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**The Effect of Axial Velocity Ratio, Turbulence
Intensity, Incidence and Leading Edge Geometry on
the Off-Design Performance of a
Turbine Blade Cascade**

By

David Richard Whitehouse

ng. (Mechanical Engineering)

A thesis submitted to
the Faculty of Graduate Studies and Research
in partial fulfilment of the requirements
for the degree of

**MASTER OF ENGINEERING
in AEROSPACE ENGINEERING**

**OTTAWA-CARLETON INSTITUTE FOR MECHANICAL
AND AEROSPACE ENGINEERING**

**DEPARTMENT OF MECHANICAL AND AEROSPACE
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submitted by David Richard Whitehouse, B.Eng.
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the degree of Master of Engineering

Thesis Co-Supervisor

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Abstract

The off-design performance of axial flow turbines is generally predicted using empirical correlations. These often reflect important trends that can be used to make blade designs more tolerant to off-design conditions. The objective of the present study is to determine the effect of axial velocity ratio, turbulence intensity, incidence and leading edge geometry on the off-design performance of a turbine cascade. The correlation of Moustapha *et al.* (1990) identified the leading edge diameter as a significant parameter affecting off-design performance. A new cascade was built which is a modification of a design extensively tested by earlier workers. The new design has a larger leading edge diameter to reduce off-design losses and a lower leading edge wedge angle to maintain a similar blade thickness and an acceptable velocity distribution.

Endplates were used to adjust the axial velocity ratio (AVR) across the blade passage to 1.0 in order to obtain conditions representative of two-dimensional flow at the blade midspan. The blade losses were found to be very sensitive to AVR at high positive incidence. Decreasing the AVR by diverging the endplates strongly increased the blade losses. Loss measurements were made at turbulence intensities of 0.3% and 3.0%. Turbulence intensity (Tu) can play a role in loss production through its influence on the location and length of boundary layer transition and separation bubbles. For the range of incidence and AVR over which measurements were made, the different values of Tu had no strong effect on the off-design loss measurements. Loss measurements were made at incidences between -20° and $+20^\circ$ from design. The total pressure loss increased sharply after $+10^\circ$ due to the formation of a separation bubble on the rear half of the blade suction surface. The two-dimensional off-design loss measurements from the new blade were almost identical to those from the old. This suggests that the leading edge wedge angle may have a stronger influence than previously thought, as the benefits of the larger leading edge diameter appear to have been cancelled out.

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Nomenclature

AVR	axial velocity ratio (see equation 2.5)
C	chord
C _x	axial chord
C _p	blade static pressure coefficient, $(P-P_1)/q_1$
C _{pb}	base pressure coefficient, $(P_b-\bar{P}_2)/\bar{q}_2$
d	blade leading edge diameter
H	blade span
i	angle of incidence, $i=(\alpha_1-\beta_1)$
P	static pressure
P _b	base pressure
P _o	total pressure
q	dynamic pressure, $\frac{1}{2}\rho V^2$
Re	Reynolds number, $\rho_1 V_1 C/\mu$
S	blade pitch or spacing
t	blade thickness
Tu	turbulence intensity
V	velocity
x	chordwise coordinate
y	pitchwise coordinate
Y	mass averaged total pressure loss coefficient, $(\bar{P}_{o2}-\bar{P}_{o1})/(\bar{P}_{o2}-P_2)$ (see equation 2.1)
z	spanwise coordinate
α	flow angle measured from axial direction
β	blade metal angle, measured from the axial direction
γ	blade stagger angle
δ	deviation angle, $\delta=(\beta_2-\alpha_2)$
μ	dynamic viscosity
ϵ	endplate knife-edge deflection

ρ	density
σ	passage throat opening
θ	camber angle
ϕ^2	kinetic energy loss (see equation 2.2)

Subscripts and Superscripts

b	base
des	design
eff	effective
ex	exit
in	inlet
l.c.	leading edge
t.c.	trailing edge
m	midspan
1	inlet measurement plane
2	exit measurement plane
—	(overbar) a mass averaged value

Chapter 1

Introduction

The aerodynamic design of the turbomachinery components in a gas turbine is a very involved process. Each must perform individually at high efficiency while operating in concert with the rest of the engine to deliver the desired output reliably and efficiently. In general, gas turbines are designed to deliver a range of output power or thrust in a relatively narrow range of flow conditions. Anticipated and unanticipated conditions can arise that lead to changes in the flow angle through the different stages of the engine. Turbine blading must be able to effectively perform its duty across a reasonable range of inlet flow angles in order to continue providing power to the upstream compressor or a shaft output.

Estimates of turbine blade performance at off-design inlet angles or incidences can be found using experimentally derived loss correlations. The most widely used correlation is that of Ainley and Mathieson (1951) which has been improved periodically to account for more modern blade designs. In a recent review of loss correlations, Tremblay (1989)

showed that this method significantly over-predicted the losses of very recent blade designs. Moustapha *et al.* (1990) presented a new correlation based on a database of blades of recent design. This method was significantly more successful at predicting blade losses than the Ainley and Mathieson correlation. The correlation's most notable feature is the inclusion of the blade leading edge diameter as a dominant factor.

The present study was undertaken to experimentally determine the effect of axial velocity ratio, turbulence intensity, incidence and leading edge geometry on the two-dimensional loss of a turbine cascade. Goobie (1989) and Tremblay (1989) obtained loss measurements from a linear turbine blade cascade (CC2) mounted in a variable incidence test rig. Measurements were made at effective incidences from -20° to $+20^\circ$. Rodger (1992) made similar measurements at -20° , design and $+20^\circ$ effective incidence while examining the cascade outlet to inlet axial velocity ratio (AVR) which was taken as a measure of the flow two-dimensionality.

The profile design of the blade used in the present study (CC3) is very similar to the one used in the earlier work. The correlation of Moustapha *et al.* (1990) suggests that increasing the turbine blade leading diameter will increase tolerance to off-design inlet flow angles. The blade manufactured for this study shares the same aerodynamic parameters as CC2 but has almost double the leading edge diameter. The database used to develop the correlation contains blades with widely varying geometric and aerodynamic parameters. The intention of this work is to examine, among other things, the effect of the leading edge geometry in order to determine the validity of the assumptions made in developing the correlation of Moustapha *et al.* (1990).

The measurements of Tremblay (1989) were performed in a wind tunnel with a free stream turbulence intensity (Tu) of 1.5%. A wind tunnel of more recent design with a Tu of 0.3% was employed by Rodger (1992) and in the present study. Rodger (1992) and Tremblay (1989) reported significantly different losses at $+20^\circ$ incidence. To determine if this difference in losses was due to the higher Tu present in the earlier tests,

loss measurements were made on CC3 at both high (3.0%) and low (0.3%) turbulence intensity. The level of free stream turbulence intensity was increased by mounting a turbulence generating grid at the test section inlet.

Chapter 2

Literature Review

The design of a gas turbine is a complex, iterative process that involves balancing aerodynamic, thermodynamic and metallurgical considerations with operating requirements. The gas triangles through the turbine blade rows are determined from the flow path using the stage inlet velocity and rotational speed. The blades of a turbine are optimized to operate at their highest efficiency during design operating conditions. Frequently, however, the engine will not be operating at this condition. For example, most aircraft engines perform with highest efficiency at cruising speed and altitude. The conditions at this operating point cannot be maintained indefinitely as the plane must land, take off and, in the case of military or aerobatic aircraft, conduct high performance manoeuvres. Given the known design operating point, the aerodynamicist must select appropriate parameters such as aspect ratio, leading edge diameter and stagger angle to perform the required duty with high efficiency. The designer must then re-examine the design to allow the gas turbine to operate with a reasonable level of efficiency at the anticipated off-design operating conditions. Care must be taken to avoid flow conditions

that can result in engine shut down, a potentially hazardous situation for the operator.

The overall effect of design changes on a gas turbine is readily determined through performance tests on a full scale engine. Unfortunately, this does not reveal the effect of the design change on the efficiency of the component modified or, in the case of blade modifications, the effect on stage efficiency. Large scale linear cascade testing is used to study the details of the flow passing through a turbine blade row. Such studies are often difficult or even impossible on an actual engine due to physical limitations. The simplified geometry of the linear cascade makes it easier to modify the experimental configuration and the large scale allows smaller flow features to be identified and examined in detail.

2.1 Loss Breakdown

Turbine blade loss can be broken down into physically meaningful components in the following manner

1. **Profile Loss** : associated with the development of the boundary layer along the blade surface.
2. **Trailing Edge Loss** : due to the sudden expansion at the trailing edge of the blade.
3. **Secondary Loss** : arising from growth of the endwall boundary layer and its interaction with the flow through the blade passage.
4. **Tip Clearance Loss** : a consequence of the finite gap between the blade tip and the casing resulting in a cross flow between the pressure and suction surfaces.

5. **Annulus Loss** : due to skin friction on the hub and casing of the blade passage.

The present study is concerned with two-dimensional losses. These arise from the direct dissipation of free stream flow energy through viscous or mixing action as it travels through the blade passage. The profile and trailing edge losses are considered two-dimensional because they do not involve the creation of significant flow in the spanwise direction.

2.2 Incidence and Loss Terminology

Blade geometry is determined using the inlet axial flow velocity and the blade rotational speed. These are used in a velocity triangle analysis to determine the blade inlet and outlet angles. The difference between the inlet flow and metal angles is referred to as the design incidence. For off-design measurements, the effective incidence is defined as the difference between the flow angle and the design flow angle. The difference between the outlet metal angle and the outlet flow angle is the deviation. The terminology is shown graphically in Figure 2.1 and in Figure A.1. The loss across a turbine blade can be presented as a mass averaged total pressure loss coefficient, Y . This is defined as the drop in mass averaged total pressure across the blade row normalized to the mass averaged dynamic pressure at the blade outlet.

$$Y = \frac{\bar{P}_{o2} - \bar{P}_{o1}}{\bar{q}_2} \quad (2.1)$$

Loss measurements can also be presented as a kinetic energy loss coefficient, ϕ^2 . This is calculated from the actual and isentropic blade outlet velocity as follows

$$\phi^2 = 1 - (V_2^2 / V_{is}^2) \quad (2.2)$$

Figure 2.1 Turbine blade incidence terminology reproduced from Tremblay (1989)

The two loss coefficients are related through the following equation

$$Y = \frac{1 - \frac{\gamma-1}{2} M_2^2 \left(\frac{1}{1 - \phi^2} - 1 \right)^{-\frac{\gamma}{\gamma-1}} - 1}{\left(1 + \frac{\gamma-1}{2} M_2^2 \right)^{-\frac{\gamma}{\gamma-1}}} \quad (2.3)$$

For incompressible flows such as the ones used in the present work, this expression reduces to the following

$$Y = \frac{1}{1 - \phi^2} - 1 \quad (2.4)$$

The off-design loss is defined as the difference between the loss at design conditions and the loss at an off-design condition. The reference (design) loss is sometimes presented as the loss at 0° incidence or the loss at the effective design incidence.

Turbine blade losses can be strongly dependent on the Reynolds number. As the Reynolds number is decreased, blade losses remain relatively constant. Below a certain value, however, the losses increase very rapidly. This value of Reynolds number is known as the critical Reynolds number. It is important when making blade loss measurements that the critical Reynolds number has been exceeded for valid comparisons with other results.

The ideal two-dimensional flow would use a blade of infinite span; however, experimental compromises must be made. In the two-dimensional flow case there is no net change in the axial velocity through the blade passage. Rodger (1992) has shown that this is a necessary condition to perform valid two-dimensional loss measurements. Experimentally this can be verified with the mass average axial velocity ratio (AVR). An axial velocity ratio of 1.0 is necessary to ensure two-dimensional flow. In a compressible flow, the axial velocity density ratio (AVDR) is used to account for changes in density across the blade. In general, an increase in AVR has a favourable effect on blade

performance due to the beneficial effect on the boundary layer stability. The AVR is defined as the ratio of mass averaged axial velocity at the blade trailing edge plane divided by the mass averaged axial velocity at the leading edge plane and simplifies to

$$AVR = \frac{\int_0^s V_{x\ te}^2 dy}{\int_0^s V_{x\ le}^2 dy} \quad (2.5)$$

2.3 Off-Design Experimental Investigations

The magnitude of profile loss is dependent on the Reynolds number, turbulence intensity, inlet angle and blade geometry. A study of the available literature over the past ten years presents few studies concerned with the off-design performance of turbine cascade blades of recent design. The range of off-design incidence for which measurements are presented is often restricted by equipment limitations, making loss measurements at high positive incidence scarce. At these inlet conditions, the high adverse pressure gradient can result in flow separation on the blade suction surface accompanied by a sudden increase in losses.

Patterson and Hoeger (1986) performed tests on three different blade profiles designed with a prescribed velocity distribution technique. Loss coefficients were measured at different Reynolds numbers. Profile loss predictions were made using the boundary layer parameters measured at the blade outlet. These measurements highlight some of the differences between front, mid and aft-loaded blade profiles.

Hodson and Dominy (1987) presented high subsonic flow ($M=0.92$) loss measurements on a turbine blade cascade at design, 8.6° and -20.3° incidence with axial velocity density ratios of 0.917, 0.923 and 0.907 respectively. The blade had a leading

edge diameter to chord ratio of 0.032 and flow turning angle of 93° . At design, a laminar separation bubble formed on the suction surface, reattaching before reaching the trailing edge. At high positive incidence the flow remained attached. At the large negative incidence a separation bubble formed on the pressure surface just downstream of the leading edge. Both the 8.6° and -20.3° profile losses presented were almost double the loss measured at design.

Yamamoto and Nouse (1988) presented a considerable amount of data for flow development through a cascade blade passage. The blade studied had thick leading and trailing edges with a leading edge diameter to chord ratio of 0.22 and a turning angle of 107° . Measurements were made at effective incidence of -53.3° , -43.3° , -23.3° , -6.2° and $+7.2^\circ$. Blade and endwall static pressure distributions, and five-hole probe measurements made in and downstream of the passage, showed how the flow developed at different incidence. Pitchwise averaged total pressure losses were presented but there was a significant level of three-dimensionality in the outlet flow, making extraction of the two-dimensional loss difficult. As flow angle was decreased a separation bubble formed on the pressure surface with a corresponding increase in total losses measured downstream. Losses decreased as the inlet angle approached design, then increased slightly at $+7.2^\circ$.

Vijayaraghavan and Kavanagh (1988) examined the effect of free stream turbulence intensity, Reynolds number and incidence on a linear cascade with a leading edge diameter to chord ratio of 0.12 and a turning angle of 118° . No positive incidences were examined, only design, -7° and -14° . Measurements were made at inlet turbulence intensities of 0.8%, 2.0%, 4.8% and 6.4%. Suction flow at the blade endwall was used to maintain a high level of two-dimensionality. Under test conditions axial velocity ratios of 0.93 to 0.97 were maintained. Profile losses were not seen to vary significantly with turbulence intensity or inlet angle owing perhaps to the narrow range of conditions tested and the well behaved nature of the boundary layer flow.

A cascade test section designed for a wide range of effective incidence

measurements was employed by Goobie (1989). Midspan loss measurements were made at design, $+15^\circ$ and -15° incidence. The blade profile had a leading edge diameter to chord ratio of 0.058 and a flow turning angle of 87° . Flow parameters were measured with a three-hole pressure probe mounted at midspan, 141% downstream of the blade leading edge. Endplates fitting over the blade profiles were used to decrease the effect of secondary flows by forcing the tunnel endwall boundary layer to bypass the cascade. A much weaker boundary layer developed from the endplate leading edge, resulting in an increased width of the low loss "bucket". This was confirmed by flow visualization and a spanwise plot of loss measurements. No information was presented on the AVR under which these tests were conducted.

The same cascade was tested by Tremblay (1989) who presented results for loss variation with Reynolds number and incidence. Measurements were made at effective incidence from -25° to $+25^\circ$ in increments of 5° . A large increase in losses was seen at higher positive incidence as the suction surface flow started to separate. Blade endplates were used to increase the level of two-dimensionality in the cascade passage flow. Spanwise loss measurements showed that the use of endplates significantly increased the width of the low loss region at off-design incidence as well as design. Loss measurements made over a range of Reynolds numbers showed that the blade was being tested at a point above the critical Reynolds number both at design and at higher positive incidence. To prevent separation at the endplate leading edge, highly convergent endplate settings were used. This resulted in an axial velocity ratio of 1.05 at design incidence and 1.08 at $+20^\circ$ incidence. A free stream turbulence intensity of 1.5% was measured at the cascade inlet.

Rodger (1992) performed tests on the same blade profile to establish the level of two-dimensionality and study the effects of axial velocity ratio. The cascade was mounted in the test section designed by Goobie (1989) which was connected to a wind tunnel with improved flow quality and a free stream turbulence intensity of 0.3%. The endplate leading edge was improved to prevent flow separation allowing a greater range of plate settings to be used. Measurements were made at effective inlet incidence of -20° , 0° and

+20°. Profile loss results for -20° and design incidence showed very good agreement with those from Tremblay(1989). The results at +20° from an axial velocity ratio of 1.008 to 1.087 were several times greater than those reported by Tremblay. It was suggested that these differences were due to the presence of different boundary layer loss mechanisms. The low free stream turbulence intensity may have inhibited the onset of natural transition, causing the flow to separate toward the trailing edge of the blade. For 0° and -20°, increasing the AVR slightly reduced losses. However, for the separated flow case, losses varied strongly, indicating that the change of pressure gradient affected the location of flow separation. The differences in turbulence intensity and axial velocity ratio make it difficult to directly compare these loss measurements and those presented earlier by Tremblay (1989).

Perdichizzi and Dossena (1992) presented kinetic energy loss coefficients for profile losses at five incidence angles from -60° to 35° and three different pitch to chord ratios. This blade had a leading edge diameter to chord ratio of 0.08 and a turning angle of 89.4°. A gradual increase in losses was seen as incidence was moved away from design. At the highest pitch to chord ratio there was a steep increase in losses at +35°. However, this was not a two-dimensional effect as secondary flow effects are reported to extend to midspan at this incidence.

2.4 Off-Design Analytical Investigations

The focus of the present study is two-dimensional losses. The ideal two-dimensional cascade flow would have an infinite span. In such a case the losses would arise from boundary layer growth around the blade profile and the mixing process downstream of the trailing edge. The two-dimensional performance of a turbine blade can be predicted using a straight forward boundary layer calculation. To implement such a calculation and properly interpret the results, it is important to have an understanding of

the factors that affect the flow development through the blade passage.

2.4.1 Blade Boundary Layer Development

The development of the blade surface boundary layer is reflected in blade profile losses. Profile losses arise from the dissipation of energy through viscous action in the laminar boundary layer and a combination of viscous and turbulent effects in the turbulent boundary layer. At off-design incidence, the blade geometry has an even more significant effect on boundary layer development. The leading edge overspeed and subsequent strong adverse pressure gradient can trigger flow separation. Further downstream, the value of the pressure and suction surface boundary layer parameters at the trailing edge contributes to the trailing edge loss. Flow in an engine environment is highly transitional and laminar to turbulent transition can take place over a significant fraction of the blade surface. This makes accurate transition prediction important in any profile loss prediction scheme or accurate calculation of skin friction and heat transfer distribution. In lieu of direct blade surface measurements using thin film probes (Halstead *et al.*, 1991) or boundary layer pitot probes (Abu-Ghannam and Shaw, 1980), computational methods can be used to predict the starting point of laminar-turbulent transition and its length.

The transition location and length are primarily functions of such flow parameters as free stream turbulence intensity, Reynolds number and pressure gradient. Other contributing factors are surface roughness, surface curvature, compressibility and heat transfer. Usually, the effect of this second group of parameters is several times less than the first.

In a gas turbine, the close spacing of the blade rows does not allow uniform flow to develop between blade rows. The impingement of wakes from upstream blade rows has a strong effect on the transition mechanism in the blade boundary layer. As the wake passes through the blade passage, a band of increased instability causes transition to

temporarily move further upstream. The changing position of the transition region with time leads to the name unsteady transition. This fluctuation of transition will cause a periodic change in blade efficiency. A detailed description of the process can be found in the work of Addison and Hodson (1991).

There are several laminar-turbulent transition mechanisms. The classic natural transition process is described in Schlichting (1979). A review of transition through boundary layer separation is found in Mayle (1991). Transition can occur in a laminar separation bubble, resulting in the flow reattaching as a turbulent boundary layer. The length of the bubble depends on the transition process within the bubble free shear layer. Longer bubbles usually develop in lower free stream turbulence intensities. Since higher losses are a result of increased bubble length, these should be avoided in turbomachinery design. Short bubbles are, however, an effective means of forcing laminar-turbulent transition over a short length. This can be done intentionally to avoid the high losses associated with the occurrence of non-reattaching turbulent separation over the aft portion of the blade.

Vijayaraghavan and Kavanagh (1988) used a differential method boundary layer prediction program to predict total pressure losses across a linear turbine blade cascade. Predictions from this program were compared to results from turbine cascade tests made over a range of turbulence levels and inlet angles. The location and length of transition predicted by the program were compared with locations determined from hot-film measurements. The predicted transition on the blade suction surface was seen to move closer to the stagnation point for each increase in turbulence intensity and each increase in inlet angle. In fact, the hot-film measurements showed that the pressure gradient rather than turbulence intensity was the dominant factor affecting transition, causing its delay until the minimum blade pressure is reached. The predicted transitions occurred in a region of strong favourable pressure gradient, suggesting that the transition model was overly sensitive to free stream turbulence intensity. Loss calculations using outlet boundary layer parameters and a method by Stewart (1955) produced very favourable

results when compared to losses measured at low turbulence intensities and design flow conditions. Vijayaraghavan and Kavanagh concluded that the method was adequate for predicting overall loss coefficients but not sufficient for obtaining actual shear stress or heat transfer distributions.

2.4.2 Trailing Edge Loss

A significant portion of the blade profile loss arises from the sudden expansion at the blade trailing edge. In modern gas turbine design this loss takes on added significance as air used in blade cooling is ejected from the trailing edge. This loss is influenced by the momentum and displacement thicknesses at the blade trailing edge, the trailing edge geometry and the pressure acting on the trailing edge (the base pressure). Accurate evaluation of the trailing edge loss in cascade measurements has been hampered by the physical difficulty of placing a static pressure tap in this location. The blade model used by Tremblay (1989) and Rodger (1992) was designed from the outset to accommodate a static pressure tap in this location.

Denton (1990) derived a trailing edge loss expression using a control volume analysis around the trailing edge of a symmetric plate with a flat trailing edge. The trailing edge boundary layer parameters, plate thickness and a base pressure coefficient are used to calculate a total pressure loss coefficient. Applying this analysis requires no knowledge of the complex mixing process itself. Tremblay (1989) implemented a loss expression developed by Stewart (1955) that uses the trailing edge boundary layer parameters and the base pressure to obtain an overall total pressure loss coefficient. The expression does not permit straightforward separation of the profile and trailing edge losses, which is necessary to analyze the relative effect of profile geometry on off-design loss. The overall blade loss calculated by this method showed reasonable agreement with experimental data.

2.4.3 Off-Design Loss Correlations

Gas turbine blades are designed to operate at a certain inlet flow angle for maximum efficiency. During the normal course of operation the inlet angle can vary significantly. Designing blades tolerant to inlet angle changes requires selection of a profile that exhibits low loss across an acceptable range. Off-design loss correlations are used for initial estimates of loss variation with inlet angle.

The most widely known and used of these correlations is that due to Ainley and Mathieson (1951). This method was refined by Dunham and Came (1970) and more recently by Kacker and Okapuu (1982). The total pressure loss in a blade cascade is divided into the profile, secondary, trailing edge and tip leakage loss components. To calculate off-design profile losses, the value for design is first obtained from figures presented in Ainley and Mathieson (1951) using the equation

$$Y_p = \{ Y_{p(\beta_1=0)} + \left(\frac{\beta_1}{\alpha_2} \right)^2 [Y_{p(\beta_1=\alpha_2)} - Y_{p(\beta_1=0)}] \} \left(\frac{t/c}{0.2} \right)^{\frac{\beta_1}{\alpha_2}} \left(\frac{Re}{2 \times 10^5} \right)^{-0.02} \quad (2.6)$$

The additional profile loss due to the change in incidence is then determined graphically from figures taken from Ainley and Mathieson as shown in Figure 2.2.

Moustapha *et al.* (1990) presented an off-design correlation developed from a database of off-design measurements from turbine blade profiles of recent design. The database used the leading edge diameter to chord ratio to account for the effect of leading edge geometry on the cascade off-design performance. It was found that this correlation predicted profile losses significantly better than that of the modified Ainley and Mathieson correlation. The off-design loss component is determined from the following equations

Figure 2.2 Determination of stalling incidence and variation of profile loss with incidence after Ainley and Mathieson (1951).

$$\Delta\phi_p^2 = 0.78 \times 10^{-5} \chi + 0.56 \times 10^{-7} \chi^2 + 0.4 \times 10^{-10} \chi^3 + 2.05 \times 10^{-19} \chi^6 \quad (2.7)$$

$$0 < \chi < 800$$

$$\Delta\phi_p^2 = -5.1734 \times 10^{-6} \chi + 7.69 \times 10^{-9} \chi^2 \quad (2.8)$$

$$-800 < \chi < 0$$

$$\chi = \left(\frac{d}{s}\right)^{-1.6} \left(\frac{\cos\beta_1}{\cos\beta_2}\right)^{-2} (\alpha_1 - \alpha_{des}) \quad (2.9)$$

Figure 2.3 shows the improved incidence loss correlation plotted against the cascade measurements used to develop the correlation. Measurements made by Tremblay (1989) that were not included in the database were predicted very well by the correlation.

2.5 Objectives of the Present Study

The objectives of this work are to determine experimentally the effect of axial velocity ratio, turbulence intensity, incidence and leading edge geometry on the performance of a turbine blade cascade. The profile incidence loss correlation presented by Moustapha *et al.* (1990) accounts for the effect of leading edge geometry through the diameter correction. It is impossible to isolate this effect due to the wide range of cascade geometries in the database. The blade to be investigated in the present study is based on a profile whose off-design performance has been documented by Rodger (1992) and Tremblay (1989). The new blade shares the earlier blade's aerodynamic parameters but has a larger leading edge diameter and a smaller wedge angle. The experimentally obtained loss measurements will be compared to those of the earlier blade to test the validity of the assumptions used to develop the improved off-design correlation.

Figure 2.3 Improved profile incidence loss correlation reproduced from Tremblay (1989).

Free stream turbulence intensity has a strong effect on turbine blade losses due to its influence on the laminar-turbulent transition process. This work will attempt to determine the effect of turbulence intensity on the performance of the blade being studied. The effect of turbulence intensity on the new blade will also be used to explain differences in the off-design losses measured by Rodger (1992) and Tremblay (1989).

Chapter 3

Test Apparatus and Software

3.1 Test Equipment

3.1.1 Wind Tunnel

Two open circuit wind tunnels are currently being used for linear cascade testing in the Carleton University Turbomachinery Lab. At any time, several research projects may be using the same tunnel to supply an air flow. Modular, interchangeable test sections allow different studies to proceed using the same wind tunnel. The older of the two wind tunnels has been used for tip leakage studies by Yaras (1990) and off-design cascade performance measurements by Tremblay (1989) and Goobie (1989). The second tunnel was designed to supply high quality low turbulence intensity flow to a turbine blade cascade test section. Tunnel commissioning was performed by Rodger (1992).

Experience gained from operation of the earlier wind tunnel resulted in several

notable design changes to improve flow quality and functionality. Debris screens at the fan outlet are easily and rapidly removed and cleaned. Flow quality has been improved. Air from the fan passes through a large settling chamber where fine screens remove large scale turbulent structures and other non-uniformities. From the settling chamber, air enters a rectangular contraction. This contraction has been designed to smoothly accelerate the flow without the occurrence of separation along the wall. The resulting flow delivered to the test section is of high quality with low turbulence intensity and uniform velocity distribution.

3.1.2 Test Section

The test section used for the present study was designed by Goobie (1989). Figure 3.1, reproduced from Rodger (1992), shows the test section with the rectilinear cascade and endplates mounted. The cross section is rectangular with an aluminum back wall and a plexiglass window. The assembly is mounted on a wheeled steel frame that allows it to be removed from the tunnel outlet as shown. Downstream of the test section inlet, vertical slots are used to remove the boundary layer that developed inside the contraction. The cascade blades are bolted to a 0.5" thick aluminum mounting plate. The plate fits flush with the wall of the test section and can be easily removed to allow access to the blades. Tests at different values of incidence are possible due to the large turntable supported by a steel shaft and two pillow block bearings. The bearing between the turntable and the rest of the test section is lined with a teflon strip. The turntable can be rotated through a turning angle of 60° . A protractor fixed on the back of the turntable allows a positioning accuracy to 0.1° . The front wall of the test section is made of plexiglass to allow visual inspection. The plexiglass just in front of the cascade is supported on its own wheeled frame and is easily pulled back to allow access to the cascade. The endplates with their "knife" leading edge can be seen. These endplates are presently used to ensure that two-dimensional flow conditions are present. A detailed description of the endplate design and development can be found in Rodger (1992).

Figure 3.1 Variable incidence cascade test section (from Rodger (1992)).

Adjustable surfaces are used to control the flow entering and leaving the cascade. These are used to make the five blades perform as if they are in an infinite cascade of identical blades. The top and bottom flaps adjust the flow above the top blade and below the lowest blade to obtain a uniform inlet flow angle. A floating wall at the bottom of the cascade compensates for changes in the cascade height as measurements are made at different angles. The two bypass valves control the amount of air that passes under the floating wall. Tailboards flush with the top and bottom blade trailing edges are used to trim the outlet flow. It has been found at higher positive incidences that the inlet control surfaces are rendered ineffective because of reduced flow past the top and bottom flap. For these angles a light perforated metal sheet was clamped between the outlet tailboards. This increased the cascade outlet back pressure and diverted more flow to the inlet control surfaces.

Flow measurements are made using three-hole pressure probes. These are mounted in planes upstream and downstream of the cascade. The probe locations are adjusted using stepper motor driven traverse gears. These are controlled through the parallel port of an IBM PC compatible. The traverse gears are bolted to the outside of the wind tunnel wall. The probe is mounted on rods which pass through vertical slots machined in the tunnel wall. On the other side of the tunnel wall the rods are fixed to the moving element of the traverse gear. Both pitchwise (vertical) and spanwise (horizontal) traverse gears can be used, allowing two degrees of freedom in the measurement plane. In the present study, only the vertical traverse gear was used for measurements. For convenience, the spanwise traverse gear was sometimes used to accurately position the probe at different spanwise locations.

3.1.3 Carleton Cascade Three

The cascade designed and manufactured for the present study is the latest of three tested at Carleton University. Tip clearance losses have been studied using Carleton

Cascade One (CC1), a front-loaded profile design. An aft-loaded blade profile of more recent design, Carleton Cascade Two (CC2), has been used in an on going study of off-design incidence profile losses. CC2 is a midspan profile of a power turbine blade from a midsize Pratt and Whitney Canada engine. The latest cascade, CC3, is the subject of the present study. The geometric parameters of this blade are presented in Figure 3.2. CC3 has been designed with the same velocity triangles as CC2 but has a leading edge radius of 16.70 mm, almost double the 9.43 mm diameter of CC2. In order to maintain the same blade thickness and have an acceptable velocity distribution, the leading edge wedge angle was reduced from 52.4° to 43° . The two profiles are shown superimposed in Figure 3.3.

The cascade was manufactured at the Carleton University Science Technology Centre using a numerically controlled milling machine. Full size blade data supplied by Pratt and Whitney Canada was scaled to the model dimensions using a CAD program. A spline was fit to these data points and the information transferred to a tool path generation program to create the part program used to machine the blade profiles. The resulting profile is machined to a surface tolerance of ± 0.05 mm.

The five cascade blades are bolted to a common aluminum plate. A fixed separation of 111 mm between the blades is maintained using locator pins. Small 1/4" pins have been press fit to the mounting plate and aligned with holes machined in the base of the blade. The blades in their test configuration can be seen in Figures 3.4 and 3.5. Each blade has been drilled through with a 3/4" hole. Bolt and pin holes corresponding to the holes at the base of the blade have also been placed in the tip face. These can be used to attach extensions to lengthen the blade and study the effect of aspect ratio changes. This was done out of consideration for possible future research and was not a part of the current program. The blade tips were covered with foam rubber to make a tight seal with the plexiglass window. This prevented tip-leakage from influencing the passage flow. The foam can be seen on the ends of the blades in Figure 3.5.

Figure 3.2 CC3 geometry and measurement plane nomenclature.

Figure 3.3 CC2 and CC3 profile and leading edge geometry comparison.

A sixth identical blade was manufactured to measure the static pressure distribution on the blade surface. Each of the previously built blades had been machined from a single piece of aluminum. The block of aluminum used for the instrumented blade was first cut into two pieces. The cut was positioned so that a row of static taps would lie exactly on the blade midspan. The two pieces were then fitted together with the same pin arrangement used to align the blades on the mounting plate. Two bolt holes were drilled from the tip face so the two pieces could be fixed together for the profile machining process. Before machining the profile, the outer half of the blade was drilled spanwise with 41 channels 2 mm in diameter and near what would become the blade surface. After these channels were complete, the profile was machined.

Static pressure taps were then drilled normal to the blade surface through to the channels. The orientation of each tap was determined from the profile data and channel locations. Using the CAD program, lines were drawn that passed through the centre of each channel and normal to the spline defining the blade surface. The intersection of the line with the blade profile and the inclination of the line from vertical was measured using the program. The inner half of the blade has been partially hollowed out to allow tygon tubing to be attached to each of the spanwise channels in the outer half. The tubing is then passed through the 3/4" hole in the inner half of the blade and the matching hole in the mounting plate. The instrumented blade can be mounted at any of the five blade positions.

There are 10 rows of 43 static pressure taps on the outer half of the instrumented blade, making it possible to examine the spanwise variation in blade loading, an important consideration in tip clearance studies. Each spanwise row of taps shares the same internal channel. To isolate a tap from its neighbours, the other taps in a row must be covered with tape. For the purpose of this study only the midspan row of taps was sampled. Static pressure distributions are an important piece of information in tip leakage studies. To accommodate future work that may be performed on CC3, the distance between tap rows is decreased near the blade tip. Graphical depictions of the static pressure taps are shown

in Appendix D, Figures D.1 and D.2. The tap coordinates and orientation are listed in Table D.1. Because of the finite thickness of the blade, tap pairs [23, 27] and [24, 26] near the trailing edge share the same connecting channel. Due to difficulties encountered in manufacturing, a leakage path exists between holes 42 and 43. Therefore these two taps are treated as if they share a channel. During sampling, one hole of a shared channel pair was taped over. After the open holes had been sampled, the test was paused. At this time the first hole was uncovered and the second was taped over. Sampling was then resumed.

3.1.4 Turbulence Generation Grid

The wind tunnel used for the present set of experiments has very good flow quality. Sometimes, such an environment is not necessary or desired. Some of the results from Rodger (1992) indicate that turbulence intensity may have a very strong influence on the two-dimensional loss at high incidences. To study the effect of turbulence intensity on cascade performance, a turbulence generating grid was designed by Abbott (1993) using a configuration similar to that of Blair *et al.* (1981). Correlations from Baines and Peterson (1951) relate the turbulence intensity to the grid solidity ratio (grid area/open area) and are used to ensure a uniform inlet flow turbulence. The grid is shown in detail in Appendix F, Figure 4. It is a square arrangement of 1/8" aluminum rods spaced 3" apart mounted in an aluminum frame. The frame slides into a holder bolted between the contraction exit and the test section inlet. The grid can be easily removed to conduct low turbulence intensity tests with the empty frame inserted in the holder. This configuration allows high and low turbulence intensity tests to be performed in quick succession on a particular endplate/cascade configuration. This became important as the setup procedure for any particular test case became more elaborate. The increased flow blockage when the grid was installed required running the fan at higher speeds, sometimes at almost the maximum permissible motor speed. This may be a constraint on future work that requires a flow with a higher Reynolds number.

3.1.5 Endplate Design

Adjustable endplates have been used by Goobie (1989), Tremblay (1989) and Rodger (1992) to control the two-dimensionality of the flow through the cascade. The endplates are manufactured from 1/8" aluminum with blade profile cut-outs to fit over the cascade blades. Boundary layer flow passes through the space between the endplates and the tunnel wall. A much weaker boundary layer develops from the endplate leading edge reducing secondary flow effects in the blade passage. Four threaded rods attached to the tunnel wall are used to set and maintain the endplate spacing. The endplates used by Tremblay (1989) and Goobie (1989) had a blunt leading edge. This caused a separation bubble to form except when highly convergent plate spacings were used. The endplates used in the present study are modelled after those developed by Rodger (1992). The plates have a flexible and adjustable "knife edge" which can be trimmed until the flow fully attaches at the endplate leading edge. The perspective view of the cascade in Figure 3.4 shows the bend in the plate and the pull-back mechanism. The leading edge assembly is mounted on a bracket that can be positioned at different heights. This is done to avoid interference with the tunnel floor and provide smooth flow when operating at higher positive incidences. In Figure 3.5 the leading edge has been positioned for tests at +20° incidence. Triangular cardboard cutouts have been fixed to the top and bottom of the endplates to further smooth out the flow.

A vertical slot has been machined through the inner endplate, 50 mm in front of the blade leading edges, to allow three-hole probe measurements inside the passage formed by the endplates. This slot is parallel to the traverse slot in the tunnel wall but is further downstream. A special rod assembly was used to connect the probe to the traverse gear for inlet traverses. This rod can be disassembled and removed without disrupting the test configuration. It was found that leaving the rod in place during an outlet measurement had a strong effect on the measured losses. For this reason, the rod is completely removed after the inlet test sequence is complete. The open slot in the endplate has a negligible effect on the inlet flow measurements but can influence loss measurements. During outlet

Figure 3.4 Perspective view of Carleton Cascade 3 (CC3)

Figure 3.5 End view of Carleton Cascade 3 (CC3) configured for +20° incidence.

measurements, the inlet slot was always covered with thin tape.

3.2 Data Acquisition Equipment

3.2.1 Pressure Measurement Equipment

Flow parameters are measured at the inlet and outlet of the cascade with three-hole pressure probes. Inlet measurements were made at midspan in a plane 60% of axial chord upstream of the cascade leading edges. Outlet measurements were made in a plane 140% of axial chord downstream of the leading edge. A probe with a short tail was used for inlet measurements to allow measurements close to the cascade without interfering with the leading edges. Colour coded tygon tubing connects the tail of the probe to a multi-port scanivalve. The scanivalve switches the pressure reading between the three holes. A typical probe is shown in Appendix F, Figure 3. The probe head is 0.7 mm thick and 2.0 mm wide and is located approximately six rod diameters ahead of the probe holder. This prevents the mounting system from interfering with the pressure measurements.

The two pressure probes used for this work were calibrated by Rodger (1992) according to the method established by Lewis (1966). The probe calibration was carried out in an open jet. The probe angle was varied from -10° to $+10^\circ$ in increments of 0.5° . At each probe position the holes of the probe were sampled and the measurement recorded to a data file. The probe angles and corresponding pressure measurements were used to obtain calibration curves for total pressure, velocity, static pressure, dynamic pressure and flow angle. A detailed description of the probe calibration procedure can be found in Rodger (1992). The probe calibration was periodically verified in case of accidental damage to the probe head.

Only two variable capacitance pressure transducers are available to convert

measured pressures to analog voltage signals. To overcome this limitation a scanivalve was used. The scanivalve is a multi-port switching device. It can handle up to 48 pressure sources as inputs. There is only one outlet pressure port. The output line can then be connected to any one input line. One transducer is directly connected to the upstream dynamic pressure measurement positioned at the test section inlet. The other is connected to the three-hole probe or the static pressure measurement blade through the scanivalve. The transducer voltages are monitored simultaneously so that each probe measurement can be referred to a common reference. The voltages are monitored using a pair of digital voltmeters. The analog voltage signals from the transducers are converted to twelve bit binary values by a Measurement Systems Analog to Digital Converter. Sampling and data storage is performed by a Sciometric CPU Unit controlled by an IBM PC compatible. The control commands and data are exchanged between the two through the PC's serial (RS-232) port.

When the connection from a pressure input is broken and the scanivalve is switched to the next port, the measurement system experiences a large pressure fluctuation. A steady reading is obtained once the pressure in the system has stabilized at the new level. The data acquisition program controlling the sampling pauses each time the scanivalve is switched. A reasonably conservative wait of 15 seconds is used. The pressure source is then sampled for 10 seconds to determine what level of gain to use with the CPU unit's internal amplifier. The data acquisition program determines and sets the correct amplification based on the largest voltage signal obtained during the 10 second sample. After completing this task the pressure source and reference dynamic pressure are sampled for 30 seconds. The CPU unit reads signals at approximately 3 samples per second. Non-dimensional pressure coefficients, C_p , for each probe hole (or static pressure tap) are calculated by normalizing the pressure read at the hole by the upstream reference pressure. The three C_p values obtained at each position of the three-hole probe are written to the data file along with the average voltage from each hole and the corresponding reference voltage.

3.2.2 Hot Wire Anemometry Equipment

Turbulence intensity measurements were made at a plane 220% of blade axial chord upstream of the blade leading edge. The free stream turbulence intensity is defined as the root mean square of the fluctuating component of velocity in the flow divided by the mean flow velocity. Velocity measurements were made with a DISA 55P11 hot-wire probe and a TSI 1053-B constant temperature anemometer. The probe was operated at a temperature of 280°C to maximize the difference with the free stream flow temperature and minimize errors due to slight fluctuations in the flow temperature. The hot-wire was calibrated in the low turbulence intensity wind tunnel for the velocity range 0 to 40 m/s. Velocity measurements were made with a three-hole probe mounted parallel to the hot-wire at the same axial and vertical position. The wind tunnel has been shown to have very uniform spanwise flow by Rodger (1992) so the two probes, when mounted a small distance either side of midspan, should see the same flow velocity. Hot-wire voltages at a series of airspeeds from 0 m/s to 40 m/s were recorded and a third order polynomial relating airspeed to velocity was obtained. The polynomial coefficients were used as settings on a TSI 1052 signal linearizer. The linearizer takes a voltage from the hot-wire constant temperature anemometer and converts it to a DC voltage on a 0 to 10 V linear scale. A 0 V output indicates 0 m/s, 10 V indicates a mean flow velocity of 40 m/s. Voltages between 0 and 10 V represent a fraction of the maximum velocity. A reading of 7.5 V would indicate a 30 m/s mean flow velocity. The DC output of the linearizer was measured using a DISA 55D31 voltmeter. This meter has a built in variable time constant to average out variations in the mean velocity. The AC signal from the linearizer was monitored on a Fluke 8050A digital multimeter. There was no time constant function on this meter so variations in the AC signal had to be removed by manual averaging. After some experience it was found that as few as ten samples were needed for a stable reading. The spectral content of the linearized signal was examined using an HP 3582A Spectrum Analyzer and hard copy was produced with an HP 7044A X-Y recorder.

3.3 Three-Hole Probe Data Acquisition and Analysis Software

The majority of experimental results presented in this work were obtained using the three-hole pressure probe. The three-hole probe tests are controlled with a IBM PC-XT compatible desktop computer. The computer is used for data sampling, storage and analysis. Previous experimental work relied on a number of programs written in Fortran and Turbo Pascal 3.0. Each of these programs performed a specific task, such as setting the initial probe position, sampling data, reducing data and calculating losses. A large number of the parameters required by these programs were the same and had to be re-entered at different points in the data acquisition and analysis process. The full procedure from test to final data was somewhat time consuming and the large number of operator inputs greatly increased the chance of introducing numerical or procedural errors. In addition, the existing programs were somewhat unforgiving and an incorrect parameter input could cause a program to crash or force the user to restart to correct an error.

Although adequate, the existing set of programs did not take full advantage of the capabilities of the PC compatible. The source code for all of the existing software was available and it was decided to consolidate the code into a single program for data acquisition and analysis. The goal of this project was to increase lab productivity and the quality of the results. Turbo Pascal 5.0 was the language of choice for the project as the majority of the existing code was written in an earlier version of this language.

3.3.1 Initial Changes to Data Acquisition Software

The first step in the project involved converting the original data acquisition program to the new language version. The use of the serial port for communication with the CPU unit required low level interaction with the PC hardware. A set of public domain asynchronous communication routines for use with Turbo Pascal 5.0 replaced the version

3.0 routines used by the earlier code. Once a stable version was available the programming effort focused on increasing functionality and user friendliness. Interactive menus were introduced to simplify parameter entry and equipment control during the set-up procedure. In conjunction with this, fault tolerant data entry routines were implemented. These routines verify that the entered value is compatible with the variable being changed. If it is not, the new value is rejected and the variable retains the value it had upon entering the input procedure. On program execution, all variables are initialized to reasonable default values. This eliminates the re-entry of items which remain constant over a series of tests. Several variables associated with the traverse set-up can also be stored in a default file. This protects against losing a test configuration due to an unexpected program termination.

3.3.2 Stepper Motor Control

Before starting a test the probe must be in its initial position. Before the current project a separate program was used. It was the responsibility of the operator to first locate the probe in a known reference position and then move it to its initial starting location. This has been integrated in the data acquisition program and is accessible from the motor control menu at any time before the test is actually initiated. There is no feedback mechanism from the probe traverse gear to the PC. The new program keeps track of the relative movement of the probe from its initial position. It is still necessary to set this initial position and periodically verify it. The scanivalve is also controlled through the PC parallel port. The scanivalve can be stepped to any one of its ports from the motor control menu while the program keeps track of which port is open. The motor control portion of the program can be compiled separately and used as a stand-alone program complete with menus.

3.3.3 Probe Traversal

The earliest version of the data acquisition program performed a traverse by specifying the number of stations and the distance between them. Several tests had to be run to fully resolve the details of the flow behind the cascade blade. This was improved upon by Rodger (1992) who modified the program to allow a series of traverses to be linked together, although a separate file was still created for each traverse. The latter method was adopted in the updated program and the data file format was modified to allow all the data to be consolidated in one file. The interface for specifying the number of stations and the step size in each range of the traverse has been designed for ease of use. The traverse is laid out in tabular form. Each row is a new traverse. Provided the probe position has been set properly, the menu tells the user exactly where the test will start and stop each specified traverse. The user must specify the number of sampling stations in each traverse, the distance between these stations and the distance to move to the start of the next traverse. Up to nine sequential traverses can be specified. Some minor modification of the program would be required to increase this number.

3.3.4 Data Reduction

Once the main sampling program had proven itself to be stable the process of adding features began. The data reduction program was converted into a series of Pascal routines from the original Fortran. This allowed data to be reduced "on the fly" by calling a data reduction procedure after sampling is finished at each probe position. The reduced data is stored in internal arrays. Existing data files could also be loaded into these arrays. Almost all experimentally obtained data must be examined in graphical form at some point. A series of graphing routines were added to the program to allow immediate inspection of results. Results from different tests can be directly compared by loading them into memory from data files. Newly reduced data, whether it has just been sampled or loaded from a file, is always added to the data already in memory. The number of

reduced data points being held is indicated to the user. It is up to the user to clear the arrays once they have been filled to the limit (300 data points). If a hard copy of a graph is required the reduced data can be saved to a file in the format used by Rodger (1992). This file can then be imported into a commercial spreadsheet or graphing program. The data reduction section can also be compiled separately and used as a stand alone program if desired.

3.3.5 Loss Calculations

Program development took place simultaneously with the experimental test program. As initial commissioning of the cascade and endplate combination progressed, outlet loss calculations were required. Two separate loss calculation programs were used. One calculated the mass averaged axial velocity and the total pressure loss coefficient at the measurement plane. The other calculated the loss generated as the downstream mixing process produced a uniform flow. The original versions of these programs required a reduced data file in the format used by Rodger (1992). These programs can be used with the reduced data files created with the new program, although this is not necessary. Since the new version of the program maintains all the loss data in memory after data reduction, integrating the loss calculations into the code was relatively straightforward.

The loss is calculated by integrating the total pressure loss distribution behind the blade with respect to a zero loss reference determined from the free stream data. The menu for loss calculations is very interactive and allows the loss data to be viewed graphically or in tabular form. To calculate the loss across one blade it is necessary to specify four points. The first and fourth point specified are the start and finish of the calculation. These are the limits across which the outlet flow losses are integrated. Both points must be free stream (zero loss) data and exactly 111 mm apart. This distance is the same as the spacing between the cascade blades in this experiment. The downstream wake of the blade must lie between these two points. The second and third points specified are

the start and end of the blade wake. These positions are easily identified graphically and verified by inspecting the actual data. Once all settings have been made the total pressure loss coefficient, the mixing-out loss coefficient and the mass averaged loss are calculated.

3.3.6 Future Use and Modification

The program in its present form is very user friendly. Much effort went into structuring menus and operating procedures to allow the operator to concentrate on the experiment. In some cases, the program was made deliberately "unfriendly" to prevent the operator from taking too many things for granted. There is enough potential for experimental error in the lab without encouraging careless operating procedures.

The new program has been thoroughly tested and debugged by comparing output with that of the earlier programs. A full listing of the Pascal source code and a description of the procedures used by the program can be found in Whitehouse (1993). The source code structure is laid out to easily allow additions and modifications to adapt to new experimental situations. Even if the user interface or specific sampling procedures are not used, there are a large number of Pascal subroutines that are easily accessible to other programmers. These subroutines control the traverse gear operations, the CPU unit and the scanivalve. Future users are encouraged to modify the program to their own needs.

Chapter 4

Experimental Procedures

4.1 Operating Point

The losses presented in this work were obtained at a Reynolds number of 300000. This is greater than the critical Reynolds number and can be achieved over a wide range of test configurations. This is also the same value used by Rodger (1992) for his two-dimensionality tests and by Tremblay (1989) and Goobie (1989) for their off-design loss measurements. The value is calculated based on the blade chord, the test section inlet velocity and the air viscosity. This last item can be determined from the inlet flow temperature and pressure. The inlet velocity is calculated from the upstream reference dynamic pressure. This is measured by a pitot tube positioned halfway up the tunnel wall and extended into the tunnel free stream flow. Directly across from this probe is a static pressure tap. Measurements made by Rodger (1992) show that there is a uniform total pressure distribution across the passage. It can therefore be assumed that the centreline dynamic pressure is the difference between these two measurements. The difference is

obtained by connecting the two taps and measuring the resulting pressure. To run the tunnel at the desired Reynolds number the indicated parameters are entered into the data acquisition program. Using the Reynolds number specified by the user, the program calculates the voltage that must be observed from the transducer connected to the upstream reference. The speed of the wind tunnel is adjusted until this voltage is indicated.

4.2 Cascade Flow Conditions

4.2.1 Flow Uniformity and Periodicity

The exact duplication of the conditions experienced by a set of blades in an actual turbine configuration would require a linear cascade having an infinite number of equally spaced blades. Since this is a practical impossibility, other means must be used in an experimental setup. The cascade being studied has five blades. Adjustable surfaces at the top and bottom of the cascade are used to control the flow through the cascade and make the three centre blades behave as if they are in an infinite series. The control surfaces are set in an iterative manner known as "trimming" the flow. First, the inlet flow to the cascade must be uniform regardless of the cascade orientation. This ensures that each blade has the same inlet angle. Then tests are performed to determine if each blade produces the same loss and turns the flow the same amount. Only when these conditions are met can a loss or deviation measurement be considered representative of a blade in an ideal cascade.

The trimming process must be performed for each incidence and endplate setting combination. Initially, all control surfaces are set to reasonable values. Rodger (1992) documented the surface and endplate settings that gave AVR close to unity. Corresponding information for CC3 can be found in Appendix F.

4.2.2 Inlet Measurements

Before loss measurements can be made it is first necessary to establish a uniform flow across the cascade inlet. Each blade should encounter a midspan flow parallel to the axial direction to ensure that each blade has the same inlet angle. Starting with the initial control surface settings, the inlet is traversed at the midspan from top to bottom with the three-hole pressure probe. It is mounted in a plane 60% of axial chord (C_x) upstream of the cascade leading edges. The spacing between stations on the initial test can be relatively large (25-50 mm) since only an indication of the amount of control surface adjustment is needed. The top and bottom flap angles are adjusted an appropriate amount and the inlet is traversed again. This process is repeated until the inlet flow angle is within $\pm 0.5^\circ$ of the probe angle. As this condition is satisfied, the spacing between stations should be reduced to increase data resolution and ensure that uniformity is present across the passage. Figure 4.1 shows an inlet flow after the trimming process has been completed. When establishing inlet uniformity at higher effective incidences a periodic variation in the inlet angle was observed. This periodicity corresponded to the blade spacing, suggesting it was caused by the upstream influence of the cascade blades.

4.2.3 Outlet Flow Uniformity

Once a uniform inlet flow has been established, the probe is moved to the outlet traverse plane and aligned with the flow. The three-hole pressure probe was calibrated for a $\pm 10^\circ$ range of flow angles. It is important, therefore, to orient the probe so that the flow angle incident on the probe never exceeds this range. Fortunately, few problems were encountered with outlet flow alignment. If the endplates are being used, the inlet traverse slot machined in the inner plate must be taped over to prevent the outlet flow from being

affected by flow through the open slot.

Before performing outlet loss measurements, the flow must be uniform and the blade losses must be periodic. An outlet angle survey can be obtained with a coarse traverse. This yields data such as that shown in Figure 4.1. Any significant variation of the flow angle across the cascade outlet can be easily identified from this data. The tailboards can be used to make minor changes to the flow angle. However, provided the initial tailboard settings are reasonable, the flow is usually quite insensitive to attempts to fine trim the angle. Deviation measurements presented in Chapter 5, however, show that under the flow conditions at $+15^\circ$ and $+20^\circ$ incidence the outlet angle with respect to the blade angle is quite different than the values seen at design. This higher flow deviation made it necessary to select quite different tailboard settings. These settings are documented in Appendix F for future reference. The periodic outlet angle fluctuation seen in Figure 4.1 was caused by the high shear flow in the wake. As the probe is moved into the wake the velocity gradient across the probe head causes an apparent shift in the flow angle. These fluctuations are accounted for by calculating the mass averaged angle.

4.2.4 Outlet Flow Loss Periodicity

The outlet loss periodicity is examined by performing a traverse across the cascade outlet that includes the wakes from the three centre blades. By comparing the wake geometries graphically or calculating the loss behind each blade, it can be determined if each blade is operating at the same level of loss. Earlier researchers performed full traverses to obtain this data. During the programming effort described in Chapter 2, a new type of test procedure was developed to assist in outlet periodicity checks. This procedure is known as the quick traverse and is discussed in detail in Section 4.5. The results of a quick outlet traverse are presented with the results from a full outlet traverse in Figure 4.2. With this technique the full outlet can be traversed in a short period of time. The result is a detailed picture of the total pressure distribution behind the blades. The wakes

Figure 4.1 Flow angle measurements at design incidence.

Figure 4.2 Full and quick outlet traverse comparison.

of the three centre blades can then be compared visually. Tailboard adjustments can be made to correct any significant loss variation across the passage. Such changes may affect conditions at the inlet so the flow uniformity would have to be retested. If necessary, inlet control surfaces are adjusted and the probe returned to the outlet. These procedures would be repeated until flow uniformity and cascade periodicity have been fully established.

4.3 Three-Hole Probe Measurements

4.3.1 Total Pressure Loss Measurements

After the inlet and outlet uniformity of the flow have been firmly established, proper measurements can be made. An inlet traverse with the three-hole probe is used to determine the mass average inlet angle. Very good inlet flow uniformities were obtained using the procedure described above. The calculated mass average inlet angle will usually have a magnitude between 0° and 0.5° . The cascade inlet angle is corrected by this amount before the mass averaged inlet velocity is calculated. The corrected inlet angle is also used when calculating the outlet mass averaged velocity and the blade total pressure loss.

The total pressure loss measurements are calculated using the data obtained from a full outlet traverse. The free stream fluid passing between the blades is sampled using a relatively coarse station spacing. When in the wake, the probe is moved only a small distance between stations in order to fully resolve the detail of the rapidly changing parameters. The total pressure loss distribution obtained from a typical full outlet traverse is shown in Figure 4.2. The midspan total pressure loss coefficient is found by numerically integrating the total pressure loss with respect to the free stream data reference. Further losses are incurred as the blade wakes are mixed out into a uniform flow far downstream of the outlet measurement plane.

The fully mixed out total pressure loss coefficient is calculated from the following equation

$$\overline{Y}_m = \frac{1}{\dot{m}} \int_0^{\delta} Y d\dot{m} + 1 - \frac{2 \int_0^1 V_1^2 d(y/s)}{(\int_0^1 V_1 d(y/s))^2} + \frac{\int_0^1 V_1^3 d(y/s)}{(\int_0^1 V_1 d(y/s))^3} \quad (4.1)$$

The first term is the total pressure loss incurred up to the measurement plane. The integral is evaluated from the data in the manner specified in Section 3.3.5. The next three terms represent the additional losses arising from the downstream mixing where V_1 is the velocity in the outlet plane. The derivation of this expression can be found in Goobie (1989). Measurements made by Goobie also show that the total pressure loss obtained from Equation 4.1 is not dependent on the axial location of the measurement plane.

4.3.2 Deviation Measurements

Flow deviation data are presented with the total pressure results in Chapter 5. The deviation is calculated by subtracting the flow angle calculated by the data analysis program from the blade metal angle. This information is very important to the turbine designer. The outlet flow angle from one blade row is the inlet flow for the following blade row and is a major factor in determining that blade's geometry.

4.3.3 Accuracy of the Three-Hole Probe

The three-hole probe is the primary instrument used in this and previous off-design loss studies. It is used to establish the inlet uniformity and the flow periodicity and to calculate total pressure loss and flow deviation. Tremblay(1989) and Goobie(1989) performed an accuracy analysis on the probe using the calibration apparatus. Flow angles were seen to have errors of $\pm 0.5^\circ$ in the -10° to $+10^\circ$ range for which the probe

calibration had been performed. The worst case dynamic pressure measurement error was found to be about 1%.

Tremblay (1989) found that the non-uniformity in the flow downstream of the cascade could have some additional effect on measurement error. The high shear flow in the trailing edge wakes causes a velocity gradient across the probe tip. Performing the analysis at design incidence, where the narrow wakes cause the steepest velocity gradient, Tremblay (1989) found the error in flow angle was less than $\pm 1^\circ$. The maximum error in dynamic pressure due to the shear effect was 3% and the total error for the mass average total pressure loss was $\pm 5\%$. It can be seen then that the error due to the shear flow appears to be negligible. Another important source of error could occur if the probe head were accidentally damaged. The probe would then have a different geometry than that for which it was calibrated. When not in use, the probe head was covered with a piece of Tygon tubing. The probe should be cleaned regularly and calibration checks should be performed periodically.

4.4 Flow Two-Dimensionality and AVR

For the purposes of the present study, the measurements to be discussed in Chapter 5 are presented as two-dimensional losses. This can only be considered valid if the flow at the location being measured can be considered two-dimensional. Much of the previous work performed on CC2 has dealt with establishing the criteria for two-dimensional flow. Ideally, measurements would be performed on a cascade of blades having infinite aspect ratio. Physical limitations and experimental necessity assure the presence of a finite passage width. The growth of boundary layer fluid on the tunnel endwall further reduces the effective width of the passage. A complex development of secondary flows is caused by the portion of the cascade blade immersed in the endwall boundary layer. The effects of these secondary flows can be seen in the plot of the spanwise loss distribution of CC3

at design incidence in Figure 4.3. Losses remain at a relatively low level near the blade midspan. Closer to the endwall, however, they begin to increase rapidly. Two-dimensional loss measurements are made at midspan to avoid the majority of the secondary loss effects. The insensitivity of the losses to the spanwise location in this area leads to the conclusion that the flow is for the most part two-dimensional.

The existence of the constant loss region was used by Tremblay (1989) and Goobie (1989) as the main criteria for flow two-dimensionality. The extent of secondary flow intrusion into the passage was also examined using oil film flow visualization. Tremblay's (1989) spanwise loss distribution, obtained using endplates, showed a widening of the low loss region with the same midspan loss coefficient. A spanwise loss distribution and corresponding flow visualization obtained at $+20^\circ$ incidence were also presented. These tests were conducted using the endplates and a large amount of flow convergence through the passage is indicated. No comparison with the blade operating at this high incidence without endplates is provided.

Rodger (1992) examined the effect of outlet to inlet axial velocity ratio on the flow two-dimensionality. If a simple control volume is applied to an ideal two-dimensional flow through a cascade the net axial velocity change across the blades is zero. The axial velocity ratio (AVR) is a common measure of the flow acceleration through the cascade passage. The improved endplates used by Rodger (1992) allowed the AVR to be varied. Blade losses were observed to increase as axial velocity ratios decreased. When AVR was decreased the losses were reduced. In some cases they appeared to level off to a constant value as the axial velocity ratio was increased above unity. The use of AVR as a test of two-dimensionality appears convincing but it should be noted that the axial velocity ratio is based on values that have been averaged across a full blade passage. There may be significantly different convergence or divergence in the boundary layer flow on opposite blade surfaces. Detailed three-dimensional measurements would be needed to determine the local effects on flow convergence.

Figure 4.3 CC3 spanwise loss distribution at 0° incidence, $AVR = 1.003$.

In the present work the endplates were used to obtain loss measurements at AVR values of unity. To trim the AVR, all of the above flow uniformity and periodicity tests must be carried out and loss measurements made on the cascade with an initial endplate spacing. Based on the axial velocity ratio obtained from this test, the endplates would be either converged or diverged. The flow trimming process would then be repeated and another loss measurement obtained. Endplate spacing matrices are presented in Appendix F. It is cautioned though, that these are only meant as a guide. It was found that the AVR could be affected by blockage in the space between the endplates and the tunnel walls. Some methods used to seal the gap between the endplates and the wind tunnel walls introduced extra blockage into the plate-wall gap, producing different values of AVR for identical cascade configurations. This has not been investigated further but may provide a means of increasing the range of AVR beyond what the plate spacing limits allow.

Ideally, the mass average axial velocity ratio would be evaluated at the blade leading and trailing edge planes. Equipment constraints make this impossible. The axial velocities are calculated from the information obtained at the measurement planes. Both of these planes are inside the endplate region. The flow will be accelerated (or decelerated) in the distance between the probe location and the blade. In this study a linear interpolation has been used to make the measured values relevant to the blade inlet and outlet planes. The procedure used is presented in Figure 4.4. Strictly speaking, the flow velocity in a converging duct does not vary linearly. However, even the largest and smallest values of AVR recorded are still relatively close to unity and the linear approximation can be considered adequate.

4.5 Outlet Flow Survey Using the Quick Traverse Technique

The quick traverse was developed to allow the rapid and accurate identification of flow phenomenon location. This information is used to optimize traverse station

$$V_{l.e.} = V_1 + \frac{(V_2 - V_1) X_1}{T}$$

$$V_{t.e.} = V_1 + \frac{(V_2 - V_1) (X_1 + Cx)}{T}$$

Figure 4.4 Schematic top view of cascade and endplates presented with the interpolation scheme used to determine the inlet and trailing edge mass averaged velocities.

spacing during full sampling to resolve important flow features with minimal sampling time. The technique is a stripped down three-hole probe sampling procedure. Rather than sample every probe hole, only the centre hole is monitored, essentially using it as a pitot tube. This makes it possible to avoid using the scanivalve and waiting the required periods of time to allow the measurement system to stabilize after switching between holes. In the quick traverse, the distance between measurement stations is kept very small to obtain high data resolution. This also helps to keep the total pressure difference between stations small, allowing the pneumatic measuring system to adjust quickly. These simplifications produce a very accurate snapshot of the outlet total pressure distribution. A quick traverse across one wake is compared with a full traverse of the same flow conditions in Figure 4.2 showing that overall shapes are almost identical. Fluctuations can be seen in the free stream data of the quick traverse. Whether these are due to natural variations in the flow or are a byproduct of the "quickness" of the technique is unknown. In any case they have no bearing on current quick traverse applications.

Total sampling times of as little as 5 seconds at each station are used, allowing a traverse across the full range of the outlet (about 300 mm) in approximately one hour. For comparison purposes, a full outlet traverse with 1 mm station spacing would take over 15 hours. Quick traverse data are written to a data file that records the probe location and the centre hole pressure coefficient, C_{p1} , at that point. It is stressed that data obtained with quick traverses has not been used to calculate losses but the possibility has not been ruled out and may warrant future investigation.

During the present study the technique was found useful in a number of situations. As can be seen in Figure 4.2, there is a sudden change in total pressure at the interface between the wake and free stream fluid. The location of this change can be easily identified with a quick outlet traverse and then used to set up a full outlet traverse. The full outlet traverse would fully resolve the wake but would not waste time over sampling in the free stream region. A quick outlet traverse can also be used to evaluate outlet periodicity and the effect of tailboard changes by comparing graphically the shape, size

and depth of the blade wakes. Once a good midspan wake periodicity has been achieved, a quick check of the periodicity at spanwise locations closer to the end plates is sometimes useful. A very large wake behind only one of the blades may indicate corner stall or flow separation.

4.6 Free Stream Turbulence Intensity Measurements

Turbulence intensity measurements were performed with the single wire hot-wire probe described in Chapter 3. The free stream turbulence intensity (Tu) is defined as the root mean square (rms) of the fluctuating component divided by the mean flow velocity. The wind tunnel in which the present set of cascade measurements were performed normally has a low free stream turbulence intensity ($Tu = 0.3\%$). This value was first established by Rodger (1992). Tremblay (1989) and Goobie (1989) performed their tests on the older wind tunnel which supplied a flow with a measured Tu of 1.5%. The high positive incidence loss measurements on both CC2 and CC3 are significantly higher than those quoted by Tremblay on CC2. Initially this increase was attributed to the effect of the different turbulence intensities on the profile boundary layer development. The turbulence generating grid designed by Abbott (1993) and described in Chapter 3 was used to raise the cascade inlet turbulence intensity. With the grid mounted at the test section inlet, free stream turbulence intensity levels were increased to 3.0% at the cascade inlet. This is a similar environment to the early CC2 tests and also closer to the level seen by a blade operating in a gas turbine.

A midspan traverse of the cascade inlet was made with a single wire hot-wire probe. The hot-wire measurements of Rodger (1992) were made between two blades and only at this one location. The tail end of the hot-wire probe holder extended into the blade passage, making it impossible to perform a vertical traverse at this axial location. To perform the traverse, the probe was moved to a plane 220% of axial chord upstream of

the cascade leading edges. This also placed the probe ahead of the endplate leading edges. Traverses were performed both with and without the grid.

The results of the two traverses are shown in Figures 4.5 and 4.6. The low turbulence intensity flow appears to be very uniform. The high turbulence intensity flow also appears to be relatively uniform except for a region of lower turbulence intensity fluid above the centre blade. The region appears in the same location regardless of grid orientation. At the drop region there was a slight increase in the mean signal and a large drop in the rms signal from the hot-wire. The indication is that the effect of the grid on the flow has not been entirely uniform. The lowest value of Tu measured using the grid is still almost twenty times any of the results of the low turbulence intensity tests. Since the purpose of the grid was to investigate the effect of a higher level of turbulence intensity on blade losses, the range of values shown in Figure 4.6 was considered reasonable.

Spectral analyses are presented of both high and low Tu flows showing dominant frequencies and the distribution of turbulent energy in the flow in Figures 4.7 through 4.10. The low Tu spectral analysis is relatively "quiet". There are some differences between the analyses presented here and those of Rodger (1992). His plot shows a very distinct spike at 230 Hz associated with the passing frequency of the ten blade fan. The same point exists at roughly the same location in Figure 4.7. However, there appears to be much more energy associated with the frequencies between 100 Hz and 200 Hz. It is believed that this is due to vibrations in the rod used to hold the probe. Rodger (1992) used a straight rod. The spectral analyses presented here were made in the same plane as that used for the hot-wire inlet traverses described earlier. The rod used to hold the probe had a crank shape in order to bend around the endplates and hence was relatively long. The vibrations in the rod caused by the airflow would cause the probe to move. These movements would appear as apparent turbulence. The error introduced by this apparent vibration is insignificant compared to the energy levels observed in the flow with the

Figure 4.5 Low Tu hot-wire traverse at cascade inlet.

Figure 4.6 High Tu hot-wire traverse at cascade inlet.

Figure 4.7 Spectral analysis of the hot-wire signal (0-500Hz, no grid).

Figure 4.8 Spectral analysis of the hot-wire signal (0-5 kHz, no grid).

Figure 4.9 Spectral analysis of the hot-wire signal (0-500 Hz, with grid).

Figure 4.10 Spectral analysis of the hot-wire signal (0-5 kHz, with grid).

screenings can be seen in Figures 4.9 and 4.10. The spectral analysis plots also show that most of the energy associated with the turbulence intensity is in the lower frequencies.

4.7 Static Pressure Measurements

Blade surface static pressure measurements were made by replacing the centre blade of the cascade with the instrumented blade. The static pressure measurements were made after the loss measurement tests had been completed. The control surface and endplate settings used for each test had been recorded. The settings that gave an axial velocity ratio close to 1.0 at each incidence were reused for the static pressure tests. To verify that the same flow conditions were present as each configuration was repeated, outlet free stream measurements were made and compared to the free stream results from the earlier tests. Pressure distributions were obtained for high and low turbulence intensity flows from 0° to +20° effective incidence in 5° steps. The only condition not tested was +5°, high T₁, as no corresponding loss measurements had been made at this incidence. Each tap is sampled in the same manner as a hole on the pressure probe: a waiting period of 15 seconds, followed by a gain check of 10 seconds and a sampling time of 30 seconds. For each hole sampled, a pressure coefficient based on the measured value and the upstream dynamic and static reference pressures is written to a data file

$$C_p = \frac{(P_s - P_{ref})}{q_{ref}} \quad (4.2)$$

This coefficient is then referenced and normalized to the cascade inlet conditions through the following equation

$$\begin{aligned} C_p &= \frac{P - P_1}{q_1} = \left(\frac{P - P_{ref}}{q_{ref}} - \frac{P_1 - P_{ref}}{q_{ref}} \right) \frac{q_{ref}}{q_1} \\ &= (C_{p \text{ mens}} - \frac{P_1 - P_{ref}}{q_{ref}}) \frac{q_{ref}}{q_1} \end{aligned} \quad (4.3)$$

P_1 and q_1/q_{ref} are extracted from the corresponding inlet flow data. Repeatability tests performed on the static pressure measurements have shown an uncertainty of less than one percent on the value of C_p .

4.8 Base Pressure Measurement

The turbine blade models used in profile loss measurement studies have a pressure tap in the blade trailing edge aligned with the outlet metal angle. The base pressure coefficient, C_{pb} is defined as

$$C_{pb} = \frac{(P_{base} - \bar{P}_2)}{\bar{q}_2} \quad (4.4)$$

This is calculated from the pressure coefficient recorded during static pressure measurements and the mass averaged static and dynamic pressures at the outlet obtained from earlier tests. The trailing edge pressure coefficient is converted to C_{pb} using the formula

$$C_{pb} = \frac{C_{p\ te} - \bar{C}_{p\ out}}{\bar{q}_{out}} \quad (4.5)$$

The overbars indicate mass averaging and the subscripts *te* and *out* refer to the trailing edge and probe measurement planes respectively.

Chapter 5

Presentation and Discussion of Results

In this chapter, measurements made on Carleton Cascade Three (CC3) and Carleton Cascade Two (CC2) are presented in graphical form to simplify analysis. The numerical values of the loss measurements are tabulated in Appendix B. Static pressure measurements can be found in Appendix D. To facilitate comparisons between the present work and that of Rodger (1992) and Tremblay (1989), Tables 5.1 and 5.2 have been compiled. In Table 5.1 the turbulence intensities and range of AVR are specified for each incidence tested. W1 and W2 refer to the old and new wind tunnels respectively. Table 5.2 shows the full range of tests that were performed on CC3. The blade loading column indicates if static pressure measurements were obtained.

Following the work of Rodger (1992), close attention was paid to the blade passage AVR in order to ensure adequate two-dimensionality. When sufficient data is available, the effect of AVR on blade losses and deviation is presented. Rodger (1992) expressed concern about the effects of turbulence intensity on the blade losses.

Author	Parameters	CC2	CC3
Tremblay (1989)	Tu	1.5% (W1)	
	i	-25° to 25°	
	AVR	1.05 to 1.08	
Rodger (1992)	Tu	0.3% (W2)	
	i	-20°, 0°, 20°	
	AVR	0.953 to 1.087	
Whitehouse (1993)	Tu		0.3%, 3.0% (W2)
	i		-20° to 20°
	AVR		0.950 to 1.019

Table 5.1 Summary of cascade off-design test programs

Effective incidence	0.3% Tu Blade Loading	3.0% Tu Blade Loading	AVR Range
-20°	No	No	0.950
-10°	No	No	0.941 to 0.968
0°	Yes	Yes	0.950 to 1.010
5°	Yes	No	1.004
10°	Yes	Yes	1.002 to 1.007
15°	Yes	Yes	0.953 to 1.009
20°	Yes	Yes	0.979 to 1.019

Table 5.2 Summary of CC3 off-design measurements

In the present study, measurements were made under two different levels of turbulence intensity so that its effect on blade performance could be studied across a range of incidence and AVR.

5.1 Effect of AVR and Tu on Design Performance

The present study is focused on determining the effect of leading edge geometry on the off-design performance of a linear cascade. Information on the effect of axial velocity ratio (AVR) on the blade performance was obtained during the process of establishing two-dimensional flow through the centre blade passage (AVR = 1.0). Further evidence of two-dimensionality can be seen in the spanwise loss measurements at design incidence as presented in Figure 4.3.

5.1.1 Flow Visualization

The difference in turbulence intensities was believed to be the cause of the large difference in off-design losses recorded by Rodger (1992), $Tu = 0.3\%$ and Tremblay (1989), $Tu = 1.5\%$. The wind tunnel used in the present study is the same as that used by Rodger (1992). The cascade inlet free stream turbulence intensity was increased to 3.0% using a grid of steel rods installed at the entrance to the test section. Flow visualization was used to examine the effect of the increased turbulence intensity on the suction surface boundary layer development. A thin film of oil coloured with pigment was applied to the blade suction surface. The blade was returned to the tunnel and subjected to the design point flow. The blade was then removed and photographed. Markings were applied to the blade suction surface to allow easy comparison between different tests. The position of these markings on the blade profile is shown in Figure 5.1.

Figure 5.1 End-view of cascade blade superimposed on suction surface flow visualization to show location of surface markings.

Oil film visualizations at 0.0° incidence with turbulence intensities of 0.3% Tu and 3.0% Tu are shown in Figure 5.2a and Figure 5.2b. Both visualizations were done at an AVR of unity. The convergence of the flow through the passage due to the endplate settings can be observed. The laminar to turbulent boundary layer transition appears as a well defined line on the blade surface where the higher shear stress sweeps more oil to the trailing edge. There is no significant difference between Figures 5.2a and 5.2b. This indicates that the turbulence intensity increase at design incidence has had little effect on the boundary layer development.

5.1.2 Blade Losses

The loss coefficients obtained for CC5 at design incidence with free stream turbulence intensities of 0.3% and 3.0% are presented in Figure 5.3. As the AVR is increased, the losses decrease. The increased AVR was obtained by converging the endplates to accelerate the flow through the blade passage. It would be expected that the increased acceleration would produce higher losses due to the increased shear stress in the blade surface boundary layer. Rodger (1992) found that as the AVR was increased, there was a corresponding decrease in the base pressure coefficient. This causes a reduction in the trailing edge loss and compensates for the increased viscous loss.

The free stream turbulence intensity of the wind tunnel ($Tu = 0.3\%$) is much lower than the level found in an actual operating engine, which is usually between 2 and 10 percent. It is also much lower than that level measured in the tunnel used by Tremblay (1989) and Goobie (1989) for their off-design measurements ($Tu = 1.5\%$). A grid of aluminum rods was installed at the inlet of the cascade to bring the cascade inlet Tu to 3.0%. At design incidence the effect of higher Tu seems to shift the losses on Figure 5.3 down slightly. This suggests that for a given axial velocity ratio the losses will be lower at higher turbulence intensities. It was expected that the higher level of turbulence intensity would cause the blade surface boundary layer to undergo laminar-turbulent

a) Layer oil flow visualization at +0.0° incidence, 0.3% Tu

b) Layer oil flow visualization at +0.0° incidence, 3.0% Tu

Figure 5.2 Suction surface flow visualization at design incidence.

Figure 5.3 CC3 loss variation at design incidence.

transition earlier. This should increase losses, as a longer portion of the blade will be under a high shear stress turbulent boundary layer. A possible explanation for this contradiction may be the effect of higher turbulence intensity on the trailing edge losses, which are highly dependent on the base pressure coefficient. The base pressure coefficient recorded at 0.3% Tu was -0.0399 while at 3.0% Tu it was increased to -0.0131.

The profile design of Carleton Cascade Three (CC3) is a modification of Carleton Cascade Two (CC2). The two blades share the same aerodynamic parameters but have different leading edge geometries. CC2 was first tested in a wind tunnel with a turbulence intensity of 1.5% by Tremblay (1989). This work was done without a rigorous examination of the axial velocity ratio, an important consideration in two-dimensional loss measurements. Rodger (1992) presented design incidence loss measurements for a range of AVR. These measurements were made at low turbulence intensity and are presented with the corresponding measurements from CC3 in Figure 5.4. These data show a decrease in losses with increasing AVR. Above an AVR of 1.0 losses stay relatively constant. The design incidence losses of CC3 are higher than those of CC2 at an AVR of unity.

5.1.3 Flow Deviation

The effect of AVR on the cascade deviation measurements is shown in Figure 5.5. The flow deviation is calculated as the outlet metal angle minus the flow angle. There is no easily identifiable effect of AVR on deviation due to the limited number of data points. The higher turbulence intensity measurements appear to decrease as the AVR is reduced.

The deviation measurements performed on CC3 and CC2 for $Tu = 0.3\%$ are presented in Figure 5.6. The results from CC2 show a gentle increase as the axial velocity ratio is increased. This trend appears to be similar for CC3 with a deviation about one degree less than that of CC2.

Figure 5.4 CC2 and CC3 loss variation at design incidence, 0.3% Tu .

Figure 5.5 CC3 flow deviation variation at design incidence.

Figure 5.6 CC2 and CC3 deviation variation at design incidence.

5.1.4 Blade Static Pressure Distribution

Static pressure measurements were made at $AVR = 1.004$ for $Tu = 0.3\%$ and at $AVR = 1.008$ for $Tu = 3.0\%$ on CC3. The two sets of data are presented in Figure 5.7. The pressure measured at each tap is referenced to the cascade inlet static pressure and normalized on the inlet dynamic pressure to produce the coefficients presented. When the data are normalized in this fashion the measurements at 0.3% and 3.0% Tu fall on the same curve. The only noticeable difference is over the last 20% of axial chord. The high turbulence intensity flow appears to be leaving the blade with slightly more velocity than the low Tu condition. These differences are possibly the result of the slightly different axial velocity ratios under which the tests were conducted. Rodger (1992) showed that increasing the AVR resulted in higher loading levels on the rear part of the blade.

A comparison between the two blades at an AVR of unity is only available for the 0.3% Tu conditions and is shown in Figure 5.8. Although the shapes of the curves are different, the overall level of loading is quite similar. The front half of the suction surface of CC2 is more highly loaded than that of CC3. The minimum pressure on the suction surface of CC2 is at 46% of axial chord (C_x) followed by an almost linear increase to the blade trailing edge. The minimum pressure on the suction surface of CC3 is at 52% C_x . After this the pressure increases gradually until 80% C_x where it starts to rise more rapidly until reaching the trailing edge. The effect of the different leading edge geometry is quite pronounced on the pressure surface. CC2 has a relatively high pressure maximum near the leading edge. After this maximum the pressure decreases gradually until reaching a similar level as CC2. The pressure increase on CC3 is much less than on CC2 and the pressure remains relatively constant until 66% C_x where the flow begins to accelerate to the trailing edge. At the stagnation point on the blade leading edge there will be a pressure coefficient of 1.0. One of the static pressure taps on CC2 appears to be very close to this point while the corresponding point on CC3 has not been picked up. This indicates that the stagnation point is further back on CC2.

Figure 5.7 CC3 midspan blade static pressure distributions at design incidence.

Figure 5.8 CC2 and CC3 midspan blade static pressure distributions at design incidence.

Although the static pressure measurements are being performed on only one blade, the pressure distribution is very much a function of the geometry of the channel. Both blades have the same inlet flow angle but because of differences in the leading edge geometry the metal angle of CC2 is lower than that of CC3. This means that CC2 is working at a negative incidence with respect to CC3.

5.2 Effect of AVR and Tu on Off-Design Performance

In this section, data on the effects of AVR and Tu on the off-design performance of CC3 are presented. A number of data points at different values of AVR were obtained during the process of acquiring an AVR of unity for purposes of two-dimensionality. Spanwise measurements taken at $+10^\circ$ incidence are presented in Figure C.1 in Appendix C. The two-dimensionality data is compared to the results of Rodger (1992) in order to obtain further information on the blade two-dimensionality.

5.2.1 Flow Visualization

The midspan loss measurements made at $+15^\circ$ and $+20^\circ$ incidence are significantly higher than the values obtained at design. The results of flow visualization at $+20^\circ$ incidence, shown in Figures 5.9 and 5.10, suggest that this is the result of separation on the aft portion of the blade suction surface. Dots of oil coloured with fine paint particles were placed on the cascade centre blade suction surface. When placed in the wind tunnel flow, the dots near the leading edge of the blade move rapidly in the flow direction under the influence of high shear stress.

Over the aft portion of the blade there is very little dot movement, indicating low shear stresses which may be associated with a large separation bubble. This corresponds

a) Dot oil flow visualization at $+20^\circ$ incidence, 0.3% Tu

b) Layer oil flow visualization at $+20.0^\circ$ incidence, 0.3% Tu

Figure 5.9 Suction surface flow visualization at $+20^\circ$ incidence, 0.3% Tu.

a) Dot oil flow visualization at $+20^\circ$ incidence, 3.0% Tu

b) Layer oil flow visualization at $+20.0^\circ$ incidence, 3.0% Tu

Figure 5.10 Suction surface flow visualization at $+20^\circ$ incidence, 3.0% Tu.

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to the flow visualization presented by Rodger (1992) and is not present in the flows examined by Tremblay (1989). This difference is used by Rodger to explain the large increase in losses on CC2 with respect to the results of Tremblay.

5.2.2 Blade Losses

5.2.2.1 Effect of Turbulence Intensity

The variation of the losses at 0° , $+15^\circ$ and $+20^\circ$ incidence with AVR is presented in Figure 5.11. The higher incidence measurements show very strong sensitivity to AVR. The changes in the midspan loss measured at design incidence are insignificant compared to those at the two higher incidences. The limited data shown here do not suggest any significant effect of turbulence intensity. The curves shown are derived from the trends shown in Rodger (1992) and indicate that the high Tu measurements have a higher loss than the low Tu case at AVR of 1.0. More data are required to verify this conclusion.

5.2.2.2 Effect of Blade Geometry

The measurements of Rodger (1992) are shown with the current low turbulence intensity measurements at $+20^\circ$ and design incidence in Figure 5.12. The two sets of data have a similar level of losses at an AVR of unity. Curves are presented showing the general trend of losses with AVR in the low turbulence intensity flow. Although most of the CC2 low Tu data are at $AVR \geq 1.01$ and the CC3 low Tu data are at $AVR \leq 1.02$, it is reasonable to assume that both sets of data follow similar paths. As the AVR is increased past unity, Rodger observed a gradual reduction in the midspan loss. This was linked to the effect of the flow acceleration on the position of flow separation. An increase in AVR would tend to move the separation point toward the blade trailing edge

Figure 5.11 CC3 loss variation at design, +15° and +20° incidence.

Figure 5.12 CC2 and CC3 loss variation at design and $+20^\circ$ incidence.

and reduce the blade losses. The low turbulence intensity counteracts the stabilizing effect of the positive pressure gradient and prevents the flow from reattaching.

No direct comparisons with the work of Tremblay (1989) have been made here due to the lack of accurate AVR data. Tremblay's loss measurements at high positive incidence are several times lower than more recent measurements on CC2 and CC3. It is believed that all these tests were made at values of AVR much larger than unity. There are also significant differences between the test configuration of Tremblay (1989) and the present one. The endplates used by Tremblay (1989) did not have a flexible knife edge and it was necessary to converge the entire plate to avoid separation at the endplate leading edge (hence the large values of AVR). During the present set of tests the gap between the profile cutouts and the blade was carefully sealed. Tremblay (1989) did not seal this gap, allowing cross flow between the blade and bypass passages. This will cause secondary flows quite different from those observed on a passage endwall with a well defined corner. Whether this had a beneficial or adverse effect on losses is unknown. The different turbulence intensities between Tremblay (1989), Rodger (1992) and the present study also make direct comparisons of the data uninformative.

5.2.2.3 Effect of Incidence

The variation of the midspan loss of CC3 with incidence is shown in Figure 5.13. All losses presented are at an AVR of 1.0. When measurements at this AVR were not available, trends based on the results of Rodger (1992) were used to interpolate reasonable values for the midspan loss at an AVR of unity. It can be seen that the losses remain relatively low, increasing slightly with incidence until $+10^\circ$. After this angle, the losses begin to rapidly increase. It should be noted that the apparent increase in losses due to the higher turbulence intensity at $+20^\circ$ incidence may be the result of the interpolation procedure used. The sudden rise in losses after $+10^\circ$ incidence suggests that the flow at this angle is on the verge of separating. The fully separated flow can be seen in the flow

Figure 5.13 CC3 midspan loss variation with incidence, AVR = 1.0.

visualization presented in Figures 5.9 and 5.10. The two-dimensional losses at the higher incidences, both at high and low turbulence intensities, are completely dominated by the separated flow.

5.2.3 Flow Deviation

The effect of AVR and Tu on deviation, δ , is presented in Figure 5.14. Again, there is not enough information available to establish definite trends with AVR for CC3. There is a relatively large increase in the deviation magnitude from design to $+15^\circ$. The $+20^\circ$ measurements have increased by one to three degrees over those at $+15^\circ$. Across the range of data available, the deviation variation is small enough that the measurement closest to an AVR of unity can be used as the value for two-dimensional flow. The separation bubble has caused increased deviation and possibly allowed some of the pressure surface flow to wrap around the trailing edge. If a region of separated flow does not attach before the blade trailing edge then there is an apparent increase in the width of the trailing edge wake resulting in an increased trailing edge loss. Evaluating this analytically is very complicated as it depends on exact knowledge of the separation location and its effect on the trailing edge flow. The level of turbulence intensity does not appear to have significantly affected the deviation.

The effect of axial velocity ratio on the deviation of CC2 and CC3 is shown in Figure 5.15 for both design and $+20^\circ$ effective incidence. As discussed earlier, there is a noticeable difference in the design incidence deviation measurements. At $+20^\circ$ incidence, there does not appear to be any difference between the deviation measurements at different values of AVR. The measurements from CC3 all appear to lie in the same range of scatter as the results from CC2.

Figure 5.14 CC3 flow deviation variation at design, $+15^\circ$ and $+20^\circ$ incidence.

Figure 5.15 CC2 and CC3 flow deviation variation at design and +20° incidence.

5.2.4 Blade Static Pressure Distribution

The full range of pressure distributions from 0° to $+20^\circ$ incidence for low turbulence intensity are shown in Figure 5.16 and the corresponding results for high turbulence intensity are shown in Figure 5.17. The trends observed for increasing incidence are the same for both curves. As the incidence is increased from design to $+20^\circ$ the overspeed at the leading edge increases. The blade pressure over the aft portion of the blade increases with incidence until there is very little pressure change over the last 15% of the blade. At $+20^\circ$ incidence the pressure values recorded in this area are virtually the same as the pressure measured at the base pressure tap.

5.3 Effect of Leading Edge Geometry on the Off-Design Performance

The focus of this section is the effect of leading edge geometry on the off-design performance of a linear turbine cascade. The effect on losses is examined in detail and blade pressure distributions are presented. Additional information on flow deviation and base pressure variation with incidence is also discussed.

5.3.1 Blade Losses

The leading edge diameter of CC3 was selected based on the trends indicated by the empirical correlation of Moustapha *et al.* (1990) to obtain lower losses than those measured on CC2. The correlation uses the blade leading edge geometry as the most significant parameter affecting off-design losses. When Tremblay (1989) applied the correlation to the experimental results from CC2 very good agreement was obtained. The off-design loss predicted by the correlation for both CC2 and CC3 is presented in Figure 5.18. If the effect of the leading edge wedge angle is secondary then, according to the

Figure 5.16 CC3 midspan blade static pressure distribution at 0.3% Tu.

Figure 5.17 CC3 midspan blade static pressure distributions at 3.0% Tu .

Figure 5.18 Off-design profile loss for CC2 and CC3 predicted using correlation of Moustapha *et al.* (1990).

trend shown, the off-design losses of CC3 should be much less than those of CC2.

The loss data obtained from CC2 and CC3 are presented in Figure 5.19 with the corresponding correlation loss predictions. The larger leading edge diameter should smooth out the initial flow acceleration and reduce the leading edge overspeed. This does not appear to have been the end result. In order to satisfy the blade thickness requirements and have similar velocity distributions across both blades it was necessary to decrease the leading edge wedge angle. This resulted in a greater percentage of the leading edge cylinder arc being exposed to the inlet flow, causing additional acceleration that counteracts the effect of the larger radius. The similarity between the results suggests that the leading edge wedge angle plays a much more significant role in determining the tolerance of a blade to off-design conditions than previously thought. In order to properly predict the off-design loss trends, a parameter that includes both aspects of the leading edge geometry must be incorporated into the correlation. Despite the increase in diameter it can be seen in Figure 3.3 that the two blades have almost identical suction side tangents at the leading edge. This may be a clue as to what extra information needs to be added to the database to reformulate the correlation.

It is believed that the measurements presented by Tremblay (1989) were made at an axial velocity ratio well above unity at both design and higher positive incidences. The loss measurement quoted for design incidence by Tremblay (1989) was found to correspond to the data obtained by Rodger (1992) at an axial velocity ratio of about 1.05. Rodger's (1992) measurements at high positive incidence were found to be consistently several times higher than those recorded in the earlier study. Direct comparisons of more recent measurements with those of Tremblay (1989) do not reveal any useful information. Differences in the experimental configuration as discussed in Section 5.2.2.2 and the large values of AVR make the older CC2 flow highly three-dimensional and not of much interest to this study.

Figure 5.19 CC2 and CC3 midspan loss variation with incidence, AVR = 1.0.

5.3.2 Flow Deviation

The increase in deviation with positive incidence for CC3 and CC2 can be seen in Figure 5.20. From 0° to $+20^\circ$ incidence, the deviation values presented are those closest to an AVR of 1.0. For CC3 the deviation increases with incidence up to $+20^\circ$. This indicates that the separation bubble is having a strong effect on the outlet flow. The deviation at -10° incidence is presented for an AVR of 0.968 for 0.3% Tu and 10.941 for 3.0% Tu. This was the highest value of AVR attainable at this incidence due to physical limitations of the equipment. In addition, some of the inlet flows are not as uniform as those at the higher incidences, increasing the uncertainty of the loss and deviation measurements. This may account for the large difference in deviation between CC3 and CC2 at this incidence. The design incidence deviation measurements for CC3 were shown in Figure 5.15. The measurement made closest to an AVR of 1.0 was used in Figure 5.20. This may have caused an exaggeration of the difference in deviation between the two blades at design incidence. Overall it can be seen from Figure 5.15 and 5.20 that the deviation of CC3 is less than that of CC2 for the range of incidence examined.

5.3.3 Blade Static Pressure Distribution

The pressure distributions at design incidence were discussed in Section 5.1.4. In that section a comparison between the design incidence measurements for CC2 and CC3 was presented in Figure 5.8. The same level of loading was observed but the shape of the curves was quite different. The pressure distributions of the 0.3% Tu, $+20^\circ$ incidence flows are presented in Figure 5.21. The velocity distributions on the pressure surface are similar and the stagnation point for both blades appears at the same location. There appears to be some difference in the suction surface measurements. Much of this can be attributed to the different axial velocity ratios. Measurements made by Rodger (1992) showed that the value of AVR strongly influences the level of blade loading. The leading edge overspeed is higher on CC3 and it has been delayed slightly. This could result in

Figure 5.20 CC3 and CC2 flow deviation variation with incidence.

Figure 5.21 CC2 and CC3 blade static pressure distributions at +20° incidence.

higher losses, cancelling the benefits of the larger leading edge diameter. The pressure increases rapidly from this peak though not as suddenly as that of CC2. At the trailing edge both blades show very little pressure recovery over the last 20% of axial chord. This apparent delay of pressure recovery on CC3 could be due to the difference in axial velocity ratios.

5.3.4 Base Pressure Coefficient

The base pressure measurements are made in the manner outlined in Chapters 3 and 4. Depending on the flow around the trailing edge, the base pressure is usually measured in a region of high mixing. It should be noted that under extreme conditions the flow may wrap around the trailing edge, producing an attached flow across the tap. In the case of suction surface separation, the displacement effect of the separation bubble may divert the cascade outlet flow enough to allow this to happen. The base pressure coefficient plays an important role in the development of trailing edge loss. In fact, in the inviscid limiting case, it becomes the dominant factor. Accurate measurement of the base pressure is often limited by the geometry of the model being tested. A base pressure tap was included from the outset in the design of the CC3 and CC2 blade models. The trailing edge tap is in line with the outlet metal angle and a pressure coefficient is calculated based on the measured base and the mixed-out dynamic and static pressures.

The variation in the base pressure coefficient of CC3 with incidence is presented in Figure 5.22. The coefficients remain relatively unchanged from 0° to $+10^\circ$ incidence. There is a sudden drop from $+10^\circ$ to $+15^\circ$ incidence and the magnitude of the coefficients increase by a factor of almost ten. This is probably due to the suction surface separation placing the base pressure tap in a region of separated flow. It is no longer accounting for the effect of the sudden expansion as fluid passes the trailing edge plane.

Figure 5.22 CC2 and CC3 base pressure coefficient variation with incidence.

Base pressure coefficients were obtained by Rodger (1992) at -20° , design and $+20^\circ$ incidence for a number of axial velocity ratios. These results for axial velocity ratios near unity are included in Figure 5.22. The coefficients at design incidence are similar. At $+20^\circ$ incidence, however, there is a large difference between the two blades. The base pressure coefficient from CC2 is actually positive. It should be noted that the measurements at $+20^\circ$ incidence were made in the proximity of suction surface separation. The effect of this separation on the flow around the blade trailing edge cannot be predicted with the information available.

Chapter 6

Conclusions

The following conclusions have been drawn from the preceding discussion and analysis

- 1) The axial velocity ratio has a very strong effect on the losses measured at off-design incidence. Reducing the AVR has caused blade losses to increase. The same trend is observable at design incidence but the effects are not as pronounced.
- 2) For the flow conditions and range of AVR examined, CC3 appears to be insensitive to the tunnel free stream turbulence intensity.
- 3) The present investigation adds further evidence that the losses presented by Tremblay (1989) can not be considered representative of two-dimensional flow conditions due to uncertainty about the range of AVR these tests were conducted under.

- 4) Flow separation is the dominant loss mechanism at high positive incidence for both CC2 and CC3 when two-dimensional flow conditions exist across the cascade outlet.
- 5) The larger leading edge radius of CC3 has not led to a reduction of the off-design two-dimensional losses. The reduction of the leading edge wedge angle appears to have cancelled out the effects of the diameter change. The two blades show almost identical losses across the range of AVR tested.
- 6) Based on the measured losses there is a discrepancy in the trends predicted by the correlation of Moustapha et al. (1990). The leading edge geometry change has not reduced the blade sensitivity to off-design losses. The leading edge wedge angle seems to play a much greater role than previously expected, cancelling out the effect of the leading edge diameter. The wedge angle is not included in the data base used to establish the correlation. The data must be revisited in order to re-formulate the correlation.

Chapter 7

Recommendations

The preceding study has highlighted gaps in the available data. As part of a future experimental program the following recommendations are made

- 1) The trends for blade loss with AVR were seen to be similar to those of CC2. However, a larger, more uniform data base of results is needed to extend the comparison between cascades CC2 and CC3. For a more complete comparison at $+20^\circ$ incidence and 0.3% Tu, loss measurements should be made at higher AVR for CC3 and at lower values of AVR for CC2.
- 2) The measurements presented suggest that there is little effect of turbulence intensity on the blade two-dimensional off-design performance. However, the effects of turbulence intensity on CC2 are still uncertain, as the measurements of Tremblay (1989) have been found to be unsatisfactory in light of Rodger's (1992) work on two-dimensionality. Additional data should be obtained for CC2 under

high turbulence intensity conditions using the new endplate design for a complete comparison of the two blades.

3) Additional measurements should be made between 0° and $+20^\circ$ incidence for CC2 under low turbulence intensity conditions. This information could then be used to determine whether the off-design losses increase at the same rate for both blades.

4) Three-dimensional measurements at the outlet plane would establish the amount of local convergence at each blade surface. From this information the use of AVR as a criterion for two-dimensionality could be further evaluated.

5) The correlation of Moustapha *et al.* (1990) should be re-examined in light of the measurements presented in this work. The original data used to develop the correlation should be re-examined to obtain information about the blade leading edge wedge angle. This parameter should then be incorporated into the correlation to account for both the differences between cascade losses in the database and the similarity between the losses of CC3 and CC2.

6) The quick traverse technique should be explored as a means of obtaining loss data. A thorough loss analysis would be required and full outlet traverses would still be a necessary tool to obtain flow angle data.

7) The effectiveness of the grid used to raise the turbulence intensity in the wind tunnel has been tested to show that there is a reasonably uniform pitchwise distribution of Tu at the cascade inlet. A more thorough analysis of the effect of the grid on the tunnel should be performed in order to allow its use in three-dimensional flow measurements.

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Appendix A

Blade and Cascade Geometry

Figure A.1 Blade and cascade geometry nomenclature.

Table A.1 Comparison of CC2 and CC3 geometry.

Table A.2 Carleton Cascade Three: Blade Profile Coordinates

Figure A.1 Blade and cascade geometry nomenclature

Table A.1 Comparison of CC2 and CC3 geometry.

	CC2	CC3
Inlet flow angle, α_1	28.35°	28.35°
Exit flow angle, α_2	57.65°	57.65°
Inlet metal angle, β_1	29.29°	25.51°
Exit metal angle, β_2	57.51°	57.51°
Design incidence, i_{des}	-0.94°	3.00°
Design deviation, δ_{des}	-0.145°	-0.145°
Stagger angle, γ	23.06°	21.6°
Camber angle, θ	86.86°	82.86°
Chord length, C	162.8 mm	162.3 mm
Axial chord length, C_x	150 mm	149.4 mm
Pitch length (spacing), S	110.71 mm	110.71 mm
Blade Span, H	200 mm	200 mm
Pitch to chord ratio, S/C	0.68	0.68
Aspect ratio (AR), H/C	1.23	1.23
Leading edge diameter, d	9.43 mm	16.7 mm
Leading edge wedge angle	52.4°	43.0°
Trailing edge thickness	4.2 mm	4.2 mm
Maximum thickness, t_{max}	29.36 mm	31.8 mm
t_{max}/C	0.182	0.196
Throat opening, σ	63.6 mm	63.6 mm

Table A.2 Carleton Cascade Three : Blade Profile Coordinates

Suction Surface		Pressure Surface	
x/c	y/c	x/c	y/c
0.0050	0.3578	0.0050	0.3578
0.0015	0.3674	0.0102	0.3490
0.0001	0.3776	0.0171	0.3415
0.0006	0.3878	0.0254	0.3355
0.0032	0.3977	0.0347	0.3312
0.0077	0.4069	0.0447	0.3288
0.0139	0.4150	0.0549	0.3285
0.0162	0.4174	0.0570	0.3286
0.0207	0.4220	0.0612	0.3289
0.0275	0.4286	0.0675	0.3293
0.0365	0.4367	0.0759	0.3298
0.0456	0.4442	0.0844	0.3303
0.0591	0.4544	0.0970	0.3310
0.0772	0.4665	0.1138	0.3319
0.0953	0.4771	0.1306	0.3326
0.1134	0.4863	0.1475	0.3332
0.1315	0.4945	0.1643	0.3337
0.1496	0.5016	0.1812	0.3340
0.1677	0.5077	0.1980	0.3342
0.1858	0.5130	0.2148	0.3343
0.2129	0.5192	0.2401	0.3340
0.2400	0.5237	0.2653	0.3334

Suction Surface		Pressure Surface	
x/c	y/c	x/c	y/c
0.2762	0.5270	0.2990	0.3317
0.3124	0.5273	0.3327	0.3290
0.3486	0.5245	0.3663	0.3251
0.3847	0.5188	0.4000	0.3199
0.4209	0.5099	0.4337	0.3133
0.4571	0.4977	0.4674	0.3052
0.4933	0.4822	0.5010	0.2955
0.5294	0.4631	0.5347	0.2840
0.5656	0.4402	0.5684	0.2707
0.6018	0.4134	0.6020	0.2555
0.6380	0.3825	0.6357	0.2381
0.6742	0.3473	0.6694	0.2184
0.7103	0.3078	0.7031	0.1961
0.7465	0.2637	0.7367	0.1709
0.7827	0.2153	0.7704	0.1424
0.8098	0.1764	0.7957	0.1186
0.8370	0.1356	0.8209	0.0922
0.8550	0.1076	0.8377	0.0731
0.8731	0.0791	0.8546	0.0525
0.8912	0.0502	0.8714	0.0303
0.9048	0.0285	0.8840	0.0125
0.9116	0.0177	0.8904	0.0032
0.9161	0.0104	0.8946	-0.0031
0.9184	0.0068	0.8967	-0.0070
0.9197	0.0038	0.8988	-0.0095
0.9203	0.0005	0.9016	-0.0114

Suction Surface		Pressure Surface	
x/c	y/c	x/c	y/c
0.9200	-0.0028	0.9047	-0.0126
0.9188	-0.0060	0.9080	-0.0128
0.9169	-0.0087	0.9113	-0.0123
0.9144	-0.0108	0.9144	-0.0108

Appendix B

Tables of Results

Table B.1	Low turbulence intensity measurements.
Table B.2	High turbulence intensity measurements.
Table B.3	Spanwise low turbulence intensity measurements $i_{eff}=0.0^\circ$ AVR = 1.003.
Table B.4	Spanwise loss measurements (with endplate gap leakage) $i_{eff}=+10.0^\circ$ AVR = 1.003.
Table B.5	Spanwise loss measurements (no endplate gap leakage) $i_{eff}=+10.0^\circ$ AVR = 1.007.

Table B.1 Low turbulence intensity measurements.

i_{eff}	AVR	Y_m	δ (°)	Cp_{base}
-20	0.950	0.0293	-3.60	
-10	0.968	0.0302	-4.01	
0	1.010	0.0250	-3.86	
0	0.970	0.0299	-3.98	
0	1.003	0.0253	-3.46	-0.0399
0	1.004	0.0252	-3.21	
5	1.004	0.0292	-3.74	-0.0173
10	1.007	0.0357	-4.95	0.0185
15	1.009	0.0744	-7.64	-0.1490
15	0.976	0.1454	-8.91	
20	1.019	0.1379	-8.90	-0.1965
20	0.985	0.2635	-9.56	
20	0.980	0.2967	-9.60	

Table B.2 High turbulence intensity measurements

i_{eff}	AVR	Y_m	δ (°)	$C_{p_{base}}$
-20	0.950	0.0302	-2.59	
-10	0.941	0.0295	-3.29	
0	1.008	0.0217	-3.47	-0.0132
0	0.950	0.0304	-2.97	
10	1.002	0.0357	-4.76	-0.0131
15	0.999	0.0990	-7.77	-0.1400
15	0.953	0.1738	-7.82	
20	1.006	0.1746	-8.95	-0.1604
20	0.990	0.2932	-11.91	
20	0.979	0.2725	-10.40	

Table B.3 Spanwise low turbulence intensity measurements
 $i_{eff} = 0.0^\circ$ AVR = 1.003

z/H	Y_m	δ (°)
0.224	0.0743	-5.2
0.293	0.0482	-4.3
0.362	0.0280	-3.5
0.500	0.0253	-3.5
0.638	0.0276	-4.0
0.707	0.0438	-4.3
0.776	0.0808	-5.4

Table B.4 Spanwise loss measurements (with endplate gap leakage)
 $i_{eff} = +10.0^\circ$ $AVR = 1.003$

z/H	Y_m	$\delta (^\circ)$
0.362	0.0954	-5.8
0.431	0.0453	-5.4
0.638	0.0407	-4.6
0.362	0.0908	-5.7
0.224	0.1024	-5.5
0.776	0.0947	-6.9
0.879	0.0375	-3.2
0.121	0.0328	-3.5
0.397	0.0561	-5.3
0.500	0.0374	-4.9

Table B.5 Spanwise loss measurements (no endplate gap leakage)
 $i_{eff} = +10.0^\circ$ $AVR = 1.007$

z/H	Y_m	$\delta (^\circ)$
0.362	0.0413	-5.5
0.638	0.0502	-5.3
0.500	0.0358	-5.0
0.741	0.1119	-7.3
0.603	0.0368	-4.9
0.397	0.0346	-4.8

Appendix C

Effect of Endplate Gap Leakage on Spanwise Measurements

Figure C.1 Spanwise loss distributions at $+10^\circ$ incidence showing the effect of leakage through the blade-endplate gap.

Appendix C

Effect of Endplate Gap Leakage on Spanwise Measurements

The existence of a symmetric low-loss region in the spanwise plot of the total pressure losses is an important condition for valid two-dimensional loss measurements. It was realized during the course of the present work that the gap between the blade surface and the endplate must be sealed very carefully. Initial results showed that running tests with no sealing caused large distortions in the outlet flow. In response to this, the gap was filled with the same type of foam rubber weather stripping used by Rodger (1992). This was used to block the movement of air between the main passage and the bypass passage. The weather stripping seal began to leak after a series of flow visualization tests had been performed on the blade. During the course of these tests the endplates had been removed several times. Subsequent measurements at $+10^\circ$ showed an asymmetric spanwise loss distribution. As seen in Figure C.1 the width of the low-loss region on the endwall side of midspan has been reduced. A review of the flow visualization results indicated that leakage through the gap may have been the cause. To

obtain a high quality seal, the plate-blade gap of the centre blade was sealed with a combination of silicone gel normally used for weather stripping and ordinary plasticine. A quick traverse at selected spanwise locations showed that the problem had been solved. However, the same quick traverses showed a great deal of non-periodicity in the outlet wakes across the full passage. This was in spite of good uniformity and periodicity being observed at midspan. The entire cascade was then sealed using silicone on the inner plate and plasticine on the outer plate. Subsequent quick traverses showed very good spanwise periodicity and a symmetric low-loss region at the outlet.

Plasticine was not used on the inner plate as it had a tendency to move when the tunnel was operating. This was not a problem when sealing the outside of the outer plate. The advantage of plasticine was that it did not hamper removal of the outer plate for access to the measurement equipment. Silicone removal tended to be messy and time consuming. Fortunately, it was possible to conduct a number of tests before resetting the plates became necessary. At least two sets of data could be obtained from one endplate setup since installing the turbulence grid did not affect the cascade configuration. Sometimes an additional two sets of data could be obtained simply by changing the incidence. The same endplate leading edge configurations were used for $+20^\circ$ and $+15^\circ$ thus the incidence could be changed without affecting the endplate configuration.

The plasticine and silicone sealing method used in the present study differed from the foam rubber used by Rodger (1992) in that it did not block the flow through the bypass passage to any great extent. This may have contributed to the lack of high axial velocity ratio data obtained. This should be investigated as it may be possible to increase AVR by introducing small obstructions between the blades. The effect would be similar to that of the perforated screen used at the outlet to force more air past the inlet control surfaces.

Figure C.1 Spanwise loss distributions at $+10^\circ$ incidence showing the effect of leakage through the blade-endplate gap.

Appendix D

Static Pressure Blade Geometry

Figure D.1 Graphical representation of the instrumented blade pressure tap and channel locations.

Figure D.2 Instrumented blade in standard position showing static pressure tap orientation nomenclature. Positive measured clockwise from vertical.

Table D.1 Static pressure tap coordinates and orientation.

Figure D.1 Graphical representation of the instrumented blade pressure tap and channel locations.

Figure D.2 Instrumented blade in standard position showing static pressure tap orientation nomenclature. Positive measured clockwise from vertical.

Table D.1 Static pressure tap coordinates and orientation.**C = True Chord****Cx = Axial Chord** **τ = Surface Normal w.r.t. Y axis when blade in standard position**

Suction Surface					
Tap #	x/Cx	x/C	y/C	τ	s/C
1	0.0014	0.0013	0.0259	-77.185	0.0341
2	0.0182	0.0168	0.0527	-45.898	0.0656
3	0.0431	0.0397	0.0741	-39.928	0.0969
4	0.0755	0.0694	0.0962	-33.511	0.1340
5	0.1170	0.1077	0.1182	-26.574	0.1781
6	0.1614	0.1485	0.1359	-20.210	0.2226
7	0.2061	0.1897	0.1487	-14.356	0.2658
8	0.2423	0.2230	0.1558	-9.833	0.2999
9	0.2961	0.2725	0.1615	-3.344	0.3497
10	0.3547	0.3264	0.1611	4.202	0.4036
11	0.4109	0.3782	0.1547	10.055	0.4558
12	0.4638	0.4269	0.1428	16.992	0.5060
13	0.5172	0.4760	0.1247	23.365	0.5584
14	0.5649	0.5199	0.1031	28.938	0.6073
15	0.6113	0.5626	0.0769	34.090	0.6574
16	0.6523	0.6003	0.0493	38.357	0.7042
17	0.6917	0.6366	0.0184	42.216	0.7518
18	0.7309	0.6727	-0.0165	45.779	0.8020
19	0.7677	0.7065	-0.0532	48.821	0.8520
20	0.8073	0.7430	-0.0971	51.720	0.9090

Tap #	x/Cx	x/C	y/C	τ	s/C
21	0.8509	0.7831	-0.1532	54.486	0.9784
22	0.8971	0.8256	-0.2124	56.470	1.0513
23	0.9410	0.8660	-0.2750	57.663	1.1258
24	0.9774	0.8995	-0.3283	58.007	1.1887
Pressure Surface					
43	0.0048	0.0044	-0.0063	-113.896	0.0550
42	0.0264	0.0243	-0.0292	-148.219	0.1167
41	0.0682	0.0628	-0.0363	176.210	0.2065
40	0.1049	0.0966	-0.0343	176.978	0.2802
39	0.1430	0.1316	-0.0327	177.775	0.3483
38	0.1928	0.1774	-0.0314	178.941	0.4185
37	0.2405	0.2213	-0.0311	-179.689	0.4860
36	0.2860	0.2632	-0.0319	-178.078	0.5594
35	0.3521	0.3241	-0.0356	-175.081	0.6346
34	0.4324	0.3979	-0.0450	-170.208	0.7091
33	0.5122	0.4714	-0.0612	-164.880	0.7700
32	0.5880	0.5411	-0.0837	-159.320	0.8120
31	0.6554	0.6032	-0.1104	-154.071	0.8558
30	0.7223	0.6647	-0.1441	-148.543	0.9017
29	0.7835	0.7210	-0.1823	-143.100	0.9368
28	0.8449	0.7776	-0.2294	-137.264	0.9706
27	0.9123	0.8396	-0.2945	-129.894	1.0104
26	0.9533	0.8773	-0.3432	-125.503	1.0412
Trailing Edge Tap					
Tap #	x/Cx	x/C	y/C	τ	s/C
25	0.9927	0.9136	-0.3653	146.633	1.2430

Appendix E Blade Static Pressure Measurements

Table E.1 Low Turbulence Intensity Blade Static Pressure Distributions.

Table E.2 High Turbulence Intensity Blade Static Pressure Distributions.

Table E.1 Low Turbulence Intensity Blade Static Pressure Distributions

Low Turbulence Intensity Static Pressure Measurements						
i_{eff}		0°	5°	10°	15°	20°
	A.V.R.	1.003	1.004	1.007	1.009	1.019
tap	x/Cx					
1	0.0014	0.0114	0.1983	0.3911	1.1877	1.6080
2	0.0182	1.5908	2.1594	2.4288	2.9304	3.6802
3	0.0431	1.5322	1.8948	1.9703	2.2054	2.6235
4	0.0755	1.6296	1.9278	1.8556	1.9324	2.2174
5	0.1170	1.8649	1.9971	1.8108	1.7975	1.9764
6	0.1614	1.9983	2.0439	1.7620	1.6795	1.7766
7	0.2061	2.0976	2.0715	1.7205	1.5909	1.6309
8	0.2423	2.1730	2.1026	1.7027	1.5384	1.5412
9	0.2961	2.2790	2.1504	1.6917	1.5001	1.4614
10	0.3547	2.3759	2.2021	1.6987	1.4727	1.4023
11	0.4109	2.4570	2.2487	1.7110	1.4627	1.3699
12	0.4638	2.5128	2.2854	1.7245	1.4564	1.3457
13	0.5172	2.5484	2.3092	1.7267	1.4433	1.3203
14	0.5649	2.5153	2.2624	1.6860	1.3978	1.2637
15	0.6113	2.4929	2.2432	1.6614	1.3599	1.2116
16	0.6523	2.4550	2.2051	1.6217	1.3093	1.1462
17	0.6917	2.3919	2.1430	1.5604	1.2364	1.0598
18	0.7309	2.3261	2.0768	1.4995	1.1533	0.9609
19	0.7677	2.2582	1.9858	1.4055	1.0532	0.8592

i_{cr}		0°	5°	10°	15°	20°
	A.V.R.	1.003	1.004	1.007	1.009	1.019
tap	x/Cx					
20	0.8073	2.1760	1.8513	1.2895	0.9371	0.7523
21	0.8509	1.9101	1.6647	1.1423	0.8113	0.6644
22	0.8971	1.6636	1.4794	0.9957	0.7192	0.6116
23	0.9410	1.5055	1.3045	0.8709	0.6531	0.5817
24	0.9774	1.4454	1.2185	0.7994	0.6241	0.5802
25	0.9927	1.3830	1.1324	0.7338	0.6129	0.5367
26	0.9533	0.6864	0.5246	0.2298	0.1846	0.0623
27	0.9123	0.3426	0.2149	-0.0158	-0.0294	-0.1698
28	0.8449	0.1001	-0.0055	-0.1923	-0.1785	-0.3330
29	0.7835	-0.0116	-0.1088	-0.2754	-0.2476	-0.4080
30	0.7223	-0.0772	-0.1697	-0.3252	-0.2894	-0.4541
31	0.6554	-0.1231	-0.2130	-0.3607	-0.3202	-0.4878
32	0.5880	-0.1511	-0.2401	-0.3851	-0.3421	-0.5133
33	0.5122	-0.1630	-0.2556	-0.4013	-0.3593	-0.5337
34	0.4324	-0.1586	-0.2577	-0.4090	-0.3709	-0.5494
35	0.3521	-0.1448	-0.2538	-0.4138	-0.3837	-0.5673
36	0.2860	-0.1300	-0.2523	-0.4228	-0.4003	-0.5913
37	0.2405	-0.1254	-0.2594	-0.4381	-0.4211	-0.6194
38	0.1928	-0.1311	-0.2813	-0.4695	-0.4585	-0.6652
39	0.1430	-0.1526	-0.3263	-0.5262	-0.5181	-0.7381
40	0.1049	-0.1688	-0.3740	-0.5894	-0.5840	-0.8147
41	0.0682	-0.1167	-0.4002	-0.6615	-0.6673	-0.9105
42	0.0264	-0.1216	-0.0015	0.1288	0.7715	1.2293
43	0.0048	-0.3392	0.1312	0.6384	1.4471	2.1179

Table E.2 High Turbulence Intensity Blade Static Pressure Distributions

High Turbulence Intensity Static Pressure Measurements					
i_{eff}		0°	10°	15°	20°
	A.V.R.	1.003	1.002	0.999	1.006
tap	x/Cx				
1	0.0014	0.0056	0.2612	0.7696	1.1992
2	0.0182	1.6274	2.3695	2.2525	3.4402
3	0.0431	1.5646	1.9234	1.7256	2.3986
4	0.0755	1.7121	1.8203	1.5276	2.0530
5	0.1170	1.8836	1.7975	1.4326	1.8293
6	0.1614	2.0172	1.7605	1.3461	1.6342
7	0.2061	2.1117	1.7248	1.2784	1.4997
8	0.2423	2.1951	1.7155	1.2364	1.4094
9	0.2961	2.2946	1.7170	1.2011	1.3219
10	0.3547	2.3975	1.7306	1.1811	1.2617
11	0.4109	2.4864	1.7499	1.1722	1.2266
12	0.4638	2.5308	1.7680	1.1630	1.1997
13	0.5172	2.5753	1.7715	1.1545	1.1723
14	0.5649	2.5449	1.7248	1.1236	1.1212
15	0.6113	2.5257	1.6966	1.0923	1.0708
16	0.6523	2.4932	1.6615	1.0502	1.0111
17	0.6917	2.4290	1.5901	0.9952	0.9309
18	0.7309	2.3585	1.5160	0.9306	0.8426
19	0.7677	2.2829	1.4259	0.8490	0.7471

i_{eff}		0°	10°	15°	20°
	A.V.R.	1.003	1.002	0.999	1.006
tap	x/Cx				
20	0.8073	2.1731	1.3183	0.7587	0.6507
21	0.8509	1.9433	1.1642	0.6593	0.5722
22	0.8971	1.7293	1.0165	0.5908	0.5293
23	0.9410	1.5532	0.9131	0.5445	0.5311
24	0.9774	1.4774	0.8386	0.5193	0.5221
25	0.9927	1.4043	0.7770	0.4986	0.4527
26	0.9533	0.6908	0.2175	0.1729	0.0197
27	0.9123	0.3433	-0.0437	0.0095	-0.2011
28	0.8449	0.0944	-0.2295	-0.1046	-0.3595
29	0.7835	-0.0193	-0.3146	-0.1579	-0.4331
30	0.7223	-0.0885	-0.3676	-0.1912	-0.4777
31	0.6554	-0.1370	-0.4049	-0.2151	-0.5103
32	0.5880	-0.1699	-0.4295	-0.2321	-0.5369
33	0.5122	-0.1842	-0.4436	-0.2433	-0.5551
34	0.4324	-0.1774	-0.4479	-0.2525	-0.5680
35	0.3521	-0.1602	-0.4492	-0.2603	-0.5856
36	0.2860	-0.1454	-0.4565	-0.2727	-0.6061
37	0.2405	-0.1402	-0.4706	-0.2865	-0.6304
38	0.1928	-0.1467	-0.5025	-0.3121	-0.6703
39	0.1430	-0.1680	-0.5617	-0.3558	-0.7342
40	0.1049	-0.1848	-0.6270	-0.4032	-0.7999
41	0.0682	-0.1310	-0.6984	-0.4648	-0.8821
42	0.0264	-0.1254	0.0149	0.5526	0.8859
43	0.0048	-0.3410	0.5219	1.0639	1.9615

Appendix F

Test Equipment and Settings

Figure F.1 Control surface locations and labelling after Rodger (1992).

Figure F.2 Endplate nomenclature and settings.

Table F.1 Control surface settings

Figure F.3 Three-hole pressure probe

Figure F.4 Turbulence generating grid

Figure F.1 Control surface locations and labelling after Rodger (1992).

The endplate spacing matrix:

X3	X1
X4	X2

Matrices used for settings in table F.2 (all lengths in mm):

-20°, -10°

30	41
145	123

0°, +5°, +10°

30	33
145	139

+15°, +20°

30	31
145	123

$$\epsilon = 10 \text{ mm}$$

Figure F.2 Endplate nomenclature and settings.

Table F.1 Control Surface Settings

Tu Turbulence intensity
 TF Top Flap
 BF Bottom Flap
 TTB Top Tailboard (absolute measurement from vertical)
 BTB Bottom Tailboard (absolute measurement from vertical)
 FW Floating Wall
 BP1 Bypass Flap One
 BP2 Bypass Flap Two
 S indicates if screen has been used at the outlet
 max. maximum opening possible (BP1)

For endplate settings see figure E.2.

i	Tu	AVR	TF	BF	TTB	BTB	FW	BP1	BP2	S
(°)	(%)		(°)	(°)	(°)	(°)	(°)	(mm)	(mm)	
-20	0.3	0.950	12.5	12.5	62	60	0	max.	0	N
-20	3.0	0.950	15.0	15.0	62	61	0	max.	0	N
-10	0.3	0.968	31.0	20.0	71	69	0	24	0	N
-10	3.0	0.941	30.0	15.0	71	71	0	max.	0	N
0	0.3	1.003	47.0	34.0	85	82	2.5	23	17	N
0	3.0	1.008	47.0	34.0	85	82	2.5	23	27	N
5	0.3	1.004	58.0	29.0	90	86	1	35	40	N
10	0.3	1.007	57.0	47.5	96	90	0	50	32	Y
10	3.0	1.002	55.0	49.0	96	90	0	50	32	Y
15	0.3	0.999	65.0	51.0	97	90	0	69	51	Y
15	3.0	1.009	66.0	52.0	97	90	0	69	51	Y
20	0.3	1.019	76.0	54.0	102	95	0	72	51	Y
20	3.0	1.006	78.0	54.0	102	95	0	72	51	Y

Figure F.3 Three-hole pressure probe.

Figure F.4 Turbulence generating grid.

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