Nicholson LS profile represents a typical first stage commercial axial turbine rotor. This profile was tested in a linear cascade with a turbulence grid installed. In Figure (5.6a), the Nusselt No. distribution on the pressure surface divided by the Reynolds No. to the power of 0.8 for three different Reynolds numbers versus the fractional wetted surface is plotted. The left hand side of the Equation (5.2.1.4) is plotted in Figure (5.6b) for $x_g = 0.53$ and value of $a_1 = 2.17E-8$.

Dring rotor is a turbine profile which has been extensively tested in a low speed rotating facility. In Figure (5.7a), the Nusselt No. distribution on the pressure surface divided by the Reynolds No. to the power of 0.8 for three different Reynolds numbers versus the fractional wetted surface is plotted. The left hand side of the Equation (5.2.1.4) is plotted in Figure (5.7b) for $x_g = 0.60$ and value of $a_1 = 7.29E-8$.

The form of Equation (5.2.1.3) suggests two unknowns are required which can be thought of as a slope and an intercept. To use the present results as a predictive tool to obtain the pressure surface heat transfer, the experimental results from two different Reynolds numbers are sufficient. Given that the present scaling was not known as a priori, for the three profiles presented, three sets of heat transfer measurements for different Reynolds numbers were used in a regression line analysis to obtain $a_1$ and $x_g$. It is observed that the correlation suggested in Equation (5.2.1.3) collapses all 3 classes of the turbine rotor heat transfer as function of the Reynolds No. It must also be noted that there is very little pressure gradient on the pressure surfaces of modern turbines, except near the trailing edge. This empirically derived formulation provides a useful prediction tool by which the heat transfer distribution on the pressure surface can be estimated for all Reynolds numbers based on two experimental test results, to determine $a_1$ and $x_g$. 
5.3 Suction Surface

The heat transfer on the suction surface of the rotor is strongly affected by the shock interaction (from upstream nozzle guide vanes). It is believed that the influence of the wake and the shock wave from the upstream nozzle guide vane results in the movement of the transition point towards the leading edge. The comparison with the calculations performed in the last chapter, demonstrated the turbulent nature of the boundary layer on the suction surface when compared to prediction.

The measured Nusselt number distribution on the suction surface versus fractional surface length for 3 different Reynold's numbers are plotted in Figure (5.8a). In Figure (5.8b), the Nusselt numbers scaled by a factor of \(( \text{Re} / \text{Re}_{\text{des}} )^{0.8}\) are plotted. It is seen that the data does collapse within a narrow band up to the impingement point \((x/s = 0.7)\) of the shock from the pressure surface of the adjacent blade. After the shock impingement the Reynolds number dependence of the heat transfer does seem to be much higher than the 0.8 scaling. Delery [76] reported that downstream of a transonic shock boundary layer interaction region, the turbulent energy level is amplified which persists over 300 \(\delta^*\). The influence of this interaction on the heat transfer process is not yet fully understood, but from the evidence of Figure (5.8b) it seems to be dependent on the Reynolds number.

5.4 Summary and Conclusions

The experimental and analytical results in Section 5.1 demonstrated that the heat transfer at the leading edge of a turbine scales with the Reynolds number to the power of 0.5 which is consistent with the observations of a cylinder in cross flow. It was also shown that in a rotating environment, the movement of the leading edge stagnation point due to the unsteady potential field interaction results in a temporal averaging of the heat transfer. This averaging, lowers the mean leading edge heat transfer when compared with the steady state cascade tests.
or cylinder in crossflow. The results suggest that the mean heat transfer to the leading edge of a rotor blade is on the order of 10% lower than the cascade measurements for the same inflow conditions.

In Section 5.2, a semi-empirical correlation with respect to the flow Reynolds number was suggested. This correlation collapsed the pressure surface heat transfer for the 3 turbine blade profiles which were tested by 4 different researchers and the range of $290,000 < \text{Re}_c < 1,670,000$. It was argued that the concave wall curvature can produce shear instabilities within the boundary layer which in turn result in Taylor-Görtler vortices. It is believed that these vortices greatly increase the mixing within the boundary layer which results in an overall increase in heat transfer. The form of the correlation in Equation (5.2.1.3), suggests that this enhancement in heat transfer scales with the growth of the boundary layer thickness. The strong Reynolds number scaling dependence of the correction term is from a fitted correlation. The correlation requires 2 empirical constants. One of the constants, $x_g$, is the effective origin of the enhancement and the second empirical constant, $a_1$, is dependent on the particular profile. In order to fully understand the physical mechanisms influencing the pressure surface heat transfer, further research is required. Meanwhile, the suggested correlation provides a useful tool for the designer to estimate the pressure surface heat transfer for any Reynolds number, from the experimental results at two different Reynolds numbers.

On the suction surface, the $\text{Re}^{0.8}$ scaling worked well, except at the last 30% of the wetted surface. The shock from the pressure surface trailing edge of the adjacent blade is reflected at the rear of the suction surface. This shock/boundary layer interaction results in a complex flow field, which in turn affects the heat transfer process. The influence of the Reynolds No. scaling on the heat transfer in a transonic shock/turbulent boundary layer interaction is not yet understood.
The time-averaged Nusselt No. distribution measured at the mid-span of the MIT ACE rotor is plotted in Figure (5.9a). The Nusselt No. at any given position is then scaled to the design Reynolds No., using the scaling for the different sections of the blade as suggested above, which is then plotted in Figure (5.9b). It is seen that the three sets of data collapse within a narrow band of the experimental uncertainty. This result suggests that the influence of the Reynolds No. on the heat transfer of the turbine blade can now be reasonably estimated.
Chapter 6

6 Influence of Nozzle Guide Vane Coolant Ejection on the Rotor Heat Transfer at Midspan

6.1 Introduction

The interaction between an uncooled rotor and the upstream uncooled nozzle guide vane, is discussed in detail in the previous chapter.

In the modern gas turbines, the nozzle guide vanes are highly cooled, both internally and externally. The nozzle guide vane coolant is injected onto the surface as film cooling and also injected through slots near the trailing edge on the pressure surface in order to cool the trailing edge region. At the trailing edge of the blade, this injected coolant combines with the boundary layer from the suction surface to form the wake. For the velocity ratio of the injected coolant to the nozzle guide vane exit velocity of lower than unity, which is typical, the shear layer results in a relatively (compared to no injection case) thick wake. It is suspected that the influence of the injection on the downstream rotor heat transfer can be examined with relation to the structure of the incoming wake. Wittig et al. [77] studied the influence of the wake thickness on the downstream rotor heat transfer by transversing a single vane upstream of a linear stationary cascade.. Wittig showed that an increase of approximately 65% on the wake thickness resulted in a 35% rise on the pressure surface heat transfer. On the suction surface, Wittig showed that the thicker wake resulted in an earlier transition from laminar to turbulent boundary layer. The thicker wake did not influence the heat transfer on the turbulent boundary layer portion of the suction surface.

An experimental study of the influence of the slot ejection of the coolant from the nozzle guide vane trailing edge of a full stage turbine on the heat transfer processes were performed by
Dunn. The influence of the nozzle guide vane trailing edge slot coolant ejection on the downstream rotor time-averaged heat transfer was reported by Dunn [78]. Dunn's results showed that the influence of injection on the downstream rotor results in an increase of heat transfer by as much as 20%. Dunn showed that the influence of the injection gas temperature on the rotor heat transfer is generally small over most of the blade. Dunns results are compared with the present measurements in the following section.

To understand the influence of the nozzle guide vane injected coolant on the downstream rotor heat transfer, a series of tests were performed. In the tests, the coolant was supplied from the coolant supply tank into a plenum inside the nozzle guide vanes, and then injected near the trailing edge on the pressure surface, see Figure (6.1). The nominal design mass flow rate through the nozzle guide vane slots is approximately 3.0% of the main turbine flow rate. The coolant mass flow rate was adjusted by varying the coolant tank pressure, temperature and the area of the discharge orifice, with the coolant pressure and temperature measured directly inside the nozzle guide vane plenum. The pressure was measured using a surface mounted Kulite pressure transducer. The temperature was measured using a fine gas thermocouple located approximately in the middle of the plenum. The thermocouple was previously calibrated and the accuracy is believed to be on the order of 1°C. The reported Nusselt numbers are based upon the rotor blade axial chord, gas property at the wall and the mass-averaged (including the injected coolant) relative total temperature.

6.2 Influence of Injection on the Time-Averaged Rotor Heat Flux

In Figure (6.2), the mean Nusselt number distribution around the rotor profile with (T56) and without (T47) upstream nozzle guide vane coolant ejection are shown. Both test cases are at the design incidence and design Reynolds number. The injected to exit velocity ratios for this test series are listed in Table (2.7a). For test (T56), the injected to exit velocity ratiois 0.80.
On the suction surface, the effect of the blowing is small, while at the leading edge and the pressure surface an increase in the measured Nusselt No. is seen. This enhancement of the heat transfer on the pressure surface rises towards the trailing edge. These results are in agreement with the Wittig's thick wake study, except near the leading edge. In order to compare the present experimental results against Dunn's, the data is presented in Figure (6.3) as the ratio of the measured Nusselt no. with and without the injection. The test conditions (and the geometry) of the present study and Dunn's are different, but in order to make an elementary comparison, the present results and Dunn's are replotted in Figure (6.4). Dunn reports a rise in the heat transfer all around the blade, especially near the leading edge on the suction surface and towards the trailing edge on the pressure surface. Dunn's observed increase on the front section of the suction surface heat transfer is consistent with Wittig's observation of an earlier laminar to turbulent boundary layer transition.

In Figure (6.4), it has been observed that the nozzle guide vane trailing edge coolant injection increases the pressure surface heat transfer by as much as 30%, which is consistent for both the present study and Dunn's report. On the suction surface, the influence of the coolant injection is small (less than 10%).

6.3- Comparison with no Injection- Unsteady Results

A parameter that influences the surface heat transfer is the level of the external freestream turbulence level. The high level of the turbulent energy in the incoming wake increases the level of heat transfer to a fully turbulent boundary layer, see Bayley and Priddy[79]. It has been shown by Pfeil et al. that the characteristic length scale of the wake does not influence the heat transfer processes. Pfeil et al showed that the heat transfer enhancement is a function of the turbulent energy level of the wake. In an unsteady wake/ boundary layer interaction case, the time-averaged heat transfer at any point is a function of the fraction of the period when the
wake is directly influencing the boundary layer. This fraction of period is directly related to the width of the wake and the convection of the chopped wakes through the blade. In Section 6.2, it was argued that the influence of the nozzle guide vane trailing edge slot injection on the downstream rotor heat transfer, could be similar to the effect of a thicker wake interaction. In this section, the measured unsteady heat transfer around the rotor profile with and without injection are compared in detail.

In the present measurements, it was shown that the influence of injection on the heat transfer to the suction surface of the rotor is small (less than ±10%). For the present test case, the nozzle guide vane wake interaction was studied in detail in Chapter 4. By studying the time-accurate calculation of the nozzle guide vane wake and the downstream rotor, it was shown that on the suction surface, the wakes from consecutive nozzle guide vanes merge. This merging ensured that at every point on the suction surface, at any time within the blade passing period, the external flow is the turbulent wake flow. Any increase in the turbulent kinetic energy of the highly turbulent (over 5% turbulence level) wake flow due to injection should have a minimal influence on the rotor heat transfer. The conclusion that could be drawn from this observation is that any increase in the width of the wake thickness should not directly affect the suction surface boundary layer.

The comparison between the Nusselt number on the suction surface with and without the injection are shown in Figures (6.5a) through (6.5f). Note that the plotted ordinate scales in Figures (6.5a) and (6.5b) are different from the other figures shown on that page. On the back of the suction surface corresponding to the Figures (6.5c) through (6.5f), the unsteady Nusselt no. distribution with and without the nozzle guide vane coolant injection are shown. It is seen that at the back of the suction surface, the heat transfer is unaffected. On the front portion of the suction surface, however, the unsteady peaks in the measured heat transfer are smaller with the nozzle guide vane injection compared to the no injection case. In Chapter 4, it
was shown that these large unsteady peaks in heat transfer are associated with shock/boundary layer interaction. Lower peaks correspond to weaker shock strengths. It is observed that the nozzle guide vane coolant injection reduces the shock strengths (proportional to $M^2-1$), which implies a lower incoming flow Mach No. Adding the coolant mass increases the blockage for the main flow, which in a supersonic flow, reduces the flow Mach no. Therefore, it is observed that overall on the suction surface, the influence of the injection is limited to the slight change of the incoming shock strength and the associated heat transfer due to the strong interaction.

On the pressure surface, the calculations showed that the nozzle guide vane wakes, once chopped, are then stretched and convected downstream. Using the surface mounted heat flux gauges and schlieren photographs, Ashworth showed that in the cascade experiment, the front of the chopped wake was moving at around 80% of the freestream velocity, while the back was being convected at only 50% of the freestream velocity. The difference in the propagation velocity of the wake fluid results in a spreading of the wake, see Doorly. Thus, the fraction of a blade passing period that any point on the pressure surface is directly influenced by the turbulent flow of a chopped wake, increases from the leading edge towards the trailing edge. Any increase in the wake thickness should also raise this fractional wake resident time. Therefore, it is anticipated that the coolant injection should influence the unsteadiness in the heat transfer. The comparison between the Nusselt No. measurements on the pressure surface with and without the injection are shown in Figures (6.6a) through (6.6e). The output of the first sensor on the pressure surface illustrated in Figure (6.6a), is seen to be relatively unaffected by the injection. The change in the unsteadiness is initially seen in Figure (6.6b). Figures (6.6c), (6.6d) and (6.6e) show the gradual increase in the influence of the coolant injection on the pressure surface heat transfer.
6.4 Influence of Change in the Blowing Parameters

The experimental results previously presented, were attained at the design injection conditions. In order to determine the influence of the injection at different conditions, a series of tests performed. The injected parameters were changed around the design conditions with the coolant to the main velocity ratios of 0.76, 0.80 and 0.85. The time-averaged effect of injection on the heat transfer for the given test conditions are plotted in Figure (6.7). Again, it is seen that the suction surface is relatively unaffected by the injection while the influence on the pressure surface is important (ie. order of +30%). The change in the rotor Reynolds number due the slot injection is less than 3% and can not account for the observed 30% variation. It is seen that for 3 test cases shown, the change in the Nusselt no. is primarily similar.

6.5 Summary and Conclusion

In this chapter, the influence of the nozzle guide vane trailing edge coolant injection on the downstream rotor heat transfer was studied in detail. It was argued that the injection reduces the nozzle guide vane exit Mach no. for a transonic stage, which corresponds to a weaker shock interacting with the rotor boundary layer. This weaker shock results in a lowering of the unsteady component of heat transfer on the front of the suction surface. The influence of the injection on the back of the suction surface was negligible. On the pressure surface, the injection increased the heat transfer. This change in the unsteadiness was shown on the measured surface heat transfer. It is expected that at some point, the combined effect of successive wakes (and injected coolant) to merge as the wake thickness is increased. At that point, no further enhancement in the heat transfer due to the wake passing is expected.
Chapter 7

7. Rotor Film Cooling

In this chapter, the experimental results of the study of heat transfer with film cooling on the rotor of a fully cooled turbine stage are presented. The objective of this work is to determine the three-dimensional heat transfer in the presence of film cooling on the rotor of a fully scaled transonic turbine. The approach is to compare the time-averaged and the time-resolved film cooled measurements with the uncooled data. The comparison of the time-resolved film cooled and uncooled heat transfer measurements are also used to quantify the influence of the periodic unsteadiness generated by the rotor/stator interaction on the film cooling process.

There are four sections in this chapter. The first section introduces the experimental approach and is followed in the second section by the presentation of the time-averaged and the time-resolved fully cooled heat transfer measurements at three span-wise positions (nominal tip, mid and hub sections) around the rotor profile. These measurements are compared against the corresponding uncooled data. In the third section, the importance of the rotor/stator periodic unsteadiness on the film cooling is shown and modelled. The predictions from this model are then compared against the available measurements and show good agreement. In the last section, the conclusions of this study are discussed.

7.1 Introduction

The need to film cool the blading and the end-walls of the turbine blade has resulted in considerable research and development. Initially, these studies focused on the fundamentals of film cooling on two-dimensional flat and curved plates in steady flow. More recently, the steady state study of linear cascades has provided detailed measurements of film cooling
performance on a three-dimensional blade surface. These cascade experiments do not simulate the unsteady rotating environment of a rotor blade, and as such, the unsteady effects have been generally ignored. The main focus of the present study is to quantify the influence of three-dimensional and unsteady effects on the rotor film cooling process.

In Figures (7.1), the composite of the positions of the heat flux sensors for the three instrumented blades are shown and the exact positions around the blade profile are listed in Table (2.4b). In this chapter the terms "tip", "hub" and "mid" are used in reference to the nominal tip, nominal hub and nominal mid-span positions. The tip and hub span-wise locations were selected to study the 3-D nature of the flow, but were deliberately selected to be far enough from the endwalls so as to lie outside of the expected [69] secondary flow and tip vortex structures. In Chapter 2, the position of the sensors and the numbering scheme were introduced. For the time-resolved measurements, the reference point in time of each rotor blade in the pitchwise nozzle guide vane relative position was different. All the time-resolved data for the hub (cH) and the tip (cT) section instrumented blades have been shifted in time to correspond to the mid (cM) section instrumented blade's relative pitchwise position. The coolant holes all had an internal diameter of 0.5 mm (2% of axial chord), but different lengths. The exit from the holes of the SS1 coolant row (see Figure (7.1)), had a "shaped" design which provided an increased lateral film coverage immediately downstream of the holes.

In the series of experiments performed, the heat transfer to the blade at different coolant and main stream conditions were measured. The conditions of these experiments were previously listed in Table (2.8a). The coolant parameters were varied to match the coolant to main mass and momentum flux ratios, coolant to wall temperature ratios and the coolant ratio of specific heats. As discussed in Chapter 1, the variation of these parameters for a fixed geometry determined the coolant flow conditions.
7.2 Cooled Heat Transfer Distribution at three Span-Wise Positions

The time-averaged distribution of heat transfer around the blade profile at three span-wise positions (nominal tip, mid and hub sections) were measured. In this section, these measurements are compared against the previously obtained mid-span and tip section uncooled rotor measurements. There are no uncooled hub-section measurements available for comparison. Therefore, only a relative parametric comparison of the hub-section heat transfer measurements in the presence of film cooling are performed.

In Figure (7.2a), the film cooled Nusselt number distribution around the blade is plotted for the three instrumented blades against the mid-span uncooled results. It is seen that film cooling reduces the heat load, especially on the suction surface. In Figure (7.2b), the Nusselt number distribution on an unwrapped view of the blade surface is shown. This figure is an interpolated contour plot of the Nusselt number distribution as measured by the twenty-six heat flux gauges. This figure gives a view of the heat transfer distribution on the blade surface, where the heat transfer is high (dark areas) and where it is low (lighter areas).

In this section, the comparison between the film cooled and uncooled measurements for each span-wise position is presented separately. The film cooled measurements at mid-span are also compared against similar cascade experiments on the same geometry and external flow parameters as reported by Rigby et al.\textsuperscript{[80]}

7.2.1 Mid Section

The uncooled mid-section instrumented test case (T61), is used for comparison with the film cooled measurements. This comparison is performed for the same Reynolds number and at \(-10^\circ\) incidence test cases. In the uncooled rotor test (T61), the nozzle guide vanes have trailing
edge coolant ejection at the same conditions as the film cooled tests. The nozzle guide vane coolant injection test was used for comparison in order to match the rotor inflow boundary conditions.

**Suction Surface:** In Figure (7.3), the comparison of Nusselt number distribution between the film cooled (T71 and T75) and the uncooled (T61) tests is shown. The two tests (tests (T71) and T75) were performed in order to check the repeatability of the time-averaged film cooled measurements. There were no film cooled measurements available on the pressure surface due to instrumentation problems. On the suction surface, film cooling results in a reduction in the heat transfer downstream of both the SS1 and SS2 sets of cooling holes.

The measured time-resolved Nusselt number for the sensor just downstream of the SS1 coolant holes (34% of fractional wetted surface) for the two film cooled consistency tests of (T71) and (T75), and the corresponding uncooled measurements (31% of fractional wetted surface) are plotted in Figure (7.4a). The film cooled unsteady measurements are seen to be consistent from test to test. The nature of the unsteady Nusselt number on the suction surface is altered by the presence of film cooling. It is believed that this change is due to the interaction of the blowing through the coolant holes and the external pressure field. The unsteady external pressure field results in a modulation of the coolant blowing ratio and hence an unsteady film cooling effectiveness, as will be discussed in Section 7.3.

Film cooled heat transfer measurement (test T71) of gauge cM18, just before the SS2 coolant holes (66% of fractional wetted surface), and the nearest uncooled measurement (65% of fractional wetted surface in test T61), are shown in Figure (7.4b). The features of the curves are similar, with some phase lag and modulation also observed.
In Figure (7.4c), the comparison between the film cooled versus uncooled measurements just
downstream of the SS2 coolant row is presented. It is observed that for approximately 30% of
the time, the coolant does not influence the heat transfer. The comparisons for the last gauge
on the suction surface (gauge cM20) downstream of the SS2 coolant row, are shown in Figure
(7.4d). It is seen that the measurements for the repeatability tests (T71 and T75) do not match
well against each other. A factor that contributes to this apparent discrepancy is the low levels
of the measured heat transfer for the cM20 gauge, resulting in relatively higher measurement
errors, see Figure (7.3) for the error bars.

**Pressure Surface:** In another test at -10° incidence (T64), a single measurement of the film
cooled heat transfer rate downstream of the PS1 coolant row was available. The coolant to
main temperature ratios for test T64 ($T_m/T_c = 1.71$) and test T71 ($T_m/T_c = 1.94$) are different.
In Figure (7.5), the measurements of heat transfer for test T64 and test (T71) are plotted
against the uncooled results. The single operating gauge (cM21) downstream of the PS1
coolant row shows no influence of film cooling.

The measured time resolved Nusselt number of gauge cM22 for the film cooled test T64 and
uncooled test (T61) are presented in Figure (7.6a). This figure shows little influence of cooling
on the heat transfer when compared to the uncooled measurements. To understand this lack of
cooling effect, film cooling at another test condition is studied. In Figure (7.6b), the
measurement of heat transfer from sensor cM22 for the film cooled test T63 and the uncooled
measurement of test (T61) are shown. The blowing ratio for the PS1 coolant hole row is 1.1
for T63 and 1.5 for T64, with the momentum ratio of order unity for both tests (see Table 7.1).
The measurements in Figure (7.6b) exhibit reduction in the level of heat transfer, as well as an
influence of film cooling on the unsteady heat transfer process. From the above observations,
it is suggested that in the high blowing ratio condition of test T64, the coolant film lifts off of
the surface and provides no cooling effect. For the lower blowing ratio of the test T63, the
coolant adheres to the surface and reduces the heat transfer. There are no film cooled heat transfer measurements available on the mid-section instrumented (cM) blade downstream of the PS2 coolant hole row.

**Cascade Results Comparison:** In this section, the time-averaged film cooled results of the present measurements are compared against the available cascade results. In a series of experiments performed by Rigby et al [80], the film cooling effectiveness was measured around the ACE profile, with the same cooling hole geometry as the present study. These tests were reported for the $-10^\circ$ incidence and the matched Reynolds number, see the test conditions in Table (7.1). In Figure (7.7), the measured Nusselt number distribution for the film cooled (T71) test case is plotted versus the uncooled (T61) measurements and is scaled by the film effectiveness measured in the Rigby [80] cascade experiments. The film effectiveness at each position is linearly extrapolated (for different blowing ratios) from the reported measurements. Downstream of the SS1 coolant holes on the suction surface, the cascade results under-predict the film effectiveness when compared to the present cooled measurements. This figure shows a better performance of film cooling for the present rotating test case when compared to a similar cascade. Therefore, it is concluded that film cooling of turbine blades can not be fully simulated in a stationary cascade.

Rigby et al [80] also measured the film cooled heat flux in the presence of simulated upstream shock and wake passage, and showed that there was an effect of simulated nozzle guide vane shock and wake passing on the performance of the two suction surface film cooling rows. They demonstrated that downstream of the SS1 coolant row, the film cooling reduces the level of the Nusselt number in the full bar passing cycle, including the sharp peak region at the base of the passing shock wave.
7.2.2 Tip Section

In this section, the comparison between the film cooled and the uncooled measurements at the tip section of the blade are presented. The measurements from only one uncooled tip-section test at -10° incidence (test T112) is available. The measurements from T112 are plotted in Figure (7.8) against the predicted laminar and turbulent Nusselt number at the same test conditions. The measured Nusselt number distribution at mid-span for test T112 is also shown for comparison. For the tip section results, it is observed that on the pressure surface, the measurements follow the turbulent solution. The leading edge measurement is substantially below the predicted levels. On the suction surface, the data seems to follow the laminar prediction for the first 50% of the wetted surface prior to the adjacent blade shock interaction region, and the turbulent prediction for the rear half. From the above observation it is suggested that the suction surface boundary layer underwent transition at around 60% of the wetted surface.

In the film cooled tests, the coolant holes trip the boundary layer [81], which effects the heat transfer. Using the Reynolds number scaling procedure of Chapter 5, the Nusselt number measurements of test T112 are scaled to match the corresponding cooled data in the following comparisons. In test T112, the nozzle guide vanes have solid trailing edges with no coolant ejection. In Chapter 6, nozzle guide vane coolant injection was shown to result in a higher pressure surface heat load which is not modelled in the present film cooled comparisons for the tip section.

The measurements of the film cooled tests (T71 and T75), are compared against the uncooled results in Figure (7.9). The test condition is at -10° incidence and nominal design blowing ratios. The data suggests low levels of film effectiveness on the rear portion of the suction
surface before and after the SS2 coolant row. The single measurement downstream of the PS1 coolant row shows a lower level of heat transfer, exhibiting the influence of the film cooling.

**Suction Surface:** The measured unsteady Nusselt number for the cT blade is compared against the uncooled test (T112) results in this section. In Figure (7.10a), the film cooled Nusselt number downstream of the SS1 coolant holes measured at sensor cT06 for the two tests (T71) and (T75) are compared against the nearest uncooled sensor from test (T112). It was previously suggested that the front suction surface boundary layer on the tip is laminar. The uncooled sensor from T112 is within the laminar region on the front part of the suction surface. In this figure it is observed that the film cooled heat transfer measurements are substantially greater than the uncooled case. This supports the observation that the coolant holes trip a laminar boundary layer and in fact result in a higher heat load than without cooling.

The first gauge downstream of the SS2 coolant row is the sensor cT07. The film cooled measurements from this gauge are compared against the nearest gauge from the uncooled test (T112) in Figure (7.10b). Film cooling is shown to result in lower heat transfer for the major part of the period. The unsteady component of the uncooled measurement contrasts with the low levels for the film cooled results. The film cooled measurement (test T71) from cT08 (last gauge downstream of the SS2 coolant row on the suction surface) is plotted against the uncooled result in Figure (7.10c). This figure shows the reduction in heat transfer due to film cooling.

Hence, for the turbulent boundary layer, film cooling clearly lowers the heat transfer; while in the laminar region (on an uncooled blade), the film cooling in fact enhances the heat transfer by tripping the boundary layer. Qualitatively, the film cooling effectiveness on the tip section seems to be lower than was observed for the mid-section.
Pressure Surface: Downstream of the PS1 coolant row on the pressure surface, film cooled heat transfer from gauge cT10 is compared against the uncooled measurements from test (T112) in Figure (7.11a). The position of the cT10 gauge is located in between two uncooled gauges, and therefore, its Nusselt number measurements are plotted versus the two nearest (downstream and upstream) uncooled gauges. In this figure, it is observed that the film cooling lowers the heat transfer, which contrasts with the mid-section results shown in Figure (7.6a). The results of Figure (7.11a) is, however, very similar to the mid-section measurements at the lower blowing ratio shown in Figure (7.6b).

In the film cooled test T63, a single measurement (sensor cT12) of the film cooled heat transfer rate downstream of the PS2 coolant row was available. T63 is at the same external flow conditions as (T71) but at lower coolant blowing ratios, see Table (7.1). In Figure (7.11b), the measurement from test T63 is plotted against the nearest uncooled measurements. This figure shows no reduction of the heat transfer due to film cooling; in fact, some enhancement of the heat transfer is observed. It was previously suggested that the uncooled results of test T112 do not simulate the influence of the upstream nozzle guide vane injection.

7.2.3 Hub Section

In this section, the film cooled measurements from the cH blade are presented. There are no uncooled measurements of the hub-section heat transfer available for comparison with the cooled tests. For the sake of reference, the uncooled heat transfer measurement at the mid-section is also included in the following figures. The incidence angle at the hub-section is very close (2 to 3 degrees higher) to that of the mid-section, and hence, the mean heat transfer distribution is expected to be similar.
The time averaged Nusselt number measurements of the film cooled tests (T71 and T75), are compared against the uncooled test (T61) results in Figure (7.12a). The test condition is at -10° incidence and nominal blowing ratios. The data suggests low levels of heat transfer on the suction surface for the film cooled tests when compared to the uncooled. Except for the first sensor (cH27) downstream of the SS1 coolant row, the other film cooled measurements are considerably lower than the uncooled mid-span results. On the pressure surface, the film cooled and uncooled mid-span measurements are similar. The comparison between the tests (T71) and (T75) measurements are well within the error bounds of the data except for the first sensor downstream of the PS2 coolant row. In Figure (7.12b), the measurements from the test (T71) at mid-span and hub-section are compared against the mid-section results of (T61). It is seen that except for the first sensor downstream of the SS1 coolant holes, the film cooled results from the mid and hub sections are very similar.

**Suction Surface**: The time-resolved Nusselt number distributions for the film cooled hub-section (cH blade) are compared against the associated (similar axial position) mid-section cooled and uncooled measurements (when available). The unsteady measurements from the first sensor (cH27) downstream of the SS1 coolant row and the nearest uncooled and film cooled mid-span data are presented in Figure (7.13a). There is little similarity between the mid-span and hub-section film cooled results. The variation in the form and phase of the unsteady signal from the cH27 gauge and the uncooled mid-span measurement might be due to the change in the external shock structure at the hub compared to the mid-section.

In Figure (7.13b), the measurement from the cH28 sensor and the repeatability tests (T71 and T75) are compared against the film cooled mid-span data. There is no uncooled mid-span measurement available at this position. In contrast to the previous figure, the film cooled mid-span and hub-section are seen to be similar in the form of the unsteadiness. The hub-section heat transfer is, however, lower than the corresponding mid-span values. Further downstream
of the SS1 coolant row, the measurements from the film cooled hub section (cH29) from tests (T71) and (T75) are compared against the uncooled mid-span data in Figure (7.13c). In Figure (7.13d), the measurement from the last sensor upstream of the SS2 coolant row (cH30) for the hub-section film cooled condition is plotted against the mid-span film cooled and uncooled data. It is observed that the mean and the unsteady level of heat transfer in the presence of film cooling seems similar for the hub and mid-span measurements. Downstream of the SS2 coolant row, the unsteady measurement from the cH31 and cH32 gauges are plotted against the film cooled and uncooled mid-span results in Figures (7.13e) and (7.13f) respectively. Some dissimilarities between the hub and mid-span film cooled results are observed, but overall, the level of unsteadiness seems to be comparable.

On the suction surface, the cooled hub blade showed considerably lower heat transfer when compared to the uncooled mid-span data with the exception of the region close to the SS1 coolant row. All the hub section measurements (except at the cH27 sensor in Figure (7.13a) ), showed a similar unsteady level of heat transfer when compared to the mid-span film cooled data. The experimental measurements showed good repeatability, as evidenced by the results of tests (T71) and (T75).

Pressure Surface: In this section, the film cooled hub-section measurements are compared against the uncooled mid-span results. There are no available mid-span film cooled measurements on the pressure surface at these test conditions. The unsteady measurements of the cH34 gauge downstream of the PS1 coolant row is presented in Figure (7.14a). In this figure, the film cooled hub measurement seems to be similar to the uncooled mid-span data. Downstream of the PS2 coolant row, the measurements from the cH35 and cH36 gauges for test (T71) are plotted against the uncooled mid-span results, as shown in Figures (7.14b) and (7.14c) respectively. The mean level of the hub section film cooled measurements are slightly lower than the mid-span uncooled measurements.
Due to the lack of uncooled hub-section measurements for comparison, the performance of film cooling at this section is unclear. Assuming that the uncooled hub-section heat transfer were at the same level as the mid-span, the film cooling would be very effective on the suction surface and have a small influence on the pressure surface. On the suction surface, the unsteady nature of the heat transfer is similar for the hub and mid-span film cooled results. The experimental measurements showed good repeatability (except for the cH35 gauge) as evidenced by the results of tests (T71) and (T75).

### 7.3 Influence of Unsteadiness on Film Cooling

#### 7.3.1 Introduction

It was shown that film cooling not only alters the mean level of the heat transfer, but changes the unsteady nature of the process. In this section, the influence of unsteadiness on film cooling is examined. It will be shown that the unsteady rotor/stator potential field interaction directly affects the cooling performance.

In Section 7.2.1, the effect of film cooling on the unsteady heat transfer rate was first introduced. By examining Figure (7.4a), it was observed that in the front part of the suction surface, the film cooling seemed to provide reduction in the heat load for some part of the blade passing period, while providing no protection in another portion. In Figure (7.15), the time resolved isothermal film effectiveness (defined as the fractional change of film cooled to uncooled Nusselt number for an isothermal wall), measured by the cM15 sensor, is plotted. Isothermal film cooling is defined as the reduction in the heat transfer due to film cooling with respect to the uncooled level normalized by the uncooled heat transfer at a constant wall
temperature. It is observed that the film effectiveness is roughly constant for a portion of the period before dropping to zero, and then returns to its previous constant level.

Previous studies of turbine heat transfer have not addressed the interaction process of unsteadiness and film cooling. Two possible mechanisms explaining this level of unsteadiness in the effectiveness are: (1) coolant unsteady lift off from the surface (2) modulation of the coolant blowing parameters by the external flow. In the bar passing cascade experiments of Rigby et al, the coolant from the SS1 row showed no signs of lift off at any blowing conditions. For the SS1 coolant row, the convex wall curvature downstream of the coolant row tends to force the coolant towards the surface, reducing the risk of a film lift off.

The second mechanism suggested is the unsteady interaction of the blowing parameters with the external flow. In Figure (7.16), the maximum, the minimum and the time-averaged blade surface pressure distribution as predicted by UNSFLO, as well as the mean coolant plenum pressure for the film cooled test (T71) are shown. At the position of the coolant holes, the mean pressure differentials across the holes and the magnitude of the peak to peak pressure fluctuations are shown. When the magnitude of the peak to peak fluctuation is much smaller than the mean pressure differential, as is for the SS2 coolant row, the influence of unsteadiness is small. It is seen that for the SS1 coolant row and both coolant rows on the pressure surface(PS1 and PS2), the peak to peak unsteady surface pressure variation is of the same order of magnitude as the mean coolant plenum to surface differential pressure, and as such, unsteadiness is important.

To analyze this phenomenon, a simple model of the flow through the coolant hole and its influence on the cooling of the surfaces downstream is developed and presented in Section 7.3.2. In Section 7.3.3, the results from the model are compared against the cooled measurements.
7.3.2 Unsteady Blowing Model

In order to investigate the influence of unsteadiness, a basic film cooling model in the presence of unsteady coolant exit pressure field is suggested.

Assume the flow from a two-dimensional coolant slot of length \( L \) and width \( d \) onto the flat plate of length \( C \), with a periodic slot exit static pressure, shown in Figure (7.17). The coolant protects the flat plate downstream of the slot. The coordinate along the plate is defined by the symbol \( S \) and the slot is positioned at an arbitrary position along the plate. The amount of the coolant mass is injected through the coolant slot is a function of the imposed pressure ratio for an unchoked flow.

The parameter which characterizes the importance of the unsteadiness is the product of the reduced frequency and the flow mean Mach number \( (M_c, \text{ based on the mean flow and speed of sound velocities in the coolant slot}) \), which is the ratio of time scale associated with the acoustic transmission through the slot \( (L/a_c) \) and the time scale of the periodic external flow \( (1/\omega) \).

\[
\text{Reduced Frequency, } \quad \Omega = \frac{\omega L}{U_c}
\]

For \( \Omega M_c << 1 \), flow through the slot is considered to be quasi-steady.

For \( \Omega M_c > 1 \), flow through the slot is highly unsteady and no perturbation is transmitted upstream. Note that in the rotor frame of reference, the unsteady time scale \( (1/\omega) \) is the upstream nozzle guide vane passing period. For the SS1 coolant hole and the test (T71) conditions, the value of the reduced frequency is 0.4. Therefore, at unchoked conditions, the coolant behavior would be dependent on the unsteady nature of the flow.
The continuity and the momentum equations for a one-dimensional, inviscid, compressible and unsteady flow are,

\[
\frac{\partial p}{\partial t} + U \frac{\partial p}{\partial x} + \rho \frac{\partial U}{\partial x} = 0 \quad \text{Eq (7.3.2.1)}
\]

\[
\frac{\partial U}{\partial t} + U \frac{\partial U}{\partial x} + \frac{1}{\rho} \frac{\partial p}{\partial x} = 0 \quad \text{Eq (7.3.2.2)}
\]

The flow parameters may be written in terms of the mean value and a perturbation term.

\[
p = \bar{p} + p'
\]

\[
p = \bar{p} + p'
\]

\[
U = \bar{U} + u = \bar{U} \left[ 1 + \frac{\partial \varphi(x,y,z,t)}{\partial x} + \ldots \right] \quad \text{Eq (7.3.2.3)}
\]

Where \( \varphi \) is the perturbation potential.

For an isentropic flow process, \( p' = \bar{a}^2 \rho' \quad \text{Eq (7.3.2.4)} \)

By manipulating the Equations (7.3.2.1) through (7.3.2.4), the perturbation terms are reduced. Linearized around the mean flow for a small potential perturbation, \( \varphi \) satisfies the following equation, see Ashley and Landahl [82].

\[
\left( 1 - \bar{M}^2 \right) \varphi_{xx} - \frac{2 \bar{M}^2}{U} \varphi_{xt} - \frac{\bar{M}^2}{U^2} \varphi_{tt} = 0 \quad \text{Eq (7.3.2.5)}
\]

In the above expression, the non-linear transonic term has been neglected; the rational for this omission is as follows: maximum Mach number in the coolant hole is one at the exit of the
holes, corresponding to choked conditions. The argument for not having the transonic terms is that the results of this section are used as a first order correction to the reduction in the mean mass flux out of the holes for the unchoked part of the period. On the pressure surface of the blade, the flow Mach number in the coolant holes is designed to be much lower than unity at (about 0.2 - 0.5) and the flow never chokes.

Assuming a harmonic unsteady perturbation of the form \( \varphi = f(x) \, e^{i\omega t} \), the above linear partial differential equation reduces to the following linear ordinary differential equation (the subscript \( x \) and \( t \) correspond with differentiation with respect to \( x \) and \( t \)):

\[
\left(1 - \frac{M^2}{U^2}\right) f_{xx} - \frac{2i \omega M^2}{U} f_x + \frac{\omega^2 M^2}{U^2} f = 0
\]

**Eq (7.3.2.6)**

and \( \frac{1}{2} C_p = \frac{P - \bar{P}}{\rho U^2} = -\frac{\varphi_x}{U} - \frac{\varphi_t}{U} \)

**Eq (7.3.2.7)**

or \( \frac{P - \bar{P}}{\rho U^2} = -\left(f_x + \frac{i\omega f}{U}\right)e^{i\omega t} \)

**Eq (7.3.2.8)**

Assume that the external pressure for the flat plate surface is given by a travelling wave with an arbitrary origin:

\[
\frac{p}{\bar{p}} = 1 + \sum_{\omega = \omega_1}^{\infty} A_\omega e^{i\left(\omega t + \lambda s\right)}
\]

**Eq (7.3.2.9)**

At the exit from the slot; \( x = 0 \) & \( S = 0 \), \( \frac{p'}{\bar{p}} = \sum_{\omega = \omega_1}^{\infty} A_\omega e^{i\omega t} \)

**Eq (7.3.2.10)**

Thus, the perturbation potential is obtained as
\[ \varphi = \frac{-A(\omega) e^{i\omega t}}{\frac{i\omega}{U}} \left[ \frac{\exp \left[ -\frac{\bar{M}}{1-M^2} \frac{i\omega L}{U} \left( \frac{x(1-M)}{L} + 1 \right) \right]}{(1+M)} \exp \left[ -\frac{\bar{M}}{1-M^2} \frac{i\omega L}{U} \right] } \right] \]

Eq (7.3.2.11)

Summing for all frequencies and ignoring the second order terms (terms involving the square of the perturbation), the correction to the mean mass flow from the slot is given by

\[ \overline{(\rho U)} = (\overline{\rho} + \overline{\rho'}) (\overline{U} + u') = \overline{\rho U} + (\overline{\rho'u'}) = \overline{\rho U} , \text{ when } (\overline{\rho'u'}) \ll \overline{\rho U} \]

Therefore,

\[ \frac{(\rho U)^'\rho}{\rho U} = \overline{\rho u + \overline{U} \rho'} \]

Eq (7.3.2.12)

At \( x = 0 \),

\[ \frac{(\rho U)^'\rho}{\rho U} = \sum_{\omega = \omega_1}^{\infty} -A(\omega) e^{i\omega t} \left[ 1 - \frac{(1-M) \exp \left[ -\frac{\bar{M}}{1-M^2} \frac{i\omega L}{U} \right]}{(1+M) + (1+M) \exp \left[ -\frac{\bar{M}}{1-M^2} \frac{i\omega L}{U} \right] } \right] \]

Eq (7.3.2.13)

The mass flux from the coolant slot including the first order correction, is written as

\[ \frac{\rho U}{\rho U} = \frac{\overline{\rho U} + (\rho U)^'\rho}{\rho U} = \sum_{\omega = \omega_1}^{\infty} (1 - A(\omega) G(\omega) e^{i\omega t}) \]

Eq (7.3.2.14)

Where,

\[ G(\omega) = 1 - \frac{(1-M) e^{i\kappa} + (1+M) e^{i\kappa}}{(1-M) e^{i\kappa} - (1+M) e^{i\kappa}} \]

and \( \kappa = \frac{\bar{M} \Omega}{1-M^2} \)

Eq (7.3.2.15)
From Equations (7.3.2.14) and (7.3.2.15), it is seen that as $\omega \to 0$, the term $G(\omega)$ also approaches zero and the quasi steady conditions are recovered. In Figure (7.18), the absolute value of $1 - G(\omega)$ (for a unit perturbation) versus the reduced frequency for different mean flow Mach numbers through the slot are plotted. Referring back to the case of interest, the amplitude of pressure waves (non-dimensionalized by the mean inlet total relative pressure) over the SS1 coolant row (as predicted by UNSFLO for test (T52)) is shown in Figure (7.19).

For a reduced frequency of 0.4, it is suggested that considerable change in the mass flux through the slot would occur due to the unsteady exit pressure.

It has been determined that for a coolant slot with an unsteady exit pressure, both the amplitude and the reduced frequency of the pressure fluctuations influences the coolant mass flux; implying an unsteady blowing ratio. The relevance of this unsteady blowing ratio on the coolant effectiveness is studied here. We introduce the parameter $\tau$ to characterize the importance of the unsteady blowing ratio on the boundary layer behavior. It is the ratio of the coolant fluid convection time scale $(S/U_\infty)$ and the time scale associated with the periodic external flow $(1/\omega)$.

$$\tau = \left( \frac{\omega S}{U_\infty} \right) = \left( \frac{\omega C}{U_\infty} \right) \left( \frac{S}{C} \right)$$  \hspace{1cm} \text{Eq (7.3.2.16)}$$

In the present application, the value $(\omega C/U_\infty)$ is approximately one (C is the blade chord). Then,

$$\tau = \left( \frac{S}{C} \right)$$  \hspace{1cm} \text{Eq (7.3.2.17)}$$

When the distance from the coolant holes is small with respect to the chord $(S<< C)$, the value of $\tau$ is small and the film cooling is quasi-steady. In this case, the correlations reported in the
literature by other investigators could be used in a quasi steady manner to determine the film cooling performance. In a study performed by Whitten \cite{83} on heat transfer to a turbulent boundary layer with non-uniform blowing, it was shown that the performance of film cooling is a strong function of local boundary layer conditions, and retains little effect of its prior history. This study supports the use of a quasi-steady film effectiveness (based on the local film condition) in the presence of a quasi-steady blowing. This approach results in a quasi-steady film cooling effectiveness which is estimated by the time history of the coolant blowing.

To estimate the adiabatic film cooling effectiveness, Eq. (1.4.14) is used with the time dependent mass flux ratio of Eq (7.3.2.14). Further, the film cooled heat transfer coefficient is assumed to be the same as the uncooled heat transfer coefficient, which is consistent with the results of section (1.4.1) away from the injection holes (for S/d >10). The important requirement from this correlation is that it should correctly represent the physics of the scaling of the effectiveness with the injected coolant mass flux; i.e. the relative accuracy of the correlation is less important. Away From the slot ( S/ d >10), Equations (1.4.14), (1.4.15) and (1.4.16) are approximated by

For \( \eta_{ad} = (T_{\infty} - T_{ad}) / (T_{\infty} - T_c) \)

\[
\eta_{ad} = 5.75 Pr^{2/3} \xi \cdot 0.8 \left( \frac{Re_c \mu_c}{\mu_m} \right)^{0.2} \beta^{-1} \tag{7.3.2.18}
\]

Where, \( \beta = 1 + \left( 1.5 \times 10^{-4} \cos \alpha \frac{Re_c \mu_c}{\mu_m} \right) \tag{7.3.2.19} \)

\[
\xi = \frac{S}{d} \left( \frac{\rho U}{m} \right) \left( \frac{\rho U}{c} \right) \tag{7.3.2.20}
\]
\( \alpha \) is the injection angle, \( Re_c \) is the Reynolds number based on the coolant fluid through the slot, and the parameter \( \beta \) is given by Eq (7.3.2.19). It is assumed that the correction to the driving temperature due the compressibility is negligible. The ratio of the coolant to the mainstream mass flux is written as

\[
\frac{(\rho U)_c}{(\rho U)_m} = \left( \frac{\rho U_c}{\rho U_m} \right) \left( \frac{U_c}{U_m} \right)
\]

Eq (7.3.2.21)

The time dependent blowing ratio is used in the expression for the adiabatic film effectiveness, and hence by combining Equations (7.3.2.14), (7.3.2.18), (7.3.2.20) and (7.3.2.21), the adiabatic effectiveness is written as,

\[
\eta(S, t) = 5.75 \ Pr^{2/3} \ (1 - \omega, S = 0) G(\omega) e^{i \omega \left( t - \frac{S}{U_f} \right)}
\]

Eq (7.3.2.22)

The time lag required for the coolant fluid to reach any point \( S \), is accounted by the term \( U_f \), which corresponds to the convection velocity of the coolant fluid. Here, the value of \( U_f \) is assumed to be 70% of the freestream velocity, which corresponds to the mean propagation velocity of a turbulent patch within the boundary layer, as measured by Ashworth [39]. The ratio of the coolant convection velocity and the mainstream velocity (\( U_f/U_m \)) is named \( \varepsilon \). Thus, the film cooling effectiveness is the function of both time and space.

\[
\eta_{ad}(S, t) = \bar{\eta}(S) \left[ 1 - A(\omega, S = 0) G(\omega) e^{i \omega \left( t - \frac{S}{\varepsilon U_m} \right)} \right]^{0.8}
\]

Eq (7.3.2.23)

Where,

\[
\bar{\eta}(S) = 5.75 \ Pr^{2/3} \ (1 - \omega, S = 0) G(\omega) e^{i \omega \left( t - \frac{S}{U_f} \right)}
\]

Eq (7.3.2.24)
The adiabatic film effectiveness at any position, $S$, is written in terms of a steady state value $\bar{\eta}(S)$, modulated by the time dependent term given in Equation (7.3.2.23). This time dependent term is a function of the external pressure perturbation and the convection velocity of the coolant fluid. In Chapter one, it was argued that the ratio of film cooled to uncooled heat transfer rate can be (from Eq (1.4.3)) expressed as

$$\frac{\dot{Q}_{fc}}{\dot{Q}_0} = \frac{h_{fc}}{h_0} \left( \frac{T_{ad} - T_w}{T_{\infty} - T_w} \right) = \frac{h_{fc}}{h_0} \left( 1 - \Theta \eta_{ad} \right)$$

Eq (7.3.2.25)

$$\Theta = \left( \frac{T_{\infty} - T_c}{T_{\infty} - T_w} \right)$$

Eq (7.3.2.26)

In section (1.4.1), it was shown that away from the coolant holes ($S/d > 10$), the ratio of the film cooled to uncooled heat transfer coefficients is close to one (i.e. $h_{fc} \approx h_0$). Assuming that the heat transfer coefficient is unchanged by the addition of the coolant, Equation (7.3.2.25) is expressed as

$$Q_{fc} = Q_0 \left( 1 - \Theta \eta_{ad} (S, t) \right)$$

Eq (7.3.2.27)

Both the film cooled and uncooled heat transfer rate in Equation (7.3.2.27) are non-dimensionalized by the conductivity of the main gas at the wall temperature, temperature difference between the wall and the freestream and the axial chord.

$$N_{u, film \ cooled, (S, t)} = N_{u, uncooled, (S, t)} \left( 1 - \Theta \eta_{ad} (S, t) \right)$$

Eq (7.3.2.28)

From the knowledge of the uncooled unsteady Nusselt number and the coolant adiabatic film effectiveness (combining Equations (7.3.2.23) and (7.3.2.28)), the unsteady film cooled Nusselt number is determined.
7.3.3 Model Comparison Against Data

In this section, the left hand side of Equation (7.3.2.28) is evaluated and compared against the film cooled measurements. For the uncooled unsteady Nusselt number, measurements of test (T61) were used. The unsteady surface pressure at the coolant exit was predicted using UNSFLO and used to evaluate the time dependent blowing ratio. The equivalent slot width for the coolant holes was determined from Eq (1.4.17).

The comparison (-10° incidence test case) between the measured uncooled (T61), measured film cooled (T75) and the predicted film cooled (from above model) for the gauge cM15 (downstream of the SS1 coolant row) is shown in Figure (7.20). This sensor is eleven coolant hole diameters downstream of the coolant row. It is seen that the predicted film cooled Nusselt number tracks the measured unsteady film cooled data. On the pressure surface, the measurement from the uncooled (T61), film cooled (T63) and the film cooled prediction from the present model for the cM22 sensor are shown in Figure (7.21). The model seems to predict the unsteady nature of the film cooled measurements, but over-estimates the cooling effect. It seemed to predict the unsteady nature of the heat transfer for the film cooled data.

The above small perturbation analysis showed that the film cooled heat transfer to the surface is dependent upon the external pressure oscillations. In Figures (7.22a) and (7.22b), the physical coupling between the unsteady blowing and propagating shock wave is illustrated. With the shock moving upstream towards the slot, the coolant exit pressure ahead of the shock is relatively low, resulting in a greater mass flux through the slot and a higher blowing ratio. In the vicinity of the shock, the boundary layer is compressed, resulting in a large compressional heating of the wall. This high level of heat transfer occurs when the local blowing ratio is high, see Figure (7.22a). As the shock passes by the coolant slot, the coolant exit pressure rises, resulting in a lower blowing ratio. Downstream of the coolant slot, the compressional
heating is reduced, resulting in a lower level of heat transfer. Thus, this low level of heat transfer occurs when the local blowing ratio is low, see Figure (7.22b). The out of phase nature of this interaction results in the greater availability of the coolant fluid when the heat transfer is high and a lower amount of the coolant fluid when the heat transfer is low. Hence, the time-averaged film effectiveness at any point downstream of the coolant hole would be higher than measured in steady state. For a downstream moving shock wave, the opposite of the above interaction process occurs, resulting in a lower time-averaged film cooling effectiveness than when compared to the steady state.

In a transonic turbine, the primary upstream nozzle guide vane trailing edge shock wave (shock 2a from Figure (4.9)) interacts with the rotor blade on the suction surface. This shock is moving upstream on the suction surface and downstream on the pressure surface relative to the blade surface. Therefore, the rotor time-averaged film cooling effectiveness would be greater on the suction surface and lower on the pressure surface than the steady state measurements from a cascade.

The coolant holes on the suction choke and unchoke periodically and the mass flux from the holes changes depending on not only the mean differential coolant pressure but also on the shape of the pressure time history. Periodic choking results in a lower mean mass flux than the steady state and as such a direct comparison of the unsteady blowing with a steady state case on the suction surface would not be very accurate. Not accounting for this effect, the unsteady coupling of the coolant blowing and the pressure field resulted in a 12% decrease in the time averaged heat transfer for the cM15 sensor on the suction surface (Figure(7.22a)) compared to the steady state blowing resulted and a 5% increase in the time averaged heat transfer for the cM22 sensor on the pressure surface (Figure(7.22b)) compared to the steady state blowing. Using Equation (7.3.2.28), it is estimated that the unsteady blowing resulted in a 5% decrease
on the suction surface and a 5% increase on the pressure surface heat transfer for the present test case.

From the results of this study, the following design guidelines could be obtained. On the suction surface, coolant holes should be positioned as close to the leading edge as possible to maximize the beneficial interaction with the downstream moving pressure wave. Coolant holes should be as short as possible (given structural constraints) and placed at or just upstream of (to avoid injeestion) the position where the unsteady external pressure results in no flow through them. On the pressure surface, coolant holes should be designed to be as long as possible to reduce the unsteady interaction with the external pressure field.

7.4 Summary and Conclusion

A series of fully cooled turbine stage experiments were performed. The experiments consisted of the measurement of time-resolved heat transfer on the surfaces of three instrumented blades. In this chapter, the data has been presented and compared against the previously obtained uncooled measurements. Both time-averaged and time-resolved measurements were repeatable. The time-averaged film cooled heat transfer showed some span-wise variation.

The film cooled comparison with the mid-span showed considerable reduction in heat transfer on the suction surface. The detailed time-resolved comparison for the suction surface heat transfer revealed that the nature of the film cooled heat transfer is considerably different from the uncooled data. A model was presented to account for the influence of the unsteady blowing from the coolant holes due to the potential field interaction between the rotor/stator. Using a one-dimensional compressible linearized model of the flow through the coolant holes, the unsteady coolant blowing ratio was estimated. The coolant hole exit pressure fluctuations were obtained using UNSFLO. By using a simple film cooling flat plate correlation, the time-
resolved film effectiveness was obtained. The prediction of this model was compared against the film cooled data. On the suction surface, the prediction closely tracked the data. On the pressure surface, the prediction slightly over-estimated the amplitude of the film effectiveness, but the correct phase relation was obtained. It is concluded that the film cooling effectiveness on the rotor is coupled to the unsteady pressure field interaction. Therefore, the estimate of the heat transfer in a rotating film cooled environment obtained from steady state cascade measurements of uncooled heat transfer and film cooling effectiveness might be inaccurate.

On the pressure surface at mid-span, it was shown that a high blowing ratio could result in negligible film effectiveness. Lowering the blowing ratio seemed to considerably change the film performance and provide a better protection. A possible cause of this phenomenon could be due to the film lift off at the high blowing ratios. This result suggests an optimum blowing ratio for the film cooling on the pressure surface.

The comparison between the steady state film cooled cascade measurements on the same blade profile obtained by Rigby and the present data was presented. It was shown that on front part of the suction surface, the cascade heat transfer measurement was about 100% higher than of the rotor data. At the back of the suction surface, the two sets of data were much closer.

The uncooled heat transfer measurements were compared against the film cooled measurements at the tip section. It was suggested that the film cooling trips the boundary layer on the front part of the suction surface. This tripping of the boundary layer resulted in a higher heat transfer with film cooling than without. Reduction in heat transfer due to film cooling on the pressure and the back of the suction surfaces was observed, which seemed to be lower than that for the mid-span comparisons.
Due to the lack of uncooled heat transfer data, the measurements from the hub section could only be compared relative to each other. The unsteady nature of the film cooled data on the suction surface of the hub instrumented blade seemed to be similar to the mid-span film cooled measurements. On the pressure surface, the influence of film cooling could not be determined. If the uncooled hub section heat transfer were at the same level as the mid-span, then the film cooling would be very effective on the suction surface and have a relatively less influence on the pressure surface.
Chapter 8

A series of experiments was performed to characterize the heat transfer to the rotor blade profile of a rotating, fully scaled, transonic turbine stage. The experimental measurements of heat transfer on the cooled rotor blades are believed to be the first time-resolved data of their kind in a fully scaled transonic turbine. The aim of this study was to identify and analyze the unsteady physical features of the flow that influence heat transfer, using this unique collection of data. The summary of the results were given at the end of each chapter. In this chapter, the main conclusions to be drawn from the present study are described in the following sections.

Conclusions and Recommendation for Future work

Film Cooled Turbine Stage

The heat transfer measurements at 3 span-wise positions on the rotor blade of a fully cooled turbine stage were presented. The acquisition of this time-resolved heat transfer data is a major contribution of the present study. This unique data, time-averaged and time-resolved, provides a database for future studies. Overall, the film cooling reduced the suction surface heat transfer substantially, while on the pressure surface, a lower film cooling levels were observed.

The unsteady nature of film cooled heat transfer was seen to be very different from the uncooled data. This difference was shown to be due to the unsteady blowing from the coolant holes which resulted on the coolant flow propagating as a travelling wave. The unsteady surface blade pressure due to the rotor/ stator interaction results in an unsteady coolant blowing ratio. This coupling between the film cooling and the unsteady pressure field influences the time-averaged heat transfer. The measurement of the film effectiveness in a steady state cascade would not account for this coupling. The present result is the first time that the influence of unsteadiness on the film cooling effectiveness has been shown.
Unsteady Rotor Heat Transfer at Mid-Span

The unsteady phenomena that influence the heat transfer to the rotor of an uncooled high pressure turbine blade were studied. One of the available tools for this study was the two-D viscous unsteady numerical code (UNSFLO). The mid-span heat transfer measurements were directly compared against the results of the computational code. It was shown that the time-averaged data exceeded the calculated turbulent boundary layer on the front portion of the blade. At the rear of the blade, the data followed the turbulent boundary layer prediction. The time-resolved calculated heat transfer were shown to be well predicted (within 20%) by UNSFLO along much of the blade surface. This is the first time that the measured time-resolved heat transfer from a fully scaled turbine rotor has been directly compared with the numerical predictions. In the the stator/rotor interaction studies, the periodic trajectory of the shock structures from the nozzle guide vanes and their reflections were tracked. It was shown that the sharp unsteady peaks in measurements are due to the interaction of the shocks with the blade surface.

Uncooled Rotor Heat Transfer with Nozzle Guide Vane Coolant Ejection

The effect of nozzle guide vane coolant injection on the heat transfer of the downstream rotor was experimentally investigated. The results of this study showed that the nozzle guide vane coolant injection increases the rotor pressure surface heat transfer. This increase was postulated to be due the thickening of the wake (by the addition of the coolant) and its subsequent unsteady interaction with the pressure surface.

Reynolds No. Correlation of Heat Transfer

A semi-empirical relationship correlating the influence of the variation of the external flow Reynolds number on the rotor blade heat transfer, was presented. The correlation treated different sections of the blade (the leading edge, suction surface and pressure surface)
separately. At the leading edge a modified version of the heat transfer to a cylinder in cross-flow and on the suction surface, the turbulent flat plate correlations seemed to scale the measurements. It was shown that by adding an enhancement term to the turbulent flat plate heat transfer correlation, the pressure surface heat transfer is scaled. The pressure surface correlation was shown to be applicable for other turbines when compared against available data in literature. Although, the physical rational of the pressure surface correlation is not well understood, the present work provides a tool for the heat transfer comparison among experiments, verification of computations, turbine design and evaluation of influence of operating flight conditions.

**Experimental Facility**
An accomplishment of the present work is the design of the coolant facility. The mass flux, coolant to wall temperature ratio and ratio of specific heat capacities of the cooling gas are fully scaled to match the engine environment. The film cooled experiments are believed to be the first time that a fully cooled turbine stage has been tested in a transient facility with all the scaling parameters matched. The use of this experimental setup facilitated detailed film cooled measurements of heat transfer on the rotor profile.

**Recommendation for Future Work:**

1. The results of UNSFLO were shown to predict the unsteady time-resolved heat transfer reasonably well. At times, the algebraic turbulence model was at shown to result in an incorrect prediction (boundary layer relaminarization predicted when no relaminarization was observed in data). This was suggested to be due to the lack of history effects in the turbulence model. A more sophisticated turbulence would capture the time lags present in the boundary layer. A more difficult proposition is the modelling of the interaction of the highly turbulent
wake flow and the boundary layer. The numerical modelling of this interaction, should be addressed at some point.

2. The BlowDown Turbine Facility simulates most of the non-dimensionalized flow parameters of an engine environment. There are two distinguishable physical phenomena that are not matched to a real engine; inlet distortion and freestream turbulence. The inlet distortion could be in the form of circumferential (hot streaks) or radial distortion. In the past few years, some measurements of the heat transfer in the presence of a radial temperature profile were obtained. The question of the influence of hot streaks on the turbine rotor blade is still to be determined. The more basic examination of the interaction of the freestream turbulence with the profile boundary layer still remains.

3. The results of Chapters 4 and 6 point to an interaction process between the pressure surface boundary layer and the incoming nozzle guide vane wakes. Some work on the modelling of this process was previously cited. Most of these studies, however, model the interaction by assuming a rapid transition of the boundary layer. The current models do not include the process of turbulent energy entrainment (from the wake) by the boundary layer, which is essential for an accurate prediction of heat transfer on the pressure surface.

4. The Reynolds number scaling of Chapter 5, was shown to provide a useful tool for scaling of heat transfer data. Despite considerable effort, the physical interpretation of the enhancement term are still unknown.

5. The film cooled measurements presented provide an extensive database for future comparisons with analysis, experiments and calculations. In terms of the present study, the comparisons between the film cooled and uncooled heat transfer measurements at the tip and the mid span-wise positions were presented. There were no hub section uncooled data
available. In the future experimental efforts at MIT Turbine BlowDown Facility, a set of uncooled hub-section heat transfer measurements should be obtained. This would allow for the comparison of the film cooled and uncooled measurements at the hub-section.

6. The results of Chapter 7 demonstrated the influence of the unsteadiness on the film cooling process and its subsequent effect on the rotor heat transfer. The presented model seemed to reasonably predict the unsteady coupling between the external potential field interaction and the coolant hole unsteady blowing. The present results should provide sufficient stimulations for the future studies of the influence of unsteadiness on film cooling. This phenomenon should be simulated and fully analyzed in a representative fashion.
Appendix I

Scaling Design of Coolant Supply

In this appendix, the parameters that influenced the design of the coolant supply tank are discussed. The coolant supply system was designed as a second blowdown facility, in which the coolant to main gas blowing mass flux and the temperature ratios were maintained at a constant level for the duration of the test. Assuming an isentropic expansion process in the supply tank and an isentropic flow through the inlet duct, Guenette [59] has derived the analytical expression for the isentropic blowdown mass flux from the main tank supply.

\[
\dot{m}_{\text{sup}} = A^* \rho_{\text{sup}}(t=0) \sqrt{(\gamma_m \, R_m \, T_{\text{sup}}(t=0))} \left( \frac{2}{\gamma_m + 1} \right)^{(\gamma_m+1)} \left( 1 + \frac{1}{\tau_{\text{sup}}} \right)^{-(\gamma_m+1)} \quad \text{Eq I.1}
\]

In the above expression, the subscript (sup) corresponds to the supply tank. The term \( t \) corresponds to the blowdown time constant of the supply tank and can be written as;

\[
\frac{1}{\tau} = \frac{(\gamma - 1) \, A^* \, A \, \sqrt{\gamma \, R \, T(t=0)}}{2 \, \text{Vol} \, \left( \frac{2}{\gamma+1} \right)^{\gamma+1}} \quad \text{Eq I.2}
\]

\( A^* \) to the effective choked area. For the supply tank, \( A^* \) corresponds to the choked throat of the NGVs plus the effective area of the boundary layer bleeds. A similar expression can be derived for the coolant supply tank.

\[
\dot{m}_{\text{cool}} = A^* c \rho_{\text{cool}}(t=0) \sqrt{\gamma_c \, R_c \, T_{\text{cool}}(t=0)} \left( \frac{2}{\gamma_c + 1} \right)^{(\gamma_c+1)} \left( 1 + \frac{1}{\tau_{\text{cool}}} \right)^{-(\gamma_c+1)} \quad \text{Eq I.3}
\]

Dividing the Equation I.3 by I.1 and combining the constant terms, the overall coolant blowing ratio of the facility as a function of time can be written as;
The aim of the design procedure was to maintain the right hand term independent of time for the test duration. The blow time constant of the blowdown facility is typically on the order of 20 seconds, while the test period is less than 1 second. Thus, the Equation 1.4 can be expanded with respect to time;

\[
\frac{\dot{m}_{\text{cool}}}{\dot{m}_{\text{sup}}} = \left( \frac{\dot{m}_{\text{cool}}}{\dot{m}_{\text{sup}}} \right)_{(t=0)} \left[ 1 + \frac{t}{\tau_{\text{cool}}} \frac{(\gamma_c+1)}{(\gamma_m+1)} \right] + O \left( \frac{1}{\tau} \right)^2
\]

Eq I.5

Where,

\[
\tau_{\text{cool}} = \tau_{\text{sup}} \frac{(\gamma_c+1)}{(\gamma_m+1)} \frac{(\gamma_m+1)^{-1}}{(\gamma_m+1)^{-1}}
\]

Eq I.6

The coolant tank supply tank was designed such as to maintain the coefficient of time in Equation I.5, as close to zero as possible. In order to scale the correct specific heat ratio for the coolant gas, a mixture of a two gases was used, one heavy (Freon 14) and one lighter (Argon). To avoid condensation of the gases at the high pressure and low temperature of the coolant supply, Freon 14 was used. For given gas properties coolant and initial coolant temperature (determined from the temperature scaling), the ratio of the tank volume to the throat area determined the time constant of the coolant supply tank. There were some engineering limitations (size of the tank for the available space and the maximum discharge area determined by the size of the coolant valve) that had to be accounted. In the final design, the volume of the tank was 0.114 m³ and the area of the choked orifice was changed to match the blowdown time constant. In a typical test (T71), the time constants for the main and coolant tanks were measured to be 23.79 and 22.8 seconds respectively. From Equation I.5, the blowing ratio is calculated to increase by 3% during the test period, from a nominal 6% at 250 ms to 6.2% at 550 ms.
Appendix II

Uncertainty Analysis

In this appendix, the uncertainty analysis for the presented time-averaged heat transfer data are discussed. The heat transfer has been measured using the heat flux gauges previously mentioned in Chapter 2. Determining the uncertainty in the data are based up on a suitable evaluation of the contributions from different sources of error. These sources of error were introduced from both the static and run time calibrations which had to be accounted.

For the duration of the test period (250 to 550 ms), the turbine is operating at a single operating condition, see Guenette [59]. During this time, the time-averaged heat flux is measured using the difference between the upper (u) and the lower (l) temperature sensors of the heat flux gauges using the following equation.

\[
\dot{Q} = \left( \frac{k}{\delta} \right)_{\text{kapton}} (T_u - T_l)
\]

Eq (II.1)

In terms of what is actually being measured, ie. scales, zeros and voltages, u and l refer to upper and lower sensors. For S corresponding to the scale factor of the sensor, V is the measured voltage and Z is a reference temperature, the Equation II.1 is written as;

\[
\dot{Q} = \left( \frac{k}{\delta} \right)_{\text{kapton}} (S_u V_u - Z_u - S_l V_l + Z_l)
\]

Eq (II.2)

Or,

\[
\dot{Q} = \left( \frac{k}{\delta} \right)_{\text{kapton}} ([S_u V_u - S_l V_l] - (Z_u - Z_l))
\]

Eq (II.3)

And,

\[
(Z_u - Z_l) = \Delta T_E
\]

Eq (II.4)
The Equation (II.2) has been rewritten in the form of the Equation (II.3) to separate the run

time (Equation (II.4) ) from the static calibration uncertainties. The term $\Delta T_E$ is the absolute

indicated difference between the upper and the lower sensor indicated temperatures at the end

of a test. At the end of a test (the rotor has stopped for almost 9 minutes), the metal
temperature of the blade (made from aluminum) is uniform. The maximum level of heat
transfer to the blade surface (mainly natural convection) is estimated to be less than 1 kW/m$^2$
which corresponds to a difference of about 0.1 $^\circ C$ between the upper and lower sensors. At
the end of the tests, a typical difference of about 0.3-0.5 $^\circ C$ was observed. This error is
causd by the slight change in the resistance of the sensors. For the sensors used in the film
cooled tests that experienced "jumps" (see figure 2.10), the correction of the jump also
contributes towards the uncertainty.

The overall uncertainty in the heat flux is calculated from a combination of different
components (Equation (II.5) ), see Bevington [84] and Moffat [85]. In writing the Equation
(II.5), there is an implicit assumption that the uncertainties are uncoupled (all the covariances of
different terms are zero).

$$
\sigma_Q^2 = \left( \frac{\partial Q}{\partial (k/d)} \right)^2 \sigma_{(k/d)}^2 + \left( \frac{\partial Q}{\partial S_1} \right)^2 \sigma_{S_1}^2 + \left( \frac{\partial Q}{\partial V_{u}} \right)^2 \sigma_{V_{u}}^2 + \left( \frac{\partial Q}{\partial V_{l}} \right)^2 \sigma_{V_{l}}^2 + \left( \frac{\partial Q}{\partial \Delta T_E} \right)^2 \sigma_{\Delta T_E}^2
$$

Eq (II.5)

Combining Equations (II.3), (II.4) and (II.5) and for $\sigma$ corresponding to the variance of the
measurement;

$$
\sigma_Q^2 = \left( \frac{Q}{k/d} \right)^2 \sigma_{(k/d)}^2 + \left( \frac{k}{d} \right)^2 \left[ V_{u}^2 \sigma_{V_{u}}^2 + V_{l}^2 \sigma_{V_{l}}^2 + S_{u}^2 \sigma_{S_{u}}^2 + S_{l}^2 \sigma_{S_{l}}^2 \right] + \sigma_{\Delta T_E}^2
$$

Eq (II.6)
In order to evaluate the uncertainty in the measured heat flux from the above equation for any given test, the unknowns are $\sigma_{k/d}$, $\sigma_{S_u}$, $\sigma_{S_I}$, $\sigma_V$, $\sigma_{V_1}$, $\sigma_T$. Using the method suggested, different levels of uncertainty are separated and studied.

There is a quantization error associated with the measurement of the raw voltages. In the present data acquisition system, 12 bit analog to digital converters are used which correspond to a quantization error of order $2^{-12}$. The values of $\sigma_V$, $\sigma_{V_1}$ are negligible compared to the other contributions (for example $\sigma_T$), and is ignored. Hence:

$$\sigma_Q^2 = \left(\frac{Q}{k/d}\right)^2 \sigma_{[k/d]}^2 + \left(\frac{k}{d}\right)^2 \left(\frac{V_2^2 \sigma_{S_u}^2}{Q} + \frac{V_1^2 \sigma_{S_I}^2}{Q}\right) + \left(\frac{k}{d}\right)^2 \sigma_T^2$$

Eq (II.7)

The scaling calibration of the sensors were observed to be very stable from test to test as well as to the bath calibrated value. The scales values for all the sensors used were checked at the end of every test. This was achieved by matching the gradient of the change in the temperature with the output from the blade resistance temperature device for the last five minutes of the post-run calibration. The sensor scaling stability was seen to be better than 0.5%.

$$\frac{\sigma_{S_I}}{S_I} \& \frac{\sigma_{S_u}}{S_I} \leq 0.005$$

Eq (II.8)

$$\left(\frac{\sigma_Q}{Q}\right)^2 = \left(\frac{\sigma_{k/d}}{Q}\right)^2 + \left(\frac{k}{d}\right)^2 \left(\frac{V_2^2 \sigma_{S_u}^2}{Q} + \frac{V_1^2 \sigma_{S_I}^2}{Q}\right) + \sigma_T^2$$

Eq (II.9)

The term $\sigma_T$ is evaluated by combining the two different variances; the difference in the indicated temperature between the sensor and the blade resistance temperature device from the bath calibration ($\sigma_{T_E,Ca,l}$), and the end of run calibration ($\sigma_{T_E,t}$) both contribute to this.
uncertainty, with the latter dominating. Assuming that the calibration uncertainty plus test to test variations on the calibration are uncorrelated;

\[ \sigma_{TE}^2 = \sigma_{T_E Cal}^2 + \sigma_{T_E t}^2 \]  

Eq (II.10)

\( \sigma_{T_E t}^2 \) is obtained from the end of test results.

The presented measurements are in the form of Nusselt number distributions. In order to correctly evaluate the overall uncertainty of the data, the contributions of the terms used in the non-dimensionalization of the measurement would also have be accounted for. The Nusselt number is determined in terms of the measured heat flux, axial chord, the conductivity of the main gas \( (k_{gas}) \) evaluated at the upper surface temperature, and the difference between the total inlet relative \( (T_{T_{rel}}) \) and the upper sensor temperatures.

\[ \text{Nu} = \frac{\dot{Q} \cdot C}{(T_{T_{rel}} - T_u) \cdot k_{gas}} \]  

Eq (II.11)

The blade surface is close to room temperature and the small test to test variations have a negligible influence on the gas conductivity at the wall. Thus, the uncertainty in the Nusselt number is evaluated from the following expression.

\[ \left( \frac{\sigma_{Nu}}{Nu} \right)^2 \approx \left( \frac{\sigma_{\dot{Q}}}{\dot{Q}} \right)^2 + \left( \frac{\sigma_{T_{T_{rel}}}}{T_{T_{rel}} - T_u} \right)^2 \]  

Eq (II.12)

The value of the total relative inlet temperature for different tests were obtained from the output of streamline curvature calculations for the turbine stage. The measured temperature and pressure values, upstream and downstream of the stage were used as the input for the
calculations. Three thermocouples in the supply tank, were used to determine the pre-test nozzle guide vane inlet gas temperatures and the difference in their indicated temperatures was typically less than $1^\circ \text{C}$. During the test time, nozzle guide vane inlet temperature was obtained from the time history of the pressure data in the supply tank and the isentropic assumptions. The uncertainty in the indicated temperatures from the supply tank thermocouples were used as the approximate variance for the inlet total relative temperature.

Combining the Equations (II.9), (II.10) and (II.12), the variance of the uncertainty in the Nusselt number measurements were calculated for all the presented data. The variance of the heat flux gauge $k/d$ were obtained from the laser calibration of the sensors. The variance of the scale factors were obtained from the temperature coefficient resistance bath calibration of the sensors. The variance of the were obtained from temperature reference shifts ($T_E$) from the temperature coefficient resistance bath calibration as well as the post-run calibrations. The error bands in the time-averaged measurements shown in the previously presented figures, correspond to the one standard deviation (68% confidence level) away from the mean. All the time-averaged data in the preceding chapters, were plotted with their associated uncertainty band. For the measurements where the size of the error bars are smaller than the plotted symbols, no bars are plotted. The time-averaged Nusselt number measurements, and their associated 68% uncertainty bands, for all the previously reported tests are also listed in the Tables (2.6b), (2.7b) and (2.8b).
Appendix III

Influence of Nozzle Guide Vane Geometric Variations on Rotor Heat Transfer

The nozzle guide vane passages are generally not identical. This non-uniformity is the result of manufacturing and installation tolerances present in all turbine stages. In this appendix, the influence of the geometric variations of the nozzle guide vanes on the downstream rotor blade heat transfer is examined.

For the time resolved measurements presented thus far, the nozzle guide vane passages were considered to be identical and a passage ensemble averaging technique was used. To study the effect of non-uniformity, however, the vane passage to passage variation around the annulus is considered, in which the unsteady heat transfer data at mid-span from the uncooled design (test T47) and -10° incidence (test T52) cases are phase lock averaged. In this averaging scheme, the raw unsteady heat transfer data is averaged over thirty rotor revolutions for each separate nozzle guide vane passage.

In Figure (III.1), these phase lock averaged measurements for one rotor revolution from the G07 sensor (11% fractional wetted suction surface) for design incidence (a) and for -10° incidence (b) are shown. The passage to passage variation of the vanes results in a variation of rotor blade heat transfer around the annulus. In Figure (III.1), the passage averaged measurements previously presented in Chapter 4 are also plotted for comparison. The spatial dependence of peak heat transfer due to shock interaction is clearly observed, and it is seen that the variation is greater for the -10° incidence test than the design case.

The Nusselt number (sensor G07) for the vane passage periods associated with the maximum and minimum peak levels of the phase lock averaged data are superimosed in Figure (III.2),
design case in (a) and -10° incidence case in (b). The passage average and the UNSFLO prediction for the same sensor position and test conditions are also plotted in Figure (III.2). One feature of these figures is that the passage average data are close to half way between the maximum and the minimum levels. It is observed that for the design test case in Figure (III.2a), the variation in the peak levels is around 14% of the mean. The prediction level from UNSFLO is considerably (50%) lower than even the minimum level from the phase lock data. For the -10° incidence case shown in Figure (III.2b), however, the variation in the peak level is about 65% of the mean level, which is considerably more than in the design case.

There are two mechanisms that could account for the above observed difference in the peak variation between the two tests. The first mechanism is the variation in vane throat areas, which would result in a change in the throat mass flux and rotor Reynolds number. This change in the mass flux and Reynolds number due to geometric variation are the same for the two test cases. Inspection of the vanes around the annulus revealed a 4% variation in the guide vane throat areas. The influence of the vane throat area variation on rotor heat transfer was examined by comparing numerical solutions from UNSFLO for the -10° incidence case with the vane pitch changed by 4%. The prediction showed an 8% change in the shock induced peak, which is considerably lower than the variation observed from the phase lock average data.

The second mechanism is the variation in the throat to exit area ratio of the guide vanes. Variation of this area ratio would change the vane exit Mach number and alter the strength of the trailing edge shock waves. This change in the shock strength would have a direct effect on the peak level of heat transfer observed on the rotor downstream. The area ratio could not be accurately measured in the present experiment, but a typical value would be about ±3% from the mean. To estimate the influence of this variation on the downstream rotor heat transfer, the following simple analysis is performed:
From one-dimensional continuity equation, the change in the vane exit Mach number \((M)\) can be related to the change in the vane exit area \((A)\) by

\[
\frac{dM^2}{M^2} = \frac{2 + (\gamma - 1)M^2}{M^2 - 1} \frac{dA}{A},
\]

Eq (III.1)

For subscript 1 corresponding to the upstream conditions, the pressure change \((\Delta P)\) across a weak shock is given by

\[
\frac{\Delta P}{P_1} = \frac{2\gamma}{\gamma + 1} (M^2 - 1)
\]

Eq (III.2)

Differentiating Equation (III.2) and combining the result with Equation (III.1)

\[
\frac{d\left(\frac{\Delta P}{P_1}\right)}{\Delta P/P_1} = \frac{M^2 \left(2 + (\gamma - 1)M^2\right)}{(M^2 - 1)^2} \frac{dA}{A}
\]

Eq (III.3)

Assuming that the shape of the pressure time history as seen on the rotor remains unchanged and only the magnitude varies; from Equation (4.1), the change in the peak Nusselt number is proportional to the magnitude of the change in the shock pressure rise.

\[
\frac{d\left(\frac{\Delta N_u}{N_u}\right)}{\Delta N_u/N_u} = \frac{M^2 \left(2 + (\gamma - 1)M^2\right)}{(M^2 - 1)^2} \frac{dA}{A}
\]

Eq (III.4)

Equation (III.4) implies that for a given variation in the vane exit area, the change in the peak heat transfer increases as the vane exit Mach number approaches one. A 1% change in the area
ratio approximately corresponds to a change in the peak level of heat transfer of 15% for the
design case (vane exit Mach number of 1.22), and 39% for the -10° incidence case (vane exit
Mach number of 1.13). Therefore, a one to two percent variation in the vane throat to exit area
ratio would account for the observed variations in the rotor heat transfer.

Of particular interest here is the influence of the vane geometric variation on the time-averaged
heat transfer of the downstream blade row. To study this, the phase lock averaged data
presented earlier in this appendix was processed using a sliding passage average scheme. In
this scheme, a sliding time window corresponding to the vane passing time is used to average
the phase lock results. The resulting data is the time history of the mean heat transfer around
the rotor for varying guide vane passages. In Figure (III.3), the maximum and the minimum
of this sliding averaged data are shown for the design (a) and the -10° incidence (b) test cases.
For comparison, the time averaged measurements and the time averaged UNSFLO prediction
of heat transfer are also plotted in Figure (III.3). The mean heat flux averaged over a passage
varies by around 13% for both test cases. The variation from the time averaged is the highest
at the front of the suction surface (±20%), and a lower value elsewhere around the blade. It is
observed that the time-averaged heat transfer data is in between (close to half-way) the
maximum and the minimum of the sliding averaged data. This suggests that the influence of
the nozzle guide vane geometric variation on the downstream rotor heat transfer is approximately
linear and does not change the total blade heat load.
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SUMMARY OF SUCTION SURFACE MEAN HEAT TRANSFER RATE

Undisturbed flow

Wake passing, 0.9 mm bar

5% Freestream Tu only

5% Tu + wake passing, 0.9 mm bar

5% Tu + wake passing, 1.7mm bar

PRESSURE SURFACE SUMMARY

Figure 4.1: Influence of flow unsteadiness (wakes and shocks from bar passing) on a blade heat transfer measured in a cascade, from Doorly [31].
Design Incidence

Symbol:
+  No wake
x  With wake passing

Effect of wake interaction on mean heat transfer to blade with free-stream turbulence = 4 percent, nominal "design" operating conditions. Inset are typical transient recorder signals at two x/s locations showing a 1 ms interval of the record together with the time-averaged value in the absence of wake interaction. (b) Effect of wake interaction on mean heat transfer to blade with low (= 0.8 percent u'^1/2) free-stream turbulence at nominal "design" operating conditions. Inset are typical transient recorder signals together with the time-averaged value in the absence of wake interaction.

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at Design Condition, Measured at MIT Blowdown Turbine

- Nusselt Number
- Fractional Wetted Surface
- Suction Surface Transition Region
- Pressure Surface Transition Region
- Guide Vane Leading Edge
- Cubic Spline Curve Fit

- measurement
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Figure 5.2: $\frac{\text{Nu}_D}{\sqrt{\text{Re}_D}}$ vs time, measured at the leading edge and predicted (for a cylinder in crossflow, Equation (5.1.12)).

Figure 5.3: Time-averaged measured Nusselt number vs $\sqrt{\text{Re}_D}$, and the dependence predicted from the Equations (5.1.13) and (5.1.14).
Figure 5.4: Distribution of the Goertler number on the pressure surface of the ACE rotor profile, calculated at the design Reynolds number and incidence.

\[ \text{Goertler Number} \]

\[ \text{Fractional pressure surface} \]

\[ \text{Goer} = \frac{\theta}{\nu} \sqrt{\frac{\theta}{R}} \]

\[ \text{Goer} = 7 \]
Figure 5.5: Pressure surface measurement (MIT ACE rotor) at three different Reynolds numbers, (a) Nusselt numbers divided by the $Re_{c}^{0.8}$, (b) left hand side of Equation (5.2.14), $x_{g} = 0$, $a_{1} = 1.95 \times 10^{-8}$. 
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Figure 5.8: Suction surface measurement (MIT ACE rotor) at three different Reynolds numbers of, (a) Nusselt number distribution, (b) Nusselt numbers divided by scaled by \((\text{Re} / \text{Re}_{\text{des}})^{0.8}\).
Figure 5.9: (a) Time-averaged Nusselt No. distribution measured at the mid-span of the MIT ACE, (b) The Nusselt number Reynolds number scaled by: leading edge \((Re/Re_{des})^{0.5}\), suction surface \((Re/Re_{des})^{0.8}\) and pressure surface from Equation (5.2.1.3).
Figure (6.1) : Nozzle guide vane coolant injection near the trailing edge on the pressure surface.
Effect of NGV T/E Coolant Injection on Rotor

Figure (6.2) : Mean measured Nusselt number distribution around the rotor profile at mid-span with (T56) and without (T47) upstream nozzle guide vane coolant ejection.

Influence of NGV T/E Coolant Injection at Mid-Span

Figure (6.3) : Ratio of the with injection (T56) and without injection (T47) measured Nusselt number at mid-span.
Figure (6.4) : Comparison between the ratios of the with injection and without injection measured heat flux at mid-span between MIT rotor and Dunn[78].
Figure (6.5): (a) through (f) Comparison between the time-resolved measurements with and without nozzle guide vane injection around the suction surface at the design incidence, for test (T47).
Figure (6.6) : (a) through (f) Comparison between the time-resolved measurements with and without nozzle guide vane injection around the pressure surface at the design incidence, for test (T47).
Figure (6.7): Influence of the variation in the nozzle guide vane injected coolant to the main velocity ratios on the time-averaged rotor Nusselt number.
Figure (7.1): Composite of the gauge positions for the three instrumented film cooled blades.
Figure (7.2 a): Time-averaged film cooled Nusselt number distribution at three span-wise positions around the blade plotted against the mid-span uncooled results.
FIGURE (7.2b): Contour plot of the time-averaged film cooled Nusselt number on the rotor blade as function of wetted surface, design incidence and nominal blowing ratio.
Figure 7.3: Time-averaged film cooled (T71 and T75) and uncooled (T61) at mid-span.

Fractional Wetted Surface

Nusselt Number

Coolant Holes

Leading Edge

\( \frac{T_m}{T_c} = 1.9 \)
Figure (7.4) : (a) through (d) Suction surface time-resolved measured Nusselt number for the film cooled (T71 and T75) and the uncooled (T61) at mid-span.

Figure 7.4a- Film Cooled versus Uncooled Measurements At Mid-Span

Mid Section (cM) Blade
Figure 7.4b- Film Cooled versus Uncooled Measurements At Mid-Span
120% Nominal Corrected Speed

![Graph showing Nusselt Number vs. time/blade passing period for Mid Section (cM) Blade. Two lines are present: one for No Film Cooling (T61-G12) and another for With Film Cooling (T71-G18).]
Figure 7.4c- Film Cooled versus Uncooled Measurements at Midspan
120% Nominal Corrected Speed

- No Film Cooling, T61-G13
- - - With Film Cooling, T71-G19
- - - - With Film Cooling, T75-G19

Mid Section (cM) Blade
Figure 7.4d - Film Cooled versus Uncooled Measurements at Midspan 120% Nominal Corrected Speed

Nusselt Number

- No Film Cooling, T61-G14
- With Film Cooling, T71-G20
- With Film Cooling, T75-G20

Mid Section (cM) Blade
Figure 7.5: Time-averaged film cooled (T71 and T64) and uncooled (T61) at mid-span.
Figure 7.6a - Film Cooled versus Uncooled Measurements at Midspan

120% Nominal Corrected Speed, High Blowing Ratio

Blowing Ratio = 1.5

Mid Section (cM) Blade
Figure 7.6b - Film Cooled versus Uncooled Measurements at Midspan

120% Nominal Corrected Speed, Low Blowing Ratio

Blowing Ratio = 1.1
Figure (7.7): Measured time-averaged Nusselt number for film cooled (T71), uncooled (T61), and the predicted film cooled (using cascade measurements of effectiveness [80]) at mid-span.
Figure 7.8: Comparison of the measured uncooled (T112) Nusselt number for nominal tip and mid section Nusselt number against the prediction.
Figure 7.9: Measured time-averaged Nusselt number film cooled (T71 and T75) and Reynolds number scaled uncooled (T112) at nominal tip section.
Figure 7.10a - Film Cooled vs. Uncooled (Re Scaled) Measurements
Tip Section 120% Nominal Corrected Speed

- Uncooled, T112-G05, Re scaled
- - With Film Cooling, T71-G06
- - - With Film Cooling, T75-G06
Figure 7.10 b - Film Cooled vs. Uncooled (Re Scaled) Measurements
Tip Section 120 % Nominal Corrected Speed

- Uncooled, T112-G07, Re scaled
- With Film Cooling, T71-G07
- With Film Cooling, T75-G07

Tip Section (cT) Blade
Figure 7.10 c - Film Cooled vs. Uncooled (Re Scaled) Measurements
Tip Section 120 % Nominal Corrected Speed

Tip Section (cT) Blade

- Uncooled, T112-G09, Re scaled
- With Film Cooling, T71-G08
Figure 7.11 a - Film Cooled vs. Uncooled (Re Scaled) Measurements
Tip Section 120 % Nominal Corrected Speed

Tip Section (cT) Blade

<table>
<thead>
<tr>
<th>Nusselt Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>2000</td>
</tr>
<tr>
<td>1500</td>
</tr>
<tr>
<td>1000</td>
</tr>
<tr>
<td>500</td>
</tr>
<tr>
<td>0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>time/ blade passing period</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
</tr>
<tr>
<td>2.5</td>
</tr>
<tr>
<td>2</td>
</tr>
<tr>
<td>1.5</td>
</tr>
<tr>
<td>1.25</td>
</tr>
<tr>
<td>1</td>
</tr>
<tr>
<td>0.75</td>
</tr>
<tr>
<td>0.5</td>
</tr>
<tr>
<td>0</td>
</tr>
</tbody>
</table>

- Uncooled, T112-G11, Re scaled
- Uncooled, T112-G13, Re scaled
- With Film Cooling, T71-G10
Figure 7.11 b - Film Cooled vs. Uncooled (Re Scaled) Measurements
Tip Section 120 % Nominal Corrected Speed

- Uncooled, T112-G14, Re scaled
- - - - -
- With Film Cooling, T63-G12

Nusselt Number

0 500 1000 1500 2000

0 0.5 1 1.5 2 2.5 3
time/blade passing period

Tip Section (cT) Blade

1 2 3 4 5 6 7 8 9 10 11 12
Figure (7.12a): Time-averaged film cooled (T71 and T75) at nominal hub-section and uncooled (T61) at mid-span.
Figure (7.12b): Time-averaged film cooled (T71) at nominal hub and mid-span and uncooled (T61) at nominal mid-span.
Figure 7.13 a - Film Cooled Measurements at Hub Section
Also: Film Cooled and Uncooled Mid Span At Same Axial Position

![Graph showing Nusselt Number vs. time/blade passing period for different cooling conditions.]

- No Film Cooling, Mid span, T61-G09
- Film Cooled, Mid Span, T71-G15
- Film Cooled, Hub Section, T71-G27
- Film Cooled, Hub Section, T75-G27

Hub Section (cH) Blade

- 26
- 25
- 33
- 34
- 28
- 29
- 30
- 31
- 35
- 36
- 32
Figure 7.13 b - Film Cooled Measurements at Hub Section. Also Plotted: Film Cooled Mid Span At Same Axial Position.

![Graph showing Nusselt Number vs. time/blade passing period for different conditions.]

- Film Cooled, Mid Span, T71-G16
- Film Cooled, Hub Section, T71-G28
- Film Cooled, Hub Section, T75-G28

Hub Section (cH) Blade
Figure 7.13 c - Film Cooled Measurements at Hub Section
Also Plotted: Uncooled Mid Span At Same Axial Position

- No Film Cooling, Mid Span, T61-G11
- Film Cooled, Hub Section, T71-G30
- Film Cooled, Hub Section, T75-G30

Nusselt Number

0 500 1000 1500 2000

0 0.5 1 1.5 2 2.5 3

time/blade passing period

Hub Section (cH) Blade
Figure 7.13 e - Film Cooled Measurements at Hub Section.
Also: Film Cooled and Uncooled Mid Span At Same Axial Position

- No Film Cooling, Mid Span, T61-G13
- Film Cooled, Mid Span, T71-G19
- - Film Cooled, Hub Section, T71-G31
- - - Film Cooled, Hub Section, T75-G31
Figure 7.13 d - Film Cooled Measurements at Hub Section. Also: Film Cooled and Uncooled Mid Span At Same Axial Position

| No Film Cooling, Mid Span, T61-G12 |
| Film Cooled, Mid Span, T71-G18 |
| Film Cooled, Hub Section, T71-G30 |
| Film Cooled, Hub Section, T75-G30 |

Nusselt Number

0 500 1000 1500 2000

0 1 1.5 2 2.5 3

time/blade passing period

Hub Section (cH) Blade
Figure 7.13f - Film Cooled Measurements at Hub Section. Also: Film Cooled and Uncooled Mid Span At Same Axial Position

- No Film Cooling, Mid Span, T61-G14
- Film Cooled, Mid Span, T75-G20
- Film Cooled, Hub Section, T71-G32
- Film Cooled, Hub Section, T75-G32

<table>
<thead>
<tr>
<th>Nusselt Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>2000</td>
</tr>
<tr>
<td>1500</td>
</tr>
<tr>
<td>1000</td>
</tr>
<tr>
<td>500</td>
</tr>
<tr>
<td>0</td>
</tr>
</tbody>
</table>

Time/blade passing period

Hub Section (cH) Blade
Figure 7.14 a - Film Cooled Measurements at Hub Section
Also Plotted: Uncooled Mid Span At Same Axial Position

- No Film Cooling, Mid Span, T61-G04
- Film Cooled, Hub Section, T71-G34
- Film Cooled, Hub Section, T75-G34
Figure 7.14 b-Film Cooled Measurements at Hub Section
Also Plotted: UnCooled Mid Span At Same Axial Position

- No Film Cooling, Mid Span, T61-G02
- Film Cooled, Hub Section, T71-G35
Figure 7.14 c-Film Cooled Measurements at Hub Section. Also Plotted: Uncooled Mid Span At Same Axial Position

**Graph Details:**
- **Y-axis:** Nusselt Number
- **X-axis:** time/blade passing period
- **Legend:**
  - Solid line: No Film cooling, Mid Span, T61-G01
  - Dashed line: Film Cooled, Hub Section, T71-G36

**Diagram Details:**
- Hub Section (cH) Blade
- Points labeled from 26 to 36

**Caption:**
- Figure 7.14 c-Film Cooled Measurements at Hub Section. Also Plotted: Uncooled Mid Span At Same Axial Position.
Figure 7.15 - Suction Surface Measurement at Mid-Span
1 - (Nu film Cooled / Nu uncooled)

Mid Section (cM) Blade

Isothermal Effectiveness, G75-15

Isothermal Effectiveness

time/blade passing period

0 0.25 0.5 0.75 1
0 0.5 1 1.5 2 2.5 3

13 14 15 16 17 18 19 20 21 22 23 24
Fig (7.16): Peak to Peak and Time Average of the Unsteady Surface Pressure Around the Blade Profile (from UNSFLO) and the Level of Coolant Plenum Pressure.
Figure (7.17): Two-dimensional coolant slot model with unsteady external pressure field.
Figure (7.18) : Absolute value of $1 - G(\omega)$ (for unit perturbation) versus the reduced frequency.

Figure (7.19) : Amplitude of pressure waves (non-dimensionalized by the mean inlet total relative pressure) over the SS1 coolant row (from UNSFLO).
Figure (7.20): Nusselt number comparison between the measured uncooled (T61), measured film cooled (T75) and the model prediction (Eq (7.3.2.28)), for the cM15 gauge on the suction surface at nominal mid-span.

![Nusselt Number Chart]

Mid Section (cM) Blade

- No Film Cooling, Mid span, T61-G09
- With Film Cooling, T75-15
- Unsteady Blowing Model prediction
Figure (7.21) : Nusselt number comparison between the measured uncooled (T61), measured film cooled (T63) and the model prediction (Eq(7.3.2.28)), for the cM22 gauge on the pressure surface at nominal Mid Section (cM) Blade mid-span.
Figure (7.22): (a) & (b) Schematic of the physical coupling between the unsteady blowing and propagating shock wave.
Fig (III.1): Measured Phase Lock Average of Rotor Heat Flux on Suction Surface (G07 sensor) Compared to Passage Average Data at (a) Design Incidence (b) -10° Incidence Test Cases.
Fig (III.2): Comparison Between Maximum, Minimum Peak Phase Lock Average Nusselt Numbers, the UNSFLO Prediction and Passage Average for Sensor G07, (a) Design Incidence (b) -10° Incidence Test Cases.
Fig (III.3): Comparison Between Maximum, Minimum Peak of Sliding Passage Average of Phase Lock Data, the UNSFLO Prediction and Time Average Heat Flux Measurement Around the Blade Profile for (a) Design Incidence (b) -10° Incidence Test Cases.
Table 2.1 MIT Blowdown Turbine Facility Scaling

<table>
<thead>
<tr>
<th></th>
<th>Full Scale Engine</th>
<th>MIT Blowdown Facility</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid</td>
<td>Air</td>
<td>Argon / Freon 12</td>
</tr>
<tr>
<td>Ratio of Specific Heats, g</td>
<td>1.28</td>
<td>1.28</td>
</tr>
<tr>
<td>Coolant Fluid</td>
<td>Air</td>
<td>Argon / Freon 14</td>
</tr>
<tr>
<td>Coolant Ratio of Specific Heats</td>
<td>1.37</td>
<td>1.37</td>
</tr>
<tr>
<td>Coolant Mass Ratio</td>
<td>13%</td>
<td>13%</td>
</tr>
<tr>
<td>Mean Metal Temperature</td>
<td>1118 K</td>
<td>295 K</td>
</tr>
<tr>
<td>Metal to Gas Temp. Ratio</td>
<td>0.63</td>
<td>0.63</td>
</tr>
<tr>
<td>Inlet Total Temperature</td>
<td>1780 K</td>
<td>478 K</td>
</tr>
<tr>
<td>Mean Coolant Temperature</td>
<td>790 K</td>
<td>212 K</td>
</tr>
<tr>
<td>True NGV chord</td>
<td>8.0 cm</td>
<td>5.9 cm</td>
</tr>
<tr>
<td>Reynolds No. (based on vane chord)</td>
<td>$2.7 \times 10^6$</td>
<td>$2.7 \times 10^6$</td>
</tr>
<tr>
<td>Inlet Pressure</td>
<td>19.6 atm</td>
<td>4.3 atm</td>
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<tr>
<td>Outlet Pressure</td>
<td>4.5 atm</td>
<td>1.0 atm</td>
</tr>
<tr>
<td>Outlet Total Temperature</td>
<td>1280 K</td>
<td>343 K</td>
</tr>
<tr>
<td>Prandtl Number</td>
<td>0.752</td>
<td>0.755</td>
</tr>
<tr>
<td>Rotor Speed</td>
<td>12,734 rpm</td>
<td>6,190 rpm</td>
</tr>
<tr>
<td>Mass Flow</td>
<td>49.00 kg/sec</td>
<td>16.55 kg/sec</td>
</tr>
<tr>
<td>Power</td>
<td>24,880 kW</td>
<td>1,078 kW</td>
</tr>
<tr>
<td>Test Time</td>
<td>Continuous</td>
<td>0.3 seconds</td>
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</table>
Table 2.2: Data Acquisition Timing Sequence for High Speed Channels

<table>
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<tr>
<th>Start and End Time</th>
<th>High Speed Channel</th>
<th>Sample Freq. /HZ</th>
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</thead>
<tbody>
<tr>
<td>0 - 250 ms</td>
<td></td>
<td>20,000</td>
</tr>
<tr>
<td>250 - 550 ms</td>
<td></td>
<td>200,000</td>
</tr>
<tr>
<td>550 - 1200 ms</td>
<td></td>
<td>5,000</td>
</tr>
<tr>
<td>1.2 s - 600 s</td>
<td></td>
<td>50</td>
</tr>
<tr>
<td></td>
<td>Supply</td>
<td>Coolant*</td>
</tr>
<tr>
<td>--------------------------</td>
<td>--------</td>
<td>----------</td>
</tr>
<tr>
<td>Static Pressure</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Main P Total</td>
<td>2</td>
<td>-</td>
</tr>
<tr>
<td>Main Gas T/C</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>Metal T/C</td>
<td>2</td>
<td>2*</td>
</tr>
<tr>
<td>Coolant P Total</td>
<td>-</td>
<td>1*</td>
</tr>
<tr>
<td>Coolant Gas T/C</td>
<td>-</td>
<td>1*</td>
</tr>
<tr>
<td>Tachometer</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Heat Flux Gauges</td>
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<td>-</td>
</tr>
<tr>
<td>Resistance</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Temp.Devices</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

* - All Cooled Tests; ** - Cooled Stage Tests Only;

T/C = Thermocouple, NGV = nozzle guide vane
Table (2.4a): Position of heat flux gauges, uncooled rotor blades

<table>
<thead>
<tr>
<th>Sensor</th>
<th>Fractional Wetted Surface</th>
<th>Y</th>
</tr>
</thead>
<tbody>
<tr>
<td># 09</td>
<td>-0.934</td>
<td>0.34</td>
</tr>
<tr>
<td># 08</td>
<td>-0.831</td>
<td>0.34</td>
</tr>
<tr>
<td># 07</td>
<td>-0.728</td>
<td>0.34</td>
</tr>
<tr>
<td># 06</td>
<td>-0.624</td>
<td>0.34</td>
</tr>
<tr>
<td># 05</td>
<td>-0.521</td>
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</tr>
<tr>
<td># 04</td>
<td>-0.418</td>
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</tr>
<tr>
<td># 03</td>
<td>-0.314</td>
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</tr>
<tr>
<td># 02</td>
<td>-0.211</td>
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<tr>
<td># 01</td>
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<td># 00</td>
<td>0.00</td>
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<tr>
<td># 11</td>
<td>0.161</td>
<td>0.34</td>
</tr>
<tr>
<td># 12</td>
<td>0.315</td>
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<td># 13</td>
<td>0.469</td>
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<td>0.777</td>
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<td># 16</td>
<td>0.931</td>
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<td># 14</td>
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<td># 13</td>
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<td>0.80</td>
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<td># 12</td>
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<td>0.80</td>
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<tr>
<td># 11</td>
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<td>0.80</td>
</tr>
<tr>
<td># 10</td>
<td>-0.43</td>
<td>0.80</td>
</tr>
<tr>
<td># 09</td>
<td>-0.313</td>
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<tr>
<td># 07</td>
<td>0.104</td>
<td>0.80</td>
</tr>
<tr>
<td># 06</td>
<td>0.00</td>
<td>0.80</td>
</tr>
<tr>
<td># 05</td>
<td>0.161</td>
<td>0.80</td>
</tr>
<tr>
<td># 04</td>
<td>0.323</td>
<td>0.80</td>
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<tr>
<td># 03</td>
<td>0.484</td>
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<td># 02</td>
<td>0.641</td>
<td>0.80</td>
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<tr>
<td># 01</td>
<td>0.798</td>
<td>0.80</td>
</tr>
</tbody>
</table>

Y is the distance from the center of the sensor to the tip of the blade (in inches).

Nominal length of Rotor blade axial chord at mid-section is at one inch.
Distance from the tip uncertainty of the measurements is ±0.015".
Fractional wetted surface uncertainty of the measurements is ±0.005".
Table (2.4b): Position of heat flux gauges, film cooled blades

<table>
<thead>
<tr>
<th>Sensor</th>
<th>Fractional Wetted Surface</th>
<th>Y</th>
</tr>
</thead>
<tbody>
<tr>
<td>cT08</td>
<td>-0.921</td>
<td>0.375</td>
</tr>
<tr>
<td>Suction surface length= 1.937&quot;</td>
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<td></td>
</tr>
<tr>
<td>cT07</td>
<td>-0.818</td>
<td>0.375</td>
</tr>
<tr>
<td>cT06</td>
<td>-0.609</td>
<td>0.323</td>
</tr>
<tr>
<td>cT03</td>
<td>-0.299</td>
<td>0.323</td>
</tr>
<tr>
<td>TIP-SECTION</td>
<td></td>
<td></td>
</tr>
<tr>
<td>cT02</td>
<td>-0.111</td>
<td>0.341</td>
</tr>
<tr>
<td>cT09</td>
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<td>0.341</td>
</tr>
<tr>
<td>Pressure surf. length= 1.297&quot;</td>
<td></td>
<td></td>
</tr>
<tr>
<td>cT10</td>
<td>0.37</td>
<td>0.321</td>
</tr>
<tr>
<td>cT11</td>
<td>0.728</td>
<td>0.33</td>
</tr>
<tr>
<td>cT12</td>
<td>0.882</td>
<td>0.33</td>
</tr>
<tr>
<td>cM20</td>
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<td>0.807</td>
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<tr>
<td>Suction surface length= 1.965&quot;</td>
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<td></td>
</tr>
<tr>
<td>cM19</td>
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<td>0.807</td>
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<td>cM18</td>
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</tr>
<tr>
<td>cM16</td>
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<td>0.82</td>
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<tr>
<td>cM15</td>
<td>-0.342</td>
<td>0.82</td>
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<tr>
<td>MID-SECTION</td>
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<td>cM14</td>
<td>-0.102</td>
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<td>cM21</td>
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<td>Pressure surf. length= 1.267&quot;</td>
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<tr>
<td>cM22</td>
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<tr>
<td>cM24</td>
<td>0.886</td>
<td>0.731</td>
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<tr>
<td>cH32</td>
<td>-0.917</td>
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<td>Suction surface length= 1.905&quot;</td>
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<tr>
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<tr>
<td>Pressure surf. length= 1.245&quot;</td>
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<tr>
<td>cH33</td>
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<td>cH34</td>
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<tr>
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<tr>
<td>cH36</td>
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<td>1.359</td>
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</tbody>
</table>

\( Y \) is the distance from the center of the sensor to the tip of the blade (in inches).

Nominal length of Rotor blade axial chord at mid-section is at one inch.

Accuracy of the measurements is \( \pm 0.005" \).
Table (2.5): Heat flux gauges k/d and (√(rho. c k)) calibration

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<td>581</td>
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<tr>
<td>cT06</td>
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<td>9232</td>
</tr>
<tr>
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<td>581</td>
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<td>cH28</td>
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<tr>
<td>cH36</td>
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<td>8840</td>
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</table>

Presented (√(rho c k))'s have units of W (√(sec)/K)/(M**2), uncertainty ±5. k/d's are in W/(M**2)/K
Table (2.6a): Test parameters for uncooled turbine stage test conditions, tests (T47) through (T53).

<table>
<thead>
<tr>
<th>TEST ID#</th>
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<th>T50</th>
<th>T51</th>
<th>T52</th>
<th>T53</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial tunnel temp (C)</td>
<td>34.4</td>
<td>30.0</td>
<td>31.7</td>
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<td>30.0</td>
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<tr>
<td>Inlet total temp (K)</td>
<td>463</td>
<td>463</td>
<td>463</td>
<td>463</td>
<td>463</td>
</tr>
<tr>
<td>Inlet total pres. (ATM)</td>
<td>3.55</td>
<td>3.55</td>
<td>1.77</td>
<td>3.56</td>
<td>4.81</td>
</tr>
<tr>
<td>Exit total press. (ATM)</td>
<td>0.86</td>
<td>0.96</td>
<td>0.43</td>
<td>0.83</td>
<td>1.16</td>
</tr>
<tr>
<td>Exit static press. (ATM)</td>
<td>0.63</td>
<td>0.66</td>
<td>0.31</td>
<td>0.65</td>
<td>0.85</td>
</tr>
<tr>
<td>Mean main flow gamma</td>
<td>1.28</td>
<td>1.28</td>
<td>1.28</td>
<td>1.28</td>
<td>1.28</td>
</tr>
<tr>
<td>Relative total temp (K)</td>
<td>403</td>
<td>406</td>
<td>403</td>
<td>399</td>
<td>402</td>
</tr>
<tr>
<td>Mid-span rotor inlet Mach#</td>
<td>0.68</td>
<td>0.85</td>
<td>0.68</td>
<td>0.46</td>
<td>0.67</td>
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<tr>
<td>Mid-span rotor incid.(DEG)</td>
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<td>63.2</td>
<td>60.5</td>
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<tr>
<td>Stage pressure ratio</td>
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<tr>
<td>Main Gas/ wall temp. ratio</td>
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<td>1.32</td>
<td>1.30</td>
<td>1.33</td>
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Table (2.6 b): Time-averaged measured mid-span rotor Nusselt numbers and their uncertainty levels for uncooled turbine stage test conditions, tests (T47) and (T53).

<table>
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<th>T50.ERR</th>
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.ERR files correspond to the one standary devation uncertainty level.

Gaps in the table correspond to failed sensors.
Table (2.7a): Test parameters for cooled nozzle guide vane and uncooled rotor test conditions, tests (T55) through (T61).

<table>
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<th>T56</th>
<th>T57</th>
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<th>T61</th>
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<td>Inlet total pres. (ATM)</td>
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<td>Vane coolant pres. (ATM)</td>
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Table (2.7 b): Time-averaged measured mid-span rotor Nusselt numbers and their uncertainty levels for cooled nozzle guide vane (with trailing edge injection) and uncooled rotor test conditions, tests (T55) and (T61).

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</table>

.ERR files correspond to the one standard deviation uncertainty level.

Gaps in the table correspond to failed sensors.
Table (2.8a) : Continued; Test parameters for fully cooled stage test conditions, tests (T63) through (T75).
Tip-Section Blade.

<table>
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<th>TEST ID NUMBERS.</th>
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<th>T73</th>
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<tr>
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<td>Inlet total temp. (K)</td>
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Table (2.8a): Test parameters for fully cooled stage test conditions, tests (T63) through (T75).

Hub-Section Blade.

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.ERR files correspond to the one standard deviation uncertainty level.
Gaps in the table correspond to failed sensors. Sensors that did not operate were deleted from the above table.
Table (2.8 b): Continued; Time-averaged measured rotor Nusselt numbers at three span-wise positions and their uncertainty levels for fully cooled turbine stage test conditions, tests (T71) and (T75).

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.ERR files correspond to the one standard deviation uncertainty level.
Gaps in the table correspond to failed sensors. Sensors that did not operate were deleted from the above table.
Table (7.1): Blowing parameters for the coolant holes

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Blowing Ratio is defined as the ratio of the coolant mass to the main flow.

Momentum Ratio is defined as the ratio of the coolant momentum to the main flow.

Position of coolant rows are shown in Figure (2.9).