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Experimental Investigation of Two Transonic Linear Turbine Cascades at Off-Design Conditions

by

Dhafer Ben Mahmoud Jouini
M. Eng. (Mechanical)

A Thesis submitted to
the Faculty of Graduate Studies and Research
in Partial fulfilment of the requirements
for the degree of

Doctor of Philosophy
in
Aerospace Engineering

Ottawa-Carleton Institute for Mechanical
and Aerospace Engineering

Department of Mechanical and Aerospace Engineering
Carleton University
Ottawa, Ontario
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acceptance of the thesis

**Experimental Investigation of Two Transonic Linear
Turbine Cascades at Off-Design Conditions**

submitted by

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In Partial fulfilment of the requirements for
the degree of Doctor of Philosophy

Chairman, Department of Mechanical and Aerospace Engineering

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إلى كافة أفراد عائلتي من صميمي قلبي

Abstract

Detailed measurements have been made of the mid-span aerodynamic performance of two transonic turbine cascades at off-design conditions. The cascades investigated were a baseline cascade, designated HS1A, and a cascade with a modified leading edge design, designated HS1B. The measurements were for exit Mach numbers ranging from about 0.5 to about 1.2 and for Reynolds numbers from 4×10^5 to 10^6 . The turbulence intensity in the test section and upstream of the cascade test section was about 4%. The profile losses were measured for the incidence values of -10° , 0.0° , $+4.5^\circ$, $+10.0^\circ$, and $+14.5^\circ$ relative to design. To aid in understanding the loss behaviour and to provide other insights into the flow physics, measurements of the blade loading, exit flow angles, trailing-edge base pressures, and the Axial Velocity Density Ratio (AVDR) were also made.

The results showed that the profile losses at transonic Mach numbers can be closely related to the behaviour of the base pressure. The losses were also found to be affected by the AVDR. The AVDRs were found to decrease with increasing positive incidence.

Moreover the results from both cascades showed that the modifications to the leading edge geometry of HS1B cascade were not successful in improving the blade performance at positive off-design incidence.

Comparisons between the present experimental data and the available correlations in the open literature were also made. These comparisons included mid-span losses at design and off-design, and exit flow angles. It was found that further improvements can still be made to the existing correlations. Furthermore, the present experimental data represents a significant contribution to the data base of results available in the open literature for the development of new and improved correlations, particularly at transonic flow conditions, at both design and off-design conditions.

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Nomenclature

English Symbols

AVDR Axial Velocity Density Ratio, $\left(= \frac{\int_0^1 (\rho_2 C_{(ax)2})_{MS} d(y/s)}{\int_0^1 (\rho_1 C_{(ax)1})_{MS} d(y/s)} \right)$

AVR Axial Velocity Ratio, $\left(= \frac{\int_0^1 (C_{(ax)2})_{MS} d(y/s)}{\int_0^1 (C_{(ax)1})_{MS} d(y/s)} \right)$

C	air flow velocity
C_b	base pressure coefficient, $(= (P_b - P_{s2})/q_2)$
C_{p0}	total pressure coefficient, $(= (P_{02} - P_{01})/q_2)$
C_{ps}	static pressure coefficient, $(= ((P_s)_x - P_{1r})/q_{1r})$
C_p	specific heat constant for air
C_x	axial chord
CFM	Mach number multiplying factor
H	blade span
K_1, K_2, K_p	correction factors used in the profile loss correlation Y_p
M	Mach number
P_0	total pressure
P_b	base pressure
Pr	Prandtl number
P_s	blade surface static pressure
P_{s2}	downstream static pressure

$\dot{Q}$	rate of heat transfer
Re	Reynolds number, $(= \rho C C_2 / \mu)$
St	Stanton number
T	static temperature
T_{aw}	adiabatic wall temperature
T_0	total temperature, $(= T + 0.5 C^2 / (2 C_p))$
Tu	turbulence intensity
T_w	wall temperature
V	air flow velocity
We	wedge angle
Y_t	total pressure loss coefficient, $(= (P_{01} - P_{02}) / q_2)$
Y_p	profile loss coefficient
$\Delta Y_{t, te}$	trailing edge total pressure loss coefficient
$f(Re)$	Reynolds number function
d	leading edge diameter
e	suction side radius of curvature from throat to trailing edge
h	enthalpy; convective heat transfer coefficient
$i_{effective}$	effective incidence, in degrees, $(= \alpha_1 - (\alpha_1)_{des} = i - i_{des})$
$\dot{m}$	mass flow rate
o	throat opening
q	dynamic pressure, $(= 0.5 \rho C^2)$
r	recovery factor
s	pitch distance
t	trailing edge thickness
w	air flow velocity; downstream wake width
x	axial distance
y	pitchwise location

Greek Symbols

α	flow angle
β	blade metal angle
δ	deviation angle, $(= \beta_2 - \alpha_2)$
δ^+	boundary layer displacement thickness
ζ	stagger angle
Θ	non-dimensional mass flow rate
θ	boundary layer momentum thickness; blade camber; misalignment angle
θ_u	uncovered turning
μ	air dynamic viscosity
ξ	energy loss coefficient, $(= (h_2 - h_{2s})/(h_{02} - h_2))$
ξ^+	energy loss coefficient, $(= (h_2 - h_{2s})/(h_{01} - h_2))$
ρ	air density
ϕ^2	energy loss coefficient, $(= 1 - C_2^2/C_{2is}^2)$
$\Delta\phi_p^2$	profile energy loss coefficient
$\Delta\phi_{TET}^2$	trailing edge energy loss coefficient
χ, χ'	profile loss incidence parameters
ω	total pressure loss coefficient, $(= (P_{01} - P_{02})/(P_{02} - P_2))$

Subscripts

0	free stream
01r	reference inlet, total
1	cascade inlet
1r	reference inlet, static
2	cascade outlet
2c	corrected
2m	measured
2s, 2is	isentropic at exit
I	impulse

MS	mid-span
N	nozzle
ax	axial
des	design
is	isentropic
le	leading edge
m	maximum
min	minimum
max	maximum
prop	property
ref	reference
t	trailing edge, total
te	trailing edge
totalref	reference total

Abbreviations

CV	control volume
PROP	property
PS	pressure surface
SS	suction surface

1. Introduction

Market requirements for compact, low cost and fuel-efficient engines generally lead to small size turbomachines such as those designed by Pratt & Whitney Canada. Typical features in the compressor turbine of these engines are a single stage turbine with high-pressure ratio and stage loading, combined with blade passages of small aspect ratio, high turning, transonic Mach numbers and very low Reynolds numbers. These turbines need to operate at maximum efficiency at the design cruise speed and altitude as well as with minimum efficiency penalty at off-design conditions arising from changes in pressure ratio, speed and altitude. The resulting three-dimensional flow field within the turbine passages, combined with the stringent structural requirements to achieve the desired target life, pose a real challenge for the turbine designer to meet the design and off-design performance requirements.

The aerodynamic design of turbine blades is one of the areas where improvements in blade performance over the range of operating conditions may lead to gains in turbine efficiency. The improvements in blade performance are measured in terms of reduction in blade losses. Thus, the goals of the blade designer are to produce blade shapes that have low losses at the design flow conditions and also to extend the envelope of the blade minimum losses to a wide range of blade operating conditions. As mentioned, these varying operating conditions include changes in operating Mach number and blade incidence.

At transonic Mach numbers, shock waves appear within the turbine blade passages. As a result, the flow behaviour is influenced by the presence of the shock waves, in addition to the viscous effects. The interaction between the Mach number effects (compressibility effects) and viscous effects (Reynolds number effects) plays an important role in determining the blade losses. This complexity presents a challenge to the designer. Adding the effects of blade incidence, the

challenge for the blade designer is even greater. Therefore, an understanding of the flow physics under these conditions is essential for any improvements in blade aerodynamic design.

In studies of turbomachinery aerodynamics, cascade test sections and rotating rigs are used to evaluate the performance of both compressor and turbine blades. Such experimental tests are used to examine the effects of variations of flow conditions and blade geometry on the losses. These losses include mid-span losses, secondary, and tip clearance losses. In this study, the focus is on the blade mid-span losses. Over the past decades, investigations in linear turbine cascades have led to the development of a number of correlations for blade losses and outlet flow angles. These correlations are intended to be used at an early stage of the design to account for the effects of some of the geometric and flow parameters on the aerodynamic performance of the blade design under consideration. Using the correlations, an initial machine geometry can be obtained and some optimization can be performed. Unfortunately, the existing correlations are largely limited to subsonic flow conditions. Moreover, transonic Mach number data is scarce in the open literature and the majority of the data are limited to the design incidence flow conditions. Hence, more data are needed in the transonic Mach number range, and in particular at off-design incidence. These data will help in the understanding of the flow physics and augment the data base available in the open literature so that improved loss and exit flow correlations at design and off-design incidence may be devised.

A joint program involving Pratt & Whitney Canada and Carleton University has been initiated to study experimentally the aerodynamic performance of turbine blades at transonic Mach numbers and low Reynolds numbers and address the lack of data under these conditions. The ultimate objective is to develop improved design rules for small, low cost and efficient turbines. The program is based on the recently commissioned high speed wind tunnel laboratory located in the Minto Case building at Carleton University. This laboratory was recently renamed the Pratt & Whitney Canada High Speed Wind Tunnel Laboratory.

Detailed measurements have been conducted of the effects of transonic Mach numbers on blade mid-span losses at design and off-design incidence values, for two cascades: a baseline cascade (designated HS1A) and a cascade with a modified leading edge design (designated HS1B).

The modified geometry was designed by Pratt & Whitney Canada, and was intended to investigate one possible approach to reducing the sensitivity of the blade to off-design incidence.

Descriptions of the wind tunnel, the cascade test section, and the test cascades are given in Chapter 3. A description of the wind tunnel and the instrumentations used in this study is also given in Chapter 3.

Chapter 4 focuses on the types of data taken and data reduction procedures.

Chapter 5 presents the experimental results, beginning with the HS1A cascade results. The results for the HS1B cascade are then presented and compared with the HS1A data. Insights to the flow physics which are associated with transonic Mach numbers and off-design incidence are provided. The loss data and exit flow angle measurements from both HS1A and HS1B cascades are then compared with predictions from existing loss and exit flow angle correlations.

Finally, conclusions from the present study and recommendations for future work are given in Chapter 6.

2. Literature Review

2.1 Introduction

As mentioned in the previous chapter, the present investigation is directed towards the understanding of the aerodynamic performance of high pressure (HP) turbine blades of modern designs at transonic Mach numbers and off-design incidence values. In addition, one of the goals is to investigate the influence of the leading edge geometry on the tolerance of the blades to off-design incidence. In this chapter a review of previous research on the aforementioned effects on the aerodynamic performance of turbine blades will be presented.

2.2 Flow in Axial Turbine Blade Passages

2.2.1 General Features of Passage Flow

Figure 2.1 (Jeffries,2000) shows a schematic diagram of the general flow features within axial turbine blade passages. In the case of a stationary blade row, the flow structures at the hub wall are basically the same as the tipwall. In the absence of tip leakage, a horseshoe vortex is formed due to the separation of the inlet endwall boundary layer at the leading edge of the blade. The horseshoe vortex wraps around the leading edge forming a suction side and a pressure side legs. Using flow visualization, Wang et al. (1997) showed that the pressure leg of the horseshoe vortex immediately moves towards the suction surface of the adjacent blade after it enters the blade passage. It entrains the main flow and the inlet boundary layer, and merges with its counter part the suction leg from the adjacent blade. At the merging point, the interaction among the horseshoe legs, the inlet boundary layer, the blade-to-blade cross flow, and the main stream forms an even stronger vortex in the passage called the passage vortex.

Figure 2.1 A Schematic Diagram of the Flow within an Axial Turbine Blade Passage (Jeffries, 2000).

As demonstrated by Wang et al. and other investigators such as Sieverding and Van Den Boshe (1983), the pressure side leg of the horseshoe vortex has the same direction as the passage vortex, and is a major part of the passage vortex.

The passage vortex stays close to the suction surface and is gradually lifted away from the endwall as it travels downstream. It also grows in size as it entrains with it the main flow and the endwall boundary layers.

Near the tip, the flow is driven by the pressure difference between the pressure surface and the suction surface of the blade. This tip leakage flow causes the formation of the tip-leakage vortex. When relative motion at the blade tip exists, the passage vortex is strengthened and is drawn further towards the suction side. Yaras and Sjolander (1992b) showed that this interaction causes a reduction in mass flow through the gap. Thus, the flow near the tip wall appears to be dominated by the tip-leakage vortex and the passage vortex.

2.2.2 Conventional Subdivision of Losses

In turbomachinery, it has been the practice to break the losses into components, typically profile losses, endwall losses, and tip leakage losses, although experimental evidence suggests that the loss sources are seldom really independent (e.g. Denton, 1993) .

Profile losses: are usually taken to be the losses generated in the blade boundary layers well away from the endwall effects. It is often assumed that the flow here is two-dimensional so the loss may be based on two-dimensional cascade tests. In addition, the losses arising due to the finite trailing edge thickness are usually included in the profile losses. The title “mid-span losses” is sometimes used to refer to the profile losses.

Endwall losses: usually refer to the losses generated from the secondary flows generated when the annulus boundary layers pass through the blade row. However as argued by Denton (1993), it is often difficult to separate the endwall losses from profile losses and leakage

losses. Therefore, the secondary losses are sometimes taken to include all the losses that cannot be accounted for.

Leakage losses: are due to the tip leakage flow over the tips of the blades and the hub clearance of stator blades. Usually there is an interaction between tip leakage and endwall flows. The detailed loss mechanisms also depend on whether the blades are shrouded or unshrouded.

The relative magnitudes of the above loss components depend on the type of the machine and on such details as blade aspect ratios and tip clearance.

2.2.3 Scope of Present Investigation

In the present experimental investigation, focus will be on the aerodynamic losses at mid-span of a linear turbine cascade where the blade boundary layers are nearly two-dimensional. As mentioned previously, the present study includes the effects of incidence and transonic Mach numbers on mid-span losses. Thus, the following sections will present a review of previous work which is particularly pertinent to this study.

2.3 Mid-Span Flow in Transonic Axial Turbines at Design Incidence

2.3.1 Overview of Sources of Losses at Mid-Span

As discussed by Mee et al. (1992), the mid-span losses in a linear transonic turbine cascade may be broken down into three components; namely boundary layer losses, shock losses, and trailing edge mixing losses. Figure 2.2 shows the effects of Mach number on these three loss components and the total resultant losses for a particular case. The results presented in Figure 2.2 are for an HP cascade test section at an exit Reynolds number of 10^6 . Further details of the cascade and flow conditions are given in Table 2.1.

Inlet Metal Angle	42.8°	Design exit Reynolds Number	10 ⁶
Inlet Mach Number	0.31	True Chord	76.9 mm
Inlet Total Temperature	285K	Axial Chord	100.0mm
Inlet Turbulence Intensity	4.1%	Span	300 mm
Nominal Exit Flow Angle	68.0°	Pitch	84.0 mm
Design Exit Mach number	0.92	Trailing Edge Diameter	2.0 mm

Table 2.1 Details of the Cascade used by Mee et al. (1992).

Figure 2.2 Variation of Mid-Span Loss Components with Mach Number for a Turbine Cascade (reproduced from Mee et al., 1992).

This figure shows that, while boundary layer losses dominate at low Mach numbers, mixing losses and shock losses become important at high Mach numbers. The figure also shows that in this case the recompression shock impinging upon the suction surface boundary layer does not apparently cause separation since it does not lead to a sharp increase in the boundary layer losses. The increase in the total losses at Mach numbers greater than one can be attributed to the trailing edge losses associated with the mixing out of the wake downstream of the trailing edge. This trailing edge loss component is about 60% of the total mid-span losses. The trailing edge losses and their relation to the base pressure is discussed in Section 2.3.4.

Although the data shown in Figure 2.2 are only limited to a particular case, the trends of the loss components behaviour are representative for turbine blades. However, the relative magnitudes of the various loss components are influenced by various geometric and flow parameters. These parameters affect the blade boundary layer developments, and the shock boundary layer interaction, particularly at Mach numbers greater than one. Of these parameters, most important are the blade incidence, blade geometry, and the Reynolds numbers.

2.3.2 Aerodynamic Influences on Losses

2.3.2.1 Significance of Axial-Velocity-Density-Ratio (AVDR)

The profile loss measurements are usually conducted at blade mid-span where the flow is assumed to be two-dimensional. The two-dimensionality of the flow through the cascade is measured by the axial velocity density ratio (AVDR) or in incompressible flows the axial velocity ratio. Physically, the AVDR is the ratio of the mass flow rates per unit area at inlet and outlet of the cascade. Thus, AVDR becomes a measure of the average contraction or expansion of the streamtube at the blade mid-span. The AVDR is therefore a necessary, but not sufficient, condition for a two-dimensional flow. The AVDR at the blade mid-span is defined using the following equation:

$$AVDR = \frac{\int_0^1 (\rho_2 C_{(ax)_2})_{MS} d\left(\frac{y}{s}\right)}{\int_0^1 (\rho_1 C_{(ax)_1})_{MS} d\left(\frac{y}{s}\right)} \quad (2.1)$$

where the subscripts 1, 2, and MS refer to the leading edge plane, trailing edge plane, and mid-span plane respectively. The subscript (ax) refers to the axial component of the velocity. A value of AVDR of 1.0 refers to a flow with constant mass flow per unit area. A flow with axial deceleration has an AVDR value less than 1.0, and a flow with axial acceleration has an AVDR value greater than 1.0.

The effect of AVDR on mid-span performance of compressor blades has received considerable attention (e.g. Heilmann, 1972, and Stark & Hoheisel, 1981). In these investigations, it was generally found that an increase in AVDR led to a decrease in total pressure loss, a slight decrease in deviation, and a slight increase in blade loading.

For turbine blades on the other hand, the influence of AVDR on the performance has not been documented particularly well. The effects of the axial velocity ratio (AVR) in incompressible flow conditions has been examined experimentally by Rodger et al. (1992) and Whitehouse et al. (1993). They found that the losses decreased with increasing AVR. This behaviour was observed for all off-design values of incidence. At transonic flow conditions, Kiock et al. (1986) found that the exit flow angle and the blade losses were essentially independent of the AVDR value, particularly in the Mach number range of 0.9 to 1.0. The Kiock et al. results are for design incidence only. Saroch (1996) conducted a computational study of the effects of AVDR in a turbine cascade at transonic flow conditions, but again just for the design incidence. Saroch used the Navier-Stokes code of Dawes (1988) to predict the effects of AVDR on the loss coefficient and deviation in a linear turbine cascade for outlet Mach numbers from 0.6 to 1.2. The loss coefficients predicted by Saroch are shown in Figure 2.3. For Mach numbers from 0.6 to 0.9, the losses were found to increase slightly with increasing AVDR. For Mach numbers from 0.9 to 1.0 the predicted losses were essentially independent of AVDR, as also observed experimentally by Kiock et al.. As the Mach

number increased beyond unity, Saroch found that the losses decreased with increasing AVDR. The effects of AVDR on deviation at transonic Mach numbers, as computed by Saroch, were similar to the ones observed for the loss coefficient. Saroch also found that, in addition to the Mach number effects, an increase in AVDR led to an increase in blade loading, particularly at the trailing edge. Pending further investigations, the findings of Saroch provide best available measure of the effects of AVDR on blade performance at transonic Mach numbers..

Figure 2.3 Computations of the Effects of AVDR on Profile Losses at Transonic Mach Numbers (reproduced from Saroch, 1996).

2.3.2.2 Compressibility Effects

Modern turbine designs are operating at transonic Mach numbers for two reasons: a) the high tip speed of the rotor stage, and b) the reduction of the number of turbine stages in order to reduce the weight (high blade loading), and the complexity of the engine. These factors cause a considerable increase of the aerodynamic loading of the turbomachine components. The resulting transonic flow fields within the vanes and blades of highly loaded turbines are dominated by two factors (Hourmouziadis, 1989): a) the shock structure and related losses, and b) the shock boundary layer interaction.

The flow phenomena involved in these two factors and their effects on losses have been examined to some extent by several authors (e.g. Graham & Kost, 1979, Kiock et al., 1986, Dietrichs et al., 1987, Hourmouziadis, 1989, Mee et al., 1992, Moustapha et al., 1993, and many others).

The first problem is associated with the development of losses arising from the shocks which cause an entropy rise without viscous effects being involved. The second problem which is concerned with the shock boundary layer interaction, has attracted much of the attention in the analytical as well as the experimental studies. This attention is mainly due to the difficulty in predicting the effects of the shocks on transition and separation.

The incident shock from the trailing edge of one of the blades meets the suction surface of the adjacent blade. In the absence of a boundary layer, the shock would be reflected from the solid boundary layer as a compression shock wave. The presence of the boundary layer leads to a shock boundary layer interaction.

If the incident shock is weak, the boundary layer does not separate, and the losses in this case are not very high. When the incident shock is strong, a separation bubble appears at the foot of the shock (at the suction surface), and depending on the shock intensity (Mach number) the boundary layer may separate, resulting in high losses. This problem needs to be more investigated since it is

still unclear, as stated by many investigators, if the viscous interaction on the airfoils suction surfaces causes boundary layer transition.

The studies on the effects of transonic Mach number on losses were for exit Mach numbers ranging from about 0.3 to about 1.2. Although valuable, these studies were in the majority limited to design values of incidence. At off-design incidence, there is an added complexity to the transonic flow field within the blade passages. Thus, additional data at transonic Mach numbers and off-design incidence are needed to help understanding the complex flow field within the transonic turbine blade passages.

2.3.2.3 Reynolds Number Effects

In an Altitude Test Facility (ATF), Koza & Yashima (1989), showed that the operating Reynolds number of an F3-30 small turbofan engine is as low as 40,000 when at altitude. The results of the measurements conducted in this test facility showed that at high altitudes, a considerable change occurs in the operating Reynolds number of the different engine components. Table 2.2 lists some typical numbers of Re at SLS and at an altitude of 50,000 ft and a Mach number of 0.5.

Component	SLS	50,000/ $M_0 = 0.5$
Fan	2.3×10^6	4.2×10^5
Compressor	1.6×10^6	3.2×10^5
HP Turbine	1.6×10^6	2.8×10^5
LP Turbine	2.5×10^5	4.4×10^4

Table 2.2 Variation of the Operating Reynolds Number for the Different Components of a Small Turbofan Engine from SLS to Cruise Conditions (Koza & Yashima, 1989).

The Reynolds numbers effects on blade cascade performance have been the subject of several investigators (e.g. Seyb, 1972, Patterson & Hoeger, 1986, Vijayaraghavan & Kavanagh, 1988, Hourmouziadis, 1989, Mayle, 1991, Ladwig & Fottner, 1993, Boyle et al., 1998, and many others). The Reynolds numbers in these studies ranged typically from about 10^5 to about 10^6 . Figure 2.4 shows typical loss variations with Reynolds numbers where the losses are seen to increase with decreasing Reynolds numbers.

Figure 2.4 Influence of Exit Reynolds Number on the Total pressure Loss Coefficient at $M_2 = 0.59$ and different degrees of Turbulence Intensities (Ladwig and Fottner, 1993).

Mayle (1991) and Hourmouziadis (1989) explained clearly the development of the blade boundary layers as a function of the Reynolds numbers, and the losses associated with it. At high Re , turbulent separation which may be occurring near the trailing edge produces mixing losses which are added to the shear layer losses. As the Re decreases, the turbulent separation disappears and the situation corresponds to a fully attached flow. The loss corresponding to this flow is the lowest. A further decrease in Re causes a short separation bubble to appear on the suction surface of the blade. With decreasing Reynolds numbers, the separation bubble extends upstream towards the blade leading edge resulting in a separated free shear layer. The rising losses in this case are caused by the growing free shear layer thickness with falling Re .

In the present investigation, a limited range of the Reynolds numbers was examined. Therefore, only a brief discussion of the effects of Reynolds numbers on losses is presented.

2.3.3 Geometric Influences on Losses

One of the components of the profile losses is the boundary layer loss. The parameters which affect the blade boundary layer development, and hence the profile losses, include the leading edge geometry and the blade suction surface curvature (e.g. Mukhtarov & Krichakin, 1969, Martelli & Boretti, 1987, and Korakianitis & Papagiannidis, 1993, and many others).

Martelli & Boretti (1987) showed that the profile loss and its evolution in relation to exit Mach number is strongly influenced by the rear suction surface curvature. They argued that the profile losses increase if the suction surface velocity gradient is irregular and steep. They found that blades with a straight rear suction surface usually exhibit a marked variation in radius of curvature. This produces an unfavorable suction surface velocity distribution with a strong gradient of decelerating velocity following an early velocity peak. The gradient may even be strong enough to separate the boundary layer. This is particularly so when the suction surface velocity peak is supersonic and the internal shock system is associated with the decrease in velocity. For blades with

a convex rear suction surface curvature, they found that there is a gradual variation in surface curvature radius which results in a smooth gradient of decelerating suction surface velocity.

Korakianitis & Papagiannidis (1993) showed that the effects of discontinuities in surface curvature may result in undesirable blade loading distributions (e.g. separation bubbles, local regions of accelerated and decelerated flows) which affect the blade boundary layer developments and hence the loss production.

The effects of surface discontinuity, particularly at the leading edge, have been examined in detail both experimentally and numerically at low speed conditions by Benner et al. (1997). Benner et al. concluded that turbine blades would be more sensitive to off-design incidence if there is large discontinuities in surface curvature at the leading edge blend points. A similar effect was observed by Walraevens & Cumpsty (1995) for compressor blades.

2.3.4 The Trailing Edge Loss and the Role of Base Pressure

As discussed in Section 2.2.2, the extra loss due to the finite trailing-edge thickness is usually included in the profile losses. Early work by Purst and Helon (1972) on blades with different trailing edge shapes and thicknesses at subsonic speeds showed that the trailing edge loss is proportional to the drag coefficient which is a function of the base pressure. Xu & Denton (1988) showed that the trailing edge loss is directly proportional to the trailing edge thickness, with the constant of proportionality being greater at transonic and supersonic speeds. As argued by Xu & Denton, the rise in the trailing edge losses coincides with a decrease in the base pressure. With some simplifying assumptions and for incompressible flows, Denton (1993) relates the mid-span losses to the trailing edge thickness and the boundary layer parameters at the trailing edge using the following equation:

$$Y_t = -\frac{C_b t}{o} + \frac{2\theta}{o} + \left(\frac{\delta^* + t}{o} \right)^2 \quad . \quad (2.2)$$

where, δ^* : blade boundary layer displacement thickness,
 θ : blade boundary layer momentum thickness,
 t : blade trailing edge thickness,
 o : throat opening, and
 C_b : base pressure coefficient.

In explaining the behaviour of the base pressure in transonic flows, Xu and Denton (1988) argue that the high entrainment rate of the fluid from the base region into the shear layer may be the cause of the low base pressure. The high entrainment rate is also associated with high shear stresses. This mechanism helps to explain the dependence of the base pressure on both the Reynolds number and the thickness of the boundary layer at the trailing edge relative to the trailing edge thickness.

Earlier methods used to predict the base pressure are based on base pressure theories (e.g. Korst 1956, Carrière, 1970, and Cox 1971). These methods are generally based on finding a solution to the problem of the confluence of two supersonic flows at the trailing edge of the blade. The assumptions which form the basis of the calculation procedures in these theories are a) treating the blade trailing edge as a sharp cut trailing edge, and b) the boundary layer is turbulent at the trailing edge point of separation. Furthermore, the assumed amount of flow turning in the confluence region is based on empirical data which have been obtained from isolated airfoils in a supersonic free stream. As argued by Xu & Denton (1988) the flows around these airfoils may not be representative of the flow around turbine blades. In addition, Denton and Xu (1990) showed that these conventional theories underpredict the base pressure when compared to the experimental values.

Perhaps the most comprehensive evidence of the importance of the base pressure in transonic flows is the study by Sieverding et al. (1980). From the limited amount of data, Sieverding et al. concluded that the knowledge of the base pressure is important for the calculation of the trailing

edge losses and the determination of the trailing edge shock system in supersonic flows. Figure 2.5, shows the evolution of the base pressure as a function of the downstream static pressure. It is clear from the figure that the behaviour of the base pressure is determined by the characteristics of the downstream flow field. Sieverding et al. used these results to devise a correlation for the base pressure as a function of the downstream static pressure, P_{s2} , the trailing edge wedge angle, W_{α} , and the unguided turning, θ_u . The unguided turning is a measure of the suction surface curvature downstream of the blade passage throat. This correlation is shown graphically in Figure 2.6.

Denton (1993), in an extensive discussion of the trailing edge losses, argued that the correlation of the base pressure presented by Sieverding et al. does not include the trailing edge blockage effects. Sieverding et al. (1980) recognize the influence of the trailing edge thickness on the base pressure. However, due to the limited amount of data gathered on the various blades tested, they were not able to include the effects of the trailing edge thickness in their correlation. These blockage effects are due to both the trailing edge thickness, t , and the displacement thickness, δ^* (see Eqn. (2.2)).

To include the effects of trailing edge blockage on blade mid-span losses, Denton suggested using Figure 2.7. The results shown in Figure 2.7 are from numerical solutions of the conservation equations. The figure shows that the base pressure required for zero loss (i.e. $Y_t = 0$) increases rapidly as the exit Mach number approaches 1.0. In this case, Denton suggested that the loss will be proportional to the difference between the actual base pressure and the base pressure for zero loss (i.e. $Y_t = 0$). Using this approach, the values for the actual base pressure and the constant of proportionality are to be determined from experiments.

In summary, additional data on base pressure, particularly at transonic Mach numbers, are needed in order to help in the understanding of the aerodynamic performance of turbine blades. These data will be useful to the designer at the early stage of the design in order to assess the trailing edge loss penalties associated with the thick trailing edges which are necessary due to the cooling and the stress requirements.

- a: Subsonic Flow,
- b: Appearance of Supersonic Flow at Trailing Edge,
- c: Transition from Normal Shock to Oblique Trailing Edge Shock System,
- d: Displacement of Shock S1 towards Trailing Edge,
- e: Final Approach of S1 to Trailing Edge, and
- f: Shock Interaction with Wake.

Figure 2.5 Evolution of the base Pressure as a Function of the Downstream Static Pressure (reproduced from Sieverding et al., 1980).

Figure 2.6 Sieverding's et al. Base Pressure Correlation
(reproduced from Sieverding et al., 1980).

Figure 2.7 Base Pressure for Zero Loss as a Function of Exit Mach Number and Trailing Edge Thickness (reproduced from Denton, 1993).

2.4 Mid-Span Flow in Transonic Axial Turbine at Off-Design Incidence

2.4.1 Overview of the Effects of Off-Design Incidence

At off-design incidence, the inlet flow velocity vectors are mismatched with the leading edge angles of the blades causing additional losses, commonly referred to as incidence losses. Figure 2.8 (Mayle, 1991) shows typical blade boundary layer as well as profile loss behaviours at extreme values of off-design incidence. As shown in the figure, the increase in losses at very large values of both negative and positive incidence is due to the massive boundary layer separation on the pressure and suction surfaces respectively. At smaller values of off-design incidence, the separated zones may reattach to form separation bubbles. The presence of such bubbles increases the losses to a much smaller degree as indicated by the gradual rise in losses away from but close to the design incidence.

Figure 2.8 Incidence Effects on Blade Boundary Layer Behaviour and Blade Profile Losses (reproduced from Mayle, 1991).

2.4.2 Aerodynamic Influences on Off-Design Losses

The effects of off-design incidence on profile losses at subsonic Mach numbers have been examined extensively by several authors (e.g. Hashimoto & Kimura, 1984, Hodson & Dominy, 1986, Venkatrayulu et al., 1989, Goobie, et al., 1989, Tremblay et al., 1990, and Benner et al., 1997, and many others). These include effects of aerodynamic and geometric parameters on profile losses. Subsequently recent advances in the development of profile loss correlations as well as exit flow angle correlations have been for subsonic flow conditions.

At transonic Mach numbers, with the exception of few investigations, the existing data on the effects of Mach numbers on profile losses are at the design values of incidence (e.g. Graham & Kost, 1976, Camus et al., 1984, Kiock et al., 1986, Kurz, 1991, Mee et al., 1992, and others). The study of Graham & Kost also included some limited data on the effects of off-design incidence values on blade mid-span losses. These studies were used to explain the loss behaviour with Mach numbers and the shock boundary layer interaction phenomenon and its influence on losses.

At off-design incidence values, the reported investigations in the open literature at transonic Mach numbers are scarce. These studies include studies in linear cascades and rotating rigs. Graham & Kost (1979) tested two blade profile over a range of exit Mach numbers from 0.7 to 1.4 and incidence valued from -20° to $+8^\circ$. The results of Graham & Kost indicated the sensitivity of the blade performance to incidence and the suction surface curvature aft of the throat. Detemple-Laake (1991) performed tests on a highly loaded rotor blade at incidence values ranging from -10° to $+25^\circ$ and exit Mach numbers ranging from 0.8 to 1.3. Detemple-Laake found that the passage vortex shifted toward the side wall until the cascade was choked and then remained the same for all supersonic Mach numbers. The shock-boundary layer interaction did not extend to the side wall region whereas the trailing edge shock did. Perdichizzi & Dossena (1993) tested a high pressure steam turbine blade with three pitch to chord ratios at incidence values ranging from -60° to $+35^\circ$ and at an exit design Mach number value of 0.7. They found that high levels of streamwise vorticity and intense crossflows at the endwall take place at large incidence angles and low solidities. Also, they found that as the incidence angle and the pitch-to-chord ratios are increased, the region covered by the secondary flows extends progressively toward the mid-span. Krieger et al. (1999) tested a

transonic single stage high pressure turbine at off-design conditions. The turbine speed was varied from 70 to 120 percent of design speed and the pressure ratio was varied from 75 to 125 percent of the design pressure ratio. The Reynolds number was also varied from a value of 32,000 to a value of 250,000. This variation in Reynolds numbers represents variations in the turbine Reynolds number from sea level conditions to cruise conditions. Krieger et al. showed that the turbine efficiency lapse rate was over 1.5 percent from equivalent sea level to cruise altitude Reynolds numbers at the design point. Also, they found that the turbine expansion efficiency decreased by nearly 3 percent per 10 percent speed at speeds below design and increased for speeds up to 15 percent above design value. In the above studies, very little is presented on the effects of off-design incidence on the base pressure and its relationship to the blade losses.

In summary, the existing transonic flow data in the literature are still not sufficient to explain the complex nature of transonic flows, and hence to develop satisfactory transonic correlations that can be used at the design stage of turbine blades.

The available profile loss correlations and exit flow angle correlations are discussed Sections 2.5.2 and 2.5.3 respectively.

2.4.3 Geometric Influences on Off-Design Losses

As mentioned in Section 2.3.3, the suction surface curvature and the leading edge geometry have a significant effect on the suction surface pressure distribution. Benner et al. (1997) found that the effects of surface curvature discontinuities at the leading edge blend points are very significant at off-design incidence values. The effects of the leading edge geometry have been investigated at subsonic conditions by several authors (e.g. Mukhtarov & Krichakin, 1969, Martelli & Boretti, 1987, and Moustapha et al., 1993). However, the most comprehensive study on the effects of the leading edge geometry is the one by Benner et al. (1997). Benner et al. argued that the leading edge wedge angle provides a good measure of the discontinuity in curvature at the leading edge blend points. In addition, they found that blades with large wedge angles will be less sensitive to off-design incidence values.

At transonic Mach numbers, Krieger et al. (1999) showed that the leading edge flow varied significantly at off-design operating conditions. The leading edge loading increased significantly at high values of positive incidence. As argued by Krieger et al., these changes to the leading edge flows may have a significant impact on the turbine efficiency. Understanding the effects of the leading edge geometry on losses may result in blade designs capable of tolerating large changes in incidence values with minimum penalties in losses. Thus additional data on the effects of the leading edge geometry, particularly at positive incidence values, may help to achieve this objective.

2.5 Profile Loss and Exit Flow Angle Correlations for Axial Turbines

2.5.1 Introduction

Mean line prediction methods continue to play an important role in the preliminary design of modern gas turbine engines. These prediction methods use experimental correlations to account for various aerodynamic and geometric effects. The present section presents the profile loss and exit flow angle correlations available in the open literature. These correlations have been modified and improved as more data that explain and help understand the flow physics become available.

In this study, the predictions from these correlations will be compared to the experimental data. This comparison is presented in Chapter 5. The comparisons may be used as a basis for any improvements to the existing correlations.

2.5.2 Profile Loss Correlations

2.5.2.1 Loss Definitions

In this study, the loss coefficients are based on the mixed-out mass-averaged flow properties at the exit of the cascade blade row. The mass-averaged flow properties are calculated using the following equation:

$$\bar{P}_{prop} = \frac{1}{\dot{m}} \int_0^s P_{prop} d\dot{m} \quad (2.3)$$

where the integral is evaluated over one blade pitch, s , and P_{prop} is the flow property.

The fully mixed-out value of the flow property is obtained using a control volume analysis in which the measured flow is assumed to mix out to uniform conditions at constant area. This control volume analysis is the procedure followed by Amecke & Šafařík (1995). The method is based on solving the four conservation equations of mass, momentum in the axial direction (parallel to the blade row), momentum in the pitchwise direction (perpendicular to the blade row), and energy. Amecke and Šafařík extended the earlier version of the data reduction system of Amecke (1970) to include both the non-uniformity of the total temperature, as suggested by Oldfield et al. (1981), and the injection into the flow of other gases as in the case of blade cooling. An outline of Amecke and Šafařík's method is presented in Appendix A.

Numerous loss coefficients are currently in use by designers and researchers (Brown, 1972; Denton, 1993). Also, the form of the loss coefficients is usually different for compressors and turbines. For turbines, perhaps the most common form is the stagnation pressure loss coefficient (Y_t) given by the following equation:

$$Y_t = \frac{P_{01} - P_{02}}{q_2} \quad (2.4)$$

where q_2 is the mixed-out mass averaged dynamic pressure at the exit of the cascade ($= 0.5\rho C^2$), and P_{01} is the mass-averaged inlet total pressure, and P_{02} is the mixed-out mass averaged total pressure at the exit of the cascade.

Changing the denominator in equation (2.4) to $(P_{02} - P_2)$ will result in another total pressure coefficient ω_t (Eqn. (2.5)) where P_2 is the mixed-out static pressure at the exit of the cascade. This pressure loss coefficient is sometimes used for compressible flow results. In the incompressible limit, $Y_t = \omega_t$. In compressible flow, the value of ω_t will be smaller than the corresponding Y_t .

$$\omega_t = \frac{P_{01} - P_{02}}{P_{02} - P_2} \quad (2.5)$$

The energy loss coefficient ξ has also been used fairly extensively to present turbine blade performance (e.g. Brown, 1972, Gostelow, 1984, Horlock, 1960, and Denton, 1993). The energy loss coefficient is given by:

$$\xi = \frac{h_2 - h_{2s}}{h_{02} - h_2} = \frac{1 - \left(\frac{P_2}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}}}{1 - \left(\frac{P_2}{P_{02}} \right)^{\frac{\gamma-1}{\gamma}}} - 1. \quad (2.6)$$

where h_2 is the actual static enthalpy, h_{2s} is the isentropic static enthalpy, and h_{02} is the total enthalpy at the exit of the cascade.

There exists an alternative definition of the energy loss coefficient, as used by Kurz (1991) and Venkatrayulu et al. (1989) as follows:

$$\xi^* = \frac{h_2 - h_{2s}}{h_{01} - h_{2s}} = 1 - \frac{1 - \left(\frac{P_2}{P_{02}} \right)^{\frac{\gamma-1}{\gamma}}}{1 - \left(\frac{P_2}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}}} \quad (2.7)$$

where h_{01} is the stagnation enthalpy at the inlet of the cascade.

For completeness, the loss coefficient was calculated using Eqns (2.4), (2.5), and (2.6) for each test case in this experimental study. The values from all loss coefficients are included in Appendices D and E for all test cases. However, the discussion presented in Chapter 5 is based on the loss coefficient Y_t .

2.5.2.2 Loss Correlations at Design Incidence

Perhaps the most widely used empirical loss system is based on a system originally proposed by Ainley & Mathieson (AM) (1951). The Ainley & Mathieson loss system is a complete set of correlations for all the loss components at design and off-design conditions.

At design incidence, Ainley & Mathieson proposed a profile loss correlation based on cascade tests of blades designed prior to 1950. The tests were performed at an exit Reynolds number of 2×10^5 (based on chord and exit velocity) and exit Mach numbers less than 0.6. The profile losses are correlated with the inlet blade angle β_1 , pitch-to-chord ratio (s/c) (or cascade solidity), exit flow angle α_2 , and the blade maximum thickness-to-chord ratio (t_m/c). The loss correlation is presented in terms of two graphs, one for nozzle blades ($\beta_1 = 0$), and one for impulse blades ($\beta_1 = \alpha_2$), as shown in Figures 2.9 and 2.10 respectively. For blades with the same inlet angle and intermediate outlet angles, the profile loss coefficient is obtained as a weighted average of the nozzle blade and impulse blade values:

$$Y_P = \left(Y_{P(\beta_1-0)} + \left(\frac{\beta_1}{\alpha_2} \right)^2 \left(Y_{P(\beta_1-\alpha_2)} - Y_{P(\beta_1-0)} \right) \right) \left(\frac{t_m/c}{0.2} \right)^{\beta_1/\alpha_2} \quad (2.8)$$

The above equation includes a correction for the effects of maximum thickness-to-chord ratio (t_m/c) which is valid only for the range of 0.15 to 0.25. If the blade has a t_m/c outside this range, the loss is taken as the value corresponding to $t_m/c = 0.25$ or 0.15.

Figure 2.9 Ainley & Mathieson Profile Loss Correlation for Nozzles (reproduced from Ainley & Mathieson, 1951).

Figure 2.10 Ainley & Mathieson Profile Loss Correlation for Impulse Blades (reproduced from Ainley & Mathieson, 1951).

In 1970, Dunham & Came (1970) modified this loss system to reflect the improved understanding of some of the aspects of the flow. For example, Dunham & Came proposed a Reynolds number correction factor for cascades operating at Reynolds numbers other than 2×10^5 :

$$Y_P = \left(Y_{P(\beta_1=0)} + \left(\frac{\beta_1}{\alpha_2} \right)^2 \left(Y_{P(\beta_1=\alpha_2)} - Y_{P(\beta_1=0)} \right) \right) \left(\frac{t_m/c}{0.2} \right)^{\beta_1/\alpha_2} \left(\frac{Re}{2 \times 10^5} \right)^{-0.02} \quad (2.9)$$

Further modifications to the Ainley/Mathieson/Dunham/Came (AMDC) loss system were suggested by Kacker & Okapuu (KO) (1982). The major change introduced by KO is the idea that for modern blades the profile losses are two-thirds the value suggested by AM due to improvements in design. They also showed that the flow compressibility can affect the profile loss (Y_P) even in subsonic regime by affecting the flow acceleration within blade channels. Therefore, they introduced a compressibility correction factor to this channel acceleration effect. At design incidence, the modified correlation is as follows:

$$Y_P = 0.914 \left(\frac{2}{3} Y_{P,AMDC} K_P \right) \quad (2.10)$$

where $Y_{P,AMDC}$ is the profile loss coefficient predicted using AMDC, and K_P is the correction factor that takes into account the effects of exit Mach number and channel flow acceleration. The correction factor is given by the following equation:

$$K_P = 1 - K_2(1 - K_1) \quad (2.11)$$

and K_1 and K_2 are given by:

$$K_1 = 1 - 1.25(M_2 - 0.2) \quad \text{for } M_2 > 0.2$$

$$K_2 = (M_1 / M_2)^2$$

As mentioned in Section 2.2.2, the losses arising due to the finite trailing edge thickness are usually included in the profile losses. The discussion presented earlier in Section 2.3.4 shows the significance of the base pressure and the trailing edge thickness in determining the trailing edge loss. Obviously, the trailing edge thickness has little or no influence on the boundary layer losses, but has a strong influence on the sudden expansion/mixing losses. Kacker & Okapuu argued that the most meaningful expression for the trailing edge loss due to the trailing edge blockage is in terms of the trailing edge thickness to throat ratio. They expressed the trailing edge losses in terms of the energy coefficient.

The energy loss coefficient ϕ^2 is defined as follows:

$$\phi^2 = 1 - \frac{C_2^2}{C_{2is}^2} \quad (2.12)$$

where C_2 is the actual exit velocity, and C_{2is} is the isentropic exit velocity.

This energy loss coefficient was correlated with the ratio of the trailing-edge-thickness to the throat area. A graphical representation of this correlation is shown for nozzles and impulse blades in Figure 2.11. The following is a curve fit of the correlation (Verhiel & Sjolander, 1994):

$$\Delta\phi_{TET,I}^2 = 0.0232143 \left(\frac{t}{o} \right) + 0.575714 \left(\frac{t}{o} \right)^2 - 0.45 \left(\frac{t}{o} \right)^3 \quad (2.13)$$

for impulse blades, and

$$\Delta\phi_{TET,N}^2 = 0.0387381\left(\frac{t}{o}\right) + 1.13857\left(\frac{t}{o}\right)^2 - 0.916667\left(\frac{t}{o}\right)^3 \quad (2.14)$$

for nozzle blades. For blades with arbitrary inlet and outlet flow angles, Eqns. (2.13) and (2.14) are interpolated using the following equation:

$$\Delta\phi_{TET}^2 = \Delta\phi_{TET,(\beta_1=0)}^2 + \left(\frac{\beta_1}{\alpha_2}\right)^2 \left(\Delta\phi_{TET,(\beta_1=\alpha_2)}^2 - \Delta\phi_{TET,(\beta_1=0)}^2\right) \quad (2.15)$$

For the kinetic energy loss coefficient computed using Eqn. (2.15), the corresponding total pressure loss coefficient due to the trailing edge thickness, $\Delta Y_{t,te}$, is given by (Kacker & Okapuu, 1982):

$$\Delta Y_{t,te} = \frac{\left(1 - \frac{\gamma-1}{2} M_2^2 \left(\frac{1}{\Delta\phi_{TET}^2} - 1\right)\right)^{-\frac{\gamma}{\gamma-1}} - 1}{1 - \left(1 - \frac{\gamma-1}{2} M_2^2\right)^{-\frac{\gamma}{\gamma-1}}} \quad (2.16)$$

where M_2 is the Mach number at the exit of the blade row. This trailing edge loss contribution is added to the profile losses computed using Eqn. (2.10) to obtain the total profile losses as defined by Mee et al. (1992).

Figure 2.11 Trailing Edge Energy Loss Coefficient versus the Trailing Edge Thickness to Throat Ratio (reproduced from Kacker & Okapuu, 1982).

At supersonic exit Mach numbers, additional shock losses generated at the trailing edge are accounted for using a profile loss multiplier. In the AMDC loss system, the multiplier assumes a supersonic loss rise according to:

$$CFM = 1 + 60(M_2 - 1)^2 \quad (2.17)$$

This multiplier is applied to the profile loss coefficient when the exit Mach number exceeds unity, and the same multiplier is used by Kacker & Okapuu.

In Eqn. (2.10), it is assumed that the profile loss coefficient Y_p is calculated at a reference Reynolds number of 2×10^5 (based on the blade true chord and the exit flow conditions). At any other Reynolds number values, the following corrections are used:

$$\begin{aligned} f_{(Re)} &= \left(\frac{Re}{2 \times 10^5} \right)^{-0.4} & \text{for } Re \leq 2 \times 10^5 \\ &= 1.0 & \text{for } 2 \times 10^5 < Re < 10^6 \\ &= \left(\frac{Re}{10^6} \right)^{-0.2} & \text{for } Re > 10^6 \end{aligned} \quad (2.18)$$

2.5.2.3 Loss Correlations at Off-Design Incidence

So far, the aforementioned improvements to the profile loss correlation of Ainley & Mathieson were only confined to design conditions.

At off-design incidence, the method of Ainley & Mathieson (1951) consisted of first calculating the stalling incidence. They found that the stalling incidence could be correlated with the exit flow angle, inlet blade angle, and the pitch-to-chord ratio. They then expressed the profile losses as a function of the stalling incidence. Mukhtarov & Krichakin (1969) showed that the incidence losses are dependent on the leading edge geometry. They expressed the off-design losses as a function of incidence, channel convergence, compressibility, and the leading edge diameter-to-pitch ratio. The channel convergence is defined as $\cos(\beta_1)/\cos(\beta_2)$ and is used as a measure of the flow acceleration in blade passage. Martelli & Boretti (1987) introduced a multiplying factor in the stalling incidence method of Ainley & Mathieson to account for the effects of the leading edge wedge angle. Chen (1987) presented an off-design loss correlation for low pressure steam turbines. Chen found that the incidence loss correlation could be expressed in terms of the flow incidence and the inlet blade angle.

Based on an extensive survey of cascade data, Moustapha et al. (1990) proposed an improved incidence loss prediction method that includes the effects of the leading edge diameter, blade spacing, and channel convergence. They expressed the loss incidence in terms of the energy loss coefficient as follows:

$$\Delta\phi_p^2 = 0.778 \times 10^{-5} \chi' + 0.56 \times 10^{-7} \chi'^2 + 0.4 \times 10^{-3} \chi'^3 + 2.054 \times 10^{-19} \chi'^6 \quad (2.19a)$$

for $800 > \chi' > 0$, and

$$\Delta\phi_p^2 = -5.1734 \times 10^{-6} \chi' + 7.6902 \times 10^{-9} \chi'^2 \quad (2.19b)$$

for $0 > \chi' > -800$,

where,

$$\chi' = \left(\frac{d}{s} \right)^{-1.6} \left(\frac{\cos \beta_1}{\cos \beta_2} \right)^{-2} (\alpha_1 - \alpha_{1(des)}) \quad (2.19c)$$

Figure 2.12 shows the incidence loss correlation of Moustapha et al. (1990).

Figure 2.12 Profile Incidence Loss Correlation of Moustapha et al. (1990)
(reproduced from Benner et al., 1997).

Further improvements to Moustapha et al. correlation were proposed by Benner et al. (1997) by including the leading edge wedge angle as an additional and more influential correlating

parameter. The improvements proposed by Benner et al. were based on recent experimental investigations that prompted the authors to re-examine the correlation of Moustapha et al.. Benner et al. argued that the leading edge wedge angle provides a good measure of the discontinuity in curvature at the blend points where the leading edge circle joins the rest of the profile. The correlation devised by Benner et al. is a functional relationship that takes into account the effects of the leading edge wedge angle (We), the leading edge diameter, defined as diameter -to-pitch ratio, the channel acceleration effects defined as $\cos(\beta_1)/\cos(\beta_2)$, and the incidence angle ($\alpha_1 - \alpha_{1(des)}$). The improved correlation is as follows:

$$\Delta\phi_p^2 = a_8 \chi^8 + a_7 \chi^7 + a_6 \chi^6 + a_5 \chi^5 + a_4 \chi^4 + a_3 \chi^3 + a_2 \chi^2 + a_1 \chi \quad (2.20a)$$

for $\chi \geq 0$, and

$$\Delta\phi_p^2 = 1.358 \times 10^{-4} \chi^2 - 8.720 \times 10^{-4} \chi \quad (2.20b)$$

for $\chi < 0$.

where,

$$\chi = \left(\frac{d}{s} \right)^{-0.05} (We)^{-0.2} \left(\frac{\cos\beta_1}{\cos\beta_2} \right)^{-1.4} (\alpha_1 - \alpha_{1(des)}) \quad (2.20c)$$

$$a_8 = +3.711 \times 10^{-7}, \quad a_7 = -5.318 \times 10^{-6},$$

$$a_6 = +1.106 \times 10^{-5}, \quad a_5 = +9.017 \times 10^{-5},$$

$$a_4 = -1.542 \times 10^{-4}, \quad a_3 = -2.506 \times 10^{-4},$$

$$a_2 = +1.327 \times 10^{-3}, \quad a_1 = -6.149 \times 10^{-5},$$

As shown in Figure 2.13, the Benner et al. correlation seems to correlate reasonably well with the existing cascade loss data.

The Benner et al. correlation does not include the effects of compressibility and Reynolds number due to the lack of off-design incidence data that include both effects. However, Benner et al. suggest that it is sufficient to use the KO corrections. The Benner et al. correlation still needs to be validated against new experimental data at both subsonic and supersonic exit Mach numbers. If proven to be reasonably accurate, the correlation may provide the basis for any improvements to include the effects of compressibility, particularly at off-design incidence.

Figure 2.13 Profile Incidence Loss Correlation of Benner et al.
(reproduced from Benner et al., 1997).

2.5.3 Exit Flow Angle Correlations

Accurate predictions of the exit flow angle is an important step in the preliminary design phase of turbine stages. The estimation of optimum work output and the inlet flow angle of successive stages of a turbine depend on accurate predictions of the exit flow angle, or the flow deviation, where deviation is defined as the difference between the exit blade angle and the exit flow angle at the trailing edge. Compared with the profile loss correlations, little attention has been given to exit flow angle or deviation correlations.

Perhaps the most widely used correlation for exit flow angles is due to Ainley & Mathieson (1951). The correlation is usually given in graphical form and it shows that the exit flow angle is a function of the gauge angle $\cos^{-1}(o/s)$, where (o) is the throat opening and (s) is the blade spacing. The correlation is provided graphically for subsonic exit Mach numbers, as shown in Figure 2.14 for straight-back blades. At $M_2=1.0$, they suggest that the exit flow angle is equal to the gauge angle, and also that the exit flow angle varies linearly for Mach numbers between the low speed condition ($M_2=0.5$) and the choking condition ($M_2=1.0$). Corrections are also provided for curved-back blades with radius of curvature (e) between the throat and suction surface trailing edge. The exit flow angle is reduced by $(4s/e)$ at $M_2=0.5$, and by $(f \sin^{-1}(o/s))$ at $M_2=1.0$ from the values obtained from Figure 2.14. The function (f) used at $M_2=1.0$ is given in Figure 2.15. Ainley & Mathieson also suggest that the exit flow angle for a curved back blade may be assumed to vary linearly for Mach numbers in the range of 0.5 to 1.0.

Wilson (1984) translated these graphical representations into analytical expressions for the different Mach number operating conditions. These expressions are given by:

a) for exit Mach numbers less than 0.5, the predicted exit flow angle is given by:

$$|\alpha_2|_{0 < M_2 < 0.5} = \frac{7}{6} \left(\left| \cos^{-1} \left(\frac{o}{s} \right) \right| - 10^\circ \right) + 4^\circ \left(\frac{s}{e} \right) \quad (2.21)$$

Figure 2.14 Relationship between Exit Flow Angle and $\cos^{-1}(o/s)$ for Straight-Backed Blades Operating at Low Mach Numbers (reproduced from Horlock, 1973).

Figure 2.15 The Function $f(s/e)$ for Determining the Difference between α_2 and $\cos^{-1}(o/s)$ when $M_2 = 1.0$ (reproduced from Horlock, 1973).

b) for exit Mach numbers between 0.5 and 1.0, the predicted exit flow angle is given by:

$$|\alpha_2|_{0.5 < M_2 < 1.0} = |\alpha_2|_{0 < M_2 < 0.5} - (2M_2 - 1) \left(|\alpha_2|_{0 < M_2 < 0.5} - |\alpha_2|_{M_2 = 1.0} \right), \quad (2.22)$$

c) where for $M_2 = 1.0$, the predicted flow angle is given by:

$$|\alpha_2|_{M_2 = 1.0} = \left| \cos^{-1} \left(\frac{o}{s} \right) \right| - \left(\frac{s}{e} \right)^{1.786 + 4.128(s/e)} \left| \sin^{-1} \left(\frac{o}{s} \right) \right| \quad (2.23)$$

The Ainley & Mathieson correlation is still being used to provide reasonable estimates of the exit flow angle for Mach numbers up to 1.0.

Deviation correlations are also used in the design of turbine blades as an alternative approach to obtaining the outlet flow angle. Deviation correlations are generally used for $M_2 \leq 0.5$. Literature shows that the most widely used correlation for deviation is the one due to Carter & Hughes (1950). Carter & Hughes correlation was originally devised for compressor blades and is expressed as a function of the cascade geometry. The correlation is given as follows:

$$\delta = m \theta \left(\frac{s}{c} \right)^n \quad (2.24)$$

where θ is the blade camber, s is the blade spacing, c is the blade chord. The multiplier m depends on the stagger angle (ζ) and the shape of the blade camber as shown in Figure 2.16.

For normal compressor blades, the authors recommended a value of 1/2 for the exponent n . However, for the accelerating flows within inlet guide vanes, they recommended using $n=1.0$. The correlation of Carter & Hughes is applied to turbine blades with a value of 1.0 since turbines have accelerating flows, like compressor inlet guide vanes.

Figure 2.16 Relationship between the Constant (m) and Stagger Angle (reproduced from Horlock, 1973).

Recently, Islam & Sjolander (1999) devised a new deviation correlation for subsonic flow conditions in turbines. They argued that the deviation is mainly a function of the blade loading towards the trailing edge of the blade. They then identified the geometric and flow parameters which influence the aft end loading of the blade. Islam & Sjolander expressed the correlation as a function of cascade geometry and some flow parameters. The correlation was based on experimental and numerical results obtained for turbine blades that reflect the recent improvements in turbine blade design. The correlation is given by the following equation:

$$\delta = \frac{(AVDR)^3 \left(\frac{s}{c} \right)^{1.1} (\alpha_1 + \beta_2)^{2.25}}{\zeta^{1.45} \left(\frac{t_m}{c} \right)^{0.3} (22 + 0.22\beta_1^{1.64})} \quad (2.25)$$

where, AVDR : axial velocity density ratio,

α_1 : inlet flow angle, in degrees,

β_2 : exit metal angle, in degrees,

ζ : stagger angle, in degrees,

t_m/c : blade maximum thickness to chord ratio, and

s/c : blade space to chord ratio.

As discussed by Islam & Sjolander, the correlation is valid for AVDRs from 0.95 to 1.05 and exit Mach numbers up to 0.7. The correlation also includes the effects of off-design incidence. Islam & Sjolander correlation is intended to replace the low speed portion of Ainley & Mathieson correlation.

3. Experimental Apparatus

3.1 Introduction

The present chapter describes the Carleton University High-Speed Wind Tunnel (HSWT), the cascade test section, the cascades, and all instrumentations used in the present experiments. The wind tunnel is located in the Minto Case Building. The wind tunnel was originally installed at McMaster University, Hamilton, Ontario. Three graduate students were involved in the recommissioning of the wind tunnel at Carleton University. Each student was responsible of a specific task. Jeffries (1994, 2000) was primarily involved in the overall set-up and testing of the facility. His task included the redesign of the wind tunnel pressure control system. The author was primarily involved in the design and fabrication of cascades of turbine blades and probe traversing mechanisms. The author was also involved with Jeffries in developing the data acquisition system. Islam (1999), was primarily responsible of the design and fabrication of the fast response pressure probes required for flow measurements in the test section.

The following sections provide brief descriptions of the HSWT and its instrumentation. More details can be found in Jeffries (2000) and Islam (1999).

3.2 High Speed Wind Tunnel

The present experimental investigation was conducted in the recently commissioned High Speed Wind Tunnel of Carleton University shown in Figure 3.1 (Jeffries et al., 1997). The wind tunnel is of the blow-down type and is used mainly for transonic turbine research. A detailed description of the wind tunnel set-up is given by Jeffries (1994, 2000).

Using a 100 horsepower two-stage reciprocating compressor, it takes about 15 minutes to pressurize the four tanks of the wind tunnel to 100 psig. The tanks have a total volume of 26 cubic meters. The air is passed through a drier prior to filling the tanks. The dryer is used to lower the dewpoint temperature of the air in the tanks to around -45°C . This delays any condensation problems that may result from the flow expansion through first the pressure control valve and later through the cascade blade row to very high cascade exit Mach numbers. When the tanks are full, the control valve is opened and the air is discharged through the test section. The control system establishes the desired test section blowing pressure within about 10 seconds and maintains it approximately constant during the 15 to 30 seconds run. The air mass flow rate can be as high as 3 kg/sec, and the cascade outlet Mach number can be as high as 1.5. Typical wind tunnel runtimes range from 15 to 45 seconds, depending on the test section blowing pressure and the Mach number level. Because of these short runtimes, a high speed data acquisition system, fast response probes, and high speed traversing mechanisms were used for the wind tunnel data gathering.

The wind tunnel exhausts to the laboratory. It is equipped with an ejector-diffuser system to allow the cascade outlet pressure to be controlled independently of the blowing pressure. This in turn allows the Mach number to be varied independently of the Reynolds number. However, the ejector-diffuser system was not used during the present measurements.

The ejector-diffuser system is mounted on casters, and in all test configurations the ejector-diffuser assembly was aligned with the outlet blade metal angle.

Figure 3.1 The Carleton University Blow Down Wind Tunnel
(Jeffries et al. (1997)).

3.3 Cascade Test Section

The cascade test section is shown schematically in Figure 3.2. The figure shows a plan view of the cascade test section interior. The figure shows a typical arrangement of the cascade test section and the ejector-diffuser assembly for a given incidence value. In addition, the relative position of measuring probe and the downstream traversing mechanism are also shown.

Figure 3.2 A Schematic Diagram of the Cascade Test Section.

The test section was designed to allow variations in incidence. The turntable of the cascade test section is supported on a worm and gear mechanism and can be rotated to vary the values of incidence from about -20° to $+20^\circ$ relative to design. A protractor mounted on the periphery of the turntable allows the incidence angle to be set to within $\pm 0.1^\circ$. A cascade of blades is mounted on the turntable. The height of the internal passage of the test section is fixed at 61 mm, and depending on the blade chord, the blade aspect ratio is then fixed. With a blade chord of 40 mm, and for the two blade geometries used in this study, as many as 11 blades can be mounted on the base plate of the cascade. However, in the present study the blades have a chord length of 40 mm and a total of 7 blades were used giving a total of 8 blade passages.

The two side walls of the test section are mounted to the turntable using a pivot system. As the incidence is changed, the two side walls move parallel to each other allowing the inlet flow passage upstream of the cascade to adjust to variations in incidence. The width of the inlet passage can be varied from a minimum of about 14 cm to a maximum of about 28 cm.

3.4 Wind Tunnel Instrumentation

3.4.1 Test Section Pressure Control System

Figure 3.3 shows a schematic diagram of the test section pressure control system. The control system consists of the following main components:

- a) a 486DX2 desktop computer;
- b) an HP3497A data acquisition and control unit;
- c) an HP44429A dual output digital to analog (D/A) converter;
- d) an HP3437A system voltmeter;

- e) and an Omega 100 psi absolute pressure transducer, which measures the total pressure at the test section inlet.

The computer is used to control the HP3497A data acquisition and control unit (DACU). The communication between the computer and the DACU is achieved using an HPIB interface (Hewlett-Packard Interface Bus). The DACU uses the HP44429A digital-to-analog (D/A) converter to control the position of the 4" ball control valve. Each channel of the HP44429 D/A converter outputs a DC voltage with a programmable range from -10.2375 volts to +10.2375 volts in increments of 2.5 millivolts.

The DACU uses the HP3437A system voltmeter to measure the voltage output from the Omega transducer. The communication between the DACU and the HP3437A system voltmeter is also obtained using an HPIB interface. The Model HP3437A system voltmeter is a microprocessor controlled 3 ½ digit, successive approximation system voltmeter, capable of sampling voltages at rates up to 5700 samples per second.

The Omega transducer is mounted downstream of the control valve on the upper wall of the cascade test section. The Omega transducer is used to measure total pressure at the test section inlet. This total pressure at the test section is the test blowing pressure.

The control algorithm used to set the test section blowing pressure is of the feed-back Proportional-Derivative-Integral (PDI) type. The present digital control system is the recent improvement to the originally analog control system and was designed by Jeffries (2000). A detailed description of the new control system and its implementation is given by Jeffries (2000).

With the use of the control system, a nearly constant test section blowing pressure is maintained during each run. However, the runtime over which the blowing pressure is uniform depends on the cascade test section blowing pressure. Figure 3.4 shows some typical variations of test section blowing pressure with time.

Figure 3.3 Test Section Pressure Control System.

Figure 3.4 Typical Variation of Cascade Test Section Blowing Pressure with Time,
i - ides = 4.5° , HS1A Cascade.

3.4.2 Data Acquisition System

Figure 3.5 shows schematically the wind tunnel data acquisition system. The system consists of the following main components:

- a) a Pentium 90 desktop computer;
- b) an HP3852A data acquisition and control unit (main frame);
- c) an HP44702B 13 bit high speed voltmeter;
- d) a SSS-48C 48 port single Scanivalve system;
- e) an XCQ-062 25 psi absolute pressure Kulite miniature fast response pressure transducer, mounted in the Scanivalve;
- f) a total temperature sensor with type K thermocouple, mounted at the cascade test section inlet;
- g) and an HP44708A 20-Channel Relay Multiplexer unit.

The computer is used to control the HP3852A data acquisition and control unit. The communication between the computer and the main frame unit is achieved using an HP-IB interface. The main frame unit uses the HP44702B 13 bit high speed voltmeter (12 bits and sign bit) to measure the transducer output voltage. The high speed voltmeter is a 3 ½ digit voltmeter which can make DC voltage measurements at rates up to 100,000 readings per second. The number of readings per seconds is dependent on the voltmeter programming mode, the type of measurement performed, and the type of controller used with the voltmeter. The voltmeter uses the successive approximation technique for analog-to-digital conversion. The effective range of the voltmeter for DC voltage measurements is from about $\pm 10 \mu\text{V}$ to $\pm 10.24 \text{ V}$.

Figure 3.5 Data Acquisition System.

The measured data by the voltmeter can be stored in the voltmeter buffer in PACKED format (i.e. 2 bytes per reading). The voltmeter buffer is capable of storing up to 65536 readings. The data stored in voltmeter buffer can be either transferred to the main frame/computer before the buffer fills, or may be overwritten as new data become available.

The 25 psia Kulite pressure transducer is used to measure all pressures in the cascade test section. The transducer is mounted inside the transducer housing of the Scanivalve module. The Scanivalve System used is capable of scan rates of 15 ports per second. The signal output from the transducer is fed to the rear panel of the HP 44702B high speed voltmeter.

The HP44708A 20-Channel Relay Multiplexer with Thermocouple Compensation (MUX/TC) is used to measure the test section inlet total temperature. The HP44708A signal output is measured by the HP44702B voltmeter used by the HP3852A main frame unit. Scanning rates of 450 channels/second can be achieved using this multiplexer unit. The multiplexer consists of a 20-channel terminal module with an isothermal connector block and a relay MUX component module. The isothermal block is used to maintain the reference junction temperature constant. The thermocouple used for the temperature measurements is a type-K thermocouple.

The sampling times and rates of the data acquisition system will be discussed in Chapter 4.

3.5 Cascade Blade Geometries

Two cascades investigated were designated HS1A (baseline blade) and HS1B (modified blade).

The HS1A blade geometry is summarized in Figure 3.6. This blade geometry represents the midspan section of a high pressure turbine of recent Pratt & Whitney Canada engine design.

The HS1B blade represents a derivative of the baseline blade and it differs mainly in the leading edge geometry. As mentioned earlier, the objective of the modifications to the leading edge geometry was to study the tolerance of the modified blade to off-design incidence, especially at positive values. A summary of all geometric parameters and some of the design flow parameters for the two blades is shown in Table 3.1. Figure 3.7 shows the two blades superimposed on each other.

As mentioned earlier in Section 3.2, the height of the blades was fixed by the height of internal passage of the cascade test section. Sieverding (1993) recommended using an aspect ratio (H/C) of about 1.6 or larger for two dimensional cascade testing in the transonic range. In the present study, an aspect ratio of 1.525 was chosen to maximize the blade chord, and thus is at the lower range recommended by Sieverding. This was necessary in order to instrument the blades for loading measurements. Using this blade aspect ratio, each cascade consisted of 7 blades and 8 complete passages.

As shown in Figure 3.8, the blades are mounted on a base plate instrumented with static taps at a downstream distance of 40% true chord (43% axial chord). A total of 48 equally spaced taps were used with only 5 static taps behind each blade passage. This total of 48 taps covers roughly the whole width of the base plate (i.e. distance along the pitchwise direction) and hence allows up to 10 blade passages to be included in the downstream base plate static pressure measurements. However, as mentioned previously, only the 8 center blade passages were used.

Cascade Parameters	
Chord Length, C	40.0 mm
Axial Chord, Cx	36.98 mm
Blade Span, H	61.0 mm
Blade Spacing, s	29.14 mm
Trailing Edge Thickness, t	1.25 mm
Aspect Ratio, H/C	1.525
Leading Metal Angle, β_1	50.5°
Trailing Metal Angle, β_2	59.0°
Leading Edge Wedge Angle, We_{le}	38.0°
Trailing Edge Wedge Angle, We_{te}	6.0°
Design Incidence, i_{des}	-4.5°
Stagger Angle, ζ	25.1°
Throat Opening, o	15.3 mm
Unguided Turning, θ_u	11.5°

Figure 3.6 Summary of HS1A Blade Geometry and Nomenclature.

Parameter	HS1A Blade	HS1B Blade
Chord Length, C	40 mm	40 mm
Axial Chord, C_x	36.98 mm	36.98 mm
Aspect Ratio, H/C	1.525	1.525
Leading Edge Metal Angle, β_1	50.5°	61.0°
Leading Edge Wedge Angle, We_{le}	38.0°	38.0°
Leading Edge Ellipse ratio	1.0	1.0
Trailing Edge Metal Angle, β_2	59.0°	59.0°
Trailing Edge Wedge Angle, We_{te}	6.0°	6.0°
Trailing Edge Thickness, t	1.25 mm	1.25 mm
Stagger Angle, ζ	25.1°	24.7°
Design Incidence, i_{des}	-4.5°	-15.0°
Unguided Turning, θ_u	11.5°	11.5°
Throat Opening, o	15.3 mm	15.3 mm
Maximum Thickness-to-Throat Ratio, (t_m/o)	0.196	0.182
Solidity (s/C)	0.728	0.728
Inlet Mach Number, M_1	0.53	0.53
Exit Mach Number, M_2	1.04	1.04
Inlet Flow Angle, α_1	46.00°	46.00°
Exit Flow Angle, α_2	58.49°	58.49°

Table 3.1 Blades Geometric and Design Flow Parameters.

Figure 3.7 Blade Geometries.

Figure 3.8 Geometry of the Cascade Assembly.

As indicated in the Figure 3.8, the static tap in the middle of the cascade was used as a cascade downstream reference pressure. This reference pressure was used to monitor the test section operating point during a run. The base plate static tap distribution was occasionally used to verify the downstream static pressure measured with the static probe, which is the standard means of obtaining the downstream static pressure. Figure 3.9 shows a sample of these measurements conducted over one pitch for HS1A cascade at an incidence of 4.5° , and an exit Mach number of 0.91.

Figure 3.9 Downstream Base Plate Static Pressure versus Static Probe Pressure, HS1A Cascade, $i_{\text{effective}} = 4.5^\circ$, $M_2 = 0.91$.

Two blades in the middle of each cascade were instrumented with static taps at midspan, one blade for the pressure surface and one blade for the suction surface. For both cascades, the blade with suction surface taps has a base pressure tap at the trailing edge. The static taps were 0.010 inch (0.254 mm) in diameter. Table 3.2 summarizes the number of static taps on each blade for both HS1A, and HS1B cascades. The location of the static taps on both HS1A and HS1B blades are tabulated together with the distributions of surface Mach number and surface static pressure coefficient in Appendices D and E respectively. On each blade the static taps were staggered. This was used to keep a minimum of roughly 50 diameter spacings between adjacent taps, as recommended by Roach & Turner (1988).

	HS1A	HS1B
Pressure Surface	19	17
Suction Surface	22	24
Base Pressure	1	1

Table 3.2 Blade Static Taps Numbers.

The blade coordinates for both cascades were provided by Pratt & Whitney Canada. These coordinates are tabulated in Appendix B. In order to maintain the accuracy in the blade coordinates, the blades were machined to the highest tolerance available with the numerical machining process at the Carleton University Science Technology Centre (STC). Inspections of all blades after polishing showed that the tolerance on all blade coordinates were within ± 0.0005 inches. In addition, the spacing between the blades mounted on the cascade base plate was inspected and was found to be equal to (1.1476 ± 0.0005) inch for both cascades.

3.6 Downstream Traversing Mechanism

The probe traversing mechanism is mounted to the downstream transition side wall as shown in Figure 3.2. The measuring probe is mounted inside a probe holder which attaches to the end of the probe stem. The stem protrudes through the transition wall through an opening that allows spanwise and pitchwise adjustment of the probe. The opening is sealed with a 1/2 inch thick sealing block and a steel plate. The steel plate is used to fasten the sealing block to the transition wall. The sealing block was machined from TEFLON and was split into an upper and a lower section which, when assembled together fit the probe stem cross-sectional area at the transition wall protrusion plane. The sealing block also provides vertical support and smooth guidance of the probe stem. The vertical traversing gear was used to adjust the probe location to mid-span and it was fixed at that position for all measurements. The vertical traversing gear and the probe stem assembly are mounted on a transverse system that allows pitchwise movement of the probe. The traversing mechanism is mounted to the cascade test section using extension arms. These arms put the traversing mechanism clear of the transition section (see Figure 3.2). Moreover, these extension arms allow the independent motion of the ejector-diffuser system for different values of incidence without affecting the position of the probe.

A stepping motor drives the traversing gear for the pitchwise probe motion. A Pentium 90 computer is used to control the motor. This is the same computer used for data acquisition. The traversing mechanism was calibrated for the pitchwise direction (see Jeffries, 2000). From the calibration data, the probe positioning accuracy is estimated at ± 0.05 mm for three blade pitches.

3.7 Pressure Measuring Probes

3.7.1 Overview

As mentioned earlier, typical runtimes using the wind tunnel range from 15 to 30 seconds. These short runtimes present an experimental challenge for obtaining reliable and accurate flow measurements in an acceptably small number of runs. Fast response probes are essential in order to minimize the settling times after a probe has been moved to a new measurement position. Moreover, small probe heads should also be used in order to have a good spatial resolution in the blade wakes, not just important for losses. The choice of probe types and dimensions and their effect on transonic flow field measurements have been the subject of many investigations (e.g. Humm et al., 1994, Naughton et al., 1993, and Sieverding et al., 1994).

The probes used in the present experiment were designed by Islam (1999). Islam designed three miniature fast response probes, namely: a static probe, a total probe, and a three hole probe. The probes were designed specifically for use in the present wind tunnel, and were optimized for both probe head and overall dimensions.

Simplified analysis of the response of pneumatic pressure measuring systems due to a change in pressure (e.g. Benedict, 1960) shows that the time constant of the system is inversely proportional to d^4 (d is the smallest internal diameter in the system), and proportional to the effective length of the tubing and the volume of the system. To minimize the diameters, tube lengths and system volume, the probes were originally designed to have miniature Kulite pressure transducers (Model XCQ-062) housed inside the probe bodies. However, experience showed that these transducers were sensitive to the changes in operating temperature experienced inside the wind tunnel. Subsequently, measurements were made with a single Kulite transducer placed inside the housing of a Scanivalve system mounted outside the test section but as close as possible to the probes. It was found that measurements made using this arrangement had acceptable probe time responses (see Jeffries, 2000).

The following sections give a brief description of the three probe geometries used in the experiment. More details about the probes can be found in Islam (1999). The calibration and accuracy of the probes is discussed in Chapter 4.

3.7.2 Carleton University Three Hole Probe

Figure 3.10 shows the three hole probe used in the present experiment.

Figure 3.10 The Carleton University Three Hole Probe
(reproduced from Islam, 1999).

The probe shown here is the second probe built. The original probe had a chamber to house three Kulite transducers, and was destroyed during testing due to a fatigue failure of the probe stem holder.

As shown in the figure, the probe consists of a 165 mm solid stainless steel rod (main body) and three stainless steel hypodermic tubes with outside diameters of 0.46 mm (0.018 inch). The main body of the probe has an outside diameter (OD) of 7 mm (0.275 inch), and is tapered down to 1.78 mm (0.07 inch) OD near the probe head. The three hypodermic tubes are soldered together and protrude 5 mm outside the main body. The two outer tubes are machined to form a 30° semi-vertex angle. Inside the main body, these hypodermic tubes are connected to vinyl tubing.

The probe tip has a width of 1.37 mm (0.054 inch) which is roughly 4.7% of the blade pitch, and a thickness of 0.46 mm (0.018 inch). This 4.7 % probe width to pitch ratio is less than the value of 6% recommended for transonic testing (Kiock et al., 1986, Mee et al., 1992, Sieverding, et al., 1994) to obtain reasonably accurate data. Comparable probe head dimensions (4.4% probe width to pitch ratio) were also used by Sieverding et al. (1994) for downstream flow measurements at exit Mach numbers around 1.05.

The probe was built with a square block at its end which is used to align the probe inside the probe holder.

3.7.3 Carleton University Static Pressure Probe

Figure 3.11 shows the static pressure probe. The probe was built using concentric stainless steel hypodermic tubes to form the main body of the probe. The probe tip outer diameter is 1.02 mm (0.04 inch) which is 3.5 % of the blade pitch. This probe tip to pitch ratio (3.5 %) is below the value of 6% recommended for midspan traverses in transonic flows (Kiock et al., 1986, Mee et al., 1992). The probe tip has a cone angle of 15°. From the tip of the probe, two static holes with 0.3 mm in diameter and separated by 180° are used to measure the static pressure. The distance from the probe tip to the static holes is roughly 12 times the probe tip diameter, which exceeds the recommended minimum value of 10 for cone angles less than 30° (e.g. Ower, 1977). The hypodermic tube with static holes is supported coaxially by a larger tube of outside diameter 1.7 mm

(0.066 inch) with its leading edge 30 mm downstream of the static holes. This is nearly 30 times the probe tip diameter which is above the minimum recommended value of 18. Literature shows that probes which satisfy the above minimum distance requirements for the location of the static holes can be used for flow with exit Mach numbers up to 1.6.

Dimensions are in mm unless otherwise specified

Figure 3.11 The Carleton University Static Pressure Probe
(reproduced from Islam, 1999).

As shown in the figure, the front end assembly of the probe is connected to a larger tube of diameter 7 mm (0.275 inch) hollow on the inside. This chamber allows access to the vinyl tubing connected at the end of the bulged stainless steel tube. The bulged tube is soldered to the tube with the static holes.

The probe was again manufactured with a square block at its end which is used to align the probe inside the probe holder.

3.7.4 Carleton University Total Pressure Probe

Figure 3.12 shows the total pressure probe.

Dimensions are in mm unless otherwise specified

Figure 3.12 The Carleton University Total Pressure Probe
(reproduced from Islam, 1999).

The construction of the total pressure probe is similar to the static pressure probe. The probe head diameter is also 1.02 mm (0.04 inch) in diameter which is 3.5% of the blade pitch. The total pressure probe was used occasionally for downstream cascade traverses. These total pressure traverses were used to cross check the total pressure measurements obtained with the three hole probe.

The total pressure probe was also manufactured with a square block at its end which is used to align it inside the probe holder.

3.8 Mechanism for Probe Angle Alignment

Figure 3.13 shows the probe alignment mechanism mounted in its position downstream of the cascade. The mechanism is used to check the probe orientation with respect to the exit blade angle. As shown in the figure, a precision plate is mounted against the trailing edges of the cascade blades. The plate is used as a reference plane to orient a precision traversing mechanism which moves in the same direction as the exit blade angle. Two tapered pins are used to secure in place the traversing mechanism to the set angle. These tapered pins are positioned using high precision machining to obtain an angle setting of $\pm 0.01^\circ$ relative to the desired value.

A 30X Pocket Microscope mounted to the traversing mechanism, as shown in the figure, is used to traverse along the probe tip body and measure its location with a resolution of ± 0.25 mm (0.01 inch). Initially, before traversing along the probe, the cross hatch on the microscope lens is positioned along one of the side edges of the probe tip body with the line having the graduations perpendicular to the probe. At the end of the traverse (see Figure 3.14), the cross hatch may be offset from the side edge of the probe. The offset distance is given by reading the graduations on the microscope lens.

Figure 3.13 Probe Alignment Mechanism.

Referring to Figure 3.14, the probe misalignment angle can be calculated using the following equation:

$$\theta = \sin^{-1}\left(\frac{y}{x}\right) \quad (3.1)$$

where (y) is the reading from the microscope lens, and x is the distance travelled along the blade exit metal angle. The corrected outlet flow angle then can be calculated using the following equation:

$$\alpha_{2c} = \alpha_{2m} + \theta \quad (3.2)$$

Figure 3.14 Probe Misalignment Angle (Not to Scale).

3.9 Transducer Calibration Apparatus

Figure 3.15 shows the transducer calibration apparatus used to calibrate the Kulite miniature transducer.

Figure 3.15 Transducer Calibration Apparatus.

The Kulite transducer is calibrated daily against a Mercury manometer. Two valves connected to a settling chamber are used to regulate the pressure inside the chamber. One of these valves is connected to a vacuum pump, and the other valve is connected to a compressed air line. The Kulite transducer is mounted inside the transducer housing of the Scanivalve system. The signal from the Kulite transducer is recorded using the data acquisition system described earlier. The manometer range over which the transducer is calibrated is from about -165.1 mm (6.5 inch) to about +571.5 mm (22.5 inch). A total of 22 data points were collected during each calibration.

4. Experimental Procedures

4.1 Introduction

The present chapter describes the procedures followed to collect and process the data. The work involved in setting up these procedures was divided between the author and Jeffries (2000). For this reason, there will be some references to Jeffries (2000) for additional information.

4.2 Three Hole Probe Calibration and Accuracy

4.2.1 Probe Calibration and Accuracy

All loss calculations were based primarily on the three hole probe data collected downstream of the cascade. The three hole probe was calibrated by Jeffries (2000) for yaw angles (i.e. angles in a plane defined by the 3 tubes of the probe) ranging from -10° to $+10^\circ$ in increments of $\pm 1^\circ$. The pitch angle of the probe was fixed to zero degrees, and the exit Mach number was varied from 0.5 to 1.0. The probe was calibrated inside the cascade test section using a calibration insert designed by Corriveau (1999). The calibration insert is shown schematically in Figure 4.1 with the probe mounted in its calibration position. The mechanism used to yaw the probe is not shown in the figure. The calibration insert was designed as a straight rectangular duct with streamlined intake to minimize intake losses. More details on the calibration apparatus can be found in Corriveau (1999).

Figure 4.1 A Schematic Diagram of the Calibration Insert (Top View).

The calibration procedure used by Jeffries (2000) follows the method of Lewis (1966). The calibration data were recorded using the data acquisition system described in Section 3.4.2. The calibration data were only used to extract the exit flow angle and the total pressure from the three hole probe data. For loss calculations, the static pressures obtained with the static pressure probe were combined with total pressures and exit flow angles obtained from the three hole probe.

Since the three hole probe calibration was limited to exit Mach numbers less than one, a normal-shock correction for the total pressure was applied to the three hole probe total pressures for Mach number greater than one. The estimated uncertainty on the calibrated exit flow angle was less than $\pm 0.5^\circ$. More details on the calibration procedure can be found in Jeffries (2000).

4.2.2 Probe Blockage Effects

Probe blockage effects are a serious problem in transonic flow measurements. Literature shows that when the blockage area of the probe is small compared to the flow area, the flow field is not significantly affected by the presence of the probe. As the size of the probe is increased relative to the flow area, blockage effects become important and erroneous results from the probe are then obtained. This effect is shown to increase rapidly with Mach number, and is most pronounced at the sonic conditions.

As reported by Sieverding (1993), one of the most comprehensive studies on probe blockage effects is the one due to Wyler (1975). Wyler quantified the blockage effects for a probe mounted in a circular jet as a change in both the Mach number and total pressure. These changes in the flow quantities are expressed as a function of the ratio of the probe head diameter (d) to the jet flow diameter (D). The graphical representation of Wyler's data, as reported by Sieverding, shows that for a diameter ratio (d/D) of 0.05 (i.e. an area blockage ratio of 0.0025), the probe blockage effects result in a maximum change of less than 2.0% of the Mach number itself (i.e. $|\delta M/M| < 2.0\%$). In the present case, the jet flow diameter was taken as the hydraulic diameter of one blade passage area at outlet which is about 48 mm. All probes used in the present measurements have a maximum probe blockage area less than 0.0009 (i.e. a diameter ratio less than 0.03) which implies a percentage change in Mach number of less than 1.0%. Moreover, Sieverding et al. (1994) suggested that if the wall static pressure in the downstream measuring plane is not significantly different in the presence of the probe compared to that in the absence of the probe then blockage effects are negligible. In the present measurements, the change in the base plate static pressure in the measuring plane due to the presence of the probe was less than 1% of the local dynamic pressure for all operating conditions.

Based on the above findings, probe blockage effects in the present experimental study have been neglected.

4.3 Transducer Calibration

The Kulite miniature transducer used for the pressure measurements was calibrated daily during testing.

The pressure range over which the transducer was calibrated was from 0.5 bars to roughly 2.75 bars. During calibration, the pressure was applied gradually in both directions in order to detect any hysteresis effects. Figure 4.2 shows calibration data collected over a period of two months. As shown, the data exhibit a good linear characteristic.

Figure 4.2 Kulite Transducer Calibration Curve (Model XCQ-62).

The uncertainty in the pressure measurements associated with the transducer calibration is estimated as less than 0.003 bars. This uncertainty was based on variations in pressure data obtained using transducer calibrations from different days.

4.4 Data Taking Procedure

To evaluate the losses, traverses of the wake downstream of the blade row were conducted. The measurement plane was located at 40% true chord (43% axial chord) downstream of the cascade blade row. The traverses were conducted using a three hole probe and a static pressure probe. Both probes were described in Section 3.7. The static pressure probe was used to measure the static pressure, and the three hole probe was used to measure the total pressure and the flow angle. In addition to the traverses downstream, a total pressure probe located at the inlet to the test section (i.e far upstream of the cascade) was used to monitor the blowing pressure at all operating conditions. This total pressure was used as a reference total pressure (P_{01r}). In addition, a static tap was located upstream of the cascade blade row at a distance of one blade true chord length. This static pressure was used as a reference inlet static pressure (P_{1r}). Both reference pressures were used to monitor the cascade inlet isentropic Mach number during a run. Before conducting any downstream traverses, a Pitot-static probe was placed at a distance of one true chord upstream of the cascade blade row. The measurements from the Pitot-static were referenced to the inlet total pressure (P_{01r}) and the inlet static pressure (P_{1r}). The Pitot-static probe was removed from the upstream location when downstream traverses were made. The inlet Mach number was monitored during a run based on the relationship between the reference pressure measurements (i.e. P_{01r} and P_{1r}) and the ones obtained using the Pitot-static probe.

As mentioned earlier, typical run times for the wind tunnel range from 15 to 30 seconds. For this reason, a downstream traverse over one pitch is accomplished in four runs. Each run took roughly 15 seconds. Figure 4.3 shows the downstream measuring plane. In each run there are 10 measuring points equally spaced. The spatial displacement for passes 2 and 3, which span the blade wake, are half the one used for passes 1 and 4. This non-uniform spacing was chosen in order to

have a better definition of the downstream wake. For each test flow condition the blade wake was first located before collecting any data. The wake was located by trial and error during preliminary runs. This procedure also provided the basis for choosing the starting location of the downstream traverse. A ruler placed along the traverse direction on the side of the traversing mechanism was used to monitor the starting position for all subsequent wake traverses. The data collected at each traversing point using the static probe was assigned to the corresponding measuring point for the three hole probe.

Figure 4.3 Cascade Blade Row Measurement Locations.

A downstream static tap located at the measuring plane on the cascade base plate, as indicated in Figure 4.3, was used to monitor the pressure ratio across the cascade blade row during each test run. This is the reference downstream static pressure (P_{2r}). The pressure ratio across the cascade was used to match the operating condition between the four runs.

Once the wake measurements were completed, blade loading as well as trailing edge base pressure data were collected. When taking loading measurements, all reference pressures (i.e. the two upstream, and the one downstream) were also collected in order to match them with the operating conditions for the wake traverses.

4.5 Cascade Test Conditions

Two cascades were investigated; HS1A and HS1B. The cascades were tested for exit Mach numbers ranging from 0.5 to 1.2 (i.e. typically about 7 Mach number points covered in this range) and incidence values ranging from -10° to $+14.5^\circ$. Table 4.1 summarizes the test matrix for the present experiment.

	HS1A					HS1B			
Incidence	-10.0°	0.0°	4.5°	10.0°	14.5°	0.0°	4.5°	10.0°	14.5°
Mach Number	0.5 $\rightarrow$ 1.2					0.5 $\rightarrow$ 1.2			
Reynolds Number	450,000 $\rightarrow$ 10^6					450,000 $\rightarrow$ 10^6			
Turbulence Intensity	4%					4%			

Table 4.1 Cascades Test Matrix.

As mentioned in Section 3.2, the wind tunnel is equipped with an ejector-diffuser system that allows independent control of the exit Mach number and the Reynolds number. This ejector was not used in the present experiment. This meant that the test section flow discharged to a constant outlet pressure (back pressure). The Reynolds number was based on the outlet cascade flow conditions and the true chord. Figure 4.4 shows a typical variation of the Reynolds number with Mach number. The variation in Reynolds number is due to the variation in total temperature at the inlet of the test section during a run. Therefore Re_{2max} and Re_{2min} in Figure 4.4 correspond to the Reynolds numbers for the minimum and maximum temperatures respectively.

The turbulence intensity upstream of the cascade test section was measured by Corriveau (1999) and it was found to be about 4% (Tu). The turbulent length scale was found to be about 1.7 cm.

The temperature in the storage tanks, and thus the approximate total temperature T_{01} in the test section, typically starts at 13° to 20° C. The total temperature drops as the air in the tanks expands during a run. Moreover, the total temperature drop is an effect of the heat transfer in the piping system between the tanks and the test section. Therefore, the temperature was measured upstream of the cascade during each run. Figures 4.5(a) and (b) show typical temperature drop variations during a run with exit Mach number. The figures are for the incidence values of 0.0° and +4.5° and are typical to the temperature drops seen during hot days (summer) and cold days (winter). It should be mentioned that, for the loss calculations, the total temperature was assumed equal at the inlet and outlet planes of the cascade, even though a small amount of heat transfer would be occurring during a run. Estimates of the total temperature variations which could occur across the cascade and the corresponding changes in the loss coefficients for different operating points are presented in Section 4.10.3.3

Figure 4.4 Reynolds Number Variation with Exit Mach Number, HS1A Cascade.

a) $i - i_{des} = 0.0^\circ$, HS1A cascade

b) $i - i_{des} = +4.5^\circ$, HS1A cascade

Figure 4.5 Typical Total Temperature Drops during a Run at Different Operating Conditions for a) Hot Day, and b) Cool Day.

4.6 Sampling Times and Rates

As a first step in commissioning the high speed wind tunnel, Jeffries et al. (1997) conducted an experimental investigation of the effects of both the sampling times and the sampling rates on the accuracy of the mean values of the flow quantities. The investigation was for both total and static pressures. In the investigation both low and high frequency contents of the pressure signal were examined. They found that the wind tunnel test section flow contained fluctuations in both pressure and velocity with a frequency of the order of 0.50 Hz. These fluctuations were traced to the wind tunnel control system. In addition to these low frequencies, the pressure signal contained high frequency fluctuations due to the turbulence of the flow. For all flow ranges, it was concluded that all pressure data must be sampled for at least 0.1 second at a minimum frequency of 2000 Hz in order to obtain good mean values. In addition, to eliminate the low frequency content, they recommended normalizing the pressure values sample by sample by the test section inlet total reference pressure (P_{01r}). The test section reference pressure is used to monitor the test section blowing pressure. The integration time of 0.1 seconds gave errors in static and total pressures less than 1% relative to the mean values obtained over a long period of integration. These error magnitudes were observed for all the flow conditions tested. In the present experimental investigation the data sampling procedure recommended by Jeffries et al. (1997) was adopted. More details on the sampling times and sampling rates can be found in Jeffries et al. (1997) and in Jeffries (2000).

4.7 Cascade Flow Conditions

4.7.1 Inlet Flow Conditions

The cascade inlet flow uniformity was examined by Jeffries (2000) as part of the commissioning of the wind tunnel. Jeffries traversed upstream of the cascade in both the pitchwise and the spanwise directions. Referring to Figure 4.3, the upstream traverse plane was located at a distance of one true chord. At this upstream plane three pitchwise locations were used for the

spanwise traverses: one upstream of blade 4, one upstream of blade 2, and one upstream of blade 6. Thus, these traverses covered roughly four pitchwise blade spacings. Jeffries found that the three spanwise traverses showed essentially identical velocity profiles. The core of the flow in the spanwise direction showed a uniform velocity distribution over more than 70% of the test section height (blade span). More details on these measurements can be found in Jeffries (2000).

4.7.2 Downstream Flow Periodicity

The first cascade tested in the wind tunnel was HS1A. Extensive measurements were performed for the HS1A cascade to document the downstream flow periodicity.

Initially, HS1A cascade was set at the positive incidence of 4.5° . For this configuration, wake traverses were performed for three blades: namely blade number 3, 4, and 5 (see Figure 4.3). The wake traverses for this configuration were examined for exit Mach numbers ranging from 0.5 to 1.2.

Figure 4.6 shows downstream total pressure traverses for HS1A cascade at the 4.5° incidence and for three exit Mach numbers. As shown, the wake profiles were very similar, indicating good periodicity in the wake flow. The wake data are presented as a total pressure coefficient defined as follows:

$$C_{po} = \frac{P_{02} - P_{01}}{q_2} \quad (4.1)$$

Following the results obtained at the 4.5° incidence, it was decided to reduce the number wake traverses to two: behind blades 3 and 4. This was the procedure used to check periodicity for the HS1A cascade for most of the incidence values. As an example, Figure 4.7 shows the downstream flow periodicity for HS1A at the design incidence and for three Mach numbers. From Figures 4.6 and 4.7, it appears that the periodicity is somewhat poorer at Mach numbers above 1.1.

The uncertainty in the downstream total pressure measurements will be discussed later in Section 4.10.3.1.

Figure 4.6 Downstream Flow Periodicity at $i - i_{des} = 4.5^\circ$, HS1A Cascade.

Figure 4.7 Downstream Flow Periodicity at design incidence, HS1A Cascade.

4.7.3 Downstream Static Pressure

Figures 4.8 and 4.9 show static pressure traverses downstream of the cascade at incidence values of 4.5° and 0° and three Mach numbers. For the same reasons mentioned in Section 4.7.2, three wakes are shown for the 4.5° versus two wakes for the design incidence.

The figures show that the downstream static pressure is quite uniform at subsonic Mach numbers. At supersonic Mach numbers, there is a gradual decrease in pressure followed by a sudden rise in pressure. The gradual decrease in static pressure is associated with the flow acceleration adjacent to the blade suction surface. The observed rise in pressure can be attributed to the crossing of a shock wave by the probe. This shock is likely to be the shock reflected from the suction surface of the adjacent blade to the right. It is noticeable that there is similar pressure rise across the shock at the Mach numbers of 1.03 and 1.12 (e.g. see Figure 4.8) which suggests that the shock is more oblique in the case of the higher Mach number.

As shown in Figures 4.8 and 4.9, the downstream static pressure distribution is more distorted at the higher values of Mach numbers (i.e. above $M_2 = 1.1$). Nevertheless, like the total pressure traverses, the downstream static pressure traverses indicate good downstream flow periodicity, particularly at Mach numbers less than 1.1.

Figure 4.8 Downstream Static Pressure Traverses at $i - i_{des} = 4.5^\circ$, HS1A Cascade.

Figure 4.9 Downstream Static Pressure Traverses at design incidence, HS1A Cascade.

4.7.4 Exit Flow Angle

Figures 4.10 and 4.11 show the downstream exit flow angle distributions at incidence values of 4.5° and 0° and three Mach numbers. The figures show some downstream angle variation, especially in the blade wake. This angle variation in the blade wake is expected since the wake is filling through the mixing process.

Across the wake, the probe tip is exposed to large velocity gradients which result in differences in the pressure sensed by the side holes of the probe. This effect may be seen in flow underturning from the pressure side and flow overturning from the suction surface, and may be erroneously interpreted as flow angularity. Although there is some uncertainty in the flow angle inferred from the three hole probe in the wake region, the flow angles shown in Figures 4.10 and 4.11 confirm that there is good downstream flow periodicity. However, as mentioned in Sections 4.7.2 and 4.7.3, there is poorer flow periodicity at higher Mach numbers (i.e. M_2 greater than 1.1). The exit flow angles inferred from the three hole probe at the Mach number values greater than unity should be treated with caution since the three hole probe was only calibrated for Mach numbers up to 1.0.

As mentioned in Section 3.8, a probe alignment mechanism was used to measure the probe alignment with the exit blade angle. For both cascades, HS1A and HS1B, the probe misalignment angle (Eqn. (3.1), Section 3.8) was estimated to be within $\pm 0.4^\circ$. The estimated uncertainty on the calibrated exit flow angle was less than $\pm 0.5^\circ$. Thus the combined uncertainty on the measured exit flow angle was estimated to be less than $\pm 0.65^\circ$.

Figure 4.10 Downstream Exit Flow Angle at $i - i_{des} = 4.5^\circ$, HS1A Cascade.

Figure 4.11 Downstream Exit Flow Angle at design incidence, HS1A Cascade.

4.8 Pressure Measurements

4.8.1 Total Pressure

Downstream total pressure measurements were obtained using the three hole probe, and for comparison purposes additional traverses were also conducted using the total pressure probe. As an example, Figure 4.12 shows a comparison between the measured total pressures obtained using the two probes for an exit Mach number of 0.81. As shown, the agreement is satisfactory.

Figure 4.12 Downstream 3Hole and Total Pressure Probes Traverses.

For outlet Mach numbers greater than unity, a bow shock was assumed to exist in front of the probe mouth. Therefore, the total pressure was corrected to obtain the pressure upstream of the shock using the following equation, obtained from normal-shock relationships:

$$\frac{P_{02c}}{P_{02}} = \left(\frac{\frac{\gamma+1}{2} M_2^2}{1 + \frac{\gamma-1}{2} M_2^2} \right)^{\frac{\gamma}{\gamma-1}} \left(\frac{\gamma+1}{2\gamma M_2^2 - (\gamma-1)} \right)^{\frac{1}{\gamma-1}} \quad (4.2)$$

Figure 4.13 shows the nomenclature adopted in the above equation. The total pressure correction was performed for each traversing point along the pitch wise direction. The corrected total pressure value is the one used in the loss calculations.

Figure 4.13 Flow in the Vicinity of the Total Probe Mouth.

4.8.2 Blade Surface Pressure Measurements

For each cascade, blade static pressure measurements were performed on the two instrumented blades described earlier in Section 3.5. The surface pressure measurements were referenced to the upstream static pressure P_{1r} and non-dimensionalized by the upstream dynamic pressure q_{1r} . Therefore the results are presented as a surface static pressure coefficient given by the following equation:

$$C_{ps} = \frac{(P_s)_x - P_{1r}}{q_{1r}} \quad (4.3)$$

where $(P_s)_x$ is the pressure at the static tap x . The results are also expressed as surface isentropic Mach number distributions. These are given by the following equation:

$$(M_{is})_x = \left(\left(\frac{2}{\gamma - 1} \right) \left(\left(\frac{P_{01r}}{(P_s)_x} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right) \right)^{\frac{1}{2}} \quad (4.4)$$

where P_{01r} is the upstream reference total pressure. The blade surface measurements were performed at all flow conditions for both HS1A and HS1B blades. The surface static pressure coefficient and the surface Mach number distributions for both HS1A and HS1B cascades are tabulated in appendices D and E respectively. Repeatability measurements of blade loading for both HS1A and HS1B cascades showed that the uncertainty in the surface Mach numbers is about ± 0.01 . Figure 4.14 shows a typical blade loading for HS1A at the design Mach number and design incidence. Figure 4.14 also shows a comparison between the measured blade loading and the blade loading predicted using the Dawes code (1988). As shown in the figure the agreement is quite good.

Figure 4.14 HS1A Blade Loading at the Design Flow Conditions.

The Dawes code predictions were used occasionally at the early stage of testing mainly to troubleshoot some initial peculiarities in the cascade experimental results. The comparison between the experimental and numerical results were limited to blade loading.

4.8.3 Base Pressure Measurements

Base pressure measurements were performed using the suction surface instrumented blades for both HS1A and HS1B. As described earlier in Section 3.5, each of these blades was instrumented with a static tap at the base of the trailing edge. The base pressure (P_b) results were referenced to the mixed-out downstream static pressure P_{s2} and non-dimensionalized by the downstream mixed-out dynamic pressure q_2 , giving the base pressure coefficient as follows:

$$C_b = \frac{P_b - P_{s2}}{q_2} \quad (4.5)$$

In addition, the base pressure measurements were normalized by the upstream stagnation pressure as (P_b/P_{01}). These non-dimensional base pressure values were plotted against Sieverding's correlation described earlier in Section 2.3.4. Tabulated values of the base pressure coefficients for both HS1A and HS1B cascades are in Appendices D and E respectively. Repeatability measurements on the base pressure, for both HS1A and HS1B cascades, showed errors less than $\pm 2\%$ of the downstream mixed-out dynamic pressure.

The relationship between the base pressure to the mixed-out losses is discussed later in Sections 5.2.3 and 5.3.4 for HS1A and HS1B respectively.

4.9 Axial Velocity Density Ratio (AVDR)

As mentioned earlier in Section 2.3.2.1, the value of the axial velocity density ratio (AVDR) is an approximate indication of the two dimensionality of the flow in the test section. Some

cascade facilities use endwall suction or blowoff to control the thickness of the endwall boundary layers and therefore the AVDR. In the present test facility, there was no control over the value of the AVDR.

Using the nomenclature shown in Figure 4.15, the AVDR can be calculated using the following equation:

$$AVDR = \frac{\int_0^1 (\rho_2 C_{(ax)_2})_{MS} d\left(\frac{y}{s}\right)}{\int_0^1 (\rho_1 C_{(ax)_1})_{MS} d\left(\frac{y}{s}\right)} \quad (4.6)$$

where $()_{MS}$ refers to the mid-span values of the flow quantities. $C_{(ax)1,2}$ are the upstream and downstream axial flow velocities respectively. ρ_1 and ρ_2 are the air densities upstream and downstream of the cascade respectively. The distance along the pitchwise direction is y , and s is the blade spacing.

Using the isentropic relations and assuming air to behave as an ideal gas, the expression for the AVDR given by equation (4.6) is rearranged to give the following expression:

$$AVDR = \frac{\int_0^1 \left(\frac{P_2}{P_{01}} M_2 \sqrt{\frac{T_{01}}{T_{02}}} (\cos \alpha_2) \left(1 + \frac{\gamma-1}{2} M_2^2 \right)^{\frac{1}{2}} \right)_y d\left(\frac{y}{s}\right)}{\int_0^1 M_1 (\cos \alpha_1) \left(1 + \frac{\gamma-1}{2} M_1^2 \right)^{-\frac{(1+\gamma)}{2(\gamma-1)}} d\left(\frac{y}{s}\right)} \quad (4.7)$$

The subscript (y) refers to location (y) along the pitchwise direction. Again, equation (4.7) is evaluated at the mid-span location. Equation (4.7) represents a convenient way to calculate the AVDR since it is expressed as a function of the measured flow quantities.

Figure 4.15 Axial Velocity Density Ratio Nomenclature.

Equation (4.7) shows that the AVDR is a function of the ratio of upstream and downstream total temperatures. In the present experimental study, no total temperature measurements were conducted downstream of the blade row. In an investigation on the total temperature field downstream of a stationary turbine cascade, Carscallen and Oosthuizen (1989) showed that for an adiabatic flow there is a redistribution of the total temperature. Their measurements showed a deficit in total temperature in the wake region, but an increase in total temperature at the boundaries of the wake. This behaviour was a function of the downstream isentropic Mach number. However, they concluded that the area averaged downstream total temperature is the same as the upstream one.

The temperature in the storage tanks, and thus the total temperature of the test section flow, drops as the air in the tanks expands during a run. Thus there may be some heat transfer

occurring between the cascade blades and the test section flow. This heat transfer may result in changes to the downstream temperature. To reduce this heat transfer, several runs of the wind tunnel are made before a series of data runs are begun. These pre-runs are used to cool the cascade blades and the walls of the test section. Consequently, it seemed a reasonable assumption to neglect the effect of the downstream total temperature on the AVDR. Following this simplification, the resulting equation used for the AVDR is as follows:

$$AVDR = \frac{\int_0^1 \left(\frac{P_2}{P_{01}} M_2 (\cos \alpha_2) \left(1 + \frac{\gamma-1}{2} M_2^2 \right)^{\frac{1}{2}} \right)_y d\left(\frac{y}{s}\right)}{\int_0^1 M_1 (\cos \alpha_1) \left(1 + \frac{\gamma-1}{2} M_1^2 \right)^{-\frac{(1+\gamma)}{2(\gamma-1)}} d\left(\frac{y}{s}\right)} \quad (4.8)$$

4.10 Mixed-Out Mass Averaged Loss Coefficients

4.10.1 Calculation Procedure

The following is an outline of the procedure used to calculate the mixed-out values of the downstream flow quantities as well as the loss coefficients. For completeness, the mid-span losses are expressed in terms of all the loss coefficients described earlier in Section 2.5.2.1. These loss coefficients, Y_t , ω_t , and ξ , are given by the following equations:

$$Y_t = \frac{P_{01} - P_{02}}{q_2}, \quad (4.9a)$$

and,

$$\omega_t = \frac{P_{01} - P_{02}}{P_{02} - P_{s2}}, \quad (4.9b)$$

$$\xi = \frac{1 - \left(\frac{P_{s2}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}}}{1 - \left(\frac{P_{s2}}{P_{02}} \right)^{\frac{\gamma-1}{\gamma}}} - 1, \quad (4.9c)$$

where:

- P_{01} : total pressure upstream of the cascade blade row,
- P_{02} : mixed-out value of the downstream total pressure,
- P_{s2} : mixed-out value of the downstream static pressure, and
- q_2 : mixed-out value of the downstream dynamic pressure.

Tabulated values of these loss coefficients for both HS1A and HS1B cascades are given in Appendices D and E respectively. However, only the loss coefficient Y_l is used in the discussion of the blade losses for both HS1A and HS1B which is presented in Chapter 5.

4.10.2 Calculation Procedure

Figure 4.16 shows a schematic diagram of the different steps used to process the measured data and calculate the mixed-out losses.

As a first step in the calculations, the transducer calibrations are applied to the static pressure traverses. For each static pressure traverse, the static pressure values are normalized with the corresponding upstream reference total pressure (P_{01r}). For a given operating point, the static pressure traverses are combined with the three hole probe data for the same operating point. Each operating point is defined by measurements of the inlet pressures, P_{01r} and P_{1r} , and the reference

downstream pressure measurement, P_{2r} . The static probe data and the three hole probe data are combined in a post processing program called PROCESS.

In the PROCESS program, the three hole probe calibration as well as the transducer calibration are applied to the three hole probe data. In the analysis, all pressure data are normalized with the corresponding upstream reference total pressure (P_{01r}). From the three hole probe data, downstream values of the total pressure, static pressure and flow angle are extracted. However, the static pressure data extracted from the three hole probe were only used for comparison purposes with the static pressure probe data, and were not used in the analysis. Using the downstream static pressure from the static probe and the total pressure from the three hole probe, the outlet downstream Mach number and dynamic pressure variations with the pitchwise distance are calculated. The output file from the PROCESS program includes tabulated values of the downstream total pressure, flow angle, Mach number, and static pressure.

To calculate the mixed-out values of the downstream flow quantities, the output file from the PROCESS program is used as input to the MIXLOSS program. As mentioned earlier in Section 2.5.2.1, the calculation procedure followed in the MIXLOSS program is based on the Amecke and Šafařík (1995) method to calculate the mixed out values. This method is outlined in Appendix A. Figure 4.17 shows a sample result of the mixed-out flow quantities as compared to the ones obtained from the downstream traverse for HS1A at an exit Mach number of 1.03 and incidence value of 4.5° .

Finally, the loss coefficients are evaluated based on the mixed-out downstream flow quantities.

Figure 4.16 Data Processing and Calculation Procedure.

Figure 4.17 A Comparison Between the Mixed-Out and the Pitchwise Variation of the Flow Quantities, HS1A Cascade, $M_2 = 1.03$, $i - i_{des} = 4.5^\circ$.

4.10.3 Measurement Uncertainties

4.10.3.1 Overall Measurement Uncertainties

The estimated uncertainties for the mixed-out total pressure loss coefficients are based on observed variations from blade to blade as well as repeatability runs made at a given operating point.

Figure 4.18 shows typical variations in blade to blade loss measurements for HS1A cascade. The blade to blade measurements were conducted mainly for the baseline cascade HS1A. However, this variation in blade to blade loss measurements is believed to be representative for both cascades.

Figure 4.19 shows the results of repeatability runs made over the period of five months. These repeatability runs were made for the HS1A cascade at an incidence value of $+4.5^\circ$. The HS1A cascade was first tested at the incidence setting of $+4.5^\circ$. For this reason, extensive data were collected at this incidence, including repeatability measurements. Later, an extra set of data was also collected at this incidence. This last set is included in Figure 4.19.

From both blade to blade and repeatability measurements, the estimated uncertainty for the mixed-out total pressure loss coefficients is ± 0.005 for exit Mach numbers less than 0.5, ± 0.003 for Mach numbers from 0.5 to 0.85, ± 0.004 for Mach numbers between 0.85 and 1.1, and ± 0.005 for Mach numbers greater than 1.1.

4.10.3.2 Reynolds Number effects

As mentioned, the total temperature in the test section, T_{01} , drops during a run. Typical temperature drops range from roughly 6°C to 20°C . Table 4.2 summarizes the temperature changes which were measured for several typical operating points. Also shown are the corresponding average Reynolds numbers and the variation in Reynolds number which corresponds to the variation in total temperature. Finally, the estimated variations in the profile loss coefficients due to the Reynolds

Figure 4.18 Blade to Blade Loss Coefficient Measurements,
 $i - i_{des} = 4.5^\circ$.

Figure 4.19 Repeatability Loss Coefficient Measurements,
 $i - i_{des} = 4.5^\circ$.

number variations are also given. These estimates are based on data for the variation in loss coefficient with Reynolds number which are presented in Section 5.2.4.

M_2	ΔT_{01} (°C)	Average Re	$\pm \Delta Re$	$Y_t \pm \Delta Y_t$
0.55	7	500,000	5,500	0.0729 ± 0.0014
0.71	12	660,000	13,000	0.0666 ± 0.0002
1.14	19	1,040,000	97,000	0.0997 ± 0.0011

Table 4.2 Typical Variations in Total Temperature and Reynolds Number
During a Run at Design Incidence, HS1A Cascade.

As shown in Table 4.2, the variation in loss coefficients due to the Reynolds number variations is less than the estimated uncertainty in the loss coefficients due to all other sources.

4.10.3.3 Effects of Changes in Total Temperature across the Cascade

For the evaluation of the mid-span loss coefficients and the axial velocity density ratios, the total temperature was assumed equal at the inlet and outlet planes of the cascade, even though a small amount of heat transfer may be occurring during a run. Several pre-runs were used to cool the cascade blades and the walls of the test section before data runs are begun and thus reduce the heat transfer during a run. These pre-runs are usually taken at exit Mach numbers around the design value (i.e. $M_2 = 1.05$), since at this Mach number the total temperature changes at the inlet of the test section are high and considerable cooling of the blades and the test section walls will be occurring under these conditions.

As shown in Section 4.5, the changes in total temperature at the inlet of the test section ranged from roughly 6 °C at exit Mach numbers around 0.5 to about 20 °C at exit Mach numbers around 1.14. The changes in static temperature at the inlet of the test section also ranged from 6 °C and exit Mach numbers around 0.5 to about 20 °C at Mach numbers around 1.14. Initially, before the runs are started, the test section walls and the blades are assumed to be at room temperature, typically at +20 °C. This room temperature represents an upper limit for the test section walls and blade temperatures. At exit Mach numbers around 1.14, the static temperature at the cascade exit plane may be as low as -60 °C. This static temperature represents a lower limit for the test section walls and blade surfaces temperatures. Thus, the test section wall and blade surfaces temperatures are assumed to vary from room temperature (i.e. +20 °C) to a lower value of -60 °C. The actual variation in test section wall and blade surfaces temperatures is expected to be smaller, since several pre-runs were used to cool the test section walls and blades before runs were started.

Using a simplified heat transfer model, estimates of the changes in total temperature across the cascade were calculated. The heat transfer model is based on the convective heat transfer coefficient for a flat plate (with zero pressure gradient) in high speed flows. The model is described in detail in Appendix C. Although, the assumptions used in the analysis are somewhat crude, the model is used to provide an estimate of the changes in total temperature that may occur across the cascade for the above wall temperature limits. For each of the surfaces that make the blade passage, the rate of heat transfer is calculated using the following equation:

$$\dot{Q} = \bar{h} A (T_w - T_{aw}) \quad (4.10)$$

where $\bar{h}$ is the average convective heat transfer coefficient over the surface, A is the surface area of the wall, T_w is the wall temperature, and T_{aw} is adiabatic wall temperature. The value of $\bar{h}$ is obtained from the flat plate analysis.

Table 4.3 shows estimates for the total temperature variations across the cascade at several typical operating points for wall temperatures equal to +20 °C and -60 °C. Also shown are the variations in loss coefficients and axial velocity density ratios that correspond to these total temperature changes. As shown from the table, the estimated variation in loss coefficients is less than the overall estimated uncertainty due to all sources. Also the variation in AVDR's due the changes in total temperature is less than the estimated uncertainty of ± 0.02 .

M_2	$T_w = 20\text{ }^\circ\text{C}$			$T_w = -60\text{ }^\circ\text{C}$		
	$T_{01}-T_{02}$	ΔY_t	ΔAVDR	$T_{01}-T_{02}$	ΔY_t	ΔAVDR
0.49	-0.02	-0.00022	0.0001	0.14	0.00154	-0.00020
0.64	-0.05	-0.00027	0.0001	0.26	0.00139	-0.00050
0.66	-0.07	-0.00035	0.0001	0.28	0.00144	-0.00040
0.81	-0.13	-0.00037	0.0002	0.36	0.00102	-0.00060
0.91	-0.16	-0.00017	0.0003	0.38	0.00043	-0.00070
0.97	-0.18	0.00001	0.0003	0.40	0.00001	-0.00070
1.03	-0.20	0.00017	0.0004	0.41	-0.00032	-0.00070
1.12	-0.21	0.00035	0.0004	0.43	-0.00068	-0.00070

Table 4.3 Typical Variations in Total Temperature across the Cascade and Corresponding Changes in Losses and AVDR, HS1A Cascade, $i - i_{des} = 4.5^\circ$.

4.10.3.4 Effects of Uncertainties in Exit Flow Angle Measurements

The uncertainty on the calibrated exit flow angle was estimated to be less than $\pm 0.5^\circ$. Also, as mentioned in Section 3.8, a probe angle alignment mechanism was used measure the probe misalignment angle, θ , relative the blade exit angle. This probe misalignment angle was used to correct the measured flow angle as per Eqn. (3.2) as follows:

$$\alpha_{2c} = \alpha_{2m} + \theta$$

where α_{2c} and α_{2m} are the corrected and the measured flow angles respectively. For all test conditions, the measured probe misalignment angle was less than $\pm 0.4^\circ$. The combined uncertainty for the exit flow angle was then estimated to be less than $\pm 0.65^\circ$. This uncertainty value of $\pm 0.65^\circ$ was used to estimate the corresponding changes in losses and AVDR values for all operating conditions.

Table 4.4 shows variations in loss coefficients and AVDRs for typical operating points. As shown in the table, the uncertainty on the measured exit flow angle has very little effect on the measured losses. However, there appears to be a significant effect on the AVDRs at all operating conditions. For all operating conditions, the changes in AVDRs were about ± 0.018 which is less than the estimated uncertainty due to all sources of ± 0.02 .

M_2	$\theta = +0.65^\circ$		$\theta = -0.65^\circ$	
	ΔY_t	$\Delta AVDR$	ΔY_t	$\Delta AVDR$
0.49	0.00003	0.0173	-0.00005	-0.0173
0.64	0.00003	0.0183	-0.00003	-0.0184
0.66	0.00004	0.0175	-0.00003	-0.0176
0.81	0.00003	0.0180	-0.00004	-0.0182
0.91	0.00005	0.0182	-0.00004	-0.0183
0.97	0.00001	0.0180	0.00001	-0.0181
1.03	-0.00004	0.0179	0.00006	-0.0180
1.12	-0.00003	0.0178	0.00004	-0.0178

Table 4.4 Effects of Probe Misalignment Angle on Loss Coefficient and AVDR Measurements, HS1A Cascade, $i - i_{des} = 4.5^\circ$.

5. Results and Discussion

5.1 Introduction

As mentioned earlier, experimental investigations were conducted to study the aerodynamic performance of two transonic turbine cascades at off-design conditions. The cascade blade geometry of the second cascade (HS1B) differed from the first cascade (HS1A) mainly in the leading edge geometry. One of the goals was to identify leading-edge design modifications which would increase the tolerance of the blades to off-design incidence. To help in understanding the physics associated with both cascade flows, the results for the base line cascade (HS1A) are first presented. These results will be used to provide a basis for the interpretation of the HS1B results. A comparison will also be made with HS1A results. The complete results for both HS1A and HS1B are tabulated in Appendices D and E respectively.

5.2 HS1A Cascade Results

5.2.1 Axial Velocity Density Ratio

Results are presented first for the axial velocity density ratio (AVDR) since the value of AVDR probably has an influence on most of the other results to be presented in the following sections.

The value of AVDR is an indication of the two-dimensionality of the flow through the cascade. Some cascade facilities use endwall suction or blowoff to control the thickness of the endwall boundary layers and therefore the AVDR. Blowoff is not currently being used in the present facility, although provision for blowoff exists on the endwalls ahead of the cascade. In the absence

of blowoff, Sieverding (1993) recommended using an aspect ratio (H/C) of about 1.6 or larger for two-dimensional cascade testing in the transonic range. The aspect ratio of 1.525 for the present cascade, which was chosen to maximize the blade chord, is thus at the lower end of the range recommended by Sieverding.

Figure 5.1 shows the measured AVDRs for all tested flow conditions. The estimated uncertainty in the value of AVDR is about ± 0.02 . The AVDRs are seen to be close to the desired value of 1.0 over the full Mach number range for moderate values of the incidence. However, the AVDR increased with large positive incidence, and decreased slightly with negative values of incidence. This should be borne in mind when interpreting the present results for larger values of off-design incidence.

Unfortunately, the precise influence of AVDR on the performance of turbine blade rows has not been documented particularly well. The effects of the axial velocity ratio (AVR) in incompressible flow conditions has been examined experimentally by Rodger et al. (1992) and Whitehouse et al. (1993). They found that the losses decreased with increasing AVR. This behaviour was observed for all off-design values of incidence. At transonic flow conditions, Kiock et al. (1986) found that the exit flow angle and the blade losses were essentially independent of the AVDR value, particularly in the Mach number range of 0.9 to 1.0. The Kiock et al. results are for design incidence only. Saroch (1996) conducted a computational study of the effects of AVDR in a turbine cascade at transonic flow conditions, but again just for the design incidence. Saroch used the Navier-Stokes code of Dawes (1988) to predict the effects of AVDR on the loss coefficient and deviation in a linear turbine cascade for outlet Mach numbers from 0.6 to 1.2. The loss coefficients predicted by Saroch are shown in the inset in Figure 5.1. For Mach numbers from 0.6 to 0.9, the losses were found to increase slightly with increasing AVDR. For Mach numbers from 0.9 to 1.0 the predicted losses were essentially independent of AVDR, as also observed experimentally by Kiock et al.. As the Mach number increased beyond unity, Saroch found that the losses decreased with increasing AVDR. Although the author is not aware of any experimental data that confirm Saroch's findings, his results have been used qualitatively in the interpretation of the present loss results where the AVDR differed significantly from 1.0.

Figure 5.1 Effects of Incidence and Mach Number on Axial Velocity Density Ratio (AVDR), HS1A Cascade.

5.2.2 Blade Loading

Blade loading measurements were performed for all test conditions. Figures 5.2, 5.3, 5.4, 5.5, and 5.6 show the effects of the exit Mach number on the surface Mach number distributions for all HS1A test flow conditions. Repeatability measurements showed that the uncertainty in the surface Mach numbers is about ± 0.01 .

Figure 5.2 Effects of Mach Number on Blade Loading at Design Incidence, HS1A Cascade.

Figure 5.3 Effects of Mach Number on Blade Loading at -10° Off-Design Incidence, HS1A Cascade.

Figure 5.4 Effects of Mach Number on Blade Loading at 4.5° Off-Design Incidence, HS1A Cascade.

Figure 5.5 Effects of Mach Number on Blade Loading at 10° Off-Design Incidence, HS1A Cascade.

Figure 5.6 Effects of Mach Number on Blade Loading at 14.5° Off-Design Incidence, HS1A Cascade.

At low exit Mach numbers, the loading distributions (e.g. Figure 5.6) show a region of nearly constant pressure on the suction side beginning at about $x/C_x = 0.75$ and following a region of diffusion. This might be interpreted as the presence of trailing edge separation. This would be consistent with the somewhat higher losses observed at the same conditions. However, trailing edge separation would be associated with reduced outlet flow angle. As shown later, the outlet angle was in fact very similar at all Mach numbers. This suggests little if any separation was present at the low Mach numbers. Once the outlet Mach number reaches about 0.95, the passage chokes and the loading remains constant forward of about 60% of axial chord for any further increases in M_2 . For all test conditions, at exit Mach numbers roughly around $M_2 = 0.96$, a nearly-normal shock also impinges on the suction side at about 75% axial distance. With further increase in Mach number, the shock becomes more oblique and the impingement point migrates towards the trailing edge.

Figures 5.7, 5.8, and 5.9 show the effects of incidence on the loading at three exit Mach numbers; high subsonic, close to sonic, and close to the design value of 1.05. The variation in the Mach numbers reflects difficulties in setting the wind tunnel operating points. Aft of the throat (at about 60% C_x) the loading distribution appears to be mainly varying as a result of the variations in the exit Mach number rather than the incidence. Although insignificant, there seems to be a slight reduction in blade loading at the aft end of the blade due to incidence. This is apparent especially for subsonic Mach numbers (see Figure 5.7). At the high positive values of incidence there may be some influence of the AVDR, although this does not appear to be very strong. On the other hand, major changes in loading distribution occur with incidence in the vicinity of the leading edge. With increasingly positive incidence a strong suction peak develops on the suction side close to the leading edge. The magnitude of the suction peak Mach number is incidence dependent. For example, at $+14.5^\circ$ the local Mach number reaches a peak value of about 1.30 at only 10% of axial chord. The flow is subsequently decelerated to close to sonic conditions without the apparent presence of a shock and without any signs of boundary layer separation. The slight disturbance in the pressure distribution on the pressure side near 30% C_x at both high values of incidence (see Figures 5.5 and 5.6) is not believed to be a sign of separation since the flow in that region is subject to a favorable pressure gradient. At -10° incidence, a region of diffusion develops on the pressure surface after the

initial acceleration aft of the stagnation point. A disturbance in the pressure distribution on the pressure side near 30% C_x may be a sign of the presence of a separation bubble here.

As mentioned in Section 4.8.2, the blade loading data were also expressed as surface static pressure coefficient distributions although they are not presented graphically here. These data are tabulated for both HS1A and HS1B cascades in Appendices D and E respectively.

Based on both experimental and computational investigations at low-speed conditions, Benner et al. (1997) concluded that turbine blades would be more sensitive to off-design incidence if there were large discontinuities in surface curvature at the leading-edge blend points. A similar effect was identified by Walraevens & Cumpsty (1995) for compressor blades. The present blade has a leading edge circle and consequently such curvature discontinuities must be present. Unfortunately, the distribution of blade-surface static taps on the present cascade was too sparse to resolve any pressure disturbances which might have originated from such curvature discontinuities. Therefore, it is not possible to confirm from the present measurements a similar effect of surface-curvature discontinuities for transonic conditions.

Figure 5.7 Effects of Incidence on Blade Loading at High Subsonic Mach Numbers, HS1A Cascade.

Figure 5.8 Effects of Incidence on Blade Loading Near Sonic Mach Number, HS1A Cascade.

Figure 5.9 Effects of Incidence on Blade Loading Near Design Mach Numbers, HS1A Cascade.

5.2.3 Base Pressure

The influence of the base pressure on the profile losses in transonic turbines is well recognized (e.g. Mee et al., 1992). Using a simple control volume analysis, Denton (1993) showed that the base-pressure contribution to the total-pressure loss coefficient is given approximately by:

$$\Delta Y_{t,te} = -C_b \left(\frac{t}{o} \right) \quad (5.1)$$

where C_b is the trailing-edge base pressure coefficient, t is the trailing edge thickness, and o is the throat opening. However, the behaviour of the base pressure coefficient is not well documented at off-design incidence. Therefore, one of the present blades was instrumented with a base pressure tap and base pressure measurements were made for all test cases.

Perhaps the most widely used correlation for the base pressure behaviour on turbine blades is that due to Sieverding et al. (1980). Sieverding et al. correlated the base pressure with the trailing edge wedge angle, the unguided turning, and the downstream static pressure. They also provided an explanation for the trend in the base pressures in terms of the appearance of shock waves in the trailing-edge region of the blades.

Figure 5.10 compares the measured base pressures for the HS1A cascade with the Sieverding et al. correlation. The measurements show good agreement with the correlation for pressure ratios P_{s2}/P_{01} larger than about 0.55 (which corresponds to outlet isentropic Mach numbers less than about 0.95). At lower values of the pressure ratio shock waves are present near the trailing edge. As seen from the figure, the correlation predicts somewhat higher base pressures than were observed for these conditions. These discrepancies do not appear to be strongly related to the incidence. Nor do they appear to be primarily due to the effects of AVDR, since comparable discrepancies were obtained at significantly different values of AVDR. Similar discrepancies have been observed at design incidence by other investigators (e.g. Xu & Denton, 1988, and Denton & Xu, 1990). Denton (1993) attributed these differences to the effects of trailing-edge blockage, which is not included as a correlating parameter in the Sieverding et al. correlation. It is not possible to confirm Denton's explanation from the present

measurements. However, they do seem to support the suggestion that one or more important geometric parameters are missing from the Sieverding et al. correlation. It should also be noted that at high Mach numbers the base pressure varies considerably with location on the trailing edge and the present single tap gives a somewhat limited picture of the base pressure behaviour.

Figure 5.10 Comparison of Measured Base Pressure with Sieverding's Correlation, HS1A Cascade.

Figure 5.11 presents the base pressure results in terms of the variation of base pressure coefficient with exit Mach number and incidence. The corresponding values of AVDR are also indicated on the figure. These results will be used along with Eqn. (5.1) in interpreting the profile loss data which will be presented next. As also found in other investigations, the C_b is generally

negative and has a strong minimum near $M_2 = 1.0$ (e.g. Carscallen et al., 1996). A few positive values of C_b are seen at low Mach numbers. The highest value occurs for an incidence value of $+14.5^\circ$. For this operating point, the Reynolds number was about 450,000.

Figure 5.11 Effects of Mach Number and Incidence on Base Pressure Coefficient, HS1A Cascade.

The lowest value of C_b occurs for an incidence of -10° and $M_2 = 0.98$. However, this point appears somewhat out of keeping with the other results for the same incidence. It is also the most anomalous point on Figure 5.10. As a result, the curve for -10° has not been forced to pass close to this point. The minimum C_b is seen to vary somewhat with incidence and AVDR. In broad terms, the weakest minima were obtained at high positive incidence and high AVDR. However, as

with Figure 5.10, the pattern is not completely consistent. Pending further investigations, the effect of AVDR seems to be the most likely reason for the variation in the minimum value of C_b .

5.2.4 Profile Losses

Figures 5.12 and 5.13 show the measured results for the profile losses. Results are presented for the incidence values of -10° , $+4.5^\circ$, $+10^\circ$, and $+14.5^\circ$. As seen earlier, for the measurements at $+14.5^\circ$ the AVDR reached about 1.25 and the results probably cannot be considered even approximately two-dimensional.

Figure 5.12 shows the variation of the total pressure loss coefficient Y_t with exit Mach number for five values of incidence. The suggested trend is similar for all five curves and is generally consistent with the variation with Mach number found by Mee et al. (1992) at design incidence. In all cases, the loss coefficient reached a minimum at about $M_2 = 0.85$, after which the losses rose sharply through an exit Mach number of unity. From Figure 5.11, it appears that the minimum losses occur at the point where the base pressure coefficient starts to decrease. For the present blade row the value of (t/o) is 0.08 and thus, from Eqn. (5.1) and Figure 5.11, just above $M_2 = 1.0$ the base pressure effect contributes about 0.025 to Y_t . This accounts for roughly half of the rise in losses through the sonic outlet condition. The remainder is presumably due mainly to direct total pressure losses occurring through the shock waves which begin to appear at M_2 of about 0.95. These effects are of course not independent since the pattern of expansion and shock waves which form at the trailing edge play an important role in determining the base pressure. Above the sonic condition, the losses briefly level off (or perhaps even decrease in the case of the design incidence). This leveling-off has also been observed before. It is evidently due to the rising base pressure and the increasing obliqueness of the shock wave which impinges on the suction surface. As the exit Mach number increases further, the shocks become stronger and the loss coefficient begins to rise again.

At low exit Mach numbers, the loss coefficient would be expected to approach asymptotically a limiting low-speed value. This asymptotic behaviour is not evident here,

particularly for the data for $+10^\circ$, and $+14.5^\circ$ of incidence. This may be partly due to the effect of AVDR. The calculations of Saroch (Figure 5.1) suggested that at low Mach numbers the observed losses would be increased by values of AVDR greater than 1.0, which applies for the $+10^\circ$, and $+14.5^\circ$ measurements. However, the main explanation probably lies in the effect of low Reynolds numbers. For the low Mach number data points shown on Figure 5.12, the Reynolds number was of the order of 500,000. The influence of Reynolds number on the losses is confirmed later in this section.

Figure 5.12 Effects of Mach Number and Incidence on Losses, HS1A Cascade.

Figure 5.13 shows the variation of Y_t with incidence for fixed values of M_2 . These curves are essentially a cross-plotting of the data from Figure 5.12, using interpolated values of Y_t at the desired values of M_2 . Data for $+14.5^\circ$ of incidence have also been included to demonstrate the likely influence of AVDR

Figure 5.13 Effects of Off-Design Incidence on Losses at Various Exit Mach Numbers, HS1A Cascade.

For subsonic exit Mach numbers, the present blade has fairly constant losses for about $\pm 10^\circ$ about the design incidence. The small rise in losses as $+10^\circ$ is approached is most likely due to increased loss production in the suction-side boundary layer on the forward part of the blade. As discussed by Denton (1993), the loss production in a boundary layer varies as the cube of the edge

velocity. A velocity overshoot near the leading edge similar to that shown in Figures 5.8 and 5.9 also occurs for lower Mach numbers (e.g. see Figure 5.7) and this probably explains the increase in losses. At its design-point Mach number of 1.05, the blade is more sensitive to incidence, as would be expected. On the other hand, the drop in losses at $+14.5^\circ$ is unexpected. However, this drop is consistent with the influence of AVDR on losses predicted by Saroch's calculations: at supersonic conditions, the high AVDR of 1.25 for the $+14.5^\circ$ measurements is predicted to reduce the losses noticeably compared with two-dimensional conditions. It is for this reason that the loss results presented for $+14.5^\circ$ should be treated with caution.

The higher than expected losses at very low Mach numbers and the apparent trailing-edge separation observed in the blade loading measurements prompted a brief investigation of the influence of Reynolds number. Measurements were made for a range of Reynolds numbers at a fixed exit Mach number of about 0.56 and an incidence of $+4.5^\circ$. The Reynolds number was varied by partially blocking the wind tunnel exhaust and thus forcing up the cascade outlet pressure. The results for the losses are shown in Figure 5.14. The loss coefficient measured at a Reynolds number of 800,000 was used as the reference value. As seen from the figure, the loss coefficient remained almost constant with decreasing Reynolds number down to about $Re = 600,000$ and then began to rise strongly. For design incidence and a different turbine cascade Ladwig and Fottner (1993) found a similar strong rise in losses below a Reynolds number of about 500,000. It is worth noting that other investigators (e.g. Patterson and Hoeger, 1986, and Kurz, 1991) found similar effects of Reynolds number on losses at design incidence. Assuming the results in Figure 5.14 are representative of the behaviour of the present cascade at other values of incidence, Reynolds number effects seem to account for the unexpectedly high values of loss coefficient seen on Figure 5.12 for low values of exit Mach number.

Figure 5.14 Reynold's Numbers Effects at $M_2 = 0.56$,
and Incidence = 4.5° .

5.2.5 Exit Flow Angle

Figure 5.15 shows the effects of both incidence and Mach number on the exit flow angle. As is generally found, the maximum exit flow angle occurs at an exit Mach number near 1.0. As the downstream flow becomes supersonic, the flow vector must swing back towards the axial to provide the increase in flow area, and this is shown by the measurements. Throughout the Mach number range, $+14.5^\circ$ of incidence produced the lowest outlet flow angles by about 0.5° . This is probably an indication of the effects of high AVDR on outlet flow angle. However, generally for the present blade geometry, there appears to be no significant effect of either incidence or AVDR on exit flow angle. Comparable

results of the effects of AVDR on the exit flow angle were also seen computationally by Saroch (1996) and experimentally by Kiock et al. (1986). However, the results reported by Kiock et al. were only for AVDR values ranging from 0.9 to 1.0. Pending further investigations, it appears that the effects of AVDR on exit flow angle may be negligible in the transonic range.

Figure 5.15 Effects of Incidence and Mach Number on Exit Flow Angle, HS1A Cascade.

5.3 HS1B Cascade Results

5.3.1 Introduction

Figures 5.16(a) and (b) compare the leading-edge geometries of the two cascades, HS1A (baseline) and HS1B (modified) respectively. The local flow direction in the vicinity of the leading edge is different from the undisturbed flow direction and is referred to as the induced flow direction. The induced flow direction can be used to determine the location of the stagnation point at the blade leading edge. As shown in Figure 5.16(a), the HS1A blade was designed with its leading edge metal angle aligned with what was estimated to be the induced flow direction. The resulting design incidence ($i_{(des)A}$) for HS1A is -4.5° .

Figure 5.16(b) shows the leading edge geometries for the two blades, HS1A and HS1B, superimposed on each other. Both HS1A and HS1B blades were designed for the same service. The difference between HS1A and HS1B lies mainly in the leading edge geometry as shown in Figure 5.16(b). The objective of the modifications to the leading edge geometry was to study the tolerance of the modified blade to positive off-design incidence. As shown in Figure 5.16(b), HS1B blade has a higher leading edge metal angle than HS1A by 10.5° . Consequently, when HS1B is operated at the design inlet flow direction, the stagnation point will tend to be on the suction side of the blade. This displacement of the stagnation point at the design point may lead to the appearance of a separation bubble at the leading edge pressure side. The design incidence for HS1B ($i_{(des)B}$) is -15.0° . The HS1B blade was investigated at the incidence values of 0.0° , $+4.5^\circ$, $+10.0^\circ$, and $+14.5^\circ$ relative to design. Since the interest is particularly in the behaviour of HS1B at positive values of incidence, negative values of incidence were not investigated, unlike for HS1A.

For comparison purposes, the data for HS1B are presented in the same format and in the same order as the data for HS1A. Since the behaviour of the two cascades was similar in many respects, the discussion in the present section is briefer and concentrates on those aspects of the flow which were different in the two cascades.

Figure 5.16 HS1A and HS1B Leading Edge Geometries and their Relationship to the Undisturbed Flow Direction.

5.3.2 Axial Velocity Density Ratio

Figure 5.17 shows the effects of both incidence and Mach number on the AVDR for the HS1B cascade. For comparison purposes, the corresponding results for HS1A are shown adjacent to Figure 5.17. As shown, the behaviour of AVDR with Mach number for HS1B cascade was similar to the one observed for the HS1A cascade, mainly that it was relatively constant with Mach number for a given incidence.

Figure 5.17 Effects of Incidence and Mach Number on Axial Velocity Density Ratio (AVDR), HS1B and HS1A Cascades.

The variation of the AVDRs with incidence for both HS1B and HS1A cascades are summarized in Table 5.1 for an outlet Mach number near design (i.e. $M_2=1.05$). Table 5.1 shows that the AVDR values for the HS1B cascade are slightly higher than the ones for the HS1A cascade at the incidence values of 0.0° , and $+4.5^\circ$ and comparable at the incidence values of $+10.0^\circ$, and $+14.5^\circ$. The differences in AVDR for HS1A and HS1B at low incidence are larger than the estimated uncertainty of ± 0.02 . Thus there seems to be some differing influence on AVDR for the two cascades at low incidence. At the incidence values of $+10.0^\circ$ and $+14.5^\circ$, the differences in AVDR for HS1A and HS1B are within the estimated uncertainty of ± 0.02 . For a given value of incidence, comparable differences in AVDRs to the ones shown in Table 5.1 are also found between HS1A and HS1B at Mach numbers other than design (i.e. $M_2=1.05$).

$i - i_{des}$	HS1A	HS1B
0.0°	1.034	1.083
4.5°	1.039	1.112
10.0°	1.126	1.141
14.5°	1.266	1.264

Table 5.1 A Comparison Between AVDRs for HS1A and HS1B Cascades at near design Mach Number (i.e. $M_2=1.05$) and Off-Design Incidence.

As mentioned in Section 2.5, there is little information available on the effects of AVDR on turbine cascade performance and the factors that influence AVDR. The effects which are thought to influence AVDR include the blade aspect ratio and the inlet endwall boundary layer thickness. However, these are very similar for HS1A and HS1B at all values of incidence. At the incidence values of 0.0° and $+4.5^\circ$ there are noticeable differences in blade loadings near the leading edge, as will be discussed in Section 5.3.3. The differences include the presence of an apparent separation

bubble on the pressure side of HS1B. For these low incidence values the AVDRs for the HS1B cascade are higher than the ones for HS1A. At $+10.0^\circ$ and $+14.5^\circ$ incidence the loading distribution for the two blades are very similar with no sign of leading edge separation (e.g. Figure 5.27). For these high values of incidence, the AVDRs were also very similar. Thus there may be some influence of the blade loading distribution on AVDR. However, the mechanism by which the loading influences the AVDR is not clear. Additional experimental data on the effects of AVDR would be valuable to interpret the physics. Computational studies using a Navier-Stokes solvers might also provide some useful insights into the factors which influence AVDR.

5.3.3 Blade Loading

Figures 5.18, 5.19, 5.20, and 5.21 show the effects of exit Mach number on surface Mach number distributions for the HS1B cascade at the incidence values of 0.0° , $+4.5^\circ$, $+10.0^\circ$, and $+14.5^\circ$ respectively. The measurements did not include the incidence case of -10.0° since, as mentioned previously in Section 5.3.1, the blade performance at positive off-design incidences is of primary interest. Figures 5.18 to 5.21 are included here for completeness and to allow direct comparisons with the corresponding figures for HS1A. These figures will not be used in the discussion that will follow.

The effects of incidence on the blade loading are shown in Figures 5.22, 5.23, and 5.24. These figures are for low, intermediate, and near design exit Mach numbers. As shown, with variations in incidence, there are changes to both the suction surface and pressure surface loadings in the vicinity of the leading edge. On the pressure side, at the incidence values of 0.0° and $+4.5^\circ$, the loading distributions shown in Figures 5.22 to 5.24 suggest the presence of a separation bubble that extends more than 20% axial chord aft of the leading edge. The inferred separation bubble is consistent with the stagnation point location for HS1B blade shown in Figure 5.16(b). The presence of the separation bubble at the incidence of $+4.5^\circ$ also suggests that the stagnation point is still on the suction surface. With increase in incidence, the stagnation point is expected to move towards the pressure side, and consequently the extent of the separation bubble is expected to be shorter. With

further increase in incidence, the incoming flow direction becomes more aligned with the blade inlet metal angle. As shown from the Figures 5.22 to 5.24, at incidence values of $+10.0^\circ$ and $+14.5^\circ$, the only sign of what appears to be a separation bubble on the pressure side is the small disturbance in the pressure distribution near 15% axial chord. This inferred separation bubble follows a region of diffusion which develops on the pressure surface after the initial acceleration aft of the stagnation point.

With increasingly positive incidence a strong suction peak develops on the suction side close to the leading edge. At $+14.5^\circ$ the local Mach number reaches a peak value of about 1.3 at roughly 10% axial chord. Figure 5.24 shows that this peak Mach number is maintained up to about 17% axial chord for the HS1B blade. The flow is subsequently decelerated to close to sonic conditions over a very short distance. This deceleration probably occurs due to an oblique shock wave, but the shock does not seem to cause boundary layer separation.

A comparison between HS1A and HS1B blade loading distributions is shown in Figures 5.25 to 5.28 for all incidence values. Each figure presents a comparison in blade loadings over a range of exit Mach numbers and for a fixed incidence. At design incidence, Figure 5.25 shows that the blade loading on the suction surface is roughly the same for both blades at exit Mach number below the choking value (which is about 0.96). Figure 5.25 shows that for given exit Mach number (higher than 0.96) the shock impingement point on the aft part of suction surface is different for the two blades: the impingement point for HS1B is located aft of the one for HS1A for a given exit Mach number. This change in the shock impingement point resulted in some change in the suction surface loading distribution in the vicinity of the trailing edge. With respect to the pressure side, Figure 5.25 shows that the only difference in the loading distributions is in the vicinity of the leading edge. As noted earlier, the HS1B loading distribution shows signs of a separation bubble that extends over the first 20% axial chord. The HS1A loading distribution in the vicinity of the leading edge show no signs of separation.

The loading distributions for the two blades at $+4.5^\circ$ incidence is shown in Figure 5.26. Compared to the design case (see Figure 5.25), at Mach numbers higher than the choking value (which is about 0.96), there seem to be significant differences in the shock impingement points on

the aft part of suction surface for the two blades. Thus, the loading distributions in the trailing edge region of the suction surface are also significantly different between the two blades. In addition, Figure 5.26 shows that the loading distribution for the two blades in the vicinity of the suction side leading edge is different. The figure shows slightly higher peak Mach numbers for the HS1B blade in the first 10% axial chord. On the pressure side, the loading distribution for HS1B also suggests the presence of a separation bubble at the leading edge pressure side. From Figures 5.25 and 5.26, the separation bubble at the $+4.5^\circ$ incidence appears to have the same extent as for the design case.

The comparison in loading distributions for the two blades at $+10.0^\circ$, and $+14.5^\circ$ incidence values is shown in Figures 5.27 and 5.28 respectively. For both incidence values, Figures 5.27 and 5.28 show that differences in loading distributions are now shifted toward the leading edge suction side. At subsonic exit Mach numbers (e.g. Figure 5.28) the loading distributions show higher local peak Mach numbers for HS1B blade in the first 20% axial chord. At Mach numbers higher than the choking value ($M_2 = 0.96$), both blade loadings show similar values of local peak Mach numbers in the vicinity of the leading edge suction side (e.g. see Figure 5.28). However the extent of the local peak Mach number region is greater for HS1B: three pressure taps for HS1B versus one tap for HS1A. For the pressure side, the loading distributions for the two blades are quite similar except for what appears to be a small pressure disturbance at the 15% axial chord shown by the HS1B pressure side loading distribution (e.g. see Figure 5.28). As discussed earlier this disturbance may be the sign of a small separation bubble.

Figure 5.18 Effects of Mach Number on Blade Loading at Design Incidence, HS1B Cascade.

Figure 5.19 Effects of Mach Number on Blade Loading at 4.5° Off-Design Incidence, HS1B Cascade.

Figure 5.20 Effects of Mach Number on Blade Loading at
10.0° Off-Design Incidence, HS1B Cascade.

Figure 5.21 Effects of Mach Number on Blade Loading at
14.5° Off-Design Incidence, HS1B Cascade.

Figure 5.22 Effects of Incidence on Blade Loading at Low Subsonic Mach Numbers, HS1B Cascade.

Figure 5.23 Effects of Incidence on Blade Loading at High Subsonic Mach Numbers, HS1B Cascade.

Figure 5.24 Effects of Incidence on Blade Loading Near Design Mach Number, HS1B Cascade.

Figure 5.25 Comparison of HS1A and HS1B Blade loadings at Design Incidence for Various Exit Mach Numbers.

Figure 5.26 HS1A versus HS1B Blade loadings at 4.5° Off-Design Incidence, and for Various Exit Mach Numbers.

Figure 5.27 HS1A versus HS1B Blade loadings at 10.0° Off-Design Incidence, and for Various Exit Mach Numbers.

Figure 5.28 HS1A versus HS1B Blade loadings at 14.5° Off-Design Incidence, and for Various Exit Mach Numbers.

5.3.4 Base Pressure

The base pressure for the HS1B cascade is plotted against Sieverding's correlation in Figure 5.29. For comparison purposes, the HS1A results (see Figure 5.10) are shown in the inset in Figure 5.29. As shown, the trend of the base pressure observed here is similar to the one for the HS1A cascade. However, the values of base pressure for HS1B cascade are slightly lower. As found with the HS1A cascade, the measurements for HS1B cascade are in best agreement with the correlation for pressure ratios P_{s2}/P_{01} less than 0.55 (which again corresponds to the choking isentropic Mach number of less than 0.96). At lower values of the pressure ratio shock waves are

present at the trailing edge and the correlation predicts somewhat higher values than observed here. These discrepancies do not appear to be strongly related to the incidence, nor do they appear to be related to the effects of AVDR. They seem to be consistent with the discrepancies seen for HS1A cascade.

Figure 5.29 Comparison of Measured Base pressure with Sieverding's Correlation, HS1B and HS1A Cascades.

Figure 5.30 presents the HS1B cascade base pressure values in terms of the base pressure coefficient as a function of exit Mach number and incidence. For comparison purposes, the HS1A cascade base pressure coefficient data are shown in the inset in Figure 5.30. As found with

the HS1A cascade, the C_b is generally negative and has a strong minimum around $M_2 = 1.0$. The highest value of C_b was also found at the incidence value of $+14.5^\circ$. The minimum C_b is seen to vary somewhat with incidence and AVDR, with the weakest minima occurring at high positive incidence and high AVDR.

Figure 5.30 Effects of Mach Number and Incidence on Base Pressure Coefficient, HS1B and HS1A Cascades.

When compared to HS1A cascade, the C_b curves for HS1B cascade are shifted downward to lower values for all incidence values. Denton & Xu (1990) suggested that higher base pressure values are usually an indication of thicker trailing edge boundary layers relative to the trailing edge thickness. It is not possible to confirm that this is the explanation for the differences in C_b for HS1A and HS1B since no measurements were conducted on the blade boundary layers.

Whatever the reasons for the differences in C_p , they have implications for the losses generated in the two cascades, as discussed in the next section.

5.3.5 Profile Losses

The profile loss coefficients for the HS1B cascade are shown in Figure 5.31 as a function of incidence and Mach number. Figure 5.32 compares the HS1A and HS1B profile losses at the values of incidence for which both cascades were tested. As shown in Figure 5.32, the profile losses for the HS1B cascade exhibit similar behaviour to the ones observed for HS1A cascade at all off-design incidence values. From Figure 5.30, it appears that the minimum losses for the HS1B cascade occur at the point where the base pressure coefficient starts to decrease. As for the HS1A cascade, the base pressure contributes to roughly half of the rise in losses through the sonic outlet condition. As noted in connection with HS1A, the higher than expected losses at low exit Mach numbers may be partly due to the effects of AVDR but are probably largely due to the effects of low Reynolds numbers.

When comparing losses from the two blades, it should be borne in mind that the AVDR values for some test flow conditions are higher in the case of HS1B, particularly at the incidence values of 0.0° and $+4.5^\circ$. Moreover, as shown in Figure 5.30, the base pressure coefficients for HS1B are lower than the ones for HS1A cascade particularly at the point where the base pressure coefficient starts to decrease (at an exit Mach number of about 0.85).

Figure 5.32 shows that the HS1B losses are slightly higher than the HS1A losses over the Mach number range of 0.7 to 0.9. The maximum difference between the two blade loss coefficients, for all incidence values, occurs at the minimum loss point and is about 0.010. As mentioned previously, with the exception of the leading edge geometry, the two cascades are almost identical, including having the same throat opening and blade trailing edge thickness. With some simplifying assumptions, Denton (1993) relates the profile losses to the boundary layer parameters and the conditions at the trailing edge as follows:

$$Y_t = -\frac{C_b t}{o} + \frac{2\theta}{o} + \left(\frac{\delta^* + t}{o} \right)^2 \quad (5.2)$$

where, δ^* : blade boundary layer displacement thickness,
 θ : blade boundary layer momentum thickness,
 t : blade trailing edge thickness,
 o : throat opening, and
 C_b : base pressure coefficient.

Figure 5.31 Effects of Mach Number and Incidence on Losses, HS1B Cascade.

Figure 5.32 A Comparison between the Profile Losses for HS1A and HS1B Cascades at all Test Flow Conditions.

For the two cascades the value of (t/o) is 0.08 and thus, from Eqn. (5.2) and Figure 5.30, at the minimum loss point ($M_2 = 0.85$) the difference in base pressure coefficients contributes about 0.008 to the difference in losses between the two cascades. This is roughly 80% of the difference. The remainder is presumably due to small differences in the losses occurring in the blade boundary layers and the effects of AVDR, particularly at the incidence values of 0.0° and 4.5° . At Mach numbers below 0.85, Figure 5.32 shows diminishing differences in losses with decreasing Mach numbers. From Figure 5.30, it appears that the base pressure coefficients for the HS1B cascade are lower than the ones for HS1A by roughly the same value over the whole range of subsonic Mach numbers. Thus, the HS1B cascade is expected to have larger base pressure loss than HS1A (as implied by Eqn. (5.2)). As shown in Figure 5.32, both cascades have very similar overall losses at the lower Mach numbers. Therefore, for HS1B the higher base pressure losses seem to be offset by a comparable reduction in the boundary losses. However, the comparison in loading distributions of both blades at subsonic Mach numbers (e.g. Figure 5.28) suggests that HS1B would probably have the largest boundary layer losses. Moreover, from Figure 5.32, the scatter in the results at $+4.5^\circ$ incidence and Mach number around 0.65 suggests that the uncertainty in the overall losses is the order of 0.01 (comparable to the difference in C_p effect between HS1A and HS1B). The explanation to this apparent contradiction may lie in the uncertainties in the measurements. Also, it is believed that the increase in base pressure contribution for HS1B is genuine since there seemed to be a systematic shift in C_p 's at all conditions of M_2 and incidence. However, the overall uncertainty means that there may not be consistency between the inferred change in boundary layer contribution (based on the changes in C_p) and the change in boundary layer contribution implied by the blade loading measurements. In addition, at these low Mach numbers, there may also be an additional effect on losses due to the differences in AVDR between the two cascades particularly at the incidence values of 0.0° and 4.5° .

At Mach numbers above the sonic condition, Figure 5.32 shows that the losses for the two blades are very similar. The comparison between the loading distributions of the two blades shown in Figures 5.25 to 5.28 shows regions with higher local Mach numbers for HS1B. As shown by Denton (1993), the loss production in a boundary layer varies roughly with the cube of the edge velocity. This suggests that at these exit Mach numbers the local boundary losses for the HS1B

cascade are somewhat higher than the ones for HS1A. Moreover, there appears to be an effect of AVDR on the HS1B losses. HS1B has somewhat higher AVDR than HS1A particularly at the incidence values of 0.0° and 4.5° . These high values of AVDR seem to reduce HS1B losses compared to HS1A values. Figure 5.32 shows that at $+10.0^\circ$ incidence and Mach numbers greater than one, HS1B losses are slightly higher than HS1A losses. At this incidence value, both HS1A and HS1B have the same AVDR and the increase in losses seen for HS1B appears to be mainly due to the loss production in the blade boundary layers.

Figure 5.33 shows the effect of incidence on profile losses for both HS1A and HS1B cascades at three representative Mach numbers. These curves are essentially a cross-plotting of the data from Figure 5.32, using interpolated values of Y_t at the desired values of M_2 . Therefore, in addition to the uncertainties in the measurements there may also be some uncertainties due to the interpolation procedure and the way the loss curves were drawn in Figure 5.32. Within the uncertainties in the measurements, HS1A and HS1B gave very similar loss results at all conditions of M_2 and incidence, with HS1B giving perhaps slightly higher losses in the minimum loss region (M_2 about 0.85). The hoped for improvement in positive incidence behaviour for HS1B clearly did not materialize.

It is worth also noting that the separation bubble inferred to be present near the leading edge of HS1B pressure surface, at the incidence values of 0.0° and 4.5° may have some deleterious effects on blade film cooling effects particularly if one or more cooling holes are located in the separated flow region. The presence of the separation bubble may reduce the effectiveness of the blade pressure side cooling.

Figure 5.33 Effects of Off-Design Incidence on Profile Losses for both HS1A, and HS1B Cascades.

5.3.6 Exit Flow Angle

Figure 5.34 shows the measured exit flow angle for HS1B cascade. For comparison purposes the exit flow angle data of the HS1A cascade is also shown in the inset in Figure 5.34. As shown from the figure, for HS1B there seems to be a definite trend in the exit flow angle with increasing incidence which was not present for HS1A. Moreover, for a given incidence, the values deviation seem to be somewhat higher for HS1B than HS1A. This raises the question of whether there was some trailing edge separation present on HS1B. From the loading distributions for the two blades, there appears to be some differences in the loading distributions in the vicinity of the trailing edge. As shown in Figures 5.25 to 5.28, the shock impingement point for HS1B is located aft of the

one for HS1A for a given exit Mach number. However, there was no apparent evidence of separation present for HS1B. And if present, the separation may be at the trailing edge or there are not enough static taps on the suction side of the blade to capture it. Whatever the exact explanation, clearly the C_p and α_2 results both suggest that there are differences in the flows near the trailing edge for the two cascades. This is of course surprising considering their geometric similarity in this region.

Figure 5.34 Effects of Incidence and Mach Number on Exit Flow Angle, HS1B and HS1A Cascades.

5.4 Comparisons with Existing Correlations

5.4.1 Introduction

As mentioned in Chapter 1, one of the objectives of the present study was to add to the open literature database of axial turbine data for purposes of developing improved design correlations for losses and outlet flow angle. Additional experimental data have been used occasionally to validate and improve the existing correlations especially if more data that reflects the recent improvements in blade design become available. As mentioned in Chapter 2, the existing correlations for mid-span losses and exit flow angle are limited to subsonic flow conditions. Therefore the comparisons with the existing correlations are only for the data obtained at the subsonic Mach numbers, although the range of Mach numbers in the present experiment extended to Mach numbers around 1.2. A survey of the loss and exit flow angle correlations of interest is given in Chapter 2.

5.4.2 Correlations for Profile Losses at Design Incidence

Figures 5.35(a) and (b) show a comparison at design incidence between the measured and predicted profile losses using the design incidence profile loss correlations discussed in Section 2.6, for both HS1A and HS1B cascades respectively. The figures show that, at Mach numbers below 0.8, the losses are overpredicted using the loss correlations of Ainley & Mathieson (AM) and Ainley/Mathieson/Dunham/Came (AMDC). However, the losses were underpredicted using the Kacker & Okapuu (KO) loss correlation.

At subsonic Mach numbers, the KO correlation predicts that modern turbine blades have considerably lower losses than those predicted by the AM and AMDC correlations. It is seen that in the high subsonic range, the measurements for HS1A and HS1B are in fact closest to the KO predictions. At lower Mach numbers, the rise in the measured losses was attributed to the effects of low Reynolds number. Although the KO loss system includes a Reynolds number correction it

clearly does not capture the significant Reynolds number effect which seemed to be present in the measurements.

As mentioned in Section 5.3.5, within the uncertainties in the measurements, HS1A and HS1B gave very similar losses results at all conditions of M_2 and incidence, with HS1B giving perhaps slightly higher losses in the minimum loss region (M_2 about 0.85). The difference in losses between HS1A and HS1B in the minimum loss region is about 0.01. As discussed previously, the HS1A and HS1B blades differ in the leading edge geometry. The leading edge metal angle is 60.0° for HS1B versus 50.5° for the HS1A blade. Thus, it appears that the slight difference in losses may be the result of the modifications to the leading edge geometry. The predicted changes in mid-span losses using the AM/AMDC and the KO prediction methods were around 0.015 and 0.01 respectively. The changes in losses predicted by the KO method are roughly 2/3 of the difference predicted using the AM/AMDC methods. This is because the KO method uses the 2/3 as a multiplying factor for the mid-span losses predicted using the AM/AMDC method. However, it appears that these prediction methods are not capturing the change in losses due to the difference in leading edge metal angle between the HS1A and HS1B cascades. From Figure 5.35, it appears that the KO predictions are closest to the measurements, and that both the AMDC and KO correctly show rise in losses with exit Mach number greater or equal to 1.0. However, the rise in mid-span losses occurs earlier in the measurements, and the rise with M_2 is much too steep in the correlations (this was recognized by KO). Moreover, the correlations do not show the flattening out of the mid-span losses at slightly supersonic exit Mach numbers. This is demonstrated by not only the present data but those of Mee et al. (1982) and others.

Figure 5.35 also shows that the Reynolds number effects are not captured by the correlations at the low values of M_2 . The KO loss system only increases the losses at Reynolds numbers below 200,000. Clearly this is not the case in the present measured data and this suggests that the Reynolds number correction in the KO loss system needs more attention.

In summary, it appears that KO correlation correctly predicts reduction in mid-span losses for modern turbine blade designs. Moreover, the influence of low Reynolds numbers on the mid-span losses at low M_2 values does not seem to be reliably predicted and needs more attention. From the above discussion, there is clearly a need for improved correlations through the transonic range. The present data can make a significant contribution to the development of such transonic loss correlations.

a) HS1A Cascade.

b) HS1B Cascade

Figure 5.35 Predicted and Measured Profile Losses at Design Incidence for a) HS1A, and b) HS1B Cascade.

5.4.3 Correlations for Profile Losses at Off-Design Incidence

An evaluation of the off-design incidence profile loss correlation of Benner et al. (1997) against both HS1A and HS1B cascade data is shown in Figures 5.36(a) and (b). As mentioned in Section 2.5.2.3, the Benner et al. correlation is a functional relationship that takes into account the effects of the leading edge wedge angle (We), the leading edge diameter-to-pitch ratio, the blade inlet and exit metal angles β_1 and β_2 respectively, and the incidence angle ($\alpha_1 - \alpha_{1(des)}$). The correlation is given in terms of $|\Delta\phi_p|^2$ (i.e. change in the kinetic energy loss coefficient relative to the design value) versus χ (the incidence parameter). As defined in Section 2.5.2.3, the incidence parameter χ is given by:

$$\chi = \left(\frac{d}{s} \right)^{-0.05} (We)^{-0.2} \left(\frac{\cos\beta_1}{\cos\beta_2} \right)^{-1.4} (\alpha_1 - \alpha_{1(des)})$$

where,

- d/s : leading edge diameter to pitch ratio,
- We : blade leading edge wedge angle,
- β_1 : blade inlet metal angle,
- β_2 : blade exit metal angle,
- α_1 : cascade inlet flow angle, and
- $\alpha_{1(des)}$: cascade inlet flow angle at design incidence.

The data shown in the figures are for exit Mach numbers less than about 0.75. This value represents the upper limit of exit Mach numbers for which the correlation is expected to be valid. Predicted values of $|\Delta\phi_p|^2$ that exceeded 0.14 have been plotted along the top edge of both figures. The figures show that reasonable agreement between predictions and measurements is only

obtained at the off-design incidence values of $-10.^\circ$ and $+4.5^\circ$. For the higher positive off-design incidence values, the incidence parameter χ used in Benner et al. correlation was close to the upper limit if not outside the range of the χ values for which the correlation was devised. Benner et al. show that for χ values higher than 4 the value of $|\Delta\phi_p|^2$ rose rapidly and this is shown in Figure 2.13 (see Section 2.5.2.3).

In the χ relationship, the ratio $\cos(\beta_1)/\cos(\beta_2)$ is intended as an approximation to the flow acceleration through the blade channel, C_2/C_1 . The form used embodies two simplifying assumptions: that the metal and flow angles are the same at the inlet and outlet of the blade row, and that the axial velocity is constant through the blade row. Neither of these simplifying assumptions are satisfied very well for the present cascades. There are substantial differences between the inlet metal angles and the design inlet flow angles: 4.5° for HS1A and 15° for HS1B. Secondly, many of the measurements were conducted for axial-velocity-density ratios significantly different from 1.0. Thus, $\cos(\beta_1)/\cos(\beta_2)$ is not a reasonable approximation of the channel acceleration for the present cascades. This is probably a major reason for the somewhat poor agreement between Benner et al. correlation and the present measurements.

Figure 5.37 shows the predictions of the energy loss coefficient using the inlet flow angle at design incidence instead of the inlet blade angle to compute the incidence parameter of Benner et al.. The exit blade angle was retained as a reasonable approximation to the exit flow angle. The modified incidence parameter is then computed as follows:

$$\chi = \left(\frac{d}{s} \right)^{-0.05} (We)^{-0.2} \left(\frac{\cos(\alpha_1)_{des}}{\cos\beta_2} \right)^{-1.4} (\alpha_1 - \alpha_{1(des)})$$

a) HS1A Cascade

b) HS1B Cascade

Figure 5.36 Evaluation of Benner et al. Correlation (Predicted versus Measured), $M_2 < 0.75$.

Using this modification, the predictions showed better agreement with the measurements. However, the predictions for the incidence values of $+10.0^\circ$ and $+14.5^\circ$ were still considerably higher than the measurements. The discrepancies may be attributed to the high AVDR values (i.e. 1.14 to 1.28) at these values of incidence. The loss data presented here for the high AVDR values could be used with additional data to include the effects of AVDR in the profile loss correlation of Benner et al.. The effect of AVDR would appear as a combined term with the cosine ratio, again raised to some power.

Any reworking of Benner et al. correlation should also examine whether it should be the channel acceleration at the particular operating point (as opposed to the design point) which should appear in the incidence parameter χ .

a) HS1A Cascade.

b) HS1B Cascade.

Figure 5.37 Predictions Using the Modified Benner et al. Correlation.

5.4.4 Computations of the Exit Flow Angle or Deviation

5.4.4.1 Evaluation of Ainley & Mathieson Exit Flow Angle Correlation

The Ainley & Mathieson (1951) method allows predictions of the outlet flow angle for Mach numbers up to the sonic condition. The Ainley & Mathieson method is usually presented in graphical form. Wilson (1984) fitted curves to the Ainley & Mathieson figures, as described in Section 2.5.3. The curve fit relationships are repeated here for convenience:

- a) for exit Mach numbers less than 0.5, the predicted exit flow angle is given by:

$$|\alpha_2|_{0 < M_2 < 0.5} = \left(\frac{7}{6} \left(\left| \cos^{-1} \left(\frac{o}{s} \right) \right| - 10^\circ \right) + 4^\circ \left(\frac{s}{e} \right) \right) \quad (5.3)$$

- b) for turbine blades with sonic exit Mach number, the predicted flow angle is given by:

$$|\alpha_2|_{M_2 = 1.0} = \left| \cos^{-1} \left(\frac{o}{s} \right) \right| - \left(\frac{s}{e} \right)^{1.786 + 4.128(s/e)} \left| \sin^{-1} \left(\frac{o}{s} \right) \right|, \quad (5.4)$$

- c) and for exit Mach numbers between 0.5 and 1.0, the exit flow angle is taken as the linearly interpolated value between those obtained from Eqns. (5.3) and (5.4):

$$|\alpha_2|_{0.5 < M_2 < 1.0} = |\alpha_2|_{0 < M_2 < 0.5} - (2M_2 - 1) \left(|\alpha_2|_{0 < M_2 < 0.5} - |\alpha_2|_{M_2 = 1.0} \right) \quad (5.5)$$

where o is the throat opening, s is the cascade pitch, e is the radius of curvature of blade suction surface downstream of the throat, and M_2 is the exit Mach number. Figure 5.38 shows the nomenclature used in the above equations. For modern turbine blades, the radius of curvature aft of the throat is usually not constant, and in order to use the Ainley & Mathieson correlations an approximate value of the radius of curvature is needed. This can be obtained from the unguided

turning, θ_u . Referring to Figure 5.38, and using the right angle consisting of sides o , s and L , the length of side L is given as follows:

$$L = s \sqrt{1 - \left(\frac{o}{s}\right)^2}$$

From Figure 5.38, the arc length of the suction surface aft of the throat can be approximated very roughly by the distance L . The distance L can then be related to the radius of curvature (e) using the following equation:

$$L = e \theta_u \frac{\pi}{180}$$

where θ_u is the unguided turning angle measured in degrees. Then the ratio (s/e) used in the Ainley & Mathieson correlations can be approximated as follows:

$$\frac{s}{e} = \frac{\theta_u}{\sqrt{1 - \left(\frac{o}{s}\right)^2}} \frac{\pi}{180} \quad (5.6)$$

Predictions of the exit flow angle for the two cascades using the Ainley and Mathieson method are shown in Figures 5.39(a) and (b) plotted against the measured exit flow angle for all incidence values. The comparison is limited to exit Mach numbers lower than 1.0 since that is the range of applicability of the correlation.

From Figures 5.39 (a) and (b), there are clearly trends in the measurements which were not captured by the Ainley & Mathieson correlation. The measurements showed variations in the outlet flow angle of over 2 degrees for various reasons, including AVDR, exit Mach number, M_2 , and incidence. However, the Ainley & Mathieson gave a very small variation in the exit flow angle and that was due exclusively to variations in M_2 .

Figure 5.38 Schematic Diagram of Cascade Exit Showing Throat and Suction Surface Curvature Parameters (Sjolander, 1999).

a) HS1A Cascade

b) HS1B Cascade

Figure 5.39 Predicted Exit Flow Angle using Ainley & Mathieson Method.

5.4.4.2 Deviation Correlation of Islam

A comparison between the measured and the predicted flow deviation angles using Islam's (1999) correlation for both HS1A and HS1B cascades is shown for exit Mach numbers less than 1.0 in Figures 5.40(a) and (b) respectively. As described in Section 2.5.3, Islam's correlation was derived for subsonic flow conditions ($M_2 \leq 0.7$) and is intended to replace the Ainley & Mathieson (AM) correlation for exit Mach numbers less than 0.5. Thus for exit Mach numbers between the values of 0.7 and 1.0, the deviation angles were predicted using the AM sonic value and the same linear interpolation scheme described in section 5.4.4.1. Islam's correlation is given by Eqn. (2.25) (see Section 2.5.3) as follows:

$$\delta = \frac{(AVDR)^3 \left(\frac{s}{c} \right)^{1.1} (\alpha_1 + \beta_2)^{2.25}}{\zeta^{1.45} \left(\frac{t_m}{c} \right)^{0.3} (22 + 0.22 \beta_1^{1.64})} \quad (2.25)$$

where,

- AVDR : axial velocity density ratio,
- α_1 : inlet flow angle, in degrees,
- β_2 : exit metal angle, in degrees,
- ζ : stagger angle, in degrees,
- t_m/c : blade maximum thickness to chord ratio, and
- s/c : blade spacing-to-chord ratio.

a) HS1A Cascade

b) HS1B Cascade

Figure 5.40 Predicted versus Measured Flow Deviation, for $M_2 \leq 1.0$.

The range of AVDR over which the correlation is valid is from 0.95 to 1.05 and as shown in the above equation the AVDR is raised to a power of 3. Over this range of AVDR, Eqn. (2.25) indicates that the deviation should vary by about $\pm 15\%$ of the deviation value for an AVDR value of 1.0. Outside the range of 0.95 to 1.05 for AVDR, the predicted effect would be of course be even stronger.

In light of this it seems that the discrepancies between the measured and predicted values are largely due to the contribution of AVDR in the formulation of Islam. Moreover the measurements showed some influence of low Reynolds number and the change in loading distribution with incidence. These effects are not included in Islam's correlation and may account for some of the discrepancies observed between the measurements and the predictions. However, for measurements which fell in the appropriate range of AVDR, Islam's correlation did quite well. Unlike the Ainley & Mathieson correlation, it clearly gives the right trends of the deviation angle with incidence.

In summary, the present data could be used as a basis for improving Islam's correlation and extending it to a wider range of AVDR values.

6. Conclusions and Recommendations

6.1 Conclusions

Detailed measurements have been presented for the midspan aerodynamic performance of two transonic turbine cascades at off-design conditions. The two cascades are a baseline cascade HS1A, and a derivative cascade, HS1B. The blade geometry of the HS1B cascade differed from HS1A mainly on the leading edge geometry. The modified geometry was intended to investigate one possible approach to reducing the sensitivity to positive off-design incidence. The blade geometry of the baseline cascade is of recent design and incorporates modern design philosophies for high pressure turbine blades.

The data cover a broad range of both incidence and exit Mach numbers. The exit Mach number ranged from roughly 0.5 to 1.2, and the incidence ranged from -10° to $+14.5^\circ$. They provide a substantial addition to the data available in the open literature on the behaviour of transonic turbine blades at off-design incidence. As such, they could be used in the development of improved correlations for profile losses, base pressure behaviour, and outlet flow angle.

In this experiment, there was no control of the cascade test section inlet endwall boundary layers. As a result, there were changes to the width of the streamtubes at the blade mid-spans of the HS1A and HS1B cascades with variations in incidence. The AVDR was used as a measure of the average contraction or expansion of the streamtube at the blade mid-span. The AVDR value varied with incidence and ranged from roughly 0.99 at -10° to roughly 1.27 at $+14.5^\circ$ incidence. Consequently, some of the mid-span results presented here are not two-dimensional. However, in actual engines the mid-span AVDR is typically different from 1.0 and the effect of this on the blade aerodynamics is in fact not well understood. Thus, the present results also represent a

useful addition to the very limited data available in the open literature on the effects of AVDR on the aerodynamic performance of turbine blades at transonic Mach numbers.

The measurements included blade loading, base pressure, exit flow angle, and blade losses at mid-span.

For the HS1A and HS1B cascades, the blade loading measurements showed that there is an increase in suction surface blade loading up to the choking Mach number (i.e $M_2 = 0.96$). At exit Mach numbers above the choking value, the loading distribution upstream of the throat remained essentially the same. At Mach numbers around 0.96 a nearly normal shock impinged on the suction surface. With further increase in exit Mach number, the shock became more oblique and the impingement point migrated towards the trailing edge. At exit Mach numbers above 1.1, the suction surface peak Mach number close to the trailing edge was about 1.5 for all incidence values. These changes in suction surface loading distributions are believed to have significant effects on the suction surface boundary layer development and loss generation.

The loading distributions for both HS1A and HS1B cascades showed an increase in blade loading in the vicinity of the leading edge with increasing positive incidence. For the present blade geometries, the peak suction surface Mach number at the leading edge reached values around 1.3. Obviously this will have an effect on the blade suction surface boundary layer development and subsequently the boundary layer loss contribution to the mid-span losses.

The contribution of the base pressure to the blade losses, particularly at transonic flow conditions, was emphasized in this study. The results showed that the drop in the base pressure coefficient through the exit Mach number range of 0.85 to 1.0 is associated with a sharp increase in mid-span losses seen for the same Mach number range. The comparisons of the base pressure measurements with Sieverding et al.'s base pressure correlation seem to support the suggestion that one or more geometric parameters are missing from that correlation.

The blade mid-span loss behaviour showed that there is a decrease in losses as the exit Mach number approached the value of 0.85. A limited investigation of the effects of Reynolds number indicated that the present blades experienced increased losses at low subsonic exit Mach

numbers (i.e. M_2 values around 0.56) for Reynolds numbers below about 600,000. The minimum losses for both cascades (i.e. HS1A and HS1B) were at $M_2 = 0.85$. The minimum losses occur at the point where the base pressure started to decrease. For the present blade geometries, the contribution of the base pressure is roughly half of the rise in losses through the sonic conditions. The remainder is presumably due to the direct total pressure losses occurring through the shocks which begin to appear at M_2 of about 0.96. These effects are of course not independent since the pattern of expansion and shock waves which form at the trailing edge play an important role in determining the base pressure. The results also showed that there is a region of nearly constant losses for Mach numbers ranging from about 1.0 to about 1.15 before the losses start to increase again at Mach numbers above 1.15.

As mentioned earlier, there was a variation in AVDR with variations in incidence. The AVDR increased with increasing positive incidence and decreased with increasing negative incidence. At subsonic Mach numbers, the losses increased slightly with increasing AVDRs. At Mach numbers above 1.0, there was a slight decrease in losses with increasing AVDRs. The computations of Saroch (1996) on the effects of AVDR on the performance of axial turbine cascades at transonic Mach numbers showed similar effects.

Within the uncertainty of the measurements, HS1A and HS1B gave very similar loss results at all conditions of M_2 and incidence, with HS1B giving perhaps slightly higher losses in the minimum loss region (i.e. $M_2 = 0.85$). Thus, the hoped for improvements in positive incidence loss behaviour for HS1B did not materialize. The comparison between HS1A and HS1B cascades could have been more conclusive if both cascades had similar AVDRs at all conditions of M_2 and incidence.

The comparisons between the measurements and the existing profile loss and exit flow angle correlations clearly show the need for improved correlation through the transonic Mach number range. At design conditions and subsonic Mach numbers, the Kacker & Okapuu (1982) correlation correctly predicts the reduction in mid-span losses for modern turbine designs. However, the influence of the Reynolds number does not seem to be reliably predicted. At off-design conditions, the Benner et al. (1997) correlation showed reasonable agreement with the measurements for AVDRs

close to 1.0. The Benner et al. correlation uses the ratio $(\cos\beta_1/\cos\beta_2)$ as a measure of the flow acceleration within the blade passage. Replacing $\cos\beta_1$ by $(\cos(\alpha_1)_{des})$ in the cosine ratio and using the same formulations as Benner et al. used, the agreement with the measurements was even better. This is therefore suggested as an improvement to the Benner et al. correlation.

At subsonic Mach numbers, the deviation correlation of Islam & Sjolander (1999) gave good agreement with the measurements for all test conditions for which the AVDRs were close to 1.0. For values of AVDR larger than about 1.10, it is clear that the dependence on AVDR for the Islam & Sjolander correlation is too strong. Thus, the present data, and perhaps with additional data on the effects of AVDR, could be used to refine the Islam & Sjolander correlation and improve its ability to predict the effects of AVDR. The Islam & Sjolander correlation may be used as a substitute to Ainley & Mathieson correlation for Mach numbers less than 0.5.

6.2 Recommendations

The present study represents a significant contribution to axial turbine research at transonic Mach numbers and off-design incidence values. However, in the course of this study several areas requiring additional investigations or improvements became apparent. The following recommendations are made regarding future work on this topic.

As mentioned repeatedly, the AVDR values in this study were different from the desirable value of 1.0. Sieverding recommended a minimum aspect ratio (H/C) of about 1.6 for two-dimensional, transonic testing of linear turbine cascades. This recommendation appears to be satisfactory for testing near the design incidence. However, for large values of off-design incidence the axial velocity density ratio in the present cascade deviated significantly from 1.0. For $+14.5^\circ$ in particular the AVDR reached about 1.25 and this apparently reduced the losses substantially below what they would have been for more nearly two-dimensional conditions.

Some cascade facilities use endwall suction or blowoff to control the thickness of the endwall boundary layers and therefore the AVDR. Blowoff was not used in the current test series,

although provision for blowoff exists on the endwalls ahead of the cascade. Thus, blowoff should be tried to control the AVDR values, particularly at high positive incidence values. The incorporation of blowoff might require some tuning of the wind tunnel control system.

Computational studies using a Navier-Stokes solver (e.g. the Dawes code (1988)) might also provide some useful insights into the effects of AVDR on the aerodynamic performance of axial turbines cascade at off-design conditions and transonic Mach numbers, and also the factors which influence AVDR.

The Navier-Stokes solver might also be used to extrapolate the current experimental data base to include other aspects of the blade geometry which may affect the blade mid-span losses at off-design conditions. The current experimental data should be used to confirm the code's ability to resolve the effects of off-design incidence.

The wind tunnel test section should be modified to include a flow visualization technique (e.g. Shadow-Graph or Schlieren system) which might help in the interpretation of the loading and subsequently the loss measurements. The flow visualization may also be used to help understand the blade boundary layer development and the shock boundary layer interaction at the transonic flow conditions.

In this study, a single tap was used to measure the trailing edge base pressure. At high Mach numbers the base pressure probably varies considerably with location on the trailing edge and the present single tap gives somewhat a limited picture of the base pressure behaviour. Thus, at a given test flow conditions, additional experiments should be conducted on blades instrumented with static taps at different locations in the trailing edge region. These experiments may be helpful for understanding the patterns of expansion and the shock waves which form at the trailing edge and their relationship to the base pressure.

At design conditions, the Reynolds numbers correction used in the profile loss correlation of Kacker & Okapuu (1982) proved to be inadequate for the present blades. The correlation could be revised to capture better the low Reynolds numbers effects observed in this study. At off-design conditions, any reworking of Benner et al. (1997) correlation should examine

whether it should be the channel acceleration at the particular operating point, as opposed to the design point, which should appear in the incidence parameter χ .

Using the present experimental data and perhaps additional computational data, the deviation correlation of Islam & Sjolander should be reexamined for the effects of AVDR.

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APPENDIX A: Loss Calculation Method

A.1 Introduction

This appendix describes the method of Amecke & Šafařík (1995) which was used in this study to calculate the mixed-out flow properties downstream of the cascade blade row. Also presented is a comparison between the loss values calculated based on flow properties determined using area averaging, mass averaging, and fully mixed-out mass averaging.

A.2 Amecke & Šafařík Method

The Amecke & Šafařík method assumes a mixing process to uniform conditions in a constant area channel with frictionless parallel walls. Figure A.1 shows the nomenclature used in the calculation procedure. Except for definition of the exit flow angle, the nomenclature shown in Figure A.1 is the same one used by Amecke & Šafařík. The method consists of replacing the measured flow quantities which are a function of the coordinate y , by homogeneous values describing the flow at a plane parallel to the cascade and far downstream. Assuming a control volume that extends from the measuring plane to the downstream plane, the homogeneous flow properties are evaluated by solving the equations of conservation of mass, momentum, and energy simultaneously

Referring to Figure A.1, the conservation of mass flux is established by the continuity equation as follows:

$$\rho_2 w_2 \cos \alpha_2 = \int_0^1 \rho_{2y} w_{2y} \cos \alpha_{2y} d(y/s) \quad (\text{A.1})$$

where ρ is the air density, w is the air flow velocity, and β is the flow angle. The subscripts (2) and (2y) refer to the mixed-out and measured flow properties respectively.

Since we are dealing with two-dimensional flow, the momentum balance is established for the axial direction (x-direction) which is normal to the cascade plane, and the pitchwise direction (y-direction) which is tangent to the cascade plane.

In the axial direction, the momentum equation is given by the following equation:

$$\rho_2 w_2^2 \cos^2 \alpha_2 + P_2 = \int_0^1 (\rho_{2y} w_{2y}^2 \cos^2 \alpha_{2y} + P_{2y}) d(y/s) \quad (\text{A.2})$$

where P is the static pressure.

In the pitchwise direction, the momentum equation is given by the following equation:

$$\rho_2 w_2^2 \cos \alpha_2 \sin \alpha_2 = \int_0^1 \rho_{2y} w_{2y}^2 \cos \alpha_{2y} \sin \alpha_{2y} d(y/s) \quad (\text{A.3})$$

The final equation is the energy equation, and is given by:

$$h_{02} \rho_2 w_2 \cos \alpha_2 = \int_0^1 h_{02y} \rho_{2y} w_{2y} \cos \alpha_{2y} d(y/s) \quad (\text{A.4})$$

where h_0 is the specific stagnation or total enthalpy.

The above equations are the general conservation equations. However, Amecke & Šafařík used a non-dimensional version of the above equations in their solution procedure.

Figure A.1 Nomenclature used in the Calculation Procedure.

To transform to a non-dimensional form the conservation of mass equation, Amecke & Šafařík used the critical mass flow rates ($\rho^* a^*$) evaluated at the upstream and downstream locations of the cascade blade row. The critical mass flow rate is used to define the non-dimensional mass flow rate as follows:

$$\Theta = \frac{\rho w}{\rho^* a^*} \quad (\text{A.5})$$

where the superscript (*) refers to the critical values, and (a) is the speed of sound. Using the non-dimensional mass flow rate given by Eqn. (A.5), the conservation of mass equation is rearranged to give the following equation:

$$\left(\frac{\rho_2^* a_2^*}{\rho_1^* a_1^*} \right) \Theta_2 \cos \alpha_2 = \int_0^1 \left(\frac{\rho_{2y}^* a_{2y}^*}{\rho_1^* a_1^*} \right) \Theta_{2y} \cos \alpha_{2y} d(y/s) \quad (\text{A.6})$$

However, the product ($\rho^* a^*$) can be expressed as a function of the total temperature and the total pressure as follows:

$$\rho^* a^* = \frac{P_0}{\sqrt{T_0}} \sqrt{\frac{\gamma}{R} \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma + 1}{\gamma - 1}}} = \frac{P_0}{\sqrt{T_0}} K \quad (\text{A.7})$$

The value of K in Eqn. (A.7) is assumed to be constant throughout the flow field. Moreover, Eqn. (A.7) assumes air to behave as an ideal gas with constant specific heat. The assumption of constant specific heats is assumed throughout the entire solution procedure.

Substituting the expressions for the critical mass flow rates given by Eqn. (A.7) corresponding to the upstream and downstream locations of the cascade blade row, the non-dimensional continuity equation is then rearranged to give:

$$\frac{K_2}{K_1} \frac{P_{02}}{P_{01}} \sqrt{\frac{T_{01}}{T_{02}}} \Theta_2 \cos \alpha_2 = I_M = \int_0^1 \frac{K_{2y}}{K_1} \frac{P_{02y}}{P_{01}} \sqrt{\frac{T_{01}}{T_{02y}}} \Theta_{2y} \cos \alpha_{2y} d(y/s) \quad (\text{A.8})$$

Although included in Eqn. (A.8), the K s are assumed constant throughout the flow field.

Similar to the treatment used for the conservation of mass equation, the energy equation is transformed into a non-dimensional form to give:

$$\frac{C_{P2}}{C_{P1}} \frac{K_2}{K_1} \frac{P_{02}}{P_{01}} \sqrt{\frac{T_{01}}{T_{02}}} \Theta_2 \cos \alpha_2 = I_E = \int_0^1 \frac{C_{P2y}}{C_{P1}} \frac{K_{2y}}{K_1} \frac{P_{02y}}{P_{01}} \sqrt{\frac{T_{01}}{T_{02y}}} \Theta_{2y} \cos \alpha_{2y} d(y/s) \quad (\text{A.9})$$

The left hand sides of Eqns. (A.8) and (A.9) are the evaluations of the integrals I_M and I_E respectively at the downstream plane.

Eqns. (A.8) and (A.9) are used to solve for the downstream mixed-out total temperature, which is given by the following equation:

$$T_{02} = T_{01} \frac{C_{P1}}{C_{P2}} \frac{I_E}{I_M} \quad (\text{A.10})$$

For the momentum equations, the dynamic pressure and the stagnation pressure were used to transform the equations into a non-dimensional form as follows:

a) in the axial direction, the non-dimensional momentum equation is:

$$\frac{P_{02}}{P_{01}} \left(2 \frac{q_2}{P_{02}} \cos^2 \alpha_2 + \frac{P_2}{P_{02}} \right) = I_A = \int_0^1 \frac{P_{02y}}{P_{01}} \left(2 \frac{q_{2y}}{P_{02y}} \cos^2 \alpha_{2y} + \frac{P_{2y}}{P_{02y}} \right) d(y/s) \quad (\text{A.11})$$

b) and, in the tangential direction, the non-dimensional momentum equation is:

$$\frac{P_{02}}{P_{01}} \left(2 \frac{q_2}{P_{02}} \cos \alpha_2 \sin \alpha_2 \right) = I_C = \int_0^1 \frac{P_{02y}}{P_{01}} \left(2 \frac{q_{2y}}{P_{02y}} \cos \alpha_{2y} \sin \alpha_{2y} \right) d(y/s) \quad (\text{A.12})$$

where the non-dimensional dynamic pressure is given by:

$$\frac{q_2}{P_{02}} = \frac{\gamma_2}{\gamma_2 - 1} \left(\frac{P_2}{P_{02}} \right)^{\frac{1}{\gamma_2}} \left(1 - \left(\frac{P_2}{P_{02}} \right)^{\frac{\gamma_2 - 1}{\gamma_2}} \right) \quad (\text{A.13})$$

Eqn. (A.13) is also used to calculate the non-dimensional dynamic pressure variation along the pitchwise direction.

To simplify further the solution procedure of the non-dimensional conservation equations, Amecke & Šafařík expressed all the dimensionless gas dynamic functions in terms of the critical Mach number as follows:

$$\Theta_2 = \left(1 - \frac{\gamma_2 - 1}{\gamma_2 + 1} M_2^{\star 2} \right)^{\frac{1}{\gamma_2 - 1}} \left(\frac{\gamma_2 + 1}{2} \right)^{\frac{1}{\gamma_2 - 1}} M_2^{\star 2}, \quad (\text{A.14a})$$

$$\frac{q_2}{P_{02}} = \left(1 - \frac{\gamma_2 - 1}{\gamma_2 + 1} M_2^{\star 2} \right)^{\frac{1}{\gamma_2 - 1}} \left(\frac{\gamma_2}{\gamma_2 + 1} \right) M_2^{\star 2}, \quad (\text{A.14b})$$

$$\frac{P_2}{P_{02}} = \left(1 - \frac{\gamma_2 - 1}{\gamma_2 + 1} M_2^{\star 2} \right)^{\frac{1}{\gamma_2 - 1}} \left(1 - \frac{\gamma_2 - 1}{\gamma_2 + 1} M_2^{\star 2} \right) \quad (\text{A.14c})$$

The solution for the system of the non-dimensional conservation equations given by Eqns. (A.8), (A.11), and (A.12) is reduced to the quadratic equation in the critical Mach number.

This quadratic equation is as follows:

$$\left(\frac{2}{\gamma_2 + 1} \right)^{\frac{2}{\gamma_2 - 1}} (1 + 2M_2^{*2} + M_2^{*4}) - \left(\frac{\gamma_2 + 1}{\gamma_2} - \frac{\gamma_2 - 1}{\gamma_2} M_2^{*2} \right) \frac{I_C^2}{I_M^{*2}} -$$

$$M_2^{*2} \frac{I_A^2 + 2I_C^2}{I_M^{*2}} + \frac{1}{\gamma_2^2} \left(\frac{\gamma_2 + 1}{2} \right)^{\frac{2\gamma_2}{\gamma_2 - 1}} \frac{I_C^2}{I_M^{*2}} \frac{I_A^2 + I_C^2}{I_M^{*2}} = 0 \quad (\text{A.15})$$

where:

$$I_M^* = \frac{P_{02}}{P_{01}} \Theta_2 \cos \alpha_2 = I_M \frac{K_1}{K_2} \sqrt{\frac{T_{02}}{T_{01}}}$$

The different solutions for the critical Mach number from Eqn. (A.15) are:

$$M_2^{*2} = \left(\frac{\gamma_2 + 1}{\gamma_2} \right)^{\frac{2}{\gamma_2 - 1}} \frac{I_A^2}{I_M^{*2}} x$$

$$\left(\frac{1}{2} - \left(\frac{2}{\gamma_2 + 1} \right)^{\frac{2}{\gamma_2 - 1}} \frac{I_M^{*2}}{I_A^2} + \frac{\gamma_2 + 1}{2\gamma_2} \frac{I_C^2}{I_A^2} \pm \sqrt{\frac{1}{4} - \left(\frac{2}{\gamma_2 + 1} \right)^{\frac{2}{\gamma_2 - 1}} \frac{I_M^{*2}}{I_A^2} + \frac{\gamma_2^2 - 1}{4\gamma_2^2} \frac{I_C^2}{I_A^2}} \right) \quad (\text{A.16})$$

If limit loading is not reached, Eqn. (A.16) should lead to only one possible solution for the critical Mach number. This solution is the minimum of the positive values obtained from Eqn. (A.16) for the critical Mach number. Amecke & Šafařík give more details on the interpretation of the different critical Mach number solutions. In addition, they discuss the choice of the critical Mach number solution for different operating conditions, particularly limit loading.

Having calculated the mixed-out critical Mach number from Eqn. (A.16), the mixed-out downstream gas dynamic properties are calculated using Eqns. (A.14a, 14b, and 14c). The mixed-out downstream total temperature is calculated using Eqn. (A.10), and the actual mixed-out downstream Mach number and exit flow angle are calculated using the following equations respectively:

$$M_2 = M_2^* \sqrt{\frac{2}{(\gamma_2 + 1) - (\gamma_2 - 1) M_2^{*2}}} \quad (\text{A.17})$$

$$\alpha_2 = \sin^{-1} \left(\frac{1}{\gamma_2 M_2^*} \left(\frac{\gamma_2 + 1}{2} \right)^{\frac{\gamma_2}{\gamma_2 - 1}} \frac{I_C}{I_M^*} \right) \quad (\text{A.18})$$

Finally, the profile losses are calculated based on the mixed-out downstream flow properties. The equations for the different loss coefficients are given in Chapter 2.

A.3 A Comparison Between the Different Averaging Methods

Other methods often used to average the downstream flow properties are area averaging and mass averaging. When used in the evaluation of the loss coefficients, the area as well as the mass averaging method lead to different results.

The area-averaged flow properties are evaluated using the following equation:

$$PROP = \frac{\int_A (PROP) dA}{\int_A dA} \quad (A.19)$$

Or using the nomenclature shown in Figure A.1, Eqn. (A.19) is rearranged to give:

$$PROP_2 = \frac{\int_0^1 (PROP_{2y}) d(y/s)}{\int_0^1 d(y/s)} = \int_0^1 (PROP_{2y}) d(y/s) \quad (A.20)$$

where (PROP) refers to any of the downstream flow properties (i.e. static pressure, total pressure, Mach number, total temperature, and flow angle).

The mass-averaged flow properties are calculated using the following equation:

$$PROP = \frac{\int_{\dot{m}} (PROP) d\dot{m}}{\int_{\dot{m}} d\dot{m}} \quad (A.21)$$

Using the nomenclature shown in Figure (A.1) and the downstream flow properties, Eqn. (A.21) is transformed into the following equation:

$$PROP_2 = \frac{\int_0^1 (PROP_{2y}) \left(P_{2y} \frac{1}{\sqrt{T_{02y}}} M_{2y} \cos \alpha_2 \left(1 + \frac{\gamma_{2y} - 1}{2} M_{2y}^2 \right)^{\frac{1}{2}} \right) d(y/s)}{\int_0^1 \left(P_{2y} \frac{1}{\sqrt{T_{02y}}} M_{2y} \cos \alpha_2 \left(1 + \frac{\gamma_{2y} - 1}{2} M_{2y}^2 \right)^{\frac{1}{2}} \right) d(y/s)} \quad (A.22)$$

To investigate the effect of the averaging scheme on the calculated loss coefficients, an investigation was conducted based on a simplified downstream wake distribution. A triangular shape wake profile was assumed in the calculations since it closely approximates a typical turbine blade profile. Initially, the calculations were performed with the assumption that the exit flow angle distribution is constant across the pitchwise direction. The downstream temperature was also assumed constant, and was taken to be equal to the one upstream of the blade row. A typical wake profile based on these assumptions is shown in Figure A.2.

Figure A.3 shows a comparison between the area averaged, mass averaged, and mixed-out loss values. The mixed-out mass averaged value is taken as the “correct” value and is used to normalize the corresponding measurement plane mass-averaged and area-averaged values. The calculations were for five values of V_{min}/V_{max} and three wake widths. The calculated losses using the three methods were found to vary with both the value of V_{min}/V_{max} and the wake width relative to the pitch distance.

Figure A.2 Example Simplified Wake Profile with Constant Flow Angle.

Figure A.3 A Comparison between the Three Loss Calculation Procedures.

The figure shows that for a symmetric wake with constant exit flow angle, the mass averaged losses underestimate and the area averaged losses overestimate the true values. For a wake width of 30% of the pitch and for a value of $V_{min}/V_{max} = 0.80$, the mass averaged loss is low by roughly 7%, and the area averaged loss is high by roughly 5% compared to the mixed out loss value. The mass averaged losses are lower since they do not account for the additional losses due to the downstream mixing of the wake. In the area averaging method, each stream tube in the wake region is assumed to carry the same mass flow rate. Consequently in this case, the computed loss is expected to be higher than the mass-averaged loss. However, the figure shows that the area averaged losses are higher than the mixed-out ones as well.

To investigate the effects of wake symmetry on the losses, two wakes were considered; one wake with asymmetry to the left and another with asymmetry to the right. Figure A.4 shows both wakes investigated. For these cases, the exit flow angle was kept constant. For the two asymmetric wakes, the calculated losses were identical to the ones shown in Figure A.3. This indicates that when the exit flow angle is constant, the wake asymmetry has no effect on the losses.

Figure A.4 Left and Right Wake Asymmetry.

The effect of the exit flow angle distribution within the wake on the calculated losses was investigated for both symmetric and asymmetric wakes.

Figure A.5 shows a typical symmetric wake with exit flow angle variation within the wake. The loss calculations were conducted for wakes with maximum angle variation ($\pm\Delta\alpha_2$) within the wake of ± 3 degrees, and ± 5 degrees. These angle flow variations are typical for turbine blade wake profiles. For each exit flow angle variation, the mixed-out losses were again taking as the true loss values. The comparison is then performed between the true losses and the losses obtained for the same wakes with and without angle variation, including the mixed-out losses with no angle variation.

Figure A.5 A Symmetric Wake with Exit Flow Angle Variation
 $\Delta\alpha_2 = \pm 3$ degrees.

Figures A.6 and A.7 show the comparison between the loss coefficients for the angle variations of ± 3 degrees, and ± 5 degrees respectively. The figures show that the mass averaged losses for wakes with and without angle variation are virtually the same. However, for higher values of angle variation, the mixed-out losses increased, and for some wake profiles the mixed-out losses were higher than the area averaged ones. The figures also show that for wakes with angle flow variation, the mass averaged losses were further reduced.

The effect of wake asymmetry was also investigated for wakes with flow angle variation. For each of the maximum flow angle variations mentioned earlier, two wake asymmetries were considered: namely wakes with the crest located to the left and located to the right of the corresponding symmetric wake centerline. Figure A.8 shows the profiles for the two asymmetric wakes considered. The figure is for a wake width of 30% relative to the pitch distance and a velocity ratio $V_{\min}/V_{\max}$ of 0.80. As before, the mixed-out loss values obtained for the wakes with the angle variation are considered to be the true values. Figures A.9 and A.10 shows the loss comparison for the asymmetric wakes considered with and without the angle variation. The figures show that when the exit flow angle is included in the analysis, the wake asymmetry has an effect on the calculated losses. For wakes with the asymmetry to the left, the mixed out losses were found to increase. This is due to the increase in mass flow rate associated with the decrease in average flow angle in the wake region. The opposite is true for wakes with the asymmetry to the right. However, in general the mixed-out losses were found to increase with increasing levels of angle flow variation within the wake region.

In conclusion, the analysis performed on the loss calculations procedures showed that the mass averaged losses underestimate the true losses whereas the area averaged losses may overestimate the losses, especially in the case of wakes with minimal to no flow angle variation within the wake region.

Figure A.6 Losses for a Symmetric Wake with Exit Flow
Angle Variation of $\Delta\alpha_2 = \pm 3$ degrees.

Figure A.7 Losses for Symmetric Wake with Exit Flow
Angle Variation of $\Delta\alpha_2 = \pm 5$ degrees.

Figure A.8 The Two Asymmetric Wake Profiles considered,
 $w/s = 0.30$, and $V_{min}/V_{max} = 0.80$.

Figure A.9(a) Losses for a Left Asymmetric Wake with Exit
Flow Angle Variation of $\Delta\alpha_2 = \pm 3$ degrees.

Figure A.9(b) Losses for a Right Asymmetric Wake with Exit
Flow Angle Variation of $\Delta\alpha_2 = \pm 3$ degrees.

Figure A.10(a) Losses for a Left Asymmetric Wake with Exit
Flow Angle Variation of $\Delta\alpha_2 = \pm 5$ degrees.

Figure A.10(b) Losses for a Right Asymmetric Wake with Exit
Flow Angle Variation of $\Delta\alpha_2 = \pm 5$ degrees.

APPENDIX B: HS1A and HS1B Blade Coordinates

Table B.1 HS1A Pratt & Whitney Blade Coordinates.

Suction Surface		Pressure Surface	
x (in)	y (in)	x (in)	y (in)
-0.26634	-0.01086	-0.26634	-0.01086
-0.26339	0.0005	-0.26339	-0.0213
-0.26044	0.00748	-0.26044	-0.02504
-0.25749	0.01379	-0.25749	-0.02746
-0.25454	0.01958	-0.25454	-0.0291
-0.25159	0.02494	-0.25159	-0.03015
-0.23684	0.04702	-0.23684	-0.02846
-0.22209	0.06397	-0.22209	-0.02021
-0.20734	0.0776	-0.20734	-0.01345
-0.19259	0.08881	-0.19259	-0.00798
-0.17784	0.09809	-0.17784	-0.00365
-0.16309	0.10576	-0.16309	-0.0003
-0.14834	0.11203	-0.14834	0.00208
-0.13359	0.11702	-0.1336	0.00362
-0.11884	0.12082	-0.11884	0.00434
-0.10409	0.12351	-0.10408	0.00428
-0.08934	0.1251	-0.08934	0.00346
-0.07459	0.12563	-0.07458	0.0019
-0.05984	0.1251	-0.05984	-0.0004

Table B.1 HS1A Pratt & Whitney Blade Coordinates, Continued.

-0.04509	0.1235	-0.04509	-0.00336
-0.03034	0.1208	-0.03034	-0.00705
-0.01559	0.11697	-0.01559	-0.01142
-0.0008	0.11194	-0.0008	-0.01649
0.01391	0.10565	0.01391	-0.02224
0.02866	0.098	0.02866	-0.02869
0.04341	0.08886	0.04341	-0.03583
0.05816	0.0781	0.05816	-0.04367
0.07291	0.06559	0.07291	-0.05222
0.08766	0.05121	0.08766	-0.06147
0.10241	0.035	0.10241	-0.07145
0.11716	0.01724	0.11716	-0.08216
0.13191	-0.00154	0.13191	-0.09361
0.14666	-0.02097	0.14666	-0.10582
0.16141	-0.04084	0.16141	-0.11881
0.17617	-0.06099	0.17617	-0.1326
0.19091	-0.08128	0.19091	-0.1472
0.20566	-0.10169	0.20566	-0.16263
0.22042	-0.12223	0.22041	-0.17891
0.23516	-0.14294	0.23517	-0.19606
0.24991	-0.16394	0.24991	-0.21407
0.26466	-0.18538	0.26466	-0.23296
0.27941	-0.20749	0.27941	-0.25276
0.29416	-0.23059	0.29416	-0.27346
0.30006	-0.2402	0.30006	-0.282

Table B.1 HS1A Pratt & Whitney Blade Coordinates, Continued.

0.30301	-0.2451	0.30301	-0.28633
0.30596	-0.25007	0.30596	-0.2906
0.30891	-0.25512	0.30891	-0.29302
0.31187	-0.26026	0.31186	-0.29406
0.31481	-0.26546	0.31481	-0.29416
0.31776	-0.27078	0.31776	-0.29334
0.32071	-0.2762	0.32072	-0.29131
0.32366	-0.28422	0.32366	-0.28422

Table B.2 HS1B Pratt & Whitney Blade Coordinates.

Suction Surface		Pressure Surface	
x (in)	y (in)	x (in)	y (in)
-0.26003	-0.01896	-0.26003	-0.01896
-0.25708	-0.00325	-0.25709	-0.0294
-0.25414	0.00683	-0.25413	-0.03314
-0.25118	0.01511	-0.25118	-0.03556
-0.24824	0.02217	-0.24823	-0.0372
-0.24528	0.02845	-0.24528	-0.03826
-0.23053	0.05253	-0.23053	-0.03656
-0.21579	0.06982	-0.21578	-0.0249
-0.20104	0.08324	-0.20103	-0.01506
-0.18629	0.09399	-0.18628	-0.00726
-0.17154	0.10268	-0.17153	-0.00111
-0.15678	0.10969	-0.15678	0.00366
-0.14203	0.11524	-0.14203	0.00721
-0.12728	0.11948	-0.12728	0.00965
-0.11253	0.12253	-0.11254	0.01109
-0.09778	0.12445	-0.09778	0.01157
-0.08303	0.12528	-0.08303	0.01114
-0.06828	0.12505	-0.06827	0.00985
-0.05353	0.12376	-0.05353	0.00771

Table B.2 HS1B Pratt & Whitney Blade Coordinates, Continued.

-0.03878	0.1214	-0.03878	0.00475
-0.02403	0.11796	-0.02403	0.00099
-0.00928	0.11339	-0.00928	-0.00356
0.00547	0.10763	0.00547	-0.00889
0.02022	0.10062	0.02022	-0.015
0.03497	0.09226	0.03497	-0.0219
0.04972	0.08244	0.04972	-0.02957
0.06446	0.07105	0.06447	-0.03803
0.07921	0.05799	0.07923	-0.04727
0.09396	0.04323	0.09397	-0.05729
0.10872	0.02683	0.10872	-0.0681
0.12347	0.00906	0.12347	-0.07971
0.13822	-0.00971	0.13822	-0.09211
0.15297	-0.02912	0.15297	-0.10532
0.16772	-0.04899	0.16772	-0.11935
0.18247	-0.06912	0.18247	-0.13419
0.19722	-0.08941	0.19722	-0.14986
0.21197	-0.10982	0.21197	-0.16634
0.22672	-0.13035	0.22672	-0.18362
0.24147	-0.15105	0.24147	-0.2017
0.25622	-0.17205	0.25622	-0.22054
0.27097	-0.1935	0.27097	-0.24015
0.28571	-0.2156	0.28572	-0.26046
0.30046	-0.23869	0.30047	-0.2815
0.30637	-0.2483	0.30637	-0.29009

Table B.2 HS1B Pratt & Whitney Blade Coordinates, Continued.

0.30931	-0.2532	0.30932	-0.29443
0.31226	-0.25817	0.31226	-0.2987
0.31521	-0.26322	0.31521	-0.30113
0.31817	-0.26836	0.31816	-0.30216
0.32111	-0.27357	0.32111	-0.30226
0.32406	-0.27889	0.32406	-0.30145
0.32701	-0.28431	0.32702	-0.29941
0.32996	-0.29233	0.32996	-0.29233

APPENDIX C: Heat Transfer Model for Blade Passage

C.1 Introduction

This appendix describes the heat transfer model used to estimate the total temperature changes across the cascade as a result of heat transfer from the blades and the wind tunnel walls to the air flow. As mentioned in Chapter 4, the temperature in the storage tanks drops as the air expands during a run. Thus, some heat transfer will be occurring between the blades, the wind tunnel walls and the air stream. Consequently, some total temperature variation will occur across the cascade. Using the heat transfer model presented here, estimates on the downstream cascade total temperatures are calculated. These total temperature estimates are used to evaluate the corresponding changes in the mid-span loss coefficients calculated with the assumption of constant total temperature through the cascade.

C.2 Heat Transfer Model

A simplified model was used in this study to estimate the total temperature changes across the cascade. For simplicity, the heat transfer coefficients for a flat plate were used to estimate the heat transfer from the blade surfaces and the test section walls. These heat transfer coefficients were based on the measured flow quantities across the cascade which account for the effects of flow acceleration within the blade passage. The heat transfer coefficients were evaluated for all experimental test cases.

Figure C.1 shows a simplified schematic diagram of the blade passage control volume.

Applying the first law of thermodynamics to the control volume, the rate of heat transfer that may occur within the blade passage may be calculated using the following equation:

$$\dot{Q} = \dot{m} \left((h_2 - h_1) + \frac{V_2^2}{2} - \frac{V_1^2}{2} \right) \quad (\text{C.1})$$

where, h_2, h_1 : are the enthalpy values upstream and downstream of the cascade,
 V_2, V_1 : are the air velocities upstream and downstream of the cascade, and
 $\dot{m}$: is the mass flow rate.

SS: Suction Surface
 PS: Pressure Surface
 CV: Control Volume

Figure C.1 Control Volume for the Heat Transfer Calculation.

The heat transfer rate in Eqn. (C.1) is the sum of all heat transfers from the four surfaces enclosing the control volume. These include the upper, lower wall of the test section, the suction and pressure surfaces approximated by the side flat plates.

Assuming air to behave as an ideal gas with constant specific heat values, Eqn. (C.1) can be rearranged to give:

$$\frac{\dot{Q}}{\dot{m}C_p} = \left(1 + M_2^2 \frac{\gamma - 1}{2} \right) T_2 - \left(1 + M_1^2 \frac{\gamma - 1}{2} \right) T_1 \quad (C.2)$$

where T_2, T_1 : the downstream and upstream static temperatures,
 M_2, M_1 : the downstream and upstream static temperatures, and
 C_p : specific heat constant for air, and
 γ : specific heat ratio for air.

Eqn. (C.2) is used to solve for the cascade outlet temperature T_2 as follows:

$$T_2 = \frac{\frac{\dot{Q}}{\dot{m}C_p} + \left(1 + M_1^2 \frac{\gamma - 1}{2} \right) T_1}{\left(1 + M_2^2 \frac{\gamma - 1}{2} \right)} \quad (C.3)$$

In Eqn. (C.3), the rate of heat transfer from the blade surfaces and the wind tunnel walls is unknown. To estimate this heat transfer rate, the procedure outlined by Holman (1986) for high speed flows over flat plates is used. This procedure is described in the following paragraphs.

In high speed flows, the viscous dissipation within the boundary layers of surfaces are significant. In this case, the surface temperature may be considerably higher than the free stream. This heating effect results from two situations: a) the increase in fluid temperature as it is brought to

rest at the surface while the kinetic energy of the fluid is converted to internal thermal energy. and b) the heating effect due viscous dissipation.

In the case of an insulated wall, the surface of the wall will be at the maximum temperature known as the adiabatic wall temperature, T_{aw} . However, in the actual case, the wall temperature, T_w , will exist at a lower value than the T_{aw} , since part of the kinetic energy of the fluid as it is brought to rest may be lost in the form of viscous work. Also, the fluid is not brought to rest adiabatically at the surface of the wall. Thus, in this case, the rate of heat transfer from the surface is calculated using:

$$\dot{Q} = \bar{h}A(T_w - T_{aw}) \quad (C.4)$$

where $\bar{h}$ is the average convective heat transfer coefficient over the surface, and A is the surface area of the wall or plate.

At start of runs, the blades and test section walls are roughly at room temperature. This room temperature is typically at 20°C. This temperature was used in the analysis as an upper limit for the wall temperatures. From the experimental results, typical variations in exit Mach numbers were from about 0.5 to about 1.14. Thus, the static temperature that corresponds to the exit Mach number of 1.14 was assumed as the lower limit for the test section walls and blade surfaces temperature. This lower temperature limit is about -60°C.

For each of the surfaces enclosing the control volume shown in Figure C.1, the heat transfer was calculated using Eqn. (C.4) for wall temperatures equal to both the upper and lower limit values (i.e. 20°C, and -60°C) respectively. The adiabatic wall temperature in Eqn. (C.4) is calculated using the recovery factor, r , which accounts for the irreversibilities within the boundary layer. The recovery factor is defined as follows:

$$r = \frac{T_{aw} - T_1}{T_{o1} - T_1} \quad (C.5)$$

where T_{o1} is the upstream total temperature. For gases with Prandtl number, Pr , close to unity such as air, the following equations are used to estimate the recovery factor:

$$\begin{aligned} r &= Pr^{1/2} && , \text{ for laminar boundary layer,} \\ r &= Pr^{1/3} && , \text{ for turbulent boundary layer} \end{aligned} \quad (C.6)$$

The value of the recovery factor from Eqn. (C.6) is used in conjunction with Eqn. (C.5) to calculate T_{aw} .

For Eqn. (C.4), the following correlations for the local convective heat transfer coefficient in high speed flows for flat plates (with zero pressure gradient) are used:

- a) laminar boundary layers, $Re_x \leq 5 \cdot 10^5$,

$$St_x \cdot Pr^{1/3} = 0.332 Re_x^{-1/2} \quad (C.7a)$$

- b) turbulent boundary layers with $5 \cdot 10^5 < Re_x \leq 10^7$, and

$$St_x \cdot Pr^{1/3} = 0.0296 Re_x^{-1/5} \quad (C.7b)$$

- c) turbulent boundary layers with $10^7 < Re_x \leq 10^9$,

$$St_x \cdot Pr^{1/3} = 0.185 \left(\log(Re_x) \right)^{-2.584} \quad (C.7c)$$

where the Stanton number S_x , and the Reynolds number Re_x are defined respectively as follows:

$$S_x = \frac{h_x}{\rho \cdot C_p \cdot V_2} \quad (C.7d)$$

$$Re_x = \frac{\rho^* V_2 x}{\mu^*} \quad (C.7e)$$

The subscript, x , in Eqns (C.7a) to (C.7e) refers to a given location along the plate surface, and the superscript, $*$, refers to the temperature, T^* , at which the fluid properties are evaluated. ρ is the fluid density, and V_2 is the fluid velocity at the downstream location. To account for the large temperature gradients within the boundary layer, as reported by Holman, Eckert (1960) suggested that the temperature T^* should be evaluated using the following:

$$T^* = T_1 + \frac{1}{2}(T_w - T_1) + 0.22(T_{aw} - T_1) \quad (C.8)$$

The average heat transfer coefficient for each surface is calculated as follows:

$$\bar{h} = \frac{\int_0^{x_{crit}} h_{lx} dx + \int_{x_{crit}}^l h_{tx} dx}{\int_0^l dx} \quad (C.9)$$

where h_{lx} , and h_{tx} are the laminar and turbulent local heat transfer coefficients for laminar and turbulent regions in the surface boundary layer respectively. x_{crit} is the distance along the surface where the boundary layer is assumed to become turbulent. x_{crit} is defined as follows:

$$x_{crit} = \frac{Re_{crit} \mu^*}{V_2 \rho^*}, \quad Re_{crit} = 500,000$$

To evaluate the average heat transfer coefficient for each surface, a few iterations may be required, since, initially an estimate of both the Prandtl number and the exit temperature are used.

Once the adiabatic wall temperature and the heat transfer coefficient are evaluated, the rate of heat transfer from the four surfaces enclosing the control volume shown in Figure (C.1) is calculated, and the downstream static temperature is calculated using Eqn. (C.3).

Finally, the downstream total temperature is calculated using the following equation:

$$T_{o2} = T_2 \left(1 + \frac{\gamma - 1}{2} M_2^2 \right) \quad (C.10)$$

Lastly, the effects of changes of the downstream total temperature on the mid-span losses are calculated using the procedure outlined in Appendix A.

APPENDIX D: HS1A Cascade Results

Table D.1(a) HS1A Cascade Performance data at Off-Design Incidence, $i - i_{des} = -10.0^\circ$.

M_2	Y_n	Y_t	ω_t	ξ_t	α_2	AVDR
0.58	0.012	0.067	0.062	0.053	57.0	0.98
0.75	0.016	0.059	0.051	0.039	57.5	1.01
0.88	0.020	0.063	0.052	0.037	58.0	0.98
0.98	0.036	0.104	0.082	0.055	57.3	0.99
1.08	0.041	0.109	0.082	0.051	57.1	0.99
1.18	0.049	0.124	0.089	0.051	56.3	0.99

M_2	P_{s2}/P_{01}	q_2/P_{01}	P_v/P_{01}	C_{bp}	$Re \pm \Delta Re$
0.58	0.788	0.184	0.793	0.030	52000 ± 6800
0.75	0.677	0.268	0.668	-0.031	670000 ± 12000
0.88	0.592	0.321	0.571	-0.067	780000 ± 19000
0.98	0.522	0.350	0.371	-0.431	870000 ± 68000
1.08	0.458	0.377	0.367	-0.242	950000 ± 84000
1.18	0.405	0.392	0.312	-0.238	1020000 ± 135000

Table D.1(b) HS1A Cascade Blade Loading at $M_2 = 0.58$, and $i - i_{des} = -10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.380	-0.224	0.066	0.329	0.076
0.050	0.480	-0.899	0.113	0.277	0.346
0.087	0.544	-1.379	0.157	0.232	0.543
0.136	0.591	-1.753	0.235	0.222	0.585
0.190	0.625	-2.029	0.283	0.220	0.591
0.252	0.685	-2.534	0.340	0.249	0.470
0.310	0.708	-2.730	0.393	0.230	0.551
0.371	0.731	-2.922	0.445	0.234	0.537
0.434	0.753	-3.112	0.504	0.253	0.455
0.493	0.708	-2.722	0.553	0.274	0.358
0.581	0.732	-2.927	0.601	0.293	0.268
0.624	0.709	-2.733	0.645	0.313	0.163
0.662	0.650	-2.236	0.695	0.334	0.050
0.699	0.620	-1.987	0.736	0.359	-0.094
0.736	0.603	-1.849	0.785	0.382	-0.234
0.774	0.596	-1.793	0.828	0.412	-0.426
0.812	0.594	-1.772	0.862	0.439	-0.611
0.847	0.598	-1.806	0.901	0.477	-0.877
0.882	0.607	-1.882	0.941	0.519	-1.189
0.917	0.616	-1.959			
0.953	0.617	-1.961			

Table D.1(c) HS1A Cascade Blade Loading at $M_2 = 0.75$, and $i - i_{des} = -10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.432	-0.201	0.066	0.391	0.008
0.050	0.573	-1.015	0.113	0.362	0.149
0.087	0.656	-1.547	0.157	0.327	0.302
0.136	0.721	-1.971	0.235	0.292	0.446
0.190	0.772	-2.313	0.283	0.280	0.492
0.252	0.852	-2.853	0.340	0.313	0.362
0.310	0.884	-3.067	0.393	0.287	0.466
0.371	0.934	-3.394	0.445	0.297	0.429
0.434	0.979	-3.683	0.504	0.315	0.356
0.493	0.937	-3.410	0.553	0.331	0.287
0.581	0.990	-3.752	0.601	0.350	0.205
0.624	0.956	-3.534	0.645	0.376	0.082
0.662	0.889	-3.099	0.695	0.407	-0.072
0.699	0.831	-2.710	0.736	0.440	-0.241
0.736	0.801	-2.511	0.785	0.481	-0.468
0.774	0.802	-2.517	0.828	0.528	-0.740
0.812	0.802	-2.519	0.862	0.562	-0.946
0.847	0.812	-2.585	0.901	0.616	-1.288
0.882	0.829	-2.697	0.941	0.689	-1.764
0.917	0.852	-2.853			
0.953	0.861	-2.911			

Table D.1(d) HS1A Cascade Blade Loading at $M_2 = 0.88$, and $i - i_{des} = -10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.455	-0.202	0.066	0.412	0.004
0.050	0.599	-0.986	0.113	0.388	0.117
0.087	0.693	-1.545	0.157	0.346	0.296
0.136	0.763	-1.970	0.235	0.299	0.479
0.190	0.820	-2.317	0.283	0.285	0.526
0.252	0.913	-2.883	0.340	0.320	0.398
0.310	0.954	-3.128	0.393	0.302	0.468
0.371	1.013	-3.471	0.445	0.310	0.436
0.434	1.085	-3.882	0.504	0.333	0.349
0.493	1.057	-3.723	0.553	0.356	0.254
0.581	1.161	-4.288	0.601	0.378	0.160
0.624	1.106	-3.993	0.645	0.401	0.057
0.662	1.063	-3.756	0.695	0.437	-0.115
0.699	0.965	-3.194	0.736	0.471	-0.284
0.736	0.912	-2.876	0.785	0.516	-0.522
0.774	0.915	-2.895	0.828	0.570	-0.821
0.812	0.926	-2.960	0.862	0.619	-1.100
0.847	0.954	-3.129	0.901	0.685	-1.495
0.882	0.994	-3.361	0.941	0.774	-2.034
0.917	1.043	-3.646			
0.953	1.096	-3.941			

Table D.1(e) HS1A Cascade Blade Loading at $M_2 = 0.98$, and $i - i_{des} = -10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.465	-0.216	0.066	0.408	0.051
0.050	0.602	-0.944	0.113	0.388	0.140
0.087	0.702	-1.523	0.157	0.346	0.315
0.136	0.773	-1.947	0.235	0.300	0.489
0.190	0.828	-2.275	0.283	0.294	0.511
0.252	0.925	-2.850	0.340	0.324	0.403
0.310	0.962	-3.062	0.393	0.299	0.492
0.371	1.026	-3.427	0.445	0.311	0.449
0.434	1.104	-3.848	0.504	0.324	0.402
0.493	1.094	-3.795	0.553	0.351	0.297
0.581	1.226	-4.467	0.601	0.378	0.185
0.624	1.206	-4.371	0.645	0.406	0.061
0.662	1.173	-4.208	0.695	0.445	-0.120
0.699	1.143	-4.056	0.736	0.480	-0.290
0.736	1.143	-4.056	0.785	0.527	-0.538
0.774	1.041	-3.510	0.828	0.579	-0.819
0.812	0.961	-3.059	0.862	0.623	-1.064
0.847	0.988	-3.214	0.901	0.696	-1.492
0.882	1.036	-3.478	0.941	0.798	-2.094
0.917	1.094	-3.799			
0.953	1.174	-4.214			

Table D.1(f) HS1A Cascade Blade Loading at $M_2 = 1.08$, and $i - i_{des} = -10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.458	-0.213	0.066	0.393	0.096
0.050	0.600	-0.985	0.113	0.378	0.163
0.087	0.699	-1.570	0.157	0.347	0.294
0.136	0.772	-2.015	0.235	0.302	0.469
0.190	0.828	-2.359	0.283	0.284	0.531
0.252	0.924	-2.937	0.340	0.325	0.381
0.310	0.966	-3.186	0.393	0.297	0.488
0.371	1.029	-3.551	0.445	0.308	0.444
0.434	1.108	-3.990	0.504	0.326	0.375
0.493	1.100	-3.947	0.553	0.344	0.305
0.581	1.224	-4.596	0.601	0.371	0.193
0.624	1.211	-4.528	0.645	0.397	0.078
0.662	1.169	-4.315	0.695	0.438	-0.118
0.699	1.143	-4.178	0.736	0.472	-0.287
0.736	1.165	-4.295	0.785	0.526	-0.569
0.774	1.233	-4.639	0.828	0.578	-0.859
0.812	1.325	-5.073	0.862	0.622	-1.115
0.847	1.411	-5.441	0.901	0.691	-1.525
0.882	1.431	-5.522	0.941	0.799	-2.181
0.917	1.207	-4.511			
0.953	1.204	-4.493			

Table D.1(g) HS1A Cascade Blade Loading at $M_2 = 1.18$, and $i - i_{des} = -10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.461	-0.219	0.066	0.392	0.107
0.050	0.605	-0.993	0.113	0.371	0.200
0.087	0.695	-1.520	0.157	0.333	0.356
0.136	0.769	-1.970	0.235	0.293	0.504
0.190	0.832	-2.350	0.283	0.286	0.530
0.252	0.921	-2.882	0.340	0.320	0.406
0.310	0.963	-3.129	0.393	0.294	0.502
0.371	1.028	-3.503	0.445	0.303	0.470
0.434	1.105	-3.928	0.504	0.321	0.402
0.493	1.108	-3.943	0.553	0.346	0.305
0.581	1.220	-4.522	0.601	0.367	0.217
0.624	1.211	-4.479	0.645	0.392	0.109
0.662	1.174	-4.290	0.695	0.436	-0.096
0.699	1.146	-4.147	0.736	0.468	-0.251
0.736	1.159	-4.217	0.785	0.521	-0.529
0.774	1.228	-4.561	0.828	0.573	-0.813
0.812	1.321	-4.996	0.862	0.616	-1.058
0.847	1.409	-5.375	0.901	0.687	-1.472
0.882	1.494	-5.704	0.941	0.791	-2.100
0.917	1.321	-4.998			
0.953	1.314	-4.966			

Table D2(a) HS1A Cascade Performance data at Design Incidence, $i - i_{des} = 0.0^\circ$.

M_2	Y_p	Y_t	ω_t	ξ_t	α_2	AVDR
0.55	0.012	0.073	0.068	0.059	56.7	1.02
0.71	0.016	0.067	0.059	0.047	57.2	1.02
0.85	0.017	0.054	0.045	0.033	57.7	1.02
0.96	0.031	0.091	0.072	0.049	57.1	1.03
1.06	0.032	0.085	0.065	0.041	57.0	1.03
1.14	0.039	0.100	0.073	0.043	57.0	1.01
1.16	0.046	0.118	0.085	0.050	55.9	1.03

M_2	P_{s2}/P_{01}	q_2/P_{01}	P_t/P_{01}	C_{bp}	$Re \pm \Delta Re$
0.55	0.807	0.168	-	-	500000 ± 5400
0.71	0.704	0.247	0.707	0.012	660000 ± 13000
0.85	0.612	0.311	-	-	800000 ± 20000
0.96	0.535	0.346	0.481	-0.156	880000 ± 50000
1.06	0.479	0.374	0.365	-0.304	970000 ± 79000
1.14	0.429	0.389	0.323	-0.272	1040000 ± 97000
1.16	0.415	0.390	-	-	1060000 ± 126000

Table D.2(b) HS1A Cascade Blade Loading at $M_2 = 0.71$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.590	-0.698	0.066	0.306	0.525
0.050	0.704	-1.296	0.113	0.232	0.744
0.087	0.774	-1.675	0.157	0.229	0.751
0.136	0.806	-1.848	0.235	0.229	0.751
0.190	0.821	-1.927	0.283	0.229	0.751
0.252	0.832	-1.987	0.340	0.247	0.704
0.310	0.852	-2.093	0.393	0.256	0.678
0.371	0.871	-2.200	0.445	0.270	0.638
0.434	0.902	-2.366	0.504	0.288	0.585
0.493	0.942	-2.579	0.553	0.304	0.532
0.545	0.939	-2.559	0.601	0.338	0.419
0.581	0.926	-2.492	0.645	0.354	0.359
0.624	0.894	-2.319	0.695	0.384	0.246
0.662	0.839	-2.027	0.736	0.412	0.133
0.699	0.774	-1.675	0.785	0.451	-0.033
0.736	0.742	-1.502	0.828	0.502	-0.266
0.774	0.744	-1.509	0.862	0.537	-0.432
0.812	0.745	-1.515	0.901	0.582	-0.658
0.847	0.753	-1.562	0.941	0.644	-0.977
0.882	0.767	-1.635			
0.917	0.768	-1.642			
0.953	0.777	-1.688			

Table D.2(c) HS1A Cascade Blade Loading at $M_2 = 0.96$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.636	-0.633	0.066	0.349	0.487
0.050	0.779	-1.285	0.113	0.270	0.711
0.087	0.867	-1.694	0.157	0.256	0.745
0.136	0.911	-1.893	0.235	0.258	0.740
0.190	0.926	-1.961	0.283	0.262	0.730
0.252	0.937	-2.010	0.340	0.272	0.706
0.310	0.960	-2.112	0.393	0.287	0.667
0.371	0.987	-2.234	0.445	0.297	0.638
0.434	1.036	-2.448	0.504	0.318	0.579
0.493	1.114	-2.774	0.553	0.343	0.506
0.545	1.177	-3.027	0.601	0.372	0.414
0.581	1.219	-3.188	0.645	0.398	0.326
0.624	1.175	-3.017	0.695	0.434	0.200
0.662	1.145	-2.901	0.736	0.471	0.058
0.699	1.115	-2.779	0.785	0.520	-0.136
0.736	0.992	-2.258	0.828	0.580	-0.389
0.774	0.936	-2.005	0.862	0.626	-0.589
0.812	0.952	-2.078	0.901	0.698	-0.915
0.847	0.987	-2.234	0.941	0.801	-1.387
0.882	1.036	-2.448			
0.917	1.084	-2.652			
0.953	1.164	-2.974			

Table D.2(d) HS1A Cascade Blade Loading at $M_2 = 1.06$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.641	-0.654	0.066	0.348	0.491
0.050	0.784	-1.308	0.113	0.256	0.745
0.087	0.870	-1.704	0.157	0.248	0.763
0.136	0.914	-1.904	0.235	0.252	0.754
0.190	0.931	-1.981	0.283	0.252	0.754
0.252	0.939	-2.017	0.340	0.269	0.713
0.310	0.959	-2.108	0.393	0.283	0.677
0.371	0.985	-2.222	0.445	0.300	0.632
0.434	1.041	-2.467	0.504	0.321	0.572
0.493	1.115	-2.776	0.553	0.343	0.504
0.545	1.178	-3.026	0.601	0.371	0.418
0.581	1.228	-3.217	0.645	0.396	0.332
0.624	1.209	-3.144	0.695	0.434	0.200
0.662	1.169	-2.989	0.736	0.470	0.064
0.699	1.145	-2.894	0.785	0.523	-0.145
0.736	1.158	-2.949	0.828	0.578	-0.377
0.774	1.202	-3.117	0.862	0.624	-0.577
0.812	1.045	-2.481	0.901	0.693	-0.890
0.847	1.009	-2.326	0.941	0.802	-1.390
0.882	1.031	-2.422			
0.917	1.088	-2.662			
0.953	1.177	-3.021			

Table D.2(e) HS1A Cascade Blade Loading at $M_2 = 1.14$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.643	-0.683	0.066	0.338	0.515
0.050	0.782	-1.327	0.113	0.254	0.744
0.087	0.872	-1.742	0.157	0.242	0.774
0.136	0.915	-1.941	0.235	0.247	0.761
0.190	0.930	-2.010	0.283	0.251	0.752
0.252	0.938	-2.049	0.340	0.265	0.718
0.310	0.961	-2.153	0.393	0.282	0.674
0.371	0.986	-2.266	0.445	0.295	0.640
0.434	1.039	-2.495	0.504	0.313	0.588
0.493	1.115	-2.819	0.553	0.339	0.510
0.545	1.178	-3.074	0.601	0.369	0.415
0.581	1.224	-3.251	0.645	0.396	0.324
0.624	1.214	-3.212	0.695	0.437	0.177
0.662	1.171	-3.044	0.736	0.471	0.048
0.699	1.139	-2.918	0.785	0.521	-0.151
0.736	1.159	-2.996	0.828	0.577	-0.389
0.774	1.230	-3.273	0.862	0.623	-0.592
0.812	1.324	-3.610	0.901	0.692	-0.908
0.847	1.405	-3.878	0.941	0.797	-1.397
0.882	1.451	-4.021			
0.917	1.246	-3.333			
0.953	1.305	-3.545			

Table D.3(a) HS1A Cascade Performance data at Off-Design Incidence, $i - i_{des} = 4.5^\circ$.

M_2	Y_p	Y_t	ω_t	ξ_t	α_2	AVDR
0.49	0.011	0.078	0.073	0.066	56.4	1.02
0.64	0.014	0.064	0.058	0.048	56.4	1.07
0.66	0.014	0.064	0.057	0.047	56.6	1.02
0.81	0.018	0.060	0.052	0.039	57.2	1.03
0.91	0.024	0.071	0.058	0.041	57.1	1.04
0.97	0.034	0.097	0.077	0.051	56.8	1.05
1.03	0.036	0.098	0.076	0.049	56.7	1.04
1.12	0.040	0.103	0.076	0.046	57.0	1.02
0.66	0.015	0.067	0.060	0.050	57.0	1.01
0.81	0.017	0.059	0.050	0.038	57.5	1.03
0.91	0.026	0.078	0.064	0.045	57.4	1.03
1.03	0.037	0.100	0.079	0.051	57.0	1.03

Table D.3(a) Continued

M_2	P_s/P_{01}	q_2/P_{01}	P_v/P_{01}	C_{bp}	$Re \pm \Delta Re$
0.49	0.841	0.140	-	-	460000 ± 4000
0.64	0.749	0.215	-	-	600000 ± 8000
0.66	0.736	0.224	-	-	630000 ± 9000
0.81	0.641	0.291	-	-	780000 ± 16000
0.91	0.568	0.333	0.563	-0.016	880000 ± 26000
0.97	0.526	0.349	-	-	920000 ± 66000
1.03	0.490	0.366	0.403	-0.241	980000 ± 90000
1.12	0.440	0.385	0.378	-0.161	1050000 ± 90000
0.66	0.736	0.224	-	-	630000 ± 9000
0.81	0.641	0.292	-	-	770000 ± 14000
0.91	0.568	0.331	0.563	-0.015	880000 ± 28000
1.03	0.491	0.365	0.403	-0.242	970000 ± 93000

Table D.3(b) HS1A Cascade Blade Loading at $M_2 = 0.91$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.814	-0.945	0.066	0.278	0.780
0.050	0.959	-1.477	0.113	0.250	0.835
0.087	1.039	-1.761	0.157	0.242	0.851
0.136	1.032	-1.737	0.235	0.252	0.831
0.190	1.002	-1.631	0.283	0.254	0.827
0.252	0.951	-1.449	0.340	0.262	0.811
0.310	0.962	-1.489	0.393	0.280	0.776
0.371	0.980	-1.552	0.445	0.296	0.740
0.434	1.025	-1.713	0.504	0.314	0.701
0.493	1.093	-1.946	0.553	0.337	0.646
0.545	1.146	-2.119	0.601	0.364	0.579
0.581	1.168	-2.190	0.645	0.390	0.508
0.624	1.088	-1.926	0.695	0.430	0.394
0.662	1.015	-1.678	0.736	0.467	0.284
0.699	0.974	-1.532	0.785	0.513	0.138
0.736	0.940	-1.410	0.828	0.566	-0.039
0.774	0.940	-1.410	0.862	0.611	-0.197
0.812	0.942	-1.418	0.901	0.681	-0.449
0.847	0.968	-1.508	0.941	0.775	-0.800
0.882	1.011	-1.662			
0.917	1.052	-1.804			
0.953	1.095	-1.950			

Table D.3(c) HS1A Cascade Blade Loading at $M_2 = 1.03$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.829	-0.989	0.066	0.287	0.762
0.050	0.978	-1.532	0.113	0.255	0.828
0.087	1.067	-1.838	0.157	0.245	0.846
0.136	1.053	-1.791	0.235	0.255	0.828
0.190	1.019	-1.674	0.283	0.261	0.817
0.252	0.963	-1.477	0.340	0.270	0.799
0.310	0.977	-1.528	0.393	0.286	0.766
0.371	0.993	-1.583	0.445	0.302	0.730
0.434	1.042	-1.755	0.504	0.323	0.682
0.493	1.113	-1.992	0.553	0.347	0.624
0.545	1.170	-2.174	0.601	0.372	0.562
0.581	1.211	-2.302	0.645	0.399	0.489
0.624	1.199	-2.265	0.695	0.439	0.372
0.662	1.130	-2.046	0.736	0.472	0.274
0.699	1.108	-1.977	0.785	0.522	0.113
0.736	1.054	-1.795	0.828	0.574	-0.062
0.774	0.997	-1.598	0.862	0.621	-0.226
0.812	0.990	-1.572	0.901	0.690	-0.474
0.847	1.012	-1.649	0.941	0.797	-0.868
0.882	1.056	-1.802			
0.917	1.109	-1.981			
0.953	1.188	-2.232			

Table D.3(d) HS1A Cascade Blade Loading at $M_2 = 1.12$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.838	-0.996	0.066	0.280	0.782
0.050	0.987	-1.533	0.113	0.249	0.843
0.087	1.076	-1.835	0.157	0.242	0.857
0.136	1.056	-1.767	0.235	0.251	0.840
0.190	1.024	-1.659	0.283	0.251	0.840
0.252	0.959	-1.434	0.340	0.265	0.812
0.310	0.973	-1.482	0.393	0.283	0.775
0.371	0.988	-1.536	0.445	0.292	0.758
0.434	1.037	-1.706	0.504	0.313	0.710
0.493	1.111	-1.951	0.553	0.341	0.646
0.545	1.169	-2.135	0.601	0.364	0.588
0.581	1.214	-2.274	0.645	0.391	0.517
0.624	1.201	-2.233	0.695	0.435	0.394
0.662	1.131	-2.016	0.736	0.469	0.292
0.699	1.116	-1.968	0.785	0.518	0.139
0.736	1.141	-2.046	0.828	0.570	-0.034
0.774	1.221	-2.294	0.862	0.616	-0.194
0.812	1.243	-2.359	0.901	0.686	-0.442
0.847	1.072	-1.822	0.941	0.788	-0.812
0.882	1.064	-1.795			
0.917	1.112	-1.954			
0.953	1.196	-2.220			

Table D.4(a) HS1A Cascade Performance data at Off-Design Incidence, $i - i_{des} = 10.0^\circ$.

M_2	Y_p	Y_r	ω_r	ξ_r	α_2	AVDR
0.49	0.015	0.109	0.102	0.091	56.4	1.15
0.68	0.017	0.074	0.066	0.054	56.8	1.15
0.81	0.019	0.063	0.054	0.040	57.3	1.15
0.92	0.022	0.066	0.054	0.038	57.7	1.14
1.00	0.039	0.110	0.086	0.057	57.0	1.14
1.08	0.041	0.109	0.082	0.051	57.1	1.13
1.15	0.048	0.123	0.089	0.052	56.5	1.13

M_2	P_{s2}/P_{01}	q_2/P_{01}	P_{t2}/P_{01}	C_{hp}	$Re \pm \Delta Re$
0.49	0.834	0.141	0.834	-0.025	440000 ± 4000
0.68	0.721	0.233	0.720	-0.033	620000 ± 9700
0.81	0.636	0.293	0.636	-0.040	740000 ± 15000
0.92	0.566	0.335	0.566	-0.046	840000 ± 24000
1.00	0.507	0.356	0.503	-0.289	920000 ± 71000
1.08	0.458	0.377	0.448	-0.150	1000000 ± 98000
1.15	0.416	0.389	0.405	-0.238	1070000 ± 116000

Table D.4(b) HS1A Cascade Blade Loading at $M_2 = 0.49$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.627	-1.289	0.066	0.185	0.804
0.050	0.672	-1.573	0.113	0.183	0.808
0.087	0.687	-1.669	0.157	0.182	0.813
0.136	0.688	-1.673	0.235	0.183	0.808
0.190	0.678	-1.611	0.283	0.219	0.711
0.252	0.668	-1.546	0.340	0.191	0.789
0.310	0.661	-1.506	0.393	0.208	0.742
0.371	0.665	-1.530	0.445	0.211	0.734
0.434	0.668	-1.548	0.504	0.228	0.684
0.493	0.633	-1.326	0.553	0.242	0.640
0.545	0.668	-1.549	0.601	0.258	0.589
0.581	0.645	-1.404	0.645	0.278	0.518
0.624	0.608	-1.169	0.695	0.298	0.442
0.662	0.578	-0.986	0.736	0.318	0.365
0.699	0.560	-0.877	0.785	0.347	0.239
0.736	0.537	-0.744	0.828	0.369	0.143
0.774	0.536	-0.737	0.862	0.398	0.005
0.812	0.536	-0.736	0.901	0.420	-0.105
0.847	0.536	-0.736	0.941	0.465	-0.337
0.882	0.536	-0.738			
0.917	0.539	-0.752			
0.953	0.539	-0.757			

Table D.4(c) HS1A Cascade Blade Loading at $M_2 = 0.68$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.813	-1.271	0.066	0.246	0.792
0.050	0.901	-1.649	0.113	0.224	0.838
0.087	0.935	-1.791	0.157	0.220	0.847
0.136	0.922	-1.740	0.235	0.234	0.818
0.190	0.893	-1.616	0.283	0.280	0.713
0.252	0.869	-1.513	0.340	0.244	0.797
0.310	0.859	-1.470	0.393	0.261	0.760
0.371	0.862	-1.484	0.445	0.271	0.736
0.434	0.884	-1.576	0.504	0.287	0.697
0.493	0.858	-1.464	0.553	0.311	0.635
0.545	0.914	-1.704	0.601	0.332	0.580
0.581	0.891	-1.605	0.645	0.348	0.534
0.624	0.823	-1.316	0.695	0.378	0.443
0.662	0.789	-1.170	0.736	0.405	0.356
0.699	0.760	-1.041	0.785	0.445	0.221
0.736	0.736	-0.937	0.828	0.490	0.063
0.774	0.734	-0.932	0.862	0.518	-0.044
0.812	0.726	-0.899	0.901	0.564	-0.221
0.847	0.735	-0.936	0.941	0.626	-0.472
0.882	0.739	-0.954			
0.917	0.746	-0.982			
0.953	0.749	-0.994			

Table D.4(d) HS1A Cascade Blade Loading at $M_2 = 0.81$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.891	-1.282	0.066	0.273	0.782
0.050	1.024	-1.769	0.113	0.254	0.819
0.087	1.076	-1.954	0.157	0.252	0.823
0.136	1.048	-1.856	0.235	0.255	0.819
0.190	1.004	-1.698	0.283	0.303	0.716
0.252	0.953	-1.512	0.340	0.269	0.791
0.310	0.945	-1.483	0.393	0.277	0.772
0.371	0.954	-1.519	0.445	0.290	0.744
0.434	1.001	-1.689	0.504	0.318	0.681
0.493	0.979	-1.607	0.553	0.340	0.627
0.545	1.085	-1.984	0.601	0.365	0.561
0.581	1.069	-1.930	0.645	0.390	0.494
0.624	0.982	-1.620	0.695	0.425	0.394
0.662	0.933	-1.438	0.736	0.453	0.309
0.699	0.895	-1.297	0.785	0.498	0.165
0.736	0.874	-1.219	0.828	0.542	0.017
0.774	0.870	-1.202	0.862	0.584	-0.132
0.812	0.874	-1.219	0.901	0.645	-0.351
0.847	0.879	-1.239	0.941	0.722	-0.640
0.882	0.906	-1.337			
0.917	0.917	-1.379			
0.953	0.918	-1.385			

Table D.4(e) HS1A Cascade Blade Loading at $M_2 = 0.92$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.916	-1.227	0.066	0.293	0.765
0.050	1.074	-1.764	0.113	0.268	0.815
0.087	1.154	-2.018	0.157	0.258	0.834
0.136	1.108	-1.874	0.235	0.267	0.816
0.190	1.051	-1.687	0.283	0.313	0.722
0.252	0.985	-1.465	0.340	0.278	0.794
0.310	0.979	-1.445	0.393	0.292	0.767
0.371	0.997	-1.507	0.445	0.311	0.726
0.434	1.041	-1.654	0.504	0.322	0.701
0.493	1.035	-1.636	0.553	0.352	0.630
0.545	1.175	-2.082	0.601	0.380	0.561
0.581	1.196	-2.145	0.645	0.404	0.498
0.624	1.115	-1.896	0.695	0.439	0.399
0.662	1.043	-1.660	0.736	0.477	0.287
0.699	1.001	-1.521	0.785	0.523	0.146
0.736	0.967	-1.403	0.828	0.579	-0.036
0.774	0.964	-1.393	0.862	0.619	-0.173
0.812	0.976	-1.435	0.901	0.692	-0.428
0.847	0.998	-1.509	0.941	0.790	-0.778
0.882	1.036	-1.638			
0.917	1.084	-1.795			
0.953	1.125	-1.925			

Table D.4(f) HS1A Cascade Blade Loading at $M_2 = 1.00$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.923	-1.227	0.066	0.294	0.767
0.050	1.087	-1.777	0.113	0.267	0.820
0.087	1.167	-2.025	0.157	0.259	0.836
0.136	1.119	-1.876	0.235	0.269	0.816
0.190	1.047	-1.645	0.283	0.318	0.716
0.252	0.984	-1.435	0.340	0.283	0.788
0.310	0.985	-1.440	0.393	0.300	0.755
0.371	1.000	-1.490	0.445	0.310	0.732
0.434	1.053	-1.665	0.504	0.328	0.693
0.493	1.053	-1.667	0.553	0.353	0.634
0.545	1.185	-2.080	0.601	0.377	0.575
0.581	1.217	-2.173	0.645	0.409	0.491
0.624	1.195	-2.110	0.695	0.442	0.398
0.662	1.132	-1.916	0.736	0.479	0.290
0.699	1.120	-1.881	0.785	0.525	0.149
0.736	1.143	-1.953	0.828	0.582	-0.035
0.774	1.132	-1.918	0.862	0.629	-0.192
0.812	1.014	-1.537	0.901	0.697	-0.428
0.847	1.027	-1.580	0.941	0.797	-0.782
0.882	1.068	-1.715			
0.917	1.127	-1.902			
0.953	1.199	-2.120			

Table D.4(g) HS1A Cascade Blade Loading at $M_2 = 1.08$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.925	-1.224	0.066	0.288	0.780
0.050	1.095	-1.789	0.113	0.263	0.830
0.087	1.184	-2.063	0.157	0.258	0.839
0.136	1.123	-1.877	0.235	0.260	0.835
0.190	1.035	-1.594	0.283	0.314	0.727
0.252	0.991	-1.448	0.340	0.278	0.801
0.310	0.987	-1.435	0.393	0.289	0.778
0.371	1.001	-1.483	0.445	0.308	0.739
0.434	1.050	-1.645	0.504	0.335	0.678
0.493	1.060	-1.677	0.553	0.360	0.619
0.545	1.185	-2.064	0.601	0.382	0.565
0.581	1.221	-2.169	0.645	0.405	0.505
0.624	1.197	-2.099	0.695	0.443	0.400
0.662	1.130	-1.899	0.736	0.473	0.312
0.699	1.121	-1.871	0.785	0.521	0.167
0.736	1.156	-1.979	0.828	0.579	-0.019
0.774	1.232	-2.202	0.862	0.622	-0.164
0.812	1.323	-2.450	0.901	0.695	-0.414
0.847	1.381	-2.595	0.941	0.796	-0.770
0.882	1.252	-2.259			
0.917	1.191	-2.083			
0.953	1.228	-2.190			

Table D.4(h) HS1A Cascade Blade Loading at $M_2 = 1.15$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.927	-1.246	0.066	0.283	0.788
0.050	1.097	-1.814	0.113	0.258	0.836
0.087	1.185	-2.085	0.157	0.250	0.851
0.136	1.112	-1.861	0.235	0.260	0.833
0.190	1.025	-1.578	0.283	0.308	0.735
0.252	0.985	-1.446	0.340	0.277	0.800
0.310	0.985	-1.445	0.393	0.290	0.774
0.371	1.000	-1.496	0.445	0.305	0.742
0.434	1.048	-1.655	0.504	0.322	0.706
0.493	1.072	-1.732	0.553	0.348	0.645
0.545	1.182	-2.077	0.601	0.376	0.575
0.581	1.218	-2.181	0.645	0.404	0.502
0.624	1.194	-2.112	0.695	0.441	0.401
0.662	1.131	-1.919	0.736	0.478	0.293
0.699	1.122	-1.894	0.785	0.528	0.139
0.736	1.156	-1.998	0.828	0.579	-0.028
0.774	1.233	-2.224	0.862	0.626	-0.185
0.812	1.318	-2.459	0.901	0.694	-0.421
0.847	1.398	-2.661	0.941	0.791	-0.766
0.882	1.403	-2.672			
0.917	1.267	-2.320			
0.953	1.322	-2.470			

Table D.5(a) HS1A Cascade Performance data at Off-Design Incidence, $i - i_{des} = 14.5^\circ$.

M_2	Y_p	Y_t	ω_t	ξ_t	α_2	AVDR
0.51	0.018	0.122	0.114	0.101	55.8	1.25
0.67	0.022	0.095	0.084	0.069	56.2	1.24
0.79	0.022	0.079	0.067	0.051	56.6	1.26
0.91	0.023	0.070	0.057	0.040	56.9	1.27
0.97	0.036	0.102	0.081	0.054	56.7	1.26
1.05	0.036	0.097	0.074	0.047	56.7	1.27
1.10	0.038	0.101	0.075	0.046	56.9	1.25

M_2	P_{s2}/P_{01}	q_2/P_{01}	P_t/P_{01}	C_{hp}	$Re \pm \Delta Re$
0.51	0.822	0.149	0.840	0.117	460000 ± 4800
0.67	0.722	0.229	0.730	0.034	620000 ± 7800
0.79	0.645	0.284	0.646	0.000	740000 ± 17000
0.91	0.574	0.330	0.569	-0.013	870000 ± 24000
0.97	0.526	0.348	0.460	-0.189	940000 ± 45000
1.05	0.478	0.371	0.411	-0.182	1020000 ± 83000
1.10	0.453	0.380	0.410	-0.113	1050000 ± 104000

Table D.5(b) HS1A Cascade Blade Loading at $M_2 = 0.51$, and $i - i_{des} = 14...5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.647	-1.545	0.066	0.165	0.840
0.050	0.667	-1.675	0.113	0.163	0.845
0.087	0.684	-1.789	0.157	0.163	0.845
0.136	0.670	-1.695	0.235	0.166	0.837
0.190	0.655	-1.594	0.283	0.199	0.750
0.252	0.641	-1.506	0.340	0.175	0.815
0.310	0.627	-1.415	0.393	0.190	0.775
0.371	0.628	-1.418	0.445	0.195	0.764
0.434	0.631	-1.440	0.504	0.217	0.699
0.493	0.605	-1.271	0.553	0.228	0.663
0.545	0.618	-1.351	0.601	0.249	0.592
0.581	0.600	-1.233	0.645	0.266	0.534
0.624	0.560	-0.982	0.695	0.286	0.458
0.662	0.544	-0.883	0.736	0.305	0.380
0.699	0.520	-0.732	0.785	0.325	0.294
0.736	0.516	-0.711	0.828	0.357	0.150
0.774	0.506	-0.653	0.862	0.375	0.062
0.812	0.505	-0.647	0.901	0.399	-0.055
0.847	0.508	-0.661	0.941	0.437	-0.257
0.882	0.506	-0.649			
0.917	0.509	-0.671			
0.953	0.517	-0.716			

Table D.5(c) HS1A Cascade Blade Loading at $M_2 = 0.67$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.868	-1.573	0.066	0.236	0.805
0.050	0.921	-1.805	0.113	0.222	0.834
0.087	0.935	-1.861	0.157	0.223	0.833
0.136	0.909	-1.751	0.235	0.231	0.817
0.190	0.874	-1.597	0.283	0.265	0.741
0.252	0.838	-1.438	0.340	0.242	0.792
0.310	0.827	-1.391	0.393	0.252	0.770
0.371	0.833	-1.418	0.445	0.265	0.742
0.434	0.847	-1.481	0.504	0.275	0.715
0.493	0.827	-1.389	0.553	0.301	0.650
0.545	0.858	-1.527	0.601	0.315	0.614
0.581	0.829	-1.400	0.645	0.333	0.562
0.624	0.765	-1.118	0.695	0.365	0.467
0.662	0.743	-1.021	0.736	0.387	0.397
0.699	0.718	-0.911	0.785	0.425	0.272
0.736	0.701	-0.835	0.828	0.464	0.133
0.774	0.700	-0.833	0.862	0.496	0.014
0.812	0.699	-0.829	0.901	0.535	-0.138
0.847	0.700	-0.833	0.941	0.592	-0.371
0.882	0.701	-0.834			
0.917	0.710	-0.874			
0.953	0.711	-0.881			

Table D.5(d) HS1A Cascade Blade Loading at $M_2 = 0.79$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	0.996	-1.592	0.066	0.265	0.809
0.050	1.086	-1.899	0.113	0.250	0.838
0.087	1.125	-2.029	0.157	0.240	0.857
0.136	1.051	-1.780	0.235	0.252	0.833
0.190	0.985	-1.554	0.283	0.288	0.761
0.252	0.942	-1.398	0.340	0.259	0.821
0.310	0.932	-1.363	0.393	0.279	0.780
0.371	0.941	-1.397	0.445	0.295	0.746
0.434	0.970	-1.498	0.504	0.311	0.711
0.493	0.962	-1.470	0.553	0.336	0.651
0.545	1.023	-1.684	0.601	0.357	0.600
0.581	0.993	-1.581	0.645	0.381	0.537
0.624	0.922	-1.326	0.695	0.416	0.440
0.662	0.883	-1.185	0.736	0.442	0.365
0.699	0.860	-1.102	0.785	0.480	0.249
0.736	0.837	-1.015	0.828	0.528	0.096
0.774	0.834	-1.004	0.862	0.568	-0.040
0.812	0.835	-1.006	0.901	0.623	-0.230
0.847	0.843	-1.039	0.941	0.697	-0.499
0.882	0.851	-1.066			
0.917	0.859	-1.097			
0.953	0.863	-1.112			

Table D.5(e) HS1A Cascade Blade Loading at $M_2 = 0.91$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	1.066	-1.632	0.066	0.287	0.791
0.050	1.169	-1.947	0.113	0.263	0.839
0.087	1.262	-2.210	0.157	0.260	0.844
0.136	1.166	-1.937	0.235	0.273	0.819
0.190	1.011	-1.458	0.283	0.308	0.748
0.252	0.988	-1.382	0.340	0.282	0.802
0.310	0.981	-1.357	0.393	0.302	0.762
0.371	0.993	-1.399	0.445	0.310	0.745
0.434	1.036	-1.538	0.504	0.327	0.709
0.493	1.048	-1.575	0.553	0.357	0.639
0.545	1.147	-1.882	0.601	0.379	0.586
0.581	1.139	-1.859	0.645	0.401	0.530
0.624	1.052	-1.587	0.695	0.446	0.410
0.662	1.009	-1.451	0.736	0.473	0.332
0.699	0.971	-1.326	0.785	0.519	0.195
0.736	0.948	-1.249	0.828	0.570	0.036
0.774	0.949	-1.251	0.862	0.615	-0.109
0.812	0.952	-1.263	0.901	0.682	-0.335
0.847	0.974	-1.334	0.941	0.772	-0.645
0.882	1.008	-1.446			
0.917	1.027	-1.509			
0.953	1.028	-1.511			

Table D.5(f) HS1A Cascade Blade Loading at $M_2 = 0.97$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{ts}	C_{ps}	x/C_x	M_{ts}	C_{ps}
0.016	1.077	-1.618	0.066	0.297	0.778
0.050	1.190	-1.953	0.113	0.269	0.832
0.087	1.299	-2.248	0.157	0.257	0.855
0.136	1.205	-1.995	0.235	0.278	0.817
0.190	1.017	-1.431	0.283	0.314	0.744
0.252	0.995	-1.358	0.340	0.291	0.790
0.310	0.995	-1.358	0.393	0.313	0.746
0.371	1.007	-1.398	0.445	0.318	0.735
0.434	1.052	-1.542	0.504	0.336	0.696
0.493	1.075	-1.612	0.553	0.362	0.637
0.545	1.183	-1.932	0.601	0.384	0.584
0.581	1.202	-1.986	0.645	0.409	0.520
0.624	1.152	-1.843	0.695	0.450	0.412
0.662	1.097	-1.678	0.736	0.483	0.319
0.699	1.096	-1.678	0.785	0.531	0.178
0.736	1.067	-1.587	0.828	0.586	0.005
0.774	1.018	-1.434	0.862	0.629	-0.134
0.812	1.011	-1.412	0.901	0.697	-0.359
0.847	1.037	-1.494	0.941	0.799	-0.705
0.882	1.075	-1.612			
0.917	1.126	-1.766			
0.953	1.196	-1.969			

Table D.5(g) HS1A Cascade Blade Loading at $M_2 = 1.05$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	1.091	-1.634	0.066	0.286	0.804
0.050	1.189	-1.921	0.113	0.268	0.839
0.087	1.309	-2.243	0.157	0.263	0.848
0.136	1.193	-1.934	0.235	0.270	0.835
0.190	1.018	-1.410	0.283	0.302	0.773
0.252	0.995	-1.338	0.340	0.279	0.817
0.310	0.994	-1.333	0.393	0.298	0.781
0.371	1.004	-1.364	0.445	0.313	0.751
0.434	1.050	-1.509	0.504	0.334	0.705
0.493	1.080	-1.602	0.553	0.358	0.650
0.545	1.187	-1.915	0.601	0.377	0.606
0.581	1.208	-1.974	0.645	0.403	0.542
0.624	1.161	-1.841	0.695	0.443	0.437
0.662	1.103	-1.673	0.736	0.474	0.352
0.699	1.120	-1.722	0.785	0.526	0.199
0.736	1.152	-1.817	0.828	0.579	0.039
0.774	1.223	-2.016	0.862	0.624	-0.104
0.812	1.249	-2.088	0.901	0.693	-0.331
0.847	1.132	-1.758	0.941	0.795	-0.675
0.882	1.098	-1.658			
0.917	1.133	-1.762			
0.953	1.208	-1.974			

Table D.5(h) HS1A Cascade Blade Loading at $M_2 = 1.10$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.016	1.082	-1.657	0.066	0.275	0.819
0.050	1.187	-1.971	0.113	0.249	0.866
0.087	1.306	-2.294	0.157	0.244	0.875
0.136	1.184	-1.962	0.235	0.258	0.849
0.190	1.015	-1.444	0.283	0.300	0.770
0.252	0.995	-1.379	0.340	0.272	0.824
0.310	0.990	-1.364	0.393	0.289	0.792
0.371	0.999	-1.394	0.445	0.300	0.770
0.434	1.047	-1.549	0.504	0.322	0.724
0.493	1.074	-1.633	0.553	0.346	0.669
0.545	1.178	-1.945	0.601	0.373	0.607
0.581	1.199	-2.003	0.645	0.396	0.549
0.624	1.151	-1.865	0.695	0.435	0.447
0.662	1.098	-1.707	0.736	0.470	0.348
0.699	1.116	-1.760	0.785	0.519	0.205
0.736	1.151	-1.866	0.828	0.574	0.033
0.774	1.225	-2.076	0.862	0.623	-0.124
0.812	1.305	-2.292	0.901	0.687	-0.339
0.847	1.381	-2.480	0.941	0.788	-0.683
0.882	1.368	-2.448			
0.917	1.217	-2.055			
0.953	1.253	-2.155			

APPENDIX E: HS1B Cascade Results

Table E.1(a) HS1B Cascade Performance data at Design Incidence, $i - i_{des} = 0.0^\circ$.

M_2	Y_p	Y_t	ω_t	ξ_t	α_2	AVDR
0.50	0.013	0.086	0.081	0.072	57.2	1.08
0.70	0.016	0.066	0.058	0.047	57.6	1.11
0.84	0.020	0.066	0.056	0.041	58.2	1.09
0.95	0.030	0.088	0.071	0.048	57.9	1.08
1.04	0.036	0.099	0.076	0.048	57.6	1.08
1.14	0.045	0.116	0.085	0.050	57.5	1.06
1.23	0.047	0.117	0.081	0.045	56.0	1.12

M_2	P_{s2}/P_{01}	q_2/P_{01}	P_t/P_{01}	C_{bp}	$Re \pm \Delta Re$
0.50	0.831	0.147	0.813	-0.126	460000 ± 4000
0.70	0.710	0.243	0.680	-0.122	640000 ± 9600
0.84	0.619	0.304	0.577	-0.138	760000 ± 15000
0.95	0.542	0.343	0.408	-0.391	850000 ± 42000
1.04	0.483	0.369	0.353	-0.354	940000 ± 81000
1.14	0.428	0.386	0.304	-0.321	1010000 ± 113000
1.23	0.378	0.400	0.297	-0.201	1070000 ± 146000

Table E.1(b) HS1B Cascade Blade Loading at $M_2 = 0.50$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.375	-0.237	0.074	0.324	0.070
0.032	0.499	-1.119	0.119	0.323	0.072
0.068	0.577	-1.740	0.174	0.317	0.109
0.113	0.625	-2.145	0.239	0.275	0.326
0.173	0.660	-2.449	0.306	0.235	0.513
0.221	0.668	-2.518	0.375	0.220	0.576
0.278	0.672	-2.553	0.430	0.230	0.535
0.341	0.684	-2.655	0.493	0.234	0.518
0.399	0.694	-2.740	0.553	0.257	0.414
0.454	0.673	-2.557	0.614	0.278	0.311
0.507	0.696	-2.758	0.668	0.304	0.177
0.554	0.684	-2.658	0.723	0.334	0.009
0.597	0.666	-2.497	0.773	0.368	-0.195
0.636	0.637	-2.250	0.824	0.400	-0.403
0.673	0.601	-1.938	0.873	0.443	-0.699
0.705	0.577	-1.738	0.917	0.475	-0.931
0.734	0.576	-1.729	0.954	0.528	-1.338
0.768	0.566	-1.651			
0.800	0.565	-1.637			
0.833	0.570	-1.679			
0.867	0.575	-1.719			
0.899	0.576	-1.732			
0.929	0.579	-1.756			
0.959	0.579	-1.757			

Table E.1(c) HS1B Cascade Blade Loading at $M_2 = 0.70$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.445	-0.217	0.074	0.394	0.034
0.032	0.623	-1.240	0.119	0.394	0.033
0.068	0.730	-1.914	0.174	0.402	-0.002
0.113	0.803	-2.385	0.239	0.351	0.232
0.173	0.846	-2.665	0.306	0.304	0.425
0.221	0.860	-2.752	0.375	0.281	0.513
0.278	0.870	-2.815	0.430	0.282	0.510
0.341	0.888	-2.934	0.493	0.291	0.474
0.399	0.902	-3.022	0.553	0.317	0.376
0.454	0.899	-3.003	0.614	0.347	0.248
0.507	0.942	-3.276	0.668	0.379	0.107
0.554	0.941	-3.266	0.723	0.419	-0.084
0.597	0.910	-3.071	0.773	0.472	-0.363
0.636	0.893	-2.967	0.824	0.523	-0.646
0.673	0.842	-2.635	0.873	0.582	-0.993
0.705	0.794	-2.326	0.917	0.637	-1.325
0.734	0.781	-2.246	0.954	0.730	-1.914
0.768	0.773	-2.194			
0.800	0.773	-2.194			
0.833	0.784	-2.265			
0.867	0.794	-2.326			
0.899	0.804	-2.392			
0.929	0.823	-2.513			
0.959	0.820	-2.498			

Table E.1(d) HS1B Cascade Blade Loading at $M_2 = 0.84$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.461	-0.152	0.074	0.420	0.039
0.032	0.657	-1.178	0.119	0.429	-0.001
0.068	0.785	-1.904	0.174	0.437	-0.038
0.113	0.871	-2.400	0.239	0.389	0.173
0.173	0.921	-2.684	0.306	0.332	0.398
0.221	0.939	-2.782	0.375	0.294	0.531
0.278	0.950	-2.845	0.430	0.305	0.495
0.341	0.970	-2.956	0.493	0.314	0.465
0.399	0.994	-3.087	0.553	0.337	0.380
0.454	1.001	-3.123	0.614	0.374	0.235
0.507	1.063	-3.458	0.668	0.407	0.093
0.554	1.086	-3.575	0.723	0.454	-0.120
0.597	1.064	-3.460	0.773	0.510	-0.391
0.636	1.048	-3.375	0.824	0.576	-0.733
0.673	1.013	-3.192	0.873	0.648	-1.122
0.705	0.970	-2.956	0.917	0.723	-1.547
0.734	0.921	-2.680	0.954	0.852	-2.289
0.768	0.909	-2.615			
0.800	0.918	-2.667			
0.833	0.937	-2.771			
0.867	0.970	-2.952			
0.899	1.001	-3.124			
0.929	1.050	-3.386			
0.959	1.074	-3.512			

Table E.1(e) HS1B Cascade Blade Loading at $M_2 = 0.95$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.465	-0.176	0.074	0.430	-0.016
0.032	0.668	-1.250	0.119	0.430	-0.012
0.068	0.795	-1.979	0.174	0.445	-0.083
0.113	0.890	-2.527	0.239	0.400	0.119
0.173	0.941	-2.816	0.306	0.340	0.364
0.221	0.958	-2.915	0.375	0.302	0.501
0.278	0.973	-2.998	0.430	0.310	0.474
0.341	0.996	-3.121	0.493	0.319	0.441
0.399	1.015	-3.228	0.553	0.347	0.338
0.454	1.031	-3.315	0.614	0.382	0.194
0.507	1.103	-3.691	0.668	0.420	0.032
0.554	1.144	-3.901	0.723	0.469	-0.197
0.597	1.157	-3.963	0.773	0.510	-0.460
0.636	1.146	-3.911	0.824	0.576	-0.786
0.673	1.129	-3.824	0.873	0.648	-1.252
0.705	1.126	-3.808	0.917	0.723	-1.726
0.734	1.133	-3.845	0.954	0.852	-2.552
0.768	1.094	-3.648			
0.800	1.033	-3.321			
0.833	1.004	-3.165			
0.867	1.025	-3.281			
0.899	1.073	-3.538			
0.929	1.123	-3.796			
0.959	1.197	-4.160			

Table E.1(f) HS1B Cascade Blade Loading at $M_2 = 1.04$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.462	-0.142	0.074	0.430	0.003
0.032	0.661	-1.175	0.119	0.431	0.000
0.068	0.795	-1.929	0.174	0.436	-0.021
0.113	0.894	-2.489	0.239	0.391	0.174
0.173	0.945	-2.776	0.306	0.337	0.387
0.221	0.961	-2.862	0.375	0.301	0.515
0.278	0.972	-2.919	0.430	0.300	0.519
0.341	0.994	-3.043	0.493	0.310	0.483
0.399	1.016	-3.157	0.553	0.340	0.377
0.454	1.034	-3.254	0.614	0.376	0.234
0.507	1.104	-3.615	0.668	0.417	0.061
0.554	1.146	-3.825	0.723	0.463	-0.145
0.597	1.159	-3.889	0.773	0.520	-0.423
0.636	1.147	-3.830	0.824	0.587	-0.768
0.673	1.127	-3.731	0.873	0.664	-1.187
0.705	1.136	-3.778	0.917	0.750	-1.675
0.734	1.166	-3.922	0.954	0.894	-2.489
0.768	1.227	-4.208			
0.800	1.292	-4.501			
0.833	1.382	-4.872			
0.867	1.353	-4.755			
0.899	1.209	-4.126			
0.929	1.181	-3.995			
0.959	1.219	-4.174			

Table E.1(g) HS1B Cascade Blade Loading at $M_2 = 1.14$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.467	-0.172	0.074	0.413	0.073
0.032	0.669	-1.230	0.119	0.418	0.049
0.068	0.802	-1.989	0.174	0.432	-0.011
0.113	0.897	-2.530	0.239	0.380	0.215
0.173	0.947	-2.809	0.306	0.321	0.442
0.221	0.962	-2.893	0.375	0.286	0.560
0.278	0.969	-2.933	0.430	0.288	0.556
0.341	0.988	-3.036	0.493	0.298	0.520
0.399	1.013	-3.172	0.553	0.327	0.419
0.454	1.032	-3.272	0.614	0.364	0.279
0.507	1.099	-3.625	0.668	0.407	0.101
0.554	1.138	-3.820	0.723	0.456	-0.123
0.597	1.158	-3.919	0.773	0.514	-0.404
0.636	1.144	-3.852	0.824	0.583	-0.762
0.673	1.124	-3.753	0.873	0.658	-1.173
0.705	1.133	-3.796	0.917	0.739	-1.627
0.734	1.163	-3.945	0.954	0.890	-2.489
0.768	1.221	-4.219			
0.800	1.291	-4.533			
0.833	1.379	-4.903			
0.867	1.481	-5.286			
0.899	1.511	-5.391			
0.929	1.371	-4.869			
0.959	1.357	-4.812			

Table E.1(h) HS1B Cascade Blade Loading at $M_2 = 1.23$, and $i - i_{des} = 0.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.465	-0.180	0.074	0.408	0.084
0.032	0.671	-1.273	0.119	0.414	0.058
0.068	0.800	-2.017	0.174	0.427	-0.005
0.113	0.899	-2.589	0.239	0.376	0.218
0.173	0.946	-2.852	0.306	0.311	0.469
0.221	0.961	-2.938	0.375	0.285	0.560
0.278	0.962	-2.940	0.430	0.288	0.547
0.341	0.987	-3.081	0.493	0.301	0.505
0.399	1.009	-3.199	0.553	0.326	0.415
0.454	1.033	-3.330	0.614	0.368	0.251
0.507	1.092	-3.644	0.668	0.407	0.087
0.554	1.141	-3.897	0.723	0.458	-0.145
0.597	1.158	-3.982	0.773	0.512	-0.410
0.636	1.141	-3.897	0.824	0.578	-0.758
0.673	1.124	-3.809	0.873	0.656	-1.188
0.705	1.133	-3.853	0.917	0.739	-1.661
0.734	1.159	-3.986	0.954	0.885	-2.509
0.768	1.215	-4.254			
0.800	1.289	-4.594			
0.833	1.375	-4.959			
0.867	1.483	-5.372			
0.899	1.579	-5.696			
0.929	1.673	-5.977			
0.959	1.655	-5.927			

Table E.2(a) HS1B Cascade Performance data at Off-Design Incidence, $i - i_{des} = 4.5^\circ$.

M_2	Y_p	Y_t	ω_t	ξ_t	α_2	AVDR
0.53	0.013	0.079	0.074	0.065	56.9	1.09
0.71	0.017	0.067	0.059	0.047	57.3	1.11
0.85	0.020	0.066	0.055	0.041	57.8	1.10
0.96	0.032	0.091	0.073	0.050	57.4	1.11
1.06	0.033	0.090	0.068	0.043	57.5	1.11
1.12	0.042	0.107	0.078	0.046	57.1	1.10
1.24	0.059	0.150	0.104	0.057	55.0	1.13

M_2	P_{s2}/P_{01}	q_2/P_{01}	P_t/P_{01}	C_{hp}	$Re \pm \Delta Re$
0.53	0.817	0.159	0.792	-0.156	480000 ± 4400
0.71	0.702	0.249	0.670	-0.127	640000 ± 10000
0.85	0.610	0.309	0.569	-0.133	770000 ± 17000
0.96	0.534	0.346	0.415	-0.344	860000 ± 53000
1.06	0.477	0.373	0.357	-0.321	950000 ± 82000
1.12	0.419	0.391	0.311	-0.276	1020000 ± 106000
1.24	0.370	0.396	0.307	-0.158	1070000 ± 141000

Table E.2(b) HS1B Cascade Blade Loading at $M_2 = 0.53$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.470	-0.490	0.074	0.329	0.253
0.032	0.607	-1.357	0.119	0.312	0.326
0.068	0.671	-1.797	0.174	0.295	0.400
0.113	0.708	-2.055	0.239	0.243	0.600
0.173	0.726	-2.179	0.306	0.223	0.668
0.221	0.725	-2.175	0.375	0.224	0.662
0.278	0.719	-2.134	0.430	0.237	0.619
0.341	0.725	-2.173	0.493	0.255	0.556
0.399	0.727	-2.186	0.553	0.272	0.491
0.454	0.716	-2.112	0.614	0.297	0.392
0.507	0.736	-2.254	0.668	0.326	0.263
0.554	0.725	-2.177	0.723	0.351	0.149
0.597	0.703	-2.021	0.773	0.393	-0.063
0.636	0.674	-1.816	0.824	0.424	-0.226
0.673	0.633	-1.538	0.873	0.467	-0.471
0.705	0.611	-1.387	0.917	0.509	-0.726
0.734	0.609	-1.370	0.954	0.561	-1.056
0.768	0.600	-1.311			
0.800	0.598	-1.297			
0.833	0.605	-1.343			
0.867	0.609	-1.371			
0.899	0.610	-1.376			
0.929	0.611	-1.385			
0.959	0.610	-1.378			

Table E.2(c) HS1B Cascade Blade Loading at $M_2 = 0.71$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.535	-0.462	0.074	0.373	0.267
0.032	0.728	-1.488	0.119	0.369	0.283
0.068	0.829	-2.049	0.174	0.350	0.353
0.113	0.889	-2.384	0.239	0.289	0.569
0.173	0.906	-2.478	0.306	0.263	0.648
0.221	0.899	-2.438	0.375	0.261	0.654
0.278	0.894	-2.414	0.430	0.275	0.610
0.341	0.898	-2.435	0.493	0.288	0.570
0.399	0.918	-2.542	0.553	0.319	0.468
0.454	0.912	-2.510	0.614	0.348	0.363
0.507	0.954	-2.736	0.668	0.380	0.236
0.554	0.949	-2.713	0.723	0.421	0.067
0.597	0.922	-2.564	0.773	0.472	-0.160
0.636	0.899	-2.439	0.824	0.525	-0.413
0.673	0.848	-2.155	0.873	0.585	-0.717
0.705	0.805	-1.918	0.917	0.646	-1.044
0.734	0.794	-1.855	0.954	0.733	-1.520
0.768	0.787	-1.818			
0.800	0.790	-1.837			
0.833	0.794	-1.858			
0.867	0.811	-1.951			
0.899	0.815	-1.975			
0.929	0.836	-2.089			
0.959	0.836	-2.090			

Table E.2(d) HS1B Cascade Blade Loading at $M_2 = 0.85$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.560	-0.404	0.074	0.406	0.240
0.032	0.780	-1.461	0.119	0.404	0.246
0.068	0.896	-2.034	0.174	0.396	0.275
0.113	0.969	-2.390	0.239	0.334	0.491
0.173	0.992	-2.498	0.306	0.289	0.628
0.221	0.976	-2.419	0.375	0.288	0.631
0.278	0.969	-2.387	0.430	0.297	0.603
0.341	0.975	-2.416	0.493	0.318	0.539
0.399	0.997	-2.520	0.553	0.344	0.459
0.454	1.009	-2.579	0.614	0.379	0.336
0.507	1.072	-2.865	0.668	0.416	0.202
0.554	1.096	-2.975	0.723	0.461	0.025
0.597	1.074	-2.874	0.773	0.512	-0.190
0.636	1.055	-2.789	0.824	0.580	-0.498
0.673	1.019	-2.625	0.873	0.653	-0.840
0.705	0.975	-2.415	0.917	0.727	-1.199
0.734	0.938	-2.239	0.954	0.857	-1.845
0.768	0.925	-2.177			
0.800	0.932	-2.212			
0.833	0.949	-2.290			
0.867	0.975	-2.417			
0.899	1.007	-2.566			
0.929	1.056	-2.795			
0.959	1.083	-2.917			

Table E.2(e) HS1B Cascade Blade Loading at $M_2 = 0.96$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.564	-0.391	0.074	0.414	0.226
0.032	0.785	-1.438	0.119	0.418	0.214
0.068	0.909	-2.037	0.174	0.407	0.254
0.113	0.992	-2.426	0.239	0.335	0.500
0.173	1.011	-2.514	0.306	0.295	0.620
0.221	0.998	-2.455	0.375	0.286	0.644
0.278	0.986	-2.397	0.430	0.305	0.591
0.341	0.996	-2.446	0.493	0.317	0.555
0.399	1.020	-2.555	0.553	0.344	0.470
0.454	1.032	-2.607	0.614	0.383	0.338
0.507	1.098	-2.901	0.668	0.421	0.201
0.554	1.142	-3.089	0.723	0.468	0.017
0.597	1.157	-3.152	0.773	0.525	-0.221
0.636	1.141	-3.084	0.824	0.590	-0.509
0.673	1.118	-2.989	0.873	0.668	-0.873
0.705	1.125	-3.016	0.917	0.748	-1.260
0.734	1.155	-3.144	0.954	0.891	-1.950
0.768	1.115	-2.974			
0.800	1.045	-2.667			
0.833	1.020	-2.557			
0.867	1.040	-2.643			
0.899	1.087	-2.853			
0.929	1.129	-3.035			
0.959	1.201	-3.333			

Table E.2(f) HS1B Cascade Blade Loading at $M_2 = 1.06$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.573	-0.430	0.074	0.419	0.209
0.032	0.788	-1.450	0.119	0.411	0.240
0.068	0.913	-2.052	0.174	0.406	0.257
0.113	0.996	-2.444	0.239	0.332	0.508
0.173	1.013	-2.521	0.306	0.295	0.621
0.221	1.005	-2.483	0.375	0.295	0.621
0.278	0.989	-2.409	0.430	0.307	0.586
0.341	0.997	-2.445	0.493	0.321	0.545
0.399	1.020	-2.553	0.553	0.348	0.459
0.454	1.038	-2.631	0.614	0.383	0.338
0.507	1.096	-2.892	0.668	0.420	0.207
0.554	1.143	-3.089	0.723	0.469	0.012
0.597	1.162	-3.170	0.773	0.526	-0.223
0.636	1.143	-3.092	0.824	0.585	-0.486
0.673	1.124	-3.008	0.873	0.668	-0.871
0.705	1.131	-3.042	0.917	0.752	-1.277
0.734	1.163	-3.174	0.954	0.894	-1.964
0.768	1.223	-3.415			
0.800	1.296	-3.693			
0.833	1.388	-4.017			
0.867	1.447	-4.208			
0.899	1.251	-3.525			
0.929	1.208	-3.356			
0.959	1.236	-3.465			

Table E.2(g) HS1B Cascade Blade Loading at $M_2 = 1.12$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.572	-0.434	0.074	0.410	0.236
0.032	0.790	-1.475	0.119	0.406	0.253
0.068	0.921	-2.110	0.174	0.402	0.267
0.113	0.997	-2.466	0.239	0.324	0.529
0.173	1.015	-2.552	0.306	0.289	0.633
0.221	1.000	-2.481	0.375	0.288	0.638
0.278	0.987	-2.423	0.430	0.302	0.596
0.341	0.994	-2.454	0.493	0.317	0.551
0.399	1.014	-2.546	0.553	0.348	0.453
0.454	1.037	-2.653	0.614	0.383	0.334
0.507	1.098	-2.923	0.668	0.420	0.198
0.554	1.142	-3.110	0.723	0.465	0.024
0.597	1.160	-3.186	0.773	0.520	-0.206
0.636	1.140	-3.104	0.824	0.588	-0.508
0.673	1.116	-3.000	0.873	0.662	-0.855
0.705	1.124	-3.037	0.917	0.742	-1.240
0.734	1.162	-3.194	0.954	0.891	-1.965
0.768	1.217	-3.421			
0.800	1.293	-3.712			
0.833	1.383	-4.031			
0.867	1.479	-4.337			
0.899	1.558	-4.567			
0.929	1.348	-3.909			
0.959	1.331	-3.850			

Table E.2(h) HS1B Cascade Blade Loading at $M_2 = 1.24$, and $i - i_{des} = 4.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.570	-0.441	0.074	0.400	0.265
0.032	0.794	-1.518	0.119	0.399	0.270
0.068	0.915	-2.113	0.174	0.388	0.310
0.113	0.992	-2.478	0.239	0.316	0.550
0.173	1.013	-2.580	0.306	0.274	0.670
0.221	0.994	-2.488	0.375	0.267	0.689
0.278	0.983	-2.438	0.430	0.284	0.643
0.341	0.984	-2.444	0.493	0.306	0.578
0.399	1.010	-2.563	0.553	0.331	0.503
0.454	1.033	-2.671	0.614	0.372	0.364
0.507	1.095	-2.950	0.668	0.410	0.229
0.554	1.138	-3.135	0.723	0.457	0.047
0.597	1.156	-3.211	0.773	0.517	-0.204
0.636	1.138	-3.137	0.824	0.581	-0.491
0.673	1.112	-3.026	0.873	0.657	-0.848
0.705	1.121	-3.062	0.917	0.740	-1.256
0.734	1.156	-3.212	0.954	0.886	-1.972
0.768	1.214	-3.454			
0.800	1.286	-3.732			
0.833	1.375	-4.056			
0.867	1.478	-4.389			
0.899	1.579	-4.681			
0.929	1.687	-4.954			
0.959	1.751	-5.100			

Table E.3(a) HS1B Cascade Performance data at Off-Design Incidence, $i - i_{des} = 10.0^\circ$.

M_2	Y_p	Y_t	ω_t	ξ_t	α_2	AVDR
0.52	0.015	0.094	0.088	0.077	56.1	1.17
0.69	0.018	0.076	0.068	0.055	56.5	1.14
0.82	0.021	0.072	0.061	0.046	57.0	1.14
0.93	0.026	0.077	0.063	0.044	57.3	1.14
1.03	0.042	0.117	0.090	0.058	56.6	1.14
1.12	0.044	0.116	0.085	0.051	56.8	1.13
1.20	0.052	0.132	0.094	0.053	55.8	1.13

M_2	P_{s2}/P_{01}	q_2/P_{01}	P_t/P_{01}	C_{bp}	$Re \pm \Delta Re$
0.52	0.819	0.156	0.800	-0.118	470000 ± 5000
0.69	0.714	0.238	0.683	-0.129	630000 ± 12000
0.82	0.630	0.296	0.594	-0.123	750000 ± 15000
0.93	0.560	0.336	0.509	-0.152	840000 ± 25000
1.03	0.487	0.363	0.390	-0.265	940000 ± 83000
1.12	0.437	0.383	0.366	-0.184	1010000 ± 94000
1.20	0.390	0.394	0.319	-0.179	1070000 ± 138000

Table E.3(b) HS1B Cascade Blade Loading at $M_2 = 0.52$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.581	-0.891	0.074	0.285	0.524
0.032	0.710	-1.661	0.119	0.207	0.764
0.068	0.755	-1.940	0.174	0.223	0.720
0.113	0.766	-2.009	0.239	0.205	0.770
0.173	0.754	-1.935	0.306	0.205	0.770
0.221	0.732	-1.796	0.375	0.207	0.765
0.278	0.709	-1.657	0.430	0.229	0.704
0.341	0.695	-1.569	0.493	0.232	0.694
0.399	0.689	-1.536	0.553	0.253	0.630
0.454	0.675	-1.452	0.614	0.282	0.534
0.507	0.687	-1.523	0.668	0.310	0.434
0.554	0.667	-1.400	0.723	0.332	0.346
0.597	0.630	-1.179	0.773	0.376	0.164
0.636	0.618	-1.107	0.824	0.410	0.009
0.673	0.594	-0.967	0.873	0.440	-0.134
0.705	0.582	-0.901	0.917	0.481	-0.344
0.734	0.576	-0.863	0.954	0.532	-0.620
0.768	0.570	-0.833			
0.800	0.579	-0.884			
0.833	0.571	-0.836			
0.867	0.580	-0.889			
0.899	0.591	-0.950			
0.929	0.582	-0.899			
0.959	0.584	-0.913			

Table E.3(c) HS1B Cascade Blade Loading at $M_2 = 0.69$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.705	-0.812	0.074	0.358	0.503
0.032	0.897	-1.639	0.119	0.274	0.727
0.068	0.993	-2.038	0.174	0.278	0.717
0.113	0.997	-2.056	0.239	0.252	0.778
0.173	0.971	-1.951	0.306	0.250	0.782
0.221	0.931	-1.783	0.375	0.252	0.778
0.278	0.885	-1.588	0.430	0.276	0.724
0.341	0.866	-1.505	0.493	0.287	0.696
0.399	0.876	-1.547	0.553	0.311	0.635
0.454	0.865	-1.502	0.614	0.346	0.538
0.507	0.898	-1.642	0.668	0.375	0.452
0.554	0.886	-1.589	0.723	0.415	0.325
0.597	0.846	-1.417	0.773	0.459	0.174
0.636	0.823	-1.317	0.824	0.512	-0.023
0.673	0.786	-1.157	0.873	0.562	-0.216
0.705	0.774	-1.106	0.917	0.619	-0.447
0.734	0.764	-1.062	0.954	0.701	-0.795
0.768	0.762	-1.056			
0.800	0.766	-1.074			
0.833	0.763	-1.061			
0.867	0.774	-1.106			
0.899	0.786	-1.159			
0.929	0.784	-1.149			
0.959	0.784	-1.151			

Table E.3(d) HS1B Cascade Blade Loading at $M_2 = 0.82$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.739	-0.739	0.074	0.383	0.502
0.032	0.978	-1.659	0.119	0.288	0.742
0.068	1.119	-2.162	0.174	0.292	0.732
0.113	1.138	-2.226	0.239	0.264	0.794
0.173	1.109	-2.129	0.306	0.262	0.797
0.221	1.014	-1.791	0.375	0.273	0.774
0.278	0.936	-1.499	0.430	0.293	0.731
0.341	0.949	-1.549	0.493	0.309	0.693
0.399	0.964	-1.606	0.553	0.334	0.631
0.454	0.965	-1.611	0.614	0.370	0.536
0.507	1.025	-1.830	0.668	0.403	0.442
0.554	1.024	-1.828	0.723	0.443	0.322
0.597	0.990	-1.703	0.773	0.498	0.144
0.636	0.963	-1.604	0.824	0.555	-0.052
0.673	0.921	-1.445	0.873	0.626	-0.310
0.705	0.898	-1.357	0.917	0.704	-0.604
0.734	0.894	-1.342	0.954	0.818	-1.046
0.768	0.887	-1.312			
0.800	0.897	-1.350			
0.833	0.903	-1.375			
0.867	0.925	-1.458			
0.899	0.950	-1.553			
0.929	0.952	-1.560			
0.959	0.954	-1.567			

Table E.3(e) HS1B Cascade Blade Loading at $M_2 = 0.93$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.752	-0.703	0.074	0.396	0.497
0.032	1.012	-1.651	0.119	0.313	0.705
0.068	1.174	-2.188	0.174	0.311	0.711
0.113	1.202	-2.274	0.239	0.283	0.771
0.173	1.179	-2.201	0.306	0.277	0.785
0.221	1.023	-1.689	0.375	0.284	0.770
0.278	0.963	-1.478	0.430	0.302	0.731
0.341	0.981	-1.542	0.493	0.320	0.689
0.399	1.001	-1.612	0.553	0.346	0.628
0.454	1.013	-1.653	0.614	0.386	0.522
0.507	1.077	-1.872	0.668	0.421	0.426
0.554	1.106	-1.970	0.723	0.465	0.295
0.597	1.094	-1.930	0.773	0.524	0.109
0.636	1.077	-1.872	0.824	0.585	-0.099
0.673	1.030	-1.713	0.873	0.659	-0.361
0.705	1.008	-1.637	0.917	0.746	-0.678
0.734	0.989	-1.570	0.954	0.883	-1.188
0.768	0.978	-1.531			
0.800	0.991	-1.576			
0.833	1.001	-1.609			
0.867	1.033	-1.724			
0.899	1.079	-1.878			
0.929	1.121	-2.018			
0.959	1.163	-2.151			

Table E.3(f) HS1B Cascade Blade Loading at $M_2 = 1.03$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.756	-0.722	0.074	0.389	0.512
0.032	1.015	-1.670	0.119	0.302	0.728
0.068	1.177	-2.210	0.174	0.307	0.717
0.113	1.220	-2.342	0.239	0.286	0.765
0.173	1.200	-2.280	0.306	0.282	0.771
0.221	1.012	-1.659	0.375	0.283	0.771
0.278	0.969	-1.509	0.430	0.304	0.723
0.341	0.982	-1.552	0.493	0.319	0.690
0.399	1.004	-1.631	0.553	0.342	0.635
0.454	1.017	-1.677	0.614	0.384	0.527
0.507	1.084	-1.907	0.668	0.419	0.428
0.554	1.113	-2.002	0.723	0.466	0.290
0.597	1.116	-2.013	0.773	0.523	0.105
0.636	1.107	-1.983	0.824	0.582	-0.091
0.673	1.093	-1.936	0.873	0.660	-0.368
0.705	1.109	-1.992	0.917	0.749	-0.698
0.734	1.150	-2.124	0.954	0.885	-1.202
0.768	1.182	-2.226			
0.800	1.136	-2.079			
0.833	1.044	-1.769			
0.867	1.057	-1.816			
0.899	1.109	-1.989			
0.929	1.151	-2.127			
0.959	1.227	-2.364			

Table E.3(g) HS1B Cascade Blade Loading at $M_2 = 1.12$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.764	-0.728	0.074	0.394	0.508
0.032	1.026	-1.674	0.119	0.314	0.708
0.068	1.184	-2.188	0.174	0.313	0.709
0.113	1.221	-2.302	0.239	0.282	0.777
0.173	1.208	-2.262	0.306	0.279	0.784
0.221	1.001	-1.586	0.375	0.286	0.769
0.278	0.974	-1.492	0.430	0.305	0.727
0.341	0.987	-1.538	0.493	0.324	0.684
0.399	1.007	-1.607	0.553	0.346	0.631
0.454	1.026	-1.672	0.614	0.385	0.531
0.507	1.084	-1.868	0.668	0.416	0.446
0.554	1.117	-1.976	0.723	0.470	0.289
0.597	1.124	-2.001	0.773	0.526	0.110
0.636	1.109	-1.950	0.824	0.588	-0.099
0.673	1.093	-1.898	0.873	0.661	-0.353
0.705	1.113	-1.964	0.917	0.739	-0.638
0.734	1.152	-2.091	0.954	0.886	-1.179
0.768	1.209	-2.267			
0.800	1.277	-2.466			
0.833	1.371	-2.718			
0.867	1.453	-2.922			
0.899	1.313	-2.564			
0.929	1.241	-2.359			
0.959	1.277	-2.465			

Table E.3(h) HS1B Cascade Blade Loading at $M_2 = 1.20$, and $i - i_{des} = 10.0^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.762	-0.730	0.074	0.387	0.523
0.032	1.026	-1.687	0.119	0.302	0.731
0.068	1.186	-2.211	0.174	0.303	0.729
0.113	1.231	-2.349	0.239	0.270	0.800
0.173	1.205	-2.270	0.306	0.271	0.797
0.221	0.995	-1.578	0.375	0.278	0.784
0.278	0.972	-1.498	0.430	0.297	0.743
0.341	0.980	-1.526	0.493	0.313	0.707
0.399	1.003	-1.608	0.553	0.337	0.650
0.454	1.028	-1.693	0.614	0.382	0.536
0.507	1.083	-1.880	0.668	0.414	0.448
0.554	1.116	-1.989	0.723	0.458	0.319
0.597	1.121	-2.005	0.773	0.520	0.125
0.636	1.108	-1.962	0.824	0.582	-0.082
0.673	1.091	-1.908	0.873	0.659	-0.353
0.705	1.110	-1.970	0.917	0.741	-0.654
0.734	1.150	-2.099	0.954	0.887	-1.192
0.768	1.208	-2.279			
0.800	1.277	-2.483			
0.833	1.366	-2.724			
0.867	1.443	-2.917			
0.899	1.381	-2.763			
0.929	1.320	-2.601			
0.959	1.335	-2.642			

Table E.4(a) HS1B Cascade Performance data at Off-Design Incidence, $i - i_{des} = 14.5^\circ$.

M_2	Y_p	Y_t	ω_t	ξ_1	α_2	AVDR
0.48	0.018	0.131	0.124	0.110	55.2	1.33
0.67	0.021	0.094	0.084	0.069	55.7	1.33
0.80	0.024	0.082	0.070	0.053	56.1	1.30
0.91	0.024	0.072	0.059	0.042	56.5	1.28
1.00	0.035	0.097	0.076	0.050	56.6	1.26
1.06	0.038	0.102	0.078	0.049	56.3	1.26
1.11	0.042	0.108	0.080	0.049	56.7	1.24
1.15	0.042	0.107	0.077	0.046	56.8	1.23

M_2	P_z/P_{01}	q_z/P_{01}	P_t/P_{01}	C_{hp}	$Re \pm \Delta Re$
0.48	0.837	0.137	0.831	-0.042	440000 $\pm$ 4000
0.67	0.726	0.226	0.698	-0.126	620000 $\pm$ 9000
0.80	0.639	0.288	0.603	-0.127	750000 $\pm$ 14000
0.91	0.570	0.331	0.532	-0.115	850000 $\pm$ 20000
1.00	0.509	0.358	0.418	-0.255	930000 $\pm$ 57000
1.06	0.472	0.372	0.375	-0.263	1000000 $\pm$ 90000
1.11	0.442	0.383	0.330	-0.292	1040000 $\pm$ 90000
1.15	0.420	0.391	0.353	-0.170	1080000 $\pm$ 86000

Table E.4(b) HS1B Cascade Blade Loading at $M_2 = 0.48$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.613	-1.135	0.074	0.268	0.568
0.032	0.715	-1.762	0.119	0.200	0.776
0.068	0.754	-2.006	0.174	0.222	0.713
0.113	0.752	-1.996	0.239	0.199	0.778
0.173	0.731	-1.865	0.306	0.196	0.786
0.221	0.706	-1.708	0.375	0.198	0.781
0.278	0.673	-1.502	0.430	0.221	0.718
0.341	0.658	-1.406	0.493	0.225	0.707
0.399	0.656	-1.394	0.553	0.242	0.655
0.454	0.627	-1.219	0.614	0.274	0.547
0.507	0.647	-1.339	0.668	0.302	0.447
0.554	0.624	-1.201	0.723	0.322	0.368
0.597	0.591	-1.000	0.773	0.358	0.217
0.636	0.578	-0.928	0.824	0.389	0.078
0.673	0.563	-0.839	0.873	0.421	-0.071
0.705	0.556	-0.798	0.917	0.462	-0.282
0.734	0.544	-0.730	0.954	0.513	-0.554
0.768	0.544	-0.728			
0.800	0.555	-0.792			
0.833	0.542	-0.720			
0.867	0.554	-0.785			
0.899	0.566	-0.858			
0.929	0.553	-0.778			
0.959	0.559	-0.817			

Table E.4(c) HS1B Cascade Blade Loading at $M_2 = 0.67$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.766	-1.062	0.074	0.332	0.580
0.032	0.957	-1.874	0.119	0.253	0.779
0.068	1.039	-2.207	0.174	0.270	0.739
0.113	1.019	-2.130	0.239	0.245	0.795
0.173	0.988	-2.002	0.306	0.242	0.801
0.221	0.902	-1.643	0.375	0.249	0.787
0.278	0.842	-1.386	0.430	0.276	0.726
0.341	0.843	-1.393	0.493	0.282	0.710
0.399	0.848	-1.413	0.553	0.299	0.669
0.454	0.825	-1.315	0.614	0.335	0.573
0.507	0.861	-1.468	0.668	0.363	0.492
0.554	0.830	-1.335	0.723	0.392	0.400
0.597	0.793	-1.177	0.773	0.445	0.226
0.636	0.775	-1.098	0.824	0.493	0.054
0.673	0.748	-0.984	0.873	0.542	-0.133
0.705	0.742	-0.958	0.917	0.600	-0.360
0.734	0.729	-0.902	0.954	0.683	-0.706
0.768	0.734	-0.924			
0.800	0.747	-0.980			
0.833	0.736	-0.934			
0.867	0.748	-0.982			
0.899	0.768	-1.069			
0.929	0.750	-0.993			
0.959	0.757	-1.020			

Table E.4(d) HS1B Cascade Blade Loading at $M_2 = 0.80$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.822	-0.944	0.074	0.376	0.556
0.032	1.081	-1.860	0.119	0.292	0.755
0.068	1.224	-2.313	0.174	0.306	0.725
0.113	1.192	-2.215	0.239	0.288	0.765
0.173	1.179	-2.175	0.306	0.280	0.782
0.221	0.970	-1.480	0.375	0.285	0.770
0.278	0.939	-1.370	0.430	0.304	0.729
0.341	0.939	-1.371	0.493	0.316	0.702
0.399	0.955	-1.427	0.553	0.343	0.638
0.454	0.946	-1.395	0.614	0.385	0.530
0.507	0.995	-1.569	0.668	0.413	0.456
0.554	0.982	-1.523	0.723	0.450	0.346
0.597	0.943	-1.384	0.773	0.506	0.173
0.636	0.927	-1.329	0.824	0.558	0.003
0.673	0.892	-1.200	0.873	0.628	-0.239
0.705	0.886	-1.178	0.917	0.698	-0.491
0.734	0.873	-1.133	0.954	0.804	-0.878
0.768	0.877	-1.147			
0.800	0.895	-1.212			
0.833	0.882	-1.164			
0.867	0.909	-1.262			
0.899	0.935	-1.355			
0.929	0.919	-1.300			
0.959	0.922	-1.307			

Table E.4(e) HS1B Cascade Blade Loading at $M_2 = 0.91$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.829	-0.874	0.074	0.380	0.573
0.032	1.105	-1.800	0.119	0.285	0.790
0.068	1.287	-2.331	0.174	0.301	0.758
0.113	1.288	-2.332	0.239	0.279	0.802
0.173	1.277	-2.302	0.306	0.277	0.805
0.221	1.010	-1.495	0.375	0.284	0.791
0.278	0.962	-1.335	0.430	0.310	0.737
0.341	0.968	-1.354	0.493	0.316	0.725
0.399	0.989	-1.425	0.553	0.340	0.670
0.454	0.993	-1.439	0.614	0.383	0.565
0.507	1.054	-1.639	0.668	0.416	0.480
0.554	1.065	-1.673	0.723	0.460	0.357
0.597	1.037	-1.582	0.773	0.522	0.171
0.636	1.024	-1.540	0.824	0.581	-0.019
0.673	0.990	-1.428	0.873	0.655	-0.267
0.705	0.980	-1.394	0.917	0.737	-0.551
0.734	0.963	-1.337	0.954	0.871	-1.019
0.768	0.965	-1.344			
0.800	0.991	-1.430			
0.833	0.991	-1.433			
0.867	1.027	-1.551			
0.899	1.074	-1.703			
0.929	1.078	-1.714			
0.959	1.088	-1.746			

Table E.4(f) HS1B Cascade Blade Loading at $M_2 = 1.00$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{ts}	C_{ps}	x/C_x	M_{ts}	C_{ps}
0.008	0.837	-0.869	0.074	0.386	0.568
0.032	1.115	-1.783	0.119	0.302	0.762
0.068	1.308	-2.330	0.174	0.318	0.727
0.113	1.317	-2.354	0.239	0.291	0.785
0.173	1.307	-2.329	0.306	0.285	0.796
0.221	1.023	-1.496	0.375	0.289	0.787
0.278	0.979	-1.349	0.430	0.312	0.740
0.341	0.978	-1.349	0.493	0.329	0.703
0.399	0.996	-1.406	0.553	0.354	0.646
0.454	1.010	-1.453	0.614	0.392	0.552
0.507	1.075	-1.660	0.668	0.428	0.457
0.554	1.096	-1.727	0.723	0.470	0.341
0.597	1.089	-1.705	0.773	0.528	0.168
0.636	1.091	-1.710	0.824	0.591	-0.032
0.673	1.076	-1.665	0.873	0.664	-0.273
0.705	1.095	-1.722	0.917	0.749	-0.564
0.734	1.123	-1.809	0.954	0.890	-1.053
0.768	1.077	-1.668			
0.800	1.058	-1.606			
0.833	1.039	-1.547			
0.867	1.076	-1.664			
0.899	1.132	-1.836			
0.929	1.161	-1.924			
0.959	1.232	-2.127			

Table E.4(g) HS1B Cascade Blade Loading at $M_2 = 1.06$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.837	-0.886	0.074	0.380	0.577
0.032	1.116	-1.811	0.119	0.292	0.779
0.068	1.308	-2.357	0.174	0.309	0.742
0.113	1.323	-2.399	0.239	0.286	0.790
0.173	1.310	-2.364	0.306	0.278	0.806
0.221	1.020	-1.506	0.375	0.287	0.789
0.278	0.980	-1.376	0.430	0.309	0.743
0.341	0.972	-1.346	0.493	0.318	0.724
0.399	0.995	-1.425	0.553	0.341	0.672
0.454	1.016	-1.493	0.614	0.384	0.569
0.507	1.070	-1.669	0.668	0.419	0.477
0.554	1.099	-1.758	0.723	0.465	0.349
0.597	1.093	-1.739	0.773	0.524	0.172
0.636	1.096	-1.748	0.824	0.588	-0.030
0.673	1.087	-1.723	0.873	0.659	-0.268
0.705	1.112	-1.801	0.917	0.744	-0.561
0.734	1.149	-1.911	0.954	0.889	-1.067
0.768	1.206	-2.078			
0.800	1.278	-2.278			
0.833	1.334	-2.426			
0.867	1.246	-2.191			
0.899	1.176	-1.992			
0.929	1.186	-2.021			
0.959	1.254	-2.214			

Table E.4(h) HS1B Cascade Blade Loading at $M_2 = 1.11$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.839	-0.891	0.074	0.382	0.574
0.032	1.120	-1.821	0.119	0.304	0.754
0.068	1.317	-2.378	0.174	0.315	0.731
0.113	1.330	-2.413	0.239	0.290	0.784
0.173	1.310	-2.360	0.306	0.283	0.798
0.221	1.027	-1.526	0.375	0.286	0.790
0.278	0.980	-1.373	0.430	0.320	0.719
0.341	0.977	-1.362	0.493	0.336	0.683
0.399	0.998	-1.431	0.553	0.353	0.644
0.454	1.017	-1.494	0.614	0.396	0.539
0.507	1.070	-1.664	0.668	0.424	0.464
0.554	1.097	-1.751	0.723	0.468	0.341
0.597	1.095	-1.744	0.773	0.529	0.158
0.636	1.095	-1.744	0.824	0.590	-0.039
0.673	1.087	-1.719	0.873	0.662	-0.277
0.705	1.113	-1.799	0.917	0.749	-0.578
0.734	1.147	-1.903	0.954	0.890	-1.066
0.768	1.206	-2.075			
0.800	1.287	-2.300			
0.833	1.361	-2.491			
0.867	1.449	-2.697			
0.899	1.372	-2.517			
0.929	1.257	-2.220			
0.959	1.303	-2.344			

Table E.4(i) HS1B Cascade Blade Loading at $M_2 = 1.15$, and $i - i_{des} = 14.5^\circ$.

Suction Surface			Pressure Surface		
x/C_x	M_{is}	C_{ps}	x/C_x	M_{is}	C_{ps}
0.008	0.833	-0.868	0.074	0.376	0.589
0.032	1.114	-1.800	0.119	0.289	0.785
0.068	1.314	-2.367	0.174	0.305	0.752
0.113	1.325	-2.396	0.239	0.273	0.817
0.173	1.314	-2.366	0.306	0.267	0.829
0.221	1.021	-1.505	0.375	0.278	0.808
0.278	0.976	-1.357	0.430	0.303	0.757
0.341	0.971	-1.340	0.493	0.316	0.729
0.399	0.996	-1.422	0.553	0.339	0.678
0.454	1.017	-1.492	0.614	0.384	0.568
0.507	1.069	-1.660	0.668	0.416	0.486
0.554	1.098	-1.750	0.723	0.458	0.369
0.597	1.092	-1.730	0.773	0.519	0.190
0.636	1.093	-1.733	0.824	0.582	-0.008
0.673	1.082	-1.699	0.873	0.655	-0.252
0.705	1.109	-1.783	0.917	0.740	-0.543
0.734	1.148	-1.902	0.954	0.863	-0.971
0.768	1.202	-2.062			
0.800	1.283	-2.286			
0.833	1.362	-2.487			
0.867	1.459	-2.715			
0.899	1.490	-2.782			
0.929	1.294	-2.314			
0.959	1.311	-2.359			