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Establishing Two-Dimensional Flow in a Large-Scale Planar Turbine Cascade

By

**Philippe Rodger
B.Eng. (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

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

**DEPARTMENT OF MECHANICAL AND AEROSPACE ENGINEERING
CARLETON UNIVERSITY
OTTAWA, CANADA
SEPTEMBER, 1992.**

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Large-Scale Planar Turbine Cascade"**

submitted by Philippe Rodger, B. Eng.
in partial fulfilment of the requirements for
the degree of Master of Engineering

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Abstract

An extensive literature survey has demonstrated the lack of data on the influence of flow convergence and divergence on the aerodynamic performance of a cascade of turbine blades. The amount of flow convergence or divergence is evaluated by measuring the axial velocity ratio (AVR) across the blade passage. A novel arrangement of end-plates has been developed to restrict the endwall boundary layer and control the AVR. The end-plates are used to investigate the influence of the flow two-dimensionality on the aerodynamic behaviour of the blade at midspan. Measurements at off-design conditions (-20° and $+20^\circ$ of incidence) were also performed with the variable incidence test section.

Results of the AVR influence on total pressure loss indicate that flow convergence has less effect than flow divergence. However, the flow trailing edge deviation angle was significantly influenced by both conditions. Blade loading measurements revealed that the AVR had some effect on the flow behaviour near the rearward part of the blade. The base pressure at the trailing edge was found to be sensitive to the AVR which would, in turn, influence the measured losses. This problem would suggest that cascades with large trailing edge losses, like those operating near transonic conditions, may be more susceptible to the lack of two-dimensionality.

The results for the total pressure loss, with respect to the AVR, were comparable to previous work performed on compressor cascades, while the AVR influence on deviation angle showed a reverse trend.

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Nomenclature

AR	Aspect ratio = H/C	
AVDR	Axial velocity density ratio (see definition in section 2.1.4)	
AVR	Axial velocity ratio (see definition in section 2.1.4)	
b.l.	Boundary layer	
C	Chord	
C_x	Axial chord	
C_p	Blade static pressure coefficient where:	$C_p = \frac{P - P_1}{q_1}$
C_{pb}	Base pressure coefficient where:	$C_{pb} = \frac{P_b - P_2}{q_2}$
h	Enthalpy	
H	Span	
i	Angle of incidence where: $i = (\alpha_1 - \beta_1)$	
k	End-plate offset distance from turntable	
l	End-plate spacing distance	
P	Static pressure	
P_b	Base pressure	
P_o	Total pressure	
q	Dynamic pressure = $\frac{1}{2} \rho V^2$	
Re	Reynolds number = $\frac{\rho V_{in} C}{\mu}$	
S	Blade pitch or spacing	
t	Blade thickness	
Tu	Turbulence intensity in the x-direction (see definition in section 5.2.1)	
V	Velocity	
u,v,w	Velocity in x, y and z direction	

x, y, z Co-ordinates in the axial, tangential and spanwise directions

Y Total pressure loss coefficient where: $Y = \frac{P_0 - P_{01}}{q_2}$

α Flow angle, measured from the axial direction

β Blade metal angle, measured from the axial direction

γ Blade stagger angle

δ Deviation angle where: $\delta = (\alpha_2 - \beta_2)$

ϵ End-plate slat curvature offset, flow deflection

θ Camber angle

ξ, ζ Total pressure loss coefficient $= \frac{h_2 - h_{2is}}{\frac{1}{2} V_2^2}$

ρ Density

σ Passage throat width

τ Shear stress

ϕ^2 Kinetic energy loss

2-D Two-dimensional

3-D Three-dimensional

Subscripts

b Base

des Design

ex Cascade exit plane

eff Effective

in Cascade inlet plane

is	Isentropic conditions
l.e.	Leading edge
m, (ms)	Midspan
max, min	Maximum, minimum value
o	Stagnation (total value)
ref	Reference measurement plane
t.e.	Trailing edge
x, y, z	Values in the axial, tangential and spanwise directions
 1	 Inlet measurement plane
2	Exit measurement plane

1 INTRODUCTION

The increase in power and sophistication of both computational hardware and software has facilitated the design of axial turbines and compressors. However, an important source of data and physical insight for designers continues to be experimental testing. The complex nature of turbomachinery flows raises several questions about the flow behaviour and the mechanisms involved. Controlled experiments on components of the machine must be performed to understand more of the flow phenomena.

Testing on actual engines is limited by several factors; high temperatures, small areas and blade speed combine to make data gathering of local parameters very difficult. Another option is to perform tests on a cascade of blades. The blades can be scaled-up to allow ease of instrumentation and increased resolution of data. The cascade of blades can be subjected to different operational conditions experienced in the engine. This allows the designer to experiment with different scenarios and predict performance. Cascade testing can also help in understanding the physical behaviour of the flow. This understanding can improve the modelling of the flow in computational fluid dynamic (CFD) programs.

The flow observed in turbomachinery is three-dimensional. Before the three-dimensional flow can be fully understood, mechanisms of two-dimensional flow must first be studied. Frequently the objective of cascade testing is to obtain the two-dimensional aerodynamic behaviour of a particular blade section. To theoretically achieve perfect two-dimensionality, the span of the cascade should be infinity. Because of physical limitations of testing facilities, most measurements are made at midspan on a blade of relatively low aspect ratio. The aspect ratio influences the degree of three-dimensionality. The presence of the endwalls physically alters the pattern of the flow and challenges the validity of the "two-dimensional" measurements. There appears to be no general agreement about either the tests required to establish two-dimensionality or the degree of deviation from two-dimensionality which is acceptable.

One possible test which can be used to investigate the two-dimensionality of the flow is the Axial Velocity Ratio (AVR hereafter). The AVR can be a measure of the acceleration or deceleration of the flow at mid-span. A full description of the AVR can be found in the next chapter. The flow acceleration in a cascade with solid endwalls is a result of the inlet boundary layer displacement effect. A corner stall can also cause considerable blockage. Tests on both compressor and turbine blades with solid endwalls report a mid-span AVR of around 1.1; this is consistent with the values obtained from measurements performed on in-house test sections with solid endwall and no boundary layer treatment. The acceleration of the flow at mid-span is a detriment to ideal two-dimensional (2-D hereafter) conditions which would measure an AVR of 1.0. To improve the two-dimensionality of the flow at mid-span, one can manipulate the boundary layer of the endwalls and consequently control the AVR.

Several methods have been used over the years to control the AVR. Some of these methods are described in the next chapter. A review of the literature revealed that all of the experimental tests were performed on cascades of compressor blades. The majority of the tests are based on outdated blade sections. A need for the investigation of the AVR on a cascade of recently designed turbine blades seemed to be warranted. The method of AVR control decided upon was the use of end-plates mounted on the blades. By adjusting the end-plates in a converging or diverging manner, the AVR can be trimmed to a desired value. End-plates were used in previous in-house tests (Goobie 1989, Tremblay 1989). These original end-plates could only be converged because of end-plate leading edge separation for the diverging setting. To allow increased flexibility of the end-plates, a modification to the leading edge was performed. An adjustable leading edge extension (to be called slat, analogy to the aircraft wing slats) was added. The slats allow the end-plates to be aligned with the oncoming flow and prevent separation. These modified end-plates will be used to perform a series of tests. The AVR can be selected as an independent variable and its effect on two-dimensionality can be evaluated.

A detailed study of the effect of the AVR on the total pressure losses, spanwise distribution of loss, flow deviation, blade static pressure and base pressure measurements is performed. The spanwise distribution of loss is also compared to the original end-plates and the no end-plates conditions. The off-design performance is also investigated with

respect to the AVR. The two angles of incidence chosen are $+20^\circ$ and -20° . The influence of Reynolds number on the total pressure losses is studied at both design and $+20^\circ$. The correlation between turbulence intensity and critical Reynolds number is examined. The results obtained are compared to existing experimental data and the theoretical models developed for compressor blades. The results can also be used to confirm the validity of previous two-dimensional testing. Finally, conclusions and recommendations for future work are discussed.

2 LITERATURE REVIEW

As mentioned earlier it is essential for the designer to understand the 2-D behaviour of the flow. The overall performance of the turbine can only be evaluated on a full scale machine. However, this analysis will only reveal the combined overall effect of several mechanisms. The physical aspect of every mechanism must be individually studied to improve the design of the turbine as a whole. The overall losses found in a turbomachine blade row can be subdivided into basic components of losses. Each of these losses are influenced by different parameters. The losses encountered in the literature review which were relevant to this study are as follows:

- 1. Profile Loss;** loss due to the boundary layer growth on the blade in a uniform 2-D flow.
- 2. Trailing Edge Loss;** loss associated with the finite trailing edge thickness of the blade. These losses are often combined with the profile losses to simplify comparison to experimental measurements.
- 3. Annulus Loss;** loss due to the skin friction on the hub and casing of the turbomachinery outside of the blade rows.
- 4. Secondary Loss;** loss due to the viscous effects of the flow on the endwalls within the blade row.
- 5. Tip Clearance Loss;** loss due to the leakage of fluid in the gap between the blade tip and the endwall.

Mixing is another source of loss, but can not be categorized as a separate loss mechanism. The mixing loss is a combination of viscous and heat transfer effects that arise from non uniform flow in the machine passage. The blade wake is a prime example of non uniform flow. Other sources of losses have been identified, such as windage loss, leaving loss and

shock loss. The latter is not of concern because of the low subsonic testing speed. The other two are not related to the total pressure losses but rather to the net stage efficiency.

This study is mainly concerned with the influence of the AVR on the profile loss. Therefore, the review of the literature will focus on the first and second type of losses. Secondary losses are also discussed since they are an inherent part of cascade testing with endwalls.

A brief section on the description of the terminology used in this work is presented. The literature review of theoretical and experimental work on the effect of the AVR and two-dimensionality will follow.

2.1 Terminology

2.1.1 Blade Incidence

The terminology used to describe the incidence of a turbine blade is summarized in figure 2.1. The design inlet flow angle $\alpha_{1(des)}$ is determined from the calculation of the velocity triangle. The inlet and outlet metal angle of the blade are β_1 and β_2 respectively. The inlet metal angle is selected by the designer to yield minimum profile losses at design conditions. This design step is iterative since the geometry of the airfoil is not yet complete and the corresponding profile losses cannot be estimated. A correlation from Aronov and Mamaev (1971) can be used during the preliminary design phase to estimate the inlet metal angle given the inlet flow angle.

The off-design incidence i is the difference between the inlet flow angle and the inlet metal angle. The difference between the design inlet flow angle and the metal angle is defined as the design incidence i_{des} . Other names for the design incidence encountered in the literature are induced incidence, minimum loss incidence or optimum incidence. The effective incidence i_{eff} is defined as the difference between the inlet flow angle and the

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

design inlet flow angle. The incidence at which the loss is double the minimum loss is called stalling incidence. The off-design losses reported in this work will specify whether the reference is the incidence i or the design incidence i_{des} . The in-house tests conducted were referenced to the latter.

2.1.2 Cascade Geometry

Figure 2.2 represents the notation commonly used for cascade geometrical parameters and which will be used in this work. Compressor notations are included since most of the literature review contains experimental work performed on compressor cascades.

The blade stagger angle γ is defined as the angle between the blade chord c and the axial direction. Other parameters defined in figure 2.2 include the camber angle θ , blade spacing S and throat width σ . Another important parameter to the designer is the flow deviation δ . In general the flow will tend to be underturned when compared to the exit blade metal angle. This resulting difference in angle is termed deviation. The deflection angle ϵ is also sometimes used to describe the flow turning. Because of the different sign convention at the exit plane, the deviation and deflection angles for a compressor are:

$$\delta = \alpha_2 - \beta_2, \quad \epsilon = \alpha_1 - \alpha_2$$

and for a turbine blade:

$$\delta = \alpha_2 - \beta_2, \quad \epsilon = \alpha_1 + \alpha_2$$

In most cases these parameters are reported as positive values. The deviation reported in this work is negative.

(a) Compressor cascade

$$\delta = \alpha_2 - \beta_2$$

(b) Turbine cascade

$$\delta = \alpha_2 - \beta_2$$

Figure 2.2: (a) Compressor cascade and (b) Turbine cascade notation, reproduced from Gostelow (1984)

2.1.3 Losses

One of the principal end results of a cascade test is the total pressure loss coefficient Y . It reports the loss in total pressure through a blade row normalized with respect to the outlet dynamic pressure. Other coefficients have been identified by Horlock (1960) for the investigation of turbine performance. They can be described mathematically as follows:

$$\bar{\omega}_t = \frac{P_{o1} - P_{o2}}{P_{o1} - P_2} \quad (\text{overall definition}),$$

special case for incompressible flow:

$$\bar{Y}_t = \frac{P_{o1} - P_{o2}}{\frac{1}{2} \rho V_2^2} \quad (\text{coefficient used in this work}),$$

$$\text{and } \bar{\xi} = \frac{h_2 - h_{2it}}{\frac{1}{2} V_2^2} = \bar{\zeta} \quad \text{is also used,}$$

where : subscript t = turbine.

A comparative analysis of the above coefficients was done by Brown (1972) and it was shown that they are equivalent for incompressible flow. Therefore any low speed data correlated in terms of one coefficient may be directly applied to the others. Using a typical turbine stage velocity triangle, Brown predicted the adiabatic and polytropic efficiencies from each of the loss coefficients as a function of Mach number. An increase in the Mach number added to the discrepancies which consisted of 4% at sonic conditions with $\bar{\xi}$ as the most reliable coefficient.

The loss coefficients for compressor cascades are similar to the above turbine coefficients except for the denominator which is referenced to the inlet plane (1), instead of the outlet plane (2).

The kinetic energy loss coefficient ϕ^2 is another way to express losses in turbomachines.

$$\phi^2 = 1 - \frac{V_2^2}{V_{2is}^2} \quad \text{where } V_{2is} = V \text{ at isentropic conditions.}$$

This coefficient can be related to the total pressure loss coefficient Y . For incompressible flow the relation is as follows:

$$Y = \frac{1}{1 - \phi^2} - 1.$$

Another method of measuring losses has been defined by Denton and Cumpsty (1987). Instead of the conventional loss coefficient, this method calculates the entropy generation in the boundary layer. This allows for a physical understanding of loss generation mechanisms which can be useful in the design of a blade section.

2.1.4 Axial Velocity Ratio (AVR)

As mentioned earlier, the axial velocity ratio is an average measure of the flow acceleration in the axial direction at midspan. The proper name for the basic equation should be the axial velocity density ratio. It is defined mathematically as :

$$AVDR = \frac{\int_0^s (\rho_2 V_{a2})_{ms} dy}{\int_0^s (\rho_1 V_{a1})_{ms} dy},$$

where: subscript ms = midspan.

When compressibility effects are negligible (low-speed tests) the simplified axial velocity ratio becomes :

$$AVR = \frac{\int_0^s (V_{a2})_{ms} dy}{\int_0^s (V_{a1})_{ms} dy} .$$

Significant effects on the downstream flow can result from using different testing techniques to vary the AVR. Therefore systematic errors can be present in the results of experimental work using specific methods.

Figure 2.3 shows the three possible values of the AVR. A flow with an $AVR < 1$ will have an axial deceleration, an $AVR > 1$ will have an axial acceleration and an $AVR = 1$ will be at constant axial velocity. Since the AVR is measured at midspan, it only represents the average value of the flow. Therefore local values of the AVR may vary at different spanwise locations.

2.2 Theoretical Review

Several papers have been written over the past 25 years on the effect of the AVR on the aerodynamic parameters relating to a cascade of airfoils. This section will deal with the theories that have evolved from various studies. The first sub-section will investigate the effect of the AVR on the boundary layer of a turbomachine blade. The following sub-section examines the theoretical effect of AVR on total pressure loss, flow turning, blade loading, etc. for specific cases. The cases are presented in a chronological order starting with the oldest. The comparison between theoretical and experimental results will be presented in the following section.

Figure 2.3: Possible values of the Axial Velocity Ratio (AVR).

2.2.1 Effect of Axial Velocity Ratio on Boundary Layers

Heilmann (1972) investigated the effect of the AVR on the boundary layer behaviour. The theoretical work concentrated on the calculation of the laminar and turbulent boundary layer applying the method of Walz. The method was modified to account for the upstream turbulence level on the position of the transition point as proposed by Schlichting. The energy dissipation in the turbulent boundary layer was determined according to Felsch. The following conclusions were obtained from Heilmann's boundary layer calculations:

- (a) The transition point position accuracy is only possible when blade pressure gradients are small.
- (b) The shift in the transition point due to AVR changes will cause shifts in the separation point.
- (c) Separation point position depends much more upon upstream Mach number than upon Reynolds number. Exact determination of the separation position which is required for the total pressure loss calculation is still not possible, especially when pressure gradients are high.

The calculation of the total pressure loss is based on the method described by Stewart and Scholz. The displacement and momentum thickness parameters used in this method are dependent on the trailing edge behaviour and can only be evaluated when the flow is fully attached. If separation occurs these parameters must be evaluated empirically.

In 1974, Dunham (1974), investigated the effect of stream surface convergence on turbomachine blade boundary layers. Consider the element of fluid (figure 2.4) bounded by the wall, by stream surfaces z apart, and by the edge of the boundary layer. In a two-dimensional flow, z is constant, but in a flow with $AVR \neq 1$ it is assumed to vary.

- Let
- U = mainstream velocity
 - u = boundary layer velocity
 - p = pressure
 - δ = boundary layer thickness

Figure 2.4: Fluid element near wall.

Then the mass flow across plane (1) is

$$\int_0^z \rho u \, dy \cdot z ,$$

and the mass flow across plane (2) is

$$\left[\int_0^z \rho u \, dy + \frac{\partial}{\partial x} \int_0^z \rho u \, dy \cdot dx \right] \times (z + dz) .$$

The mass flow coming into the boundary layer between (1) and (2) is therefore:

$$\frac{\partial}{\partial x} \int_0^z \rho u \, dy \cdot dx \cdot z + \int_0^z \rho u \, dy \cdot dz ,$$

and this flow arrives at a velocity U . The momentum equation can therefore be written as:

$$\begin{aligned} & \frac{\partial}{\partial x} \int_0^{\delta} \rho u^2 dy \cdot dx \cdot z + \int_0^{\delta} \rho u^2 dy \cdot dz \\ & - U \left\{ \frac{\partial}{\partial x} \int_0^{\delta} \rho u dy \cdot dx \cdot z + \int_0^{\delta} \rho u dy \cdot dz \right\} \\ & = - \int_0^{\delta} \frac{\partial p}{\partial x} \cdot dx \cdot dy \cdot z - \tau z dx , \end{aligned}$$

where τ = frictional stress.

Now,
$$\frac{\partial p}{\partial x} = -\rho U \frac{\partial U}{\partial x}$$

Substituting.

$$\frac{1}{z} \frac{\partial}{\partial x} \left[z \int_0^{\delta} \rho (u - U) dy \right] + \frac{\partial U}{\partial x} \left[\int_0^{\delta} \rho (U - u) dy \right] = -\tau ,$$

and if δ^* = displacement thickness,

θ = momentum thickness,

H = shape factor δ^*/θ .

The momentum equation becomes:

$$\frac{1}{z} \frac{d}{dz} (z\theta) + (H+2) \frac{\theta}{U} \frac{dU}{dz} = \frac{\tau}{\rho U^2} . \quad \dots (2.1)$$

The analysis of the boundary layer development also follows the axisymmetric derivation proposed by Thwaites. The momentum equation becomes:

$$\frac{U_o \theta^2}{\nu c} = 0.45 \left(\frac{\mu}{U_o} \right)^6 \left(\frac{z}{z_1} \right)^2 \left| \int_0^x \left(\frac{z}{z_1} \right)^2 \left(\frac{\mu}{U_o} \right)^5 d\left(\frac{x}{c} \right) \right| \quad \dots (2.2)$$

where: U_o = reference velocity (e.g. upstream flow),
 ν = kinematic viscosity,
 c = blade chord.

For the turbulent boundary layer, Head's entrainment method can easily be adapted. The flow in the boundary layer can be defined as:

$$Q' = zU (\delta - \delta^*) , \text{ (where } Q' : \text{volumetric flow rate)}$$

and Head's assumption is that the entrainment

$$\frac{dQ'}{dx} = zUF ,$$

where F is a function of the shape factor. The development of the function,

$$Q = U (\delta - \delta^*) , \text{ (where } Q : \text{volumetric flow rate per unit width)}$$

is calculated and from the above equations it will be seen that

$$\frac{dQ}{dx} = UF - \frac{Q}{z} \frac{dz}{dx} . \quad \dots (2.3)$$

It is assumed that F is unchanged as in the axisymmetric case. Some assumptions must now be made about the way z varies within the blade row, knowing only the AVR across it. The first critical assumption made is that the stream surface thickness z is constant across the blade passage (this assumption is not always valid). The variation along the passage can then be represented by :

$$\frac{1}{z} \frac{dz}{dx} = A , \quad \dots (2.4)$$

where A is an arbitrary constant. Integrating and re-arranging,

$$A = \frac{\ln \left(\frac{z_2}{z_1} \right)}{x_2 - x_1} .$$

If the chord of the blade is taken as a unit of length,

$$x_2 - x_1 \approx 1.1$$

measuring x along the flow, so that

$$A = -0.91 \ln(\text{AVR}) . \quad \dots (2.5)$$

The end wall boundary layer grows progressively and the convergence might take place in a stronger fashion as the flow progresses along the passage. Alternatively the expression can be written as follows:

$$\frac{1}{z} \frac{dz}{dx} = A(x_1 - x_2) . \quad \dots (2.6)$$

In which case,

$$A = -1.653 \ln(\text{AVR}) . \quad \dots (2.7)$$

Several blade surface boundary layers measurements have been made in a carefully controlled two-dimensional flow. For a flow with stream surface thickness $z=\text{constant}$ and $\text{AVR}=1$ the well known form of equations (2.1) to (2.3) can be used to give an accurate prediction of the momentum thickness and shape factor. To account for a change in AVR,

the alternative assumptions mentioned above (2.4), (2.5), (2.6) and (2.7) were both examined and gave similar results. The latter was retained and compared to experimentally measured data (figure 2.5). It can be seen that the flow convergence slightly increases the boundary layer thickness for a relatively high AVR of 1.433. The experimental values in the figure are from the suction surface of a compressor blade at 8.4° incidence, an AVR of 1.433, and at low speed, performed by Shaalan (1967). The large discrepancy in the momentum thickness was explained by flow visualisation which showed severe end-wall stall. The increased blockage created by the end-wall effects resulted in a large local value of the AVR. Although the pitchwise average of AVR was 1.433, it varied from unity on the pressure surface to a value of around 4 for the suction surface. The analysis presented above could not be confidently applied to such a high value of AVR, but figure 2.5 shows good agreement between the experimental data and the values predicted for a value of $AVR=4$.

Figure 2.5: Effect of the AVR on the momentum thickness and shape factor H , reproduced from Dunham (1974).

In 1981, Starke (1981), investigated the theoretical behaviour of the trailing edge momentum thickness with respect to the axial velocity density ratio (AVDR) using a modified version of White's integral method. The results can be seen in figure 2.6. The modified method predicts that the momentum thickness θ at the trailing edge varies linearly with the AVDR (μ in this case). The modified White method seems to agree perfectly with another independent method proposed by Mellor and Herring.

Figure 2.6: The AVDR effect on the trailing edge momentum thickness, reproduced from Starke (1981).

2.2.2 Effect of Axial Velocity Ratio on Aerodynamic Parameters

In 1962, Kubota (1962) investigated the effect of axial velocity change through a cascade. The change in axial velocity was reproduced by distributing sinks and sources in a potential flow within a blade passage. An additional circulation is added to satisfy the Kutta Joukowski condition at the trailing edge. The theory was tested on a cascade of flat plates and a NACA-65 series compressor blades. The turning angle corrections for the cascades were evaluated as a function of AVR and incidence.

Pollard and Horlock's (1963) theoretical analysis of the AVR is very similar to Kubota's (1962). The method of singularities, used for the determination of cascade performance, is modified to account for a change in axial velocity. Sinks and sources are used in limited regions of the potential flow field to control the AVR. The effect of AVR on the blade lift and air outlet angle was investigated for a cascade of compressor blades. The results of the theoretical investigation can be found in figure 2.7. Calculations showed that the blade lift and the air outlet angle decreased when the axial velocity was increased through the cascade.

In 1972, Stark (1972), investigated the effect of AVR on aerodynamic coefficients of two compressor cascades with NACA 65-(10) 06 profiles. The potential flow method from Pollard and Horlock (1963) was selected to calculate the performance of the cascades. Boundary layer calculations were performed according to the methods of Thwaites and Truckenbrodt to modify the potential flow velocity distributions. The final calculations of the aerodynamic coefficients from the boundary layer parameters at the blade trailing edge was done by the method of Speidel and Scholz. The theoretical results will be presented with the experimental results in the following section.

Also in 1972, Fottner (1972), studied the behaviour of a cascade of slightly cambered airfoils in a transonic flow including axial mass flow variation. The method used was a variation of the incompressible potential flow velocity distribution scheme proposed by Pollard and Horlock (1963). Correction for compressible flow was made by linearization using the Prandtl-Glauert rule. A supersonic flow field was determined using an empirical correction of the exact Prandtl-Meyer expansion function. The calculations

Figure 2.7: Effect of AVR on blade lift and deviation
 Reproduced from Pollard and Horlock (1963)

show that the total pressure loss decreases with the increase in AVR while the exit angle increases (or flow deviation increases).

In 1976, Galapatti and Hawthorne (1976), investigated a linearized theory for a steady, incompressible, inviscid flow through a linear cascade of airfoils set in a channel with expanding or contracting endwalls. The theoretical approach was based on the conventional linearized thin airfoil theory which was extended to three-dimensions and allowed for the shape of the walls and an inlet shear flow. Previous studies, which assumed a uniform spanwise axial velocity distribution, did not account for the change in circulation in the spanwise direction. The spanwise variation of the AVR for a flow with contracting endwalls is shown in figures 2.8 and 2.9. The variation of the circulation was included in the calculations of aerodynamic parameters. The theory predicts a decrease in both the lift and outlet flow angle for an increase in AVR at midspan.

In 1981, Starke (1981), determined the quasi-two-dimensional, potential flow at the midspan of a compressor cascade using the methods of Pollard and Horlock (1963) and Mani-Acosta for the calculation of the overall cascade parameters. A modified Martensen method was used to calculate the pressure distribution. The correlation prescribed by Leiblein was used to determine the profile losses. Starke assumes in his theoretical analysis that the flow remains quasi-two-dimensional if $0.8 < AVDR < 1.2$, which corresponds to a linear variation of the AVDR in the contraction (or expansion). Boundary layer calculations were performed according to Thwaites method for the laminar portion and a modified version of White's method was used for the turbulent boundary layer (as mentioned earlier). The study concluded that the effect of AVDR on the flow turning can not be precisely predicted by the use of potential theories because viscous effects are not included. However the Pollard and Horlock (1963) method predicted the almost linear relationship between flow turning and AVDR, provided the incidence was relatively close to the design case. The slope of the curve is determined at the $AVDR=1$ condition for a specific airfoil profile. Theoretical results are compared to experimental data in the following section.

Also in 1981, Stark and Hoheisel (1981), investigated the effect of both the AVDR and the aspect ratio on compressor cascade performance in incompressible and compressible flow. The theoretical analysis used is similar to the previously mentioned

Figure 2.8: Variation of axial velocity in cascade with linear endwall contraction, reproduced from Galapatti & Hawthorne (1976)

Figure 2.9: Variation of axial velocity across cascade with linear endwall contraction at different axial stations, reproduced from Galapatti and Hawthorne (1976)

cases. An axial velocity flow field is created with contracting endwalls. The induced velocities, which arise from the presence of the blades, are introduced using the Schlichting method, thus giving a method for the calculation of cascade exit angle and blade pressure distributions in an axial accelerating or decelerating flow. A modified Martensen (MM) method was also used to calculate the blade pressure distributions. The results obtained from the MM method are in better agreement with measured results than those calculated using the Schlichting method. The drawback to the MM method is that it requires that the exit flow angle be prescribed. The authors claim that the aspect ratio defines the axial velocity distribution through the cascade at a given wall shape and contraction ratio. The axial velocity distribution, in turn, decisively determines the local value of pressure distribution, turning angle and loss coefficient. The authors conclude that, at given values of $AVDR \neq 1$, a considerable effect of the aspect ratio on cascade turning angles and pressure distribution is demonstrated.

2.3 Experimental Review

This section contains experimental investigations of the effect of the AVR on aerodynamic parameters for a cascade of airfoils. The theoretical curves presented with the experimental results in the following figures can be referenced to the previous section. Once again, the cases are presented in a chronological order starting with the oldest.

2.3.1 Effect of Axial Velocity Ratio

In 1962, Kubota (1962), investigated the results of several experimental tests performed on NACA 65 series compressor cascades using low-speed wind tunnels. The author plotted turning angle (deflection) and AVR against angle of attack. The results showed, for all cases, that an increase in angle of attack (incidence) produced an increase in the AVR. However, the AVR seemed to be an independent variable rather than the controllable parameter.

In 1967, Pollard and Gostelow (1967), evaluated the status of experimental testing of compressor cascades at low speeds. The investigation was directed toward the analysis and improvement of the airflow and testing techniques. Porous sidewalls with suction were used to control the boundary layer size and correspondingly the AVR. Variation of axial velocity, aspect ratio, Reynolds number, and turbulence level were performed independently to study their influence on aerodynamic parameters. Results showed that most of the change in axial velocity occurred within the blade row especially when the AVR is near unity (figure 2.10). To obtain a value of $AVR = 1.0$ at $Re = 1.95 \times 10^5$, approximately 3% of the mainstream flow was removed through the porous walls. An increase in AVR across a cascade reduces the static pressure rise through it and causes the deviation to decrease.

Figure 2.10: Variation of axial velocity through a cascade at $Re = 1.95 \times 10^5$, reproduced from Pollard and Gostelow (1967).

The change in deviation with axial velocity ratio is shown in figure 2.11 for two different cascade geometries at a Reynolds number above critical. The test on the cascade with a space/chord ratio $s/c = 1$ was performed at a turbulence level of 2.68% and for the other spacing $s/c = 0.875$, the turbulence level was 0.4%. The authors state that the difference in turbulence level does not appear to affect the slope of the AVR vs δ curve.

Figure 2.11: Change in deviation with axial velocity ratio, reproduced from Pollard and Gostelow (1967).

The curves can be approximated by the following relationship:

$$\delta = \delta_0 - 10(\text{AVR} - 1)$$

where δ_0 is the deviation at $\text{AVR} = 1.0$. The experimental results agreed with the previous theoretical investigation performed by Pollard and Horlock (1963). The blade static pressure distribution variation with axial velocity ratio is shown in figure 2.12. The reduction in pressure rise through the cascade is reflected in a reduction in the pressure gradient on the suction surface of the blade profile. The authors also found that an increase

in free-stream turbulence promoted early boundary layer transition and hence delayed the increase in loss generally associated with a reduction in Reynolds number. The results also showed that for a test section with solid endwalls, an aspect ratio greater than 3 must be used to avoid the influence of the corner stall and secondary flow in the measuring plane. Even with an aspect ratio of 5, the AVR in the midspan plane was measured at a value greater than unity.

Figure 2.12: Pressure distribution variation with AVR at $R_e = 1.96 \times 10^5$, reproduced from Pollard and Gostelow (1967).

In 1972, Stark (1972), investigated the effect of AVR on the aerodynamic coefficients of two compressor cascades with NACA 65-(10) 06 profiles. Suction was used on the endwalls to control the AVR. The aspect ratio of the blades was 3.75. Trip wires were used to avoid laminar separation. The free-stream turbulence was 0.3% at a Reynolds number $= 2 \times 10^5$. The results of the experimental investigation are compared to theory (previous section). Figure 2.13 shows the effect of AVR (μ) and inlet-air angle on flow deflection for a space-chord ratio $t/l = 0.8$. The general trend seems to be an increase in deflection for an increase in the AVR. The effect on total pressure loss (ζ) is presented in figure 2.14, the influence of the AVR seems to be dependant on the inlet air angle. Blade static pressure distributions are shown in figure 2.15.

Figure 2.13: Effect of AVR on flow deflection angle, reproduced from U. Stark (1972).

Figure 2.14: Effect of AVR on the total pressure loss coefficient, reproduced from U. Stark (1972).

Figure 2.15: Effect of AVR on the blade-surface pressure distribution, reproduced from U. Stark (1972).

An increase in AVR (μ) causes a systematic shift of the pressure distribution to lower values in both the theoretical and experimental curves. An increase in value of AVR also diminishes the adverse pressure gradients on the suction side.

In 1972, Heilmann (1972), studied the influence of axial velocity density ratio (Ω) on compressor cascade performance. The effect on blade pressure distributions is shown in figure 2.16, for a case with and without a turbulence wire. It should be noted that the blade static pressure coefficient (S) is defined as follows:

$$S = \frac{P_{tot1} - P_{local}}{P_{tot1} - P_1}$$

where other pressure coefficients such as C_p are defined as :

$$C_p = \frac{P_{local} - P_1}{q_1}$$

The results from figure 2.16 show the same trends as the previously mentioned studies. Figure 2.17 shows the effect of AVR (Ω) on both the total pressure losses and the flow deflection. An increase in the AVR reduces the total pressure losses, especially at higher inlet flow angles, and mildly increases the deflection angle.

Also in 1972, Fottner (1972), reported on the effect of axial mass flow rate variation across a compressor blade row. The experimental results, for a NGTE profile 10 C4/30C50, with respect to total pressure loss and the exit angle are compared with theoretical values in figure 2.18. An increase in AVDR (Ω) results in a decrease in total pressure loss (ω) and an increase in outlet flow angle (β_2). The influence of a higher AVR (>1) seems to have a weaker effect on the total pressure loss.

In 1973, Ohuonu and Shaw (1973), investigated several cascades of compressor blades under the influence of the endwall boundary layers. The results presented are similar to the above cases. An increase in AVR produces a decrease in exit flow deviation (or an decrease in exit flow angle). An increase in the AVR also shows a systematic shift of the pressure distribution to a lower value with the same pressure gradient trends as discussed

Figure 2.16: The effect of AVDR (Ω) on the blade static pressure distribution, reproduced from Heilmann (1972).

Figure 2.17: The effect of AVDR (Ω) on the total pressure loss (top) and the flow deflection (θ) (bottom), reproduced from Heilmann (1972).

Figure 2.18: The effect of AVDR (Ω) on the total pressure loss and the outlet flow angle (β_2), reproduced from Fottner (1972).

Figure 2.19: Deviation factor $(\delta_0 - \delta)/(AVR - 1)$ against incidence i for various camber and stagger angles, reproduced from Ohuonu and Shaw (1973).

by Stark (1972). A deviation factor $(\delta_2 - \delta_1)/(AVR - 1)$ is calculated and plotted with respect to the AVR in figure 2.19 for different camber and stagger angles. The authors also conclude that the deviation decreases linearly with increase in AVR. The rate of decrease of deviation depends on the camber angle, stagger angle and incidence angle and is much more significant at positive incidence when suction surface separation is inhibited by the increase in axial velocity.

In 1973, Masek and Norbury (1973), determined the effect of varying axial velocity on the performance of a prescribed velocity distribution on a compressor cascade. The experiments covered a range of incidences over 20° at low speed with a Reynolds number of 1.5×10^5 and a turbulence intensity of 2-3%. The effect of AVR on the blade pressure distribution follows the previously discussed trends. An increase in AVR causes a linear increase in the flow deflection with a higher slope for positive incidence values ($+4.6^\circ$, $+9.6^\circ$). The effect of AVR on loss for different incidences is shown in figure 2.20a) and the effect of incidence on loss for different AVRs is shown in figure 2.20b). A change in AVR affects drastically the loss bucket for positive incidences, and moderately for negative incidences. The authors conclude that an increase in AVR reduces the separation giving an increase in deflection and a reduction in loss. The increase in AVR also causes a slight decrease in static pressure rise across the cascade.

Also in 1973, Ilyas and Norbury (1973), described the effect of axial velocity variation and compressibility on the performance of a compressor cascade. The AVR was controlled by variable suction applied to the porous endwalls. Figure 2.21a) and 2.21b) show the effect of AVR on total pressure loss (ξ) and flow deviation angle ($=\alpha_2 - \beta_2$) respectively. A decrease in total pressure loss was noticed for an increase in the AVR, up to 1.05, for low Mach numbers. Above $AVR=1.05$, the loss would rise again giving a minimum loss value of AVR. The authors blame the thickening of the endwall boundary layers, which causes increasingly strong secondary flows which, in turn, result in losses in total pressure outside the viscous wake of the blade. An increase in Mach number also increases the losses, this may be considered to be the result of compressibility effects. A small reduction in deviation is noticed as AVR increases. For $M_1=0.13$, there is a pronounced minimum above which the deviation increases with the AVR. This is attributed

Comparison of values of a.v.r.

		i	-10.4	-5.4	0	+4.6	+9.6
a.v.r.	Liverpool CT2		1.04	1.08	1.14	1.21	1.30
	DFL		1.02	1.05	1.09	1.15	1.23

Figure 2.20: a) top, the effect of AVR, and b) Bottom, the effect of incidence, on total pressure loss (ξ), reproduced from Masek and Norbury (1973).

Figure 2.21: The effect of AVR and inlet Mach number M_1 on: a) top, loss (ξ), and b) bottom, deviation (δ), reproduced from Ilyas and Norbury (1973).

to the low aspect ratio of the cascade and the increased influence of the secondary flows as suction is reduced.

In 1975, Starken et al. (1975), experimented with a subsonic cascade with a stator blade row. Tests were performed with different inlet Mach numbers, inlet flow angles and axial velocity density ratio. Once again a suction system was used to control and vary the AVR. A plot of total pressure loss coefficient (ω) versus AVDR (Ω) is shown in figure 2.22 for different inlet Mach numbers. The effect of a lower AVR (<1) increases the losses while the losses seem to level off at values of AVR >1 . The reason for the loss increase is explained by the increasing deceleration of the flow on the suction surface which causes the boundary layer separation point to move upstream, and a constant pressure region to develop on the rear blade surfaces. The right hand shift of the curve is attributed to the separation point movement for higher inlet flow Mach numbers. A small reduction in exit flow angle was reported for an increase in AVR. The effect of the AVR on the blade static pressure distribution is similar to the previously mentioned cases. The influence of AVR on the losses for different inlet flow angles is presented in figure 2.23. Higher loss is noted at a positive incidence angle (i.e. $\beta_1=44^\circ$). The authors conclude that the reduction in AVDR at positive incidences resulted in severe boundary layer separation from the blade suction surface.

In 1976, Galapatti and Hawthorne (1976), reported on an experimental study of a cascade of NACA 0012 symmetrical airfoils in a porous endwall tunnel, in a tunnel with contracting endwalls, and also with a thickened endwall boundary layer. Results show the spanwise influence from the presence of the endwalls; the AVR value increases from 1.0 near the endwalls to up to 1.25 at midspan. These differences influence the exit flow angle and lift coefficient for the three cases. Both the lift and outlet flow angle are decreased with an increase in the midspan AVR.

In 1981, Starke (1981), used a wall build-up approach to produce a geometrical contraction to provide variation of the axial velocity density ratio (AVDR). The 9C7/32.5C50 blades were tested with different blade spacings and blade angles. The data presented shows that an increase in AVDR causes a slight increase in the flow turning

Figure 2.22: The effect of AVDR (Ω) and inlet Mach number M_1 on the total pressure loss coefficient (ω), reproduced from Starken et al. (1975).

Figure 2.23: The effect of AVDR (Ω) and inlet flow angles (β_1) on the total pressure loss coefficient (ω), reproduced from Starken et al. (1975).

$(\beta_2 - \beta_1)$, which is supported by theory. An increase in AVDR also reduces the total pressure losses; this function was noted to be stronger at positive values of incidence. The blade static pressure distribution reported for different values of AVDR is consistent with the previous papers.

Also in 1981, Stark and Hoheisel (1981), investigated the combined effect of axial velocity density ratio and aspect ratio on compressor cascade performance in incompressible and compressible flows. Several tests were performed on two different cascades, one with 9C7/32.5C50 blades and the other with DCA 56.8-0.7 blades. The tests concluded that the secondary flow influenced the thickness of the boundary layer, heat transfer coefficients, and losses at the midspan of an airfoil in a compressor cascade. They also showed the effect of the AVDR (μ) on total pressure loss (ζ_{v1}), turning angle ($\Delta\beta$) and diffusion factor (D) (presented in figure 2.24). The diffusion factor is defined as follows:

$$D = 1 - \frac{u_2}{u_1} + \frac{1}{2} \frac{S}{C} \left(\sin\beta_1 - \frac{u_2}{u_1} \sin\beta_2 \right) + \frac{t_{\max}}{C}$$

where : S : blade spacing
 C : chord length
 $t_{\max}$: maximum blade thickness

These results echo the same trends observed in the previously mentioned experimental works and theoretical analyses. The authors stress the importance of accounting for both the AVDR and aspect ratio in the analysis of the experimental results.

In 1983, Sharma and Graziani (1983), examined the effect of endwall secondary flows on the suction surface boundary layers in turbine and compressor cascades. They concluded that the AVDR effect, in itself, is insufficient to explain the influence of endwall flows on the aerodynamic and heat transfer performance of the suction surface of the blade.

Finally in 1985, Kiock and al. (1985), tested the same gas turbine rotor blade section in four different wind tunnels with different test sections and instrumentation. A

Figure 2.24: The effect of AVDR (μ) on the Diffusion factor (D), flow turning ($\Delta\beta$), and total pressure loss coefficient (ζ_{v1}), reproduced from Stark et al. (1981).

comparison of the results from blade pressure measurements reveals the order of magnitude of tunnel effects. One of the conclusions reached by the authors is that the axial velocity density ratio indicates nearly two-dimensional flow only in two facilities. The other two tunnels give lower values due to the inlet geometry. They also concluded that the AVDR does not appear to affect the loss and exit flow angle in the range of AVDR of 0.9 to 1.0, unlike in compressor cascades.

2.3.2 Summary of Literature Survey

For the sake of clarity, a summary of the findings from the literature survey is included in table 2.1. The effect of the axial velocity density ratio on different aerodynamic parameters is presented in a comparative fashion between studies. The table also identifies the other parameters which were varied along with the AVDR for each specific study. The cases are presented in chronological order and are identified as Theoretical (T) or Experimental (E) works. The method used to control the AVDR is also identified. It should be noted that the nomenclature and the conventions used in reporting parameters vary from paper to paper. The nomenclature used for identification of the flow deviation at the exit plane for a specific paper is reported in parenthesis. Other abbreviations used include:

incr	: increase	Tu	: Turbulence intensity in x-direction
decr	: decrease	S/C	: blade Spacing to Chord ratio
i	: incidence	M	: Mach number
AR	: Aspect Ratio	b.l.	: boundary layer
Re	: Reynolds number.		

Most of the cases encountered were on a cascade of compressor blades. The general consensus is that compressor blades are more sensitive than turbine blades to the secondary flows, and correspondingly to the AVR. Suction of the boundary layer through porous endwalls was used in most of the test to control the AVR. Contracting endwalls were sometimes used to increase the AVR above the values achieved with parallel solid walls. The effect of the AVR on the different aerodynamic parameters is the same for most cases with a few exceptions. The increase in AVR generally seems to cause a decrease in total pressure loss, a slight linear decrease in flow deviation (or a slight linear increase in flow

Table 2.1: Summary of Literature Survey

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author	year	Theoretical Experimental	AVR control method	blade section investigated	AVR increase effect on Y_t	AVR increase effect on deviation	AVR increase effect on C_p	other parameters examined
1 Kubota	1962	T&E	none	compressor	-	-	-	incr i = incr AVR
2 Pollard et al.	1963	T	-	compressor	-	linear decr	lower values	-
3 Pollard et al.	1967	E	suction	compressor	-	linear decr	lower values	AR, Re, Tu
4 Stark	1972	T&E	suction	compressor	decr for +i incr for -i	linear decr (incr in deflection)	lower values	i, s/c
5 Heilmann	1972	T&E	suction	compressor	strong decr +i weak decr -i	linear decr (incr in turning)	lower values	i, M, Tu
6 Fotner	1972	T&E	suction	compressor	decr	decr (decr outlet angle)	-	M, i
7 Ohuonu et al.	1973	E	suction	compressor	-	linear decr	lower values	i, camber, stagger
8 Masek et al.	1973	E	suction	compressor	strong decr +i weak decr -i	linear decr (incr in deflection)	lower values	i, Tu
9 Ilyas et al.	1973	E	suction	compressor	decr AVR < 1.05 incr AVR > 1.05	decr (incr AVR > 1.1 at M=0.13)	lower values	M
10 Dunham	1974	T	-	compressor	-	-	-	b.l. vs AVR
11 Starken et al.	1975	E	suction	compressor	decr	decr (decr outlet angle)	lower values	M, i
12 Galapati et al.	1976	T&E	suction, contra- cting endwalls	compressor	-	decr	lower values	AR, s/c
13 Starke	1981	T&E	contracting endwalls	compressor	strong decr +i weak decr -i	linear decr (incr in turning)	lower values	i, b.l. thickness, diffusion ratio
14 Stark et al.	1981	T&E	suction	compressor	decr	linear decr (incr in turning)	lower values	AR, i, M
15 Sharma et al.	1983	E	-	compressor & turbine	-	-	-	b.l. behavior
16 Kiock et al.	1985	E	solid parallel walls	turbine	no effect AVR=0.9 to 1.0	no effect AVR=0.9 to 1.0	-	wind tunnel and instrumentation

deflection), a thickening of the midspan boundary layer, and a shift of the blade static pressure curve to lower values with a reduction in the adverse pressure gradients on the suction side. The strength of these relationships vary from study to study, but the general trends remain the same.

Limited data were found on the influence of the AVR on the two-dimensionality or performance of a cascade of turbine blades. Sharma et al. (1983) concluded that the AVDR could be used as a tool, but not the determining factor, to explain the effect of endwall flows on a turbine blade section surface performance. Kiock et al. (1985) concluded that for an AVDR between 0.9 and 1.0, the effect on the loss and exit angle was negligible, unlike the compressor cascade. Unfortunately, both studies did not include the data to collaborate their conclusions.

2.3.3 Present Investigation

The present experimental investigation includes a study of a cascade of turbine blades in a low-speed wind tunnel. A novel arrangement of end-plates is used to control the convergence and divergence of the flow across the blade row. The aerodynamic parameters measured and reported include: the axial velocity ratio, total pressure loss, flow deviation, base pressure coefficient, Reynolds number, turbulence intensity and blade static pressure distribution. The independent variables, which are separately controllable, include the axial velocity ratio, the incidence and the Reynolds number. The relationships between the parameters are described in the presentation of the results in chapter 6.

3 COMMISSIONING OF NEW WIND TUNNEL

The increased cascade testing performed at Carleton University warranted the design and construction of a new open jet wind tunnel to work in parallel to the original tunnel (#1). The new wind tunnel will be referred to as tunnel #2. The design philosophy of the wind tunnel was to use the latest technology to achieve a well-behaved uniform flow with low turbulence levels. This chapter briefly describes the tunnel requirements, hardware and installation of the components. Some operational, maintenance and safety issues are also discussed.

3.1 New Tunnel Requirements

The tunnel is designed to meet certain requirements. The mass flow rate from the fan must be sufficient to provide a Reynolds number of above 3.0×10^5 in the test section. The quality of the air flow must be superior to tunnel #1. The motor and fan noise must be less than the original wind tunnel. The exit to the wind tunnel must provide a flexible interface to allow for the attachment of both test sections (turbine and compressor cascade). The location of the air intake into the bellmouth of the fan should be somewhat distant from the wall. Nearby obstructions could cause the ingestion of vortices. The exhaust flow from the test section should be free of obstructions (wall, ducts, lights, etc.), since these can create an unwanted back pressure. Ergonomics are also considered in the placement of some of the components for ease of access and maintenance. Safety precautions must also be taken with the motor, electrical supply, fan belt and air intake into the fan.

3.2 Hardware

A schematic of the new wind tunnel #2 can be found in figure 3.1. The components are identified and labelled. A great deal of importance is placed on the proper treatment of the flow. The overall length of the flow treatment section from the beginning of the honeycomb to the exit of the contraction is 4.6 meters. A bellmouth inlet to the fan is constructed from plastic resin to insure a smooth intake of air. The centrifugal fan is driven by a kevlar fan belt connected to a three phase AC motor. The fan is identical to the one in tunnel #1 and can deliver a mass flow rate of 5 kg/s at a total pressure of 12 cm of water (gauge). The swirl in the flow is removed with the help of a honeycomb section just downstream of the fan with a length to cell diameter ratio (l/d) of over 20. The flow is then diffused from an area of (418 mm x 418 mm) to (987 mm x 987 mm) over a distance of 840 mm to the settling chamber. The diffusion process is encouraged by 5 screens bowed normal to the flow as shown in figure 3.2. Once in the settling chamber (1465 mm wide, 1275 mm high and 1790 mm long), the flow is filtered through four more screens, with the last screen at a slightly lower open area ratio. The flow is then contracted 14:1 to the exit of the tunnel. The exit flow area of the tunnel is 203 mm wide by 675 mm high. The maximum exit velocity is about 30 m/s which corresponds to a Mach number of 0.09. The motor operation is controlled by a frequency controller with a remote keypad. The acceleration and deceleration along with other parameters can be programmed into the unit. The frequency of the signal sent to the four pole motor can easily be selected and entered via the remote control. The RPM of the fan can be calculated as follows:

$$\text{RPM} = \frac{\frac{f_{\text{input}}}{\text{number of poles}}}{2} \times 60$$

The power to the frequency controller should be turned off when the tunnel is not in use or when maintenance of the components is required. The circuit breaker is located on the wall behind the fan.

Figure 3.1: Schematic of new wind tunnel #2 with components.

Figure 3.2: Picture of flow screens in diffuser (note screens bowed normal to flow direction).

3.3 Installation

The new wind tunnel is located in Room 126 of the MacKenzie building, along the south wall and parallel to tunnel #1. Careful attention was given to the layout of the tunnel to avoid any obstructions in the flow path, which can be detrimental to the flow quality, as mentioned before. All of the components were anchored to the floor with a vibration control pad inserted between the components and the floor. The pads are designed to absorb vibration and retard metal fatigue and diminish noise levels. A kevlar fan belt is used to reduce noise and increase life span. The tension on the belt must be adjusted to give a 1/2" deflection at midheight with a 30-35 lb force. This adjustment can be done with a scale, rope and ruler. The alignment of the belt can be done by adjusting the motor position. The motor can be moved laterally with some amount of play by loosening the four retaining bolts. Shims can be used to provide adjustments in the other two axes. A proper alignment of the belt is crucial to its safe operation and long life. Periodic greasing of the fan and motor bearings is also required. A cage is installed over the pulleys and belt for safety reasons. The power supply should be switched off before removing the cage for maintenance and adjustments.

The air supplied to the tunnel is contaminated with dirt and dust particles. Evidence of this was found on the screens after several hours of operation. The fouling of the screens can influence the measurements. The dirt can also change the calibration of the probe and plug static pressure taps. The effect of fouled screens on the downstream measurements was examined by Tremblay (1989) for tunnel #1. A maintenance door was designed into the diffuser to simplify the cleaning of the screens (figure 3.2). A cleaning of the upstream diffuser screens once a month, and the settling chamber screens every four months should be sufficient under normal operating conditions. The contraction must be removed along with the spacers before the settling chamber screens can be withdrawn. The honeycomb section should also be cleaned regularly. The edge of the honeycomb must be treated with care since it is made of thin aluminium and can easily be damaged.

The interface between the tunnel and the test section is achieved with a mounting flange. The horizontal alignment of the flange to the inlet of the test section can be carried out by adjusting the height of the contraction with the threaded support leg on the casters. It

is suggested to level the test section first, then adjust the contraction height to match. Some iteration of this procedure might be required to get an acceptable fit. Insulation foam and rubber is used at the interface of all the components to prevent leakage.

3.4 Flow Quality

The flow quality at the exit to the wind tunnel is measured in the upstream reference measurement slot on the test section (see figure 4.2). The spanwise velocity profile at different heights is measured with a pitot tube referenced to the static pressure tap in the same measurement plane. This value is non-dimensionalized by the pressure difference in the contraction. The results of this investigation at a Reynolds number of 3.0×10^5 are shown in figure 3.3. The scale of the measurements is omitted and the curves are shifted in the vertical axis to their respective heights for clarity. The uniformity of the flow from the wind tunnel is very good. The velocity remains constant over the entire flow field with little variation. The size of the boundary layer at the measurement plane is constant over both endwalls with an average value of about 12 mm. It should be noted that these tests were performed with the upstream boundary layer bleed slots closed and sealed. The boundary layer size is smaller (≈ 6 -8 mm) with the removal of part of the boundary layer with the bleed slots. The bleed slots are used in the normal operation of the test section. The boundary layer size on the floor and ceiling of the section is measured at less than 12 mm. This boundary layer size is not as crucial as the endwall values since the flow will be evacuated through the top and bottom flaps.

One of the main goals of the new wind tunnel is to reduce the turbulence levels. A honeycomb section, 9 flow screens, longer and larger settling chamber, and a smooth and large ratio contraction were all designed to provide low turbulence. The turbulence intensity is measured at 0.3% just downstream of the tunnel exit. The value measured in tunnel #1 was about 1%-1.5%. A more thorough discussion of the turbulence intensity and its measurement will be presented in the following chapters.

$$\text{where } V_c = \sqrt{(P_{\text{contraction inlet}} - P_{\text{contraction exit}}) \frac{2}{\rho}}$$

Figure 3.3: Spanwise velocity profile at different heights, in reference plane of test section at $Re = 3.0 \times 10^5$.

4 CASCADE APPARATUS AND INSTRUMENTATION

This chapter describes the test section, instrumentation and data acquisition system used in this work. The end-plates geometry, along with the equipment set-up for the three-hole probe and the blade static pressure measurements are also presented.

4.1 Description of Test Section

The test section which includes the rectilinear cascade of turbine blades is shown in figure 4.1. The section was designed by Goobie (1989) to study the effect of several parameters such as the incidence, pitch to chord ratio, tip clearance and aspect ratio on the aerodynamic performance of the blades. Two boundary layer bleed slots are located far upstream at the inlet and can be adjusted to minimize the boundary layer size on the endwalls. The front wall of the section is made of plexiglass to allow for flow visualisation and is mounted on a separate frame which can be removed to permit access to the blades and control surfaces. The set of blades is fixed on a separate mounting plate; this allows for the geometry of the cascade to be varied. However, it should be noted that the geometry was kept constant in this work. The mounting plate is attached to the large turntable supported by a shaft and two pillow block bearings. The incidence of the blades can be adjusted to 0.1° with the protractor and vernier scale scribed onto the back of the turntable. Nine control surfaces are mounted on the section to enable the flow to be modified and adjusted to meet uniformity, periodicity and convergence requirements. A more detailed description of the components and the dimensions of the test section can be found in Goobie (1989).

Figure 4.1: Schematic of variable incidence cascade test section.

The blade section being studied is a large scale model of a midspan section of a power turbine blade from a midsize Pratt & Whitney Canada turboprop engine. The scaled blade has a 163 mm chord and a 200 mm span. The cascade geometry and measurement plane nomenclature is shown in figure 4.2. The classification (#1, #2 or ref) is used as a subscript on the parameters to identify the measurement plane location. The instrument slot far downstream of the test section was not used in this work and is therefore ignored. It should be noted that the inlet measurement plane (#1) is located 60% axial chord length upstream of the blade leading edge plane to account for the presence of the end-plates. The implication of this will be discussed in the following sections. More details on the blade and cascade geometry can be found in appendix A.

In addition to the five blades, a sixth interchangeable blade was manufactured and instrumented with 11 rows of 43 static pressure taps on half the span of the blade. The distribution of the taps are as follows: 25 taps on the suction surface, 17 taps on the pressure surface, and one base pressure tap at the trailing edge. The x and y blade coordinates for the static pressure taps can be found in appendix C. The instrumented blade is mounted in the middle blade position where the flow conditions are better controlled. The pressure tap pairs (#1 and #42), (#2 and #41) near the trailing edge and, (#25 and #26) near the leading edge share the same internal pressure tubes. Therefore, one of the taps in the pair must be sealed while the other is being monitored. This is done by splitting the measurement into two, one measurement of taps #1 to #25 with taps #26, #42 and #43 sealed, and the second from #26 to #43 with taps #1, #2 and #25 sealed. The internal pressure tubes are also shared by the other spanwise taps from the 11 rows. Since, only midspan measurements of static pressure distribution are of interest, the remaining rows are sealed.

4.2 Instrumentation and Data Acquisition

The instrument used to measure flow velocity, flow angle and total pressure is the three-hole pressure probe as shown in figure 4.3. The main dimensions along with the probe tip schematic are presented. The yaw and pitch angle nomenclature is also included. The probe

Figure 4.2: Cascade geometry and measurement plane nomenclature.

Note: scale 1:1

Figure 4.3: Three-hole pressure probe configuration and nomenclature.

head is 0.7 mm thick and 2.0 mm wide corresponding to less than half a trailing edge thickness. The probe is mounted in the test section to a rod with a holder socket. The tip of the probe is set at a distance greater or equal to five holder diameters upstream. A second identical three-hole probe was manufactured with a shorter overall length for the inlet measurements. A shorter probe is required between the end-plates to clear the leading edges of the blade row. The angle of the probe is adjusted to 0° (level) at the inlet, and to the blade exit metal angle at the outlet, with an adjustable protractor. The blade metal angle relative to the horizontal varies with incidence value due to the rotation of the turntable. The mounting rod and socket align the probe in the pitch direction, although this value is not as crucial as the yaw alignment. A further discussion of the probe and its calibration can be found in chapter 5.

A hot-wire anemometer is used to measure the velocity fluctuations and the turbulence intensity of the flow. The anemometer is a DISA probe type 55P11 with a Tungsten wire of 6 micrometers in diameter and a temperature coefficient of resistance of 0.36%. The hot-wire probe is controlled with a TSI (Thermo Systems Inc.) constant temperature anemometer Model 1053A (inv. #925, C.U. #001726). Included in the same unit is a TSI variable decade probe resistance Model 1056, a TSI signal linearizer Model 1052, and a TSI power supply Model 1051-6. A Spectroscope real-time analyzer Model SD330 (serial #OP-S-078-020, inv #818) is used to investigate the fluctuations at different frequency levels. The voltage signals from the hot-wire are measured with a Fluke Model 8840A multimeter (serial # 3698273). The mean velocity is measured from the DC signal and the fluctuating component of velocity is measured from the RMS value of the AC signal. More details on the hot-wire probe and its calibration can be found in chapter 5.

The movement of the probes is possible with a traversing system attached to the test section. The mounting rod is attached to a 2-axis traverse gear which provides movement in the vertical and horizontal directions. The access of the rod into the section is done via an instrument slot which allows movement of the rod without leakage of the passage flow. Two stepping motors drive the traverse gear mechanism and are powered by a motor controller unit connected to a microcomputer via an amplifier. The microcomputer is an IBM compatible machine.

The pressure from the probes and static pressure taps are monitored using two pressure transducers from Rosemount (serial #591 and 633). Tygon pressure tubing is used to carry the pressure signal. The tubing is flexible and can easily be connected and disconnected while keeping a good seal. The calibration of the transducers was verified periodically using a micromanometer. Since the calibration curve is linear, only a few points within the tunnel operational range were examined to check the calibration. The transducers are very stable with almost no zero drift. A scanivalve pressure management system was used to accommodate the number of pressures scanned. The scanivalve unit can manage up to 48 independent pressures and output one value at a time. The unit is driven by a scanivalve controller connected to the microcomputer. The analog to digital conversion of the pressure signal is done with a Sciometric Instruments A/D converter Model 81/902 serial # 9023J/19006. The converter is connected directly to the microcomputer via the serial port and communication is carried out with a special set of commands. The resolution of the analog to digital conversion is 12 bits or 0.024% (1.2 mV/gain). The pressure signal received is then adjusted for the zero reading and transformed into a pressure value using the calibration factors for the transducers.

The sampling time required to achieve a good average for the signal was investigated. The waiting time required for the pressure to stabilize after a pressure step from the scanivalve going from a high positive to a high negative value is about 15 seconds. This represents the worst case scenario with a long length of pressure tube. The pressure signal is then averaged over 20 seconds (after the waiting period), reduced to pressure values, non-dimensionalized by reference values, and stored in a formatted file. All of this data acquisition and management is automated and performed by the computer program.

The calibration apparatus used to calibrate the three-hole probes and hot-wire anemometer is shown in figure 4.4. It consists of a blower box powered by two fans controlled independently. The flow is discharged, through a circular nozzle, to open atmosphere. The total pressure inside the box is monitored with a pressure pick-up on the side of the box. The probes are mounted on a two axis turntable system which can provide variations in the yaw and pitch angle without changing the absolute tip position of the probe. Two stepping motors provide the rotation of the turntables through a worm gear system with virtually no backlash. The motors are controlled in the same manner as the traverse gear system.

Figure 4.4: Calibration apparatus.

4.3 Modified End-Plates

4.3.1 Introduction

The end-plates were first used by Goobie (1989) to allow the inlet wall boundary layer to pass in the space in between the plates and the endwalls. This would increase the region of 2-D flow, since the boundary layer on the plates would be thinner than on the endwalls. The higher blockage in the outer passages, however, forces the flow to the centre passage (between the end-plates). This results in a small angle of attack on the end-plates leading edge which was shown to create a separation region with a sizeable increase in the boundary layer size. To eliminate this angle of attack, Tremblay (1989) converged the end-plates until no separation was discovered and the 2-D region was deemed wide enough. This resulted in an unwanted acceleration of the freestream flow and an increase in the blade boundary layer thickness due to the pinching effect from the plates.

4.3.2 Development of End-Plates

The solution to the separation problem, when the plates were parallel or even diverged, appeared to lie in the end-plate leading edge configuration. Figure 4.5 shows the development of the end-plate leading edge configuration. The first attempted configuration was a cylindrical leading edge tangent to the inner surface of the end-plate. Surface oil flow visualization was used to examine the flow behaviour at the leading edge for a case with the end-plates slightly diverged. The flow remained attached over most of the cylinder but separated just before the flat portion of the end-plate. A smoother airfoil shape leading edge was then investigated, but a separation region was still noticed. The next step was to investigate a knife-edge leading edge slat. The slat was designed to adjust to the oncoming flow angle and gradually guide the flow along the end-plates. This design was successful in preventing the separation bubble at the leading edge. This could probably be attributed to the fact that, since the slat extends upstream, the boundary layer has time to reach a transitional (or possibly turbulent) state which provides better mixing in the direction normal to the plate, and thus prevent separation in the diverging portion.

Figure 4.5: Development of the end-plate leading edge configuration.

A schematic of the end-plates configuration is shown in figures 4.6 and 4.7. The leading edge extends 150% C_x upstream of the blade leading edge plane. The blade cutouts allow the plates to move in the spanwise direction. The holes are slightly oversized to permit the convergence or divergence of the plates and are sealed with insulating foam to prevent leakage of the flow from the outer passage. The leading edge slat assembly can be adjusted (in 20 mm increments) in the pitchwise direction to adapt to any angle of incidence. The flexible portion of the leading edge slat is 60 mm. The angle of attack of the slat is controlled by three independent adjustment mechanisms. These mechanisms are made of a threaded rod connected from the leading edge and anchored to the end-plates. Two nuts are used to adjust the tension on the rod and thus the angle of attack of the slat. The convergence or divergence of the end-plates is regulated by four threaded rods (2 upstream and 2 downstream) running spanwise through the end-plates and attached to the test section (figure 4.7). The plates are held in place by the adjustment nuts at the desired location. The nomenclature used in identifying the position of the end-plates is also shown in figure 4.7. The values (l_1 , k_1 , l_2 , and k_2) are measured with a ruler, and the slat angle of attack is recorded as the offset distance ϵ of the leading edge to a straight edge positioned flush on the end-plate flat section.

4.3.3 Positioning of the End-Plates

The offset distance of the end-plates from the endwall should be as small as possible to increase the effective aspect ratio of the blades. The limiting factor is the size of the boundary layer that has developed from the inlet to the test section. A study of the boundary layer thickness near the leading edge plane was carried out for the worst case $+20^\circ$ incidence (i.e. longest distance from the test section inlet). The results of the boundary layer velocity profile are shown for two pitchwise locations in figure 4.8. Both curves show a boundary layer thickness of about 17 mm. The average offset value experienced by the end-plates in this work is about 25 mm, which is safely above the worst case boundary layer size. The convergence and divergence settings are established on a trial and error basis. The positions used for the different test cases can be found in appendix E. The adjustment of the leading edge slat angle of attack is made with the help of surface flow visualization. A uniform coating of oil is applied to the slat as shown in figure 4.9a. The

Figure 4.6: New end-plate configuration (side view).

Figure 4.7: Schematic and nomenclature of new end-plates (top view).

Figure 4.8: Endwall boundary layer velocity profile at $+20^\circ$ incidence near slat leading edge plane for $y/S = 1.3$ and -2.0 .

Figure 4.9: Slat surface oil flow visualization for a) initial oil coating, b) incorrect position, and c) correct position.

flow is brought up to speed and the leading edge behaviour is observed. Figure 4.9b is an example of the slat in the incorrect position, an accumulation of oil is detected near the leading edge indicating a misalignment. The angle of attack is increased until the oil is smoothly removed from the entire leading edge with no accumulation (figure 4.9c).

4.3.4 Inlet Measurements

The extension of the slat leading edge upstream resulted in the obstruction of the instrument slot used for the inlet measurements. The problem was solved by using a probe mounting rod with an offset (see top of figure 4.7). A slot was cut out from the far side end-plate (closest to the turntable) to allow for the vertical travel of the new mounting rod. The slot is sealed when not in use. A more accurate measure of the upstream conditions to the blade is one of the benefits resulting from the new inlet measurement plane. This plane is located inside the end-plates passage rather than upstream (outside of the end-plates as in previous works) where the conditions are expected to be different due to the convergence/divergence of the flow.

Measurements in the axial direction were performed with the three-hole probe inside the end-plates passage to examine the flow behaviour, and determine a proper inlet measurement plane. The location of the inlet measurement plane is chosen to be insensitive to the presence of the slats upstream and the blade leading edges downstream. Figure 4.10 shows the variation of the velocity with axial location inside the end-plates passage for the design incidence near the middle blade ($y/S = 0$). The end-plates are positioned in parallel to study the influence of the slat curvature on the axial flow variation. The influence of the blade leading edge can be seen in the reduction of velocity from $x/C_x = 20\%$ to 0% due to the stagnation point. The velocity is increased slightly in the region where the slats converge the flow near the leading edge of the end-plates. The velocity remains constant for the range $x/C_x = 20\%$ to 65% . A value of $x/C_x = 60\%$ is chosen to maximize the distance between the probe tip and the probe holder, and to allow for a vertical traverse with no interference between the blade leading edges and the tail of the probe.

Figure 4.10: Variation of inlet velocity with axial position in the end-plates passage for design incidence, $y/S=0$ and parallel end-plates.

4.4 Test Set-Up

This section describes the layout of the data acquisition system , traverse gear, scanivalve and pressure instruments used in the measurements performed with the three-hole probe and the instrumented blade.

4.4.1 Three-Hole Probe Measurements

A schematic of the layout for the three-hole probe measurements is shown in figure 4.11. The traverse gear is shown in the downstream measurement plane #2. The pressures from the probe's three ports are measured via the scanivalve with the same transducer referenced to the upstream static pressure tap. The total pressure is measured at the reference plane with a pitot tube positioned outside the boundary layer and connected to the second transducer and referenced to the static pressure tap. Previous tests have shown a constant static pressure across the passage; thus this measurement system provides a good approximation of the upstream dynamic pressure. All of the pressure values measured with the three-hole probe are non-dimensionalized by this upstream dynamic pressure.

4.4.2 Blade Static Pressure Measurements

Figure 4.12 shows the schematic of the data acquisition system for the blade static pressure measurements with the instrumented blade. The test set-up is similar to the three-hole probe measurements. The pressures from the instrumented blade are measured with the first transducer via the scanivalve. The blade static pressures are referenced to the upstream static pressure and non-dimensionalized by the upstream dynamic pressure as in the above case. The sampling time required for these measurements is reduced to 15 seconds after a 10 second waiting period, which is still a conservative value. The actual scanning of the blade pressures is divided into two segments, the suction surface and the pressure surface, for reasons mentioned in section 4.1.

Figure 4.11: Schematic of data acquisition system for three-hole probe measurements.

Figure 4.12: Schematic of data acquisition system for blade static pressure measurements.

5 CASCADE TESTING TECHNIQUES

This chapter describes the methods used to measure the aerodynamic parameters of interest. The calibration procedure for the three-hole probe and the hot-wire anemometer is described. The operating point, inlet flow quality and outlet flow periodicity are examined. The data reduction schemes to extrapolate the blade static pressure distribution and the axial velocity ratio are also presented.

5.1 Three-Hole Probe Calibration

The three-hole probe was the main instrumentation used to measure most of the data. Great care was taken to insure the reliability of the results from the probe. The probe was previously calibrated by Tremblay (1989) and Goobie (1989) using a curve fit method to the calibration data which encompassed a wide range of yaw values (-70° to $+70^\circ$). The flows observed in this work usually do not exceed $\pm 5^\circ$, especially when the probe is lined up with the oncoming flow. For this reason, it was decided to recalibrate the probe in a range of $\pm 10^\circ$. The probe was calibrated at a 0.5° interval (compared to the previous 1°) over the range to increase the accuracy. A linear interpolation scheme was adopted instead of the curve fit method to increase the local accuracy of the measurements. Periodic checks on the calibration were also made to verify the validity of the calibration.

5.1.1 Procedure

Both the inlet (short) three-hole probe and the original three-hole probe are calibrated, with the apparatus described in chapter 4, using the same procedure. The velocity of the flow in the calibration apparatus is adjusted to 30 m/s, which is the average value experienced by the probe in the test section. The turbulence intensity of the calibration flow is about 2%. The Reynolds number effect on the probe will be negligible since the operating range of the probe is not expected to exceed $\pm 5^\circ$; this minimizes the possibility of a separation region (which is R_e sensitive) on the probe. The probe is installed in the centre of the jet flow with a holder on the calibration apparatus turntable. The three pressures from the probe are monitored via the scanivalve with one of the transducers referenced to atmospheric pressure. The total pressure of the calibration flow is measured, from a pressure pickup on the side of the blower box, with the second transducer also referenced to atmosphere. Since the calibration flow is open to atmosphere, an accurate measure of the dynamic pressure at the probe can be taken from the total pressure readings. The calibration data acquisition is automatic and is controlled by the microcomputer. The angle measured by the three-hole probe is yaw, not to be confused with the data reduced angle reported by the computer program for a probe traverse in the test section, which is pitch (i.e. opposite sign convention). The yaw angle is used in the calibration of the three-hole probe.

The first step in the calibration procedure is to align the probe with the jet flow direction. Several measurements are taken near the zero yaw mark at a roll angle of 0° . The probe is then rotated to a roll angle of 180° and a second set of measurements is taken. The data is then plotted as the difference in yaw port pressures $(P_2 - P_3)/q$ versus yaw angle β for both roll angles as shown in figure 5.1. The intersection of the two curves reveals the probe misalignment of about $\psi_0 = 0.2^\circ$ with the oncoming flow. The manufacturing defect in the probe head (β_0) can also be estimated at 0.7° . The alignment of the probe is now readjusted and the probe can now be calibrated at 0° roll. It should be noted that the probe is visually aligned in the pitch direction. Results have shown that the probe is not sensitive in this direction (only a 2% drop in dynamic pressure at $\pm 10^\circ$). The probe is then yawed from -10° to $+10^\circ$ at a 0.5° interval. The pressures are recorded at each station versus yaw angle. The mechanical accuracy of the turntable is better than 0.1° with virtually no backlash.

Probe tip at 0° roll angle

ψ_0 = misalignment angle = 0.2°

β_0 = manufacturing defect = 0.7° ($2\beta_0 = 1.4^\circ$)

Figure 5.1: Yaw port pressure differential $(P_2 - P_3)/q$ versus yaw angle β for a roll angle of 0° and 180° .

5.1.2 Calibration Curves

The data from the calibration is then reduced to form coefficients according to the method of Lewis (1966) as follows,

$$K_1 = \frac{P_1 - P_s}{q}$$

$$K_{12} = \frac{P_1 - P_2}{q}$$

$$K_{13} = \frac{P_1 - P_3}{q}$$

$$K_{\phi 2} = \frac{P_1 - P_2}{P_1 - P_3}$$

$$K_{\phi 3} = \frac{P_1 - P_3}{P_1 - P_2}$$

where: P_1 = Pressure port #1
 P_2 = Pressure port #2
 P_3 = Pressure port #3
 and q = Dynamic pressure at probe

These coefficients are graphically presented in figures 5.2 to 5.6 for the original probe. The results for the short inlet probe are similar. The two curves for each coefficient show the repeatability of independent calibrations performed on two separate dates.

Once these calibration curves are determined, the static pressure, dynamic pressure (velocity), total pressure and flow direction can be obtained from the three pressure outputs from the probe (for the probe in the valid dynamic and angular range). The sign of the flow angle is first determined by comparing $P_2 - P_3$ (i.e. positive and negative yaw). The magnitude of the angle is determined from $K_{\phi 2}$ or $K_{\phi 3}$ which can be calculated from the three pressures from the probe. A closer look at figures 5.2 and 5.3 reveals that for positive values of yaw the $K_{\phi 2}$ coefficient is better suited (notice steeper slope) to interpolate the

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PM-1 3½"x4" PHOTOGRAPHIC MICROCOPY TARGET
NBS 1010a ANSI/ISO #2 EQUIVALENT

Figure 5.2: Variation of $K_{\phi 2}$ with yaw angle β .

Figure 5.3: Variation of $K_{\phi 3}$ with yaw angle β .

Figure 5.4: Variation of K_{12} with yaw angle β .

Figure 5.5: Variation of K_{13} with yaw angle β .

Figure 5.6: Variation of K_1 with yaw angle β .

magnitude of the angle. The opposite is true for the $K_{\theta 3}$ coefficient. The dynamic pressure can now be determined using the K_{12} and K_{13} coefficients and the flow angle. The curve in figure 5.4 (K_{12}) appears to be better suited for negative values of yaw where less scatter is encountered, and similarly, K_{13} (figure 5.5) is more suitable for positive values. The value of K_{12} or K_{13} is linearly extrapolated from the calibration data, for the known flow angle, and can now be used to calculate the dynamic pressure as follows (for positive yaw):

$$q_{\text{probe}} = \frac{(P_1 - P_3)_{\text{probe}}}{(K_{13})_{\text{calibration}}}$$

The static pressure can be calculated from the flow angle, the dynamic pressure and the curve of K_1 versus flow angle in figure 5.6. The value of K_1 is extrapolated from this curve from the flow angle. The static pressure is then determined as follows:

$$(P_s)_{\text{probe}} = (P_1)_{\text{probe}} - (K_1)_{\text{calibration}} (q)_{\text{probe}}$$

The total pressure of the flow is the simple addition of the static and dynamic pressure sensed by the probe. The data reduction method as described above was adopted and incorporated in the three-hole probe reduction program included in appendix F.

5.1.3 Accuracy of the Three-Hole Probe Measurements

The following section discusses the accuracy of the measurements taken with the three-hole probe. As mentioned earlier, the three-hole probe is the main instrument used to measure the most important parameters such as inlet and outlet flow conditions, flow periodicity, profile loss, and flow deviation. These parameters are calculated from the pressures and flow direction reported from the probe.

An accuracy analysis of the flow angle and dynamic pressure for the original probe in a uniform flow was done by Tremblay (1989) and Goobie (1989). The investigation was performed on the calibration apparatus as discussed earlier. The flow angle was compared with the reading from the protractor located on the turntable. Errors in yaw of $\pm 0.5^\circ$ on the flow angle were noticed for pitch angles in the range of $\pm 10^\circ$. The error in the measured

dynamic pressure from the probe with respect to $P_o - P_{atm}$ measured from the apparatus was about 1% in the worst case.

A complete analysis of the error on the total pressure loss coefficient due to the non-uniform flow experienced downstream of the blades is included in Tremblay (1989). The shear flow associated with the wake downstream of the trailing edge affects the readings from the three-hole probe. The velocity gradient across the probe tip can be falsely interpreted as a flow angularity. Evidence of this will be presented later with the exit flow measurements. The analysis on the loss was performed for the worst case, at design incidence, where the velocity gradients are the greatest (narrower wake). The error in exit angle was found to be less than $\pm 1^\circ$; however this value would be reduced by the omission of the data points which fall in the wake region. The total pressure measured from P_1 will be insensitive to the wake and the value of K_1 for the yaw angle range of interest will still remain near unity (figure 5.6). The maximum error in dynamic pressure due to the shear flow effect was 3%, while the error reported for the mass average total pressure loss was $\pm 5\%$. Therefore the error due to the shear flow effect appears to be negligible.

It should also be noted that the probe was used over a long period of time without changing its calibration. This can be seen from the repeatability in the calibration data presented earlier. However the probe should be handled with care. Periodic cleaning and calibration checks should be performed.

5.2 Hot-Wire Anemometer Measurements

This section describes the parameters measured with the hot-wire anemometer, the calibration procedure and the measurements performed in the test section.

5.2.1 Turbulence Intensity

An important parameter in wind tunnel testing is the turbulence intensity. It is a measure of the fluctuating changes in velocity. This value, along with the Reynolds number, influences the growth of the boundary layer which, in turn, affects the aerodynamic parameters of interest (e.g. 2-D profile loss). The hot-wire anemometer is used to measure the level of turbulence intensity in the wind tunnel and test section. The turbulence intensity is defined as follows:

$$T_u = \frac{\sqrt{u'^2}}{\bar{u}} \times 100\%$$

where: u' = Fluctuating component of velocity in x-direction
 $\bar{u}$ = Mean velocity in x-direction

The turbulence intensity T_u is measured in the x-direction only. A study by Schlichting (1960) concluded that the nature of the turbulence in a wind tunnel of conventional type, such as the present one, with proper settling chamber, honeycombs, screens and contractions can be expected to be homogeneous and nearly isotropic. Therefore, at a certain distance downstream of the flow screens, the mean square velocity fluctuations (RMS values) in the three coordinate directions are assumed to be equal.

$$u'^2 = v'^2 = w'^2$$

Therefore the measure reported from herein is the turbulence intensity T_u using the average of the x and y component of fluctuations since the single hot-wire is sensitive to these two components.

5.2.2 Calibration

The hot-wire anemometer is calibrated using the apparatus described in chapter 4. The hot-wire design is a simple one. The wire is kept at a constant temperature by applying a voltage across it. The heat exchange between the wire and surrounding air is increased with an increase in velocity. To compensate, the voltage across the wire must also be increased. Therefore a measure of the velocity can be achieved by monitoring the voltage

across the wire. Since the diameter of the wire is small (6 micrometers), it is possible to observe the response of the system to velocity fluctuations at high frequencies.

The first step in the calibration procedure is to determine the probe leg resistance and a suitable probe operating temperature. A high value of temperature is desirable to improve the accuracy of the measurements and to reduce errors from the fluctuations in the flow temperature. But the upper limit of temperature is dictated by the probe manufacturer at 300°C. Therefore an operating temperature of 280°C is chosen. Unfortunately, the relationship between the voltage and the velocity is a non-linear one. The hot-wire must therefore be calibrated over a velocity range. The range is chosen to match the operating conditions experienced in the wind tunnel, 10 m/s to 50 m/s. The hot wire is placed in the core of the calibration jet. The velocity of the jet is calculated from the dynamic pressure of the flow, which is equal to the total pressure measured with the transducer referenced to atmosphere. A micromanometer is also used to verify the pressure values. The hot-wire is subjected to the range of velocities and the corresponding voltages are measured with a voltmeter and recorded. A fourth order polynomial is fitted to the data and the coefficients are entered manually into the signal linearizer. The calibration of the hot-wire is verified and the results show a linear output signal of 0 Volts at 0 m/s to 10 volts at 50 m/s.

5.2.3 Test Section Measurements

The hot-wire anemometer is placed in measurement slot number 1 located about 5 chord lengths upstream of the blade leading edge plane. The average value of velocity was calculated from the DC component of the voltage signal measured with the voltmeter. The fluctuation component was calculated from the RMS value. The frequency spectrum of the RMS value, measured in this work, is that of the voltmeter. The spectral range of frequency which must be applied in the analysis of the turbulence intensity still remains unclear. To investigate the intensity of the fluctuations at different frequencies, a plot of the AC signal component from the hot-wire is shown in figure 5.7 for the 0 to 500Hz range. The bulk of the fluctuations appear to be at lower values of frequency. It is interesting to notice the local increase (spike) in fluctuations at around 230 Hz which is the frequency of the ten blades in the centrifugal blower fan operating at 1380 RPM. The fluctuations for the frequency range from 0 to 10 KHz is shown in figure 5.8. The results show that most of

Figure 5.7: Frequency spectrum of the hot-wire signal (0 - 500Hz).

Figure 5.8: Frequency spectrum of the hot-wire signal (0 - 10KHz).

the fluctuations contributing to the turbulence intensity are well below 10 KHz. One can assume that higher energy fluctuations are not likely to appear above 10 KHz, and that this value is a safe upper limit. A frequency spectrum analysis of the hot-wire signal was also performed by Tremblay (1989) in tunnel #1. The results show a higher level of fluctuations at the low frequencies below 5 KHz which is reflected in the higher turbulence intensity measurement in the tunnel. The fluctuations, however, are still negligible at 10 KHz.

The hot-wire measurements were repeated at a different axial location. The value of turbulence was measured in slot number 2 (141% axial chord upstream of the leading edge). The calibration procedure was also repeated prior to the new measurements. The results of the turbulence intensity will be presented in chapter 6.

5.3 Blade Static Pressure Measurements

The blade static pressure measurements were performed using the instrumented blade as described in chapter 4. The results of the pressures from the blade are referenced to the upstream reference static port P_{ref} and non-dimensionalized by the upstream dynamic pressure q_{ref} . Therefore the reported value is:

$$(C_p)_{measured} = \frac{P_x - (P_s)_{ref}}{q_{ref}}$$

where : P_x = Blade pressure at location x

The reference values used in the above equation are located far upstream of the blade leading edge plane and do not represent the true inlet conditions experienced by the blade between the end-plates. The inlet conditions between the end-plates are known since they are measured during the inlet flow uniformity tests. These values are also referenced to the far upstream conditions P_{ref} and q_{ref} . The proper value of C_p can now be calculated:

$$\begin{aligned}
 C_p &= \frac{P_x - P_1}{q_1} \\
 &= \left(\frac{P_x - P_{ref}}{q_{ref}} - \frac{P_1 - P_{ref}}{q_{ref}} \right) \frac{q_{ref}}{q_1} \\
 &= \left((C_p)_{measured} - \frac{P_1 - P_{ref}}{q_{ref}} \right) \frac{q_{ref}}{q_1}
 \end{aligned}$$

The terms in the latter equation are readily available from the blade static pressure measurement and the inlet flow uniformity data. Repeatability tests made on the blade static pressure measurements have shown a small error in C_p of less than 1%. The data reduction program used in the calculation of the C_p values can be found in Tremblay (1989).

5.4 Flow Quality

The main reason behind the testing of scaled blades in a cascade, is to relate the experimental results to the full scale conditions. The inlet and outlet flow of the blades must be similar in both cases to allow for a good correlation. The flow quality into the cascade must therefore be verified and adjusted to simulate the conditions experienced by the blades in the real engine. The operating point, inlet flow uniformity, exit flow periodicity and the measure of the axial velocity ratio will now be discussed.

5.4.1 Operating Point

The operating point of the wind tunnel is determined in terms of the Reynolds number. The Reynolds number is calculated as follows:

$$Re = \frac{\rho V_{in} C}{\mu}$$

where:

ρ	= Density
V_{in}	= Inlet centreline velocity
C	= Blade chord
μ	= Viscosity

The flow density and viscosity are calculated from the atmospheric pressure and temperature. The inlet centreline velocity is calculated from the dynamic pressure at the reference plane (≈ 5 chord lengths upstream of the blade row). The dynamic pressure is measured with a pitot tube, positioned outside the endwall boundary layer, and referenced to the static pressure tap at that location. A traverse of the flow at the reference plane shows that the flow is uniform with a constant static pressure across the passage. The velocity measured at the reference plane is then related to the inlet conditions, just upstream of the leading edge plane (between the end-plates), using the results from the inlet uniformity measurements.

The Reynolds number used in the tests is 3.0×10^5 ; this value is somewhat larger than the real engine's operating condition. The actual Reynolds number varies from 1.5×10^5 at sea level to 0.7×10^5 for the design altitude. Although the numbers differ, both are above the so called "critical value". The turbulence levels experienced by the real engine are much higher than those in the wind tunnel. The extent of the influence of the Reynolds number and turbulence intensity is discussed in chapter 6.

5.4.2 Inlet Flow Uniformity

The inlet flow uniformity is very important to insure constant angularity and velocity across each blade. Several control surfaces on the test section are used to adjust the flow as shown in figure 5.9. A pitchwise traverse inside the end-plates passage with the three-hole probe is carried out at 60% axial chord upstream of the leading edge plane. The probe is horizontally aligned and positioned to measure pitch. To save time in the coarse adjustment period, the pressure from port #2 and port #3 are connected to the same transducer and the difference in pressures is monitored to estimate the flow angle. The flaps and floating wall are adjusted to give a zero pressure differential (accounting for the zero reading) near the bottom blade and near the top blade. Since the change in one control surface position

Figure 5.9: Schematic and terminology of the control surfaces.

affects the entire flow, the procedure must be repeated until the pitchwise angular variation is small. The next step is a full pitchwise traverse to measure the flow angle and velocity. The control surfaces are then fine-tuned to give the best possible flow. Figure 5.10 and 5.11 show a typical inlet flow angle and inlet velocity profile for the design condition after the adjustment of the control surfaces. The flow angularity at the inlet to the cascade is constant with a variation of about $\pm 0.5^\circ$. The actual angle seen by the blades, however, can vary by $\pm 0.5^\circ$ of the reported value due to the accuracy of the probe alignment. This would show up as a systematic shift of the flow angle curve. The flow uniformity can also be seen in the velocity profile (figure 5.11). The flow near the top of the cascade is more difficult to adjust to uniform conditions because of the presence of only one flap. The bleeding of the air out of the top flap caused a positive flow angle, relative to the midpitch, due to the proximity of the flap to the measurement plane.

The inlet flow uniformity is verified at each angle of incidence. The -20° case requires few iterations from the design conditions to achieve acceptable uniformity. At $+20^\circ$ of incidence, uniformity is only possible with the addition of a perforated plate attached to the tailboards. The perforated plate is used to raise the back pressure downstream of the blade row which, in turn, provides more control to the flaps and floating wall. All of the measurements at $+20^\circ$ are performed with the perforated plate.

A description of the influence of each control surface and the settings used in the present work are presented in appendix E.

5.4.3 Exit Flow Periodicity

The aerodynamic behaviour of the blades is influenced by the presence of the adjacent blades. In the real engine, the blades are located around a circumference. This would require an infinite number of blades to model this in a planar cascade. The size of the cascade is limited to five blades, by the mass flow generated by the blower fan, for the required velocity, aspect ratio and blade spacing. To properly model the real engine case, tailboards downstream of the trailing edges of the extreme blades are used to guide the flow. The positioning of the tailboards is carried out in an iterative manner.

Figure 5.10: Design incidence pitchwise inlet flow angle α_1 variation.

Figure 5.11: Design incidence pitchwise inlet velocity V_1 variation.

A coarse downstream traverse at 40% axial chord downstream of the trailing edge is performed with the three-hole probe. The total pressure and the flow angle is observed and plotted. The position of the tailboards is then fine tuned until periodicity is achieved for the three middle blades. Once periodicity has been achieved and the width of the wake determined, the traverse is repeated with a smaller step size across the wake. The total pressure loss from the wake is calculated for all three blades and compared along with the flow angle values. Typical results for a downstream traverse at design incidence are shown in figure 5.12 and 5.13. The wake size can quickly be examined for the three blades from the total pressure profile. It should be noted that the second blade from the bottom showed a slightly deeper wake than the other two. This was also noticed at the other angles of incidence and could not be explained. Although a slight difference in wake depth was noticed, the flow angle across the three blades was constant ($\pm 0.5^\circ$) in the non-wake region. This would suggest that the flow has good periodicity, even with this localized problem at the lower blade. Only the measurements from the middle blade are used in the calculation of the parameters, once acceptable periodicity has been achieved. Some flow angle distortions were recorded in the wake region due to the velocity gradient across the probe tip. The effect of this can be seen in flow underturning and overturning as shown in figure 5.14. Some underturning is expected from the pressure side of the blade where a positive velocity gradient is present and the reverse is noticed from the suction surface.

The influence of the tailboard settings on the inlet flow uniformity was also examined and no coupling effects were noticed. The initial position of the tailboards must, however, be carefully chosen. A more detailed description of the tailboard influence and the set-up procedure can be found in appendix E.

5.4.4 Axial Velocity Ratio (AVR)

The main parameter being controlled in this work is the axial velocity ratio (AVR). It is a measure of the axial flow acceleration or deceleration across the blade passage as defined in chapter 2. To eliminate the dependency of the results on the measurement plane chosen, the AVR is calculated from the axial velocity values at the trailing edge plane and the leading edge plane. This will also facilitate the comparison of the results to other studies.

Note: α_2 is referenced to blade axial direction (not test section horizontal)

Figure 5.12: Design incidence pitchwise exit flow angle α_2 variation.

$$\text{where; } C_p = \frac{(P_{tot})_2 - (P_{tot})_{ref}}{q_{ref}}$$

Figure 5.13: Design incidence pitchwise exit total pressure profile.

Note: Scale exaggerated for clarity

Figure 5.14: Schematic of wake structure and its influence on three-hole probe angle measurements.

The velocity and flow angle profile, for both the inlet and exit plane are taken from the flow uniformity measurements. The mass average values for the velocity and flow angle, over one pitch (middle blade), are used to calculate the axial velocity at the measurement plane as follows:

$$V_{a1} = \bar{V}_1 \cos \bar{\alpha}_1$$

$$V_{a2} = \bar{V}_2 \cos \bar{\alpha}_2$$

The inlet and exit measurement planes are located inside the flat portion of the end-plates passage (figure 5.15). However, it should be noted that the change in axial velocity for a converging or diverging flow is not linear; other physical mechanism, like mixing, between the trailing edge and the measurement plane can be difficult to include in the calculation. Therefore, a linear interpolation scheme is used to estimate the values from the measurement planes to the leading and trailing edge of the blade. The final value calculated is the axial velocity ratio,

$$AVR = \frac{V_{a t.e.}}{V_{a l.e.}},$$

and is the value reported in this work. The AVR value, however, should be treated with caution since it only represents the average streamtube convergence or divergence at midspan, calculated over one blade pitch. A local convergence of the flow on the suction side may be counterbalanced by a divergence on the pressure surface to give a calculated AVR=1.0. This effect will be smaller for higher aspect ratio cascades. Many cascades, such as the present one, use smaller aspect ratios to maximize the Reynolds number, allow the use of instrumented blades, and facilitate midspan measurements. Therefore the values of AVR should be treated carefully and supplemented with other evidence, such as flow visualization and blade loading measurements.

Figure 5.15: Linear interpolation scheme to calculate the AVR relative to the leading edge (l.e.) and trailing edge (t.e.) plane.

6 PRESENTATION AND DISCUSSION OF RESULTS

Most of the results in this chapter are presented in a graphical form to simplify the analysis. The numerical values for all of the results can be found in the tables located in appendix B.

6.1 Influence of Reynolds Number on Profile Losses

The growth of the boundary layer on a blade surface is affected by the free-stream Reynolds number. The boundary layer growth, in turn, is associated with the two-dimensional loss mechanisms. For this reason, the investigation of the Reynolds number influence on profile loss is important. The effect of favourable and adverse pressure gradients, inherent in turbine blades, results in a widely varying boundary layer state for different Reynolds numbers.

A low Reynolds number will cause the laminar boundary layer to separate in the region of adverse pressure gradient. The flow has difficulty reattaching itself to the suction surface, due to the low y -momentum (transverse) transfer associated with laminar boundary layers. This will result in a large separation bubble and a high profile loss. An increase in the Reynolds number will cause the transition point (laminar to turbulent) to move towards the leading edge of the blade. A turbulent boundary layer provides better mixing in the y -direction (or normal to the blade). This will allow an earlier reattachment (or avoid separation altogether) of the flow and a smaller separation bubble, resulting in a lower profile loss. A reduction in loss will be noticed with an increase in Reynolds number to the point where the flow remains fully attached over the entire suction surface of the

blade. This point is often called the critical Reynolds number. Beyond this point, the Reynolds number has a weaker influence on the profile loss; an increase of Re will only push the transition point closer to the leading edge resulting in a small increase in the size of the downstream boundary layer. There will not be a drastic change in the boundary layer state (i.e. separation bubble) since the flow remains attached. A subsequent rise in the profile loss can occur, for large values of Reynolds number, if a separation of the turbulent boundary layer begins near the trailing edge.

6.1.1 Design Incidence

The bulk of the measurements were made at a Reynolds number of 3.0×10^5 based on the full chord and the inlet velocity. The tunnel turbulence intensity was 0.3%. To verify that these tests were performed above the critical Reynolds number, the total pressure loss at midspan Y_m was measured at different values of Reynolds number at design incidence (figure 6.1). The losses decrease with an increase in Reynolds number up to a value of $\approx 2.8 \times 10^5$, above which the losses seem to level off. Higher values of Reynolds number larger than 3.5×10^5 were not achievable due to the physical limitations of the wind tunnel.

Oil with white paint pigments was used for flow visualization on the suction surface of the blades. Figure 6.2a shows the blade at a Reynolds number of 3.0×10^5 and $AVR \approx 1$. The flow visualization reveals a laminar boundary layer region which is well structured with low shear stress for the first half of the blade. The completion of the transition region is believed to occur at $x/C_x \approx 0.75$; it is usually associated with a region of high shear stress which would remove the oil layer on the blade, and finally a fully turbulent boundary layer is noticed from $x/C_x \approx 0.75$ to 1.0. The case for a $Re = 1.5 \times 10^5$ is presented in figure 6.2b. The flow visualization is not as revealing at the higher Reynolds number case. The flow seems to remain laminar, with a possible laminar separation at $x/C \approx 0.5$ and a separation bubble where little or no movement of the oil is observed. The rear part of the blade shows a possible reattachment of the turbulent flow at $x/C \approx 0.8$ where some movement of the oil is again noticed. These observations are consistent with the results from (Tremblay 1989) who performed similar tests on the same blade section in a different wind tunnel (#1).

Figure 6.1: Influence of Reynolds number on the profile loss Y_m for design and +20° incidence.

a) $Re = 3.0 \times 10^5$ b) $Re = 1.5 \times 10^5$

Figure 6.2: Suction surface flow visualization at design incidence for
a) $Re = 3.0 \times 10^5$ and b) $Re = 1.5 \times 10^5$

It appears from the findings that the operational Reynolds number is slightly above the critical value. Results from Tremblay (1989) show a moderately lower critical Reynolds number of 2.5×10^5 with a turbulence intensity of 1.5%. The Reynolds number experienced by the blade in the real engine is lower than the large scale model blades tested, but the free-stream turbulence intensity is much higher. Therefore the engines critical Reynolds number would be lower, and the losses measured in the tests are a good approximation of the profile loss in the engine.

6.1.2 +20° incidence

The Reynolds number influence on the profile loss was also studied for a high positive value of incidence (+20°). A large increase in the profile loss is noticed. The value of the midspan profile loss versus Reynolds number for the +20° incidence case is also shown in figure 6.1. The results show that an increase in Reynolds number produces a decrease in losses up to the highest achievable value of $Re = 3.5 \times 10^5$. In this case a "critical Reynolds number" cannot be identified, because a steady decrease in losses was noticed with no stabilization. Flow visualization for this condition revealed extensive trailing edge separation. This would explain the relatively high increase in losses and the dependency of these losses on the Reynolds number beyond 3.0×10^5 . The scatter in the data could also be attributed to this trailing edge separation. Tremblay (1989) reported much lower values of losses at +20° and also did not observe any trailing edge separation. He also found that the losses started to slowly level off at $Re = 2.5 \times 10^5$. Because of the high positive incidence, a small separation bubble was noticed near the leading edge on the suction side. This can probably be attributed to the leading edge overspeed followed by an adverse pressure gradient. This leading edge separation bubble is present at the operating $Re = 3.0 \times 10^5$. Therefore the losses would still be somewhat dependent on the Reynolds number for high positive incidences, even without trailing edge separation.

6.2 Turbulence Intensity

Free-stream turbulence intensity has a major effect on the flows that occur in turbomachinery. This is mainly due to its influence on the transition of a laminar boundary layer to a turbulent one. The velocity and pressure fluctuations in the flow are responsible for the onset of transition, which in turn shapes the development and the behaviour of the boundary layer. A high level of fluctuations will cause the boundary layer to reach its transitional stage earlier. Therefore an increase in turbulence intensity will have the same effect on the profile loss as an increase in Reynolds number. An earlier transition point will reduce the separation bubble and reduce the loss until the critical value is reached. Trip wires are often used in these situations to increase the turbulence and bump the Reynolds number above the critical value. The effect of Reynolds number and turbulence intensity are closely related. An increase in either will lead to a higher rate of exchange of momentum between the free-stream and the boundary layer.

The measured value of turbulence intensity in the x-direction, T_u , was 0.3% with the hot-wire anemometer at the measurement plane, 5 chord lengths upstream of the leading edge plane. A value of $T_u=0.3\%$ was recorded at a later date for the location one chord upstream of the leading edge plane. As mentioned earlier, Tremblay (1989) performed tests on the same blade section in a different wind tunnel. The levels of turbulence measured in his tunnel (tunnel #1) were around 1.5%. The profile loss versus Reynolds number for the two cases is presented in figure 6.3. The typical loss curve is seen for both tunnels, although the lower turbulence has shifted the curve to the right. This shift is expected, since an increase in turbulence has the same effect as increasing the Reynolds number, as explained earlier. This analysis would also strengthen the argument relating the experimental losses to those of the real engine running at a higher turbulence intensity, and therefore, a lower critical Reynolds number.

Some work on the effect of the turbulence intensity on loss coefficients has been done by Blight and Howard (1952) for a compressor cascade. Figure 6.4 shows the effect of Reynolds number on the loss coefficient ϵ_1 for a normal wind tunnel and one with

Figure 6.3: Influence of Reynolds number and turbulence intensity on the profile loss Y_m for design incidence.

Figure 6.4: Influence of Reynolds number and turbulence intensity on the loss coefficient ϵ_1 for a compressor cascade, according to Blight and Howard (1952), reproduced from Gostelow (1984).

increased turbulence. The curves are almost identical to the turbine cascade case presented above in figure 6.3. The critical Reynolds number for both the compressor and the turbine seem to be approximately the same, $Re \approx 2.8 \times 10^5$ for the low turbulence tunnels, and a much lower value for the tunnels with higher turbulence.

6.3 Effect of End-Plates

This section deals with the influence of the end-plates on the midspan parameters measured at design incidence. The results from Tremblay (1989) were used to compare the new end-plates configuration to the converging end-plates and the solid parallel endwall conditions (no end-plates).

6.3.1 Profile Loss

The profile loss of a particular blade section is an important parameter used to calculate the performance in turbomachinery. The profile loss was calculated using the results from the three-hole probe traverse 40% axial chord downstream of the trailing edge plane as described in chapter 5. This location was chosen to reduce the shear flow effects on the probe. Since the probe is of finite width, a velocity gradient across it could influence the probe results. A small influence is even noticed at the measurement plane in the exit flow angle, α_2 , as discussed in chapter 5. The losses measured with the probe at this plane are comprised of the profile loss and a fraction of the mixing loss.

To remove the dependency of the losses on the location of the measurement plane, the loss value is adjusted to compensate for the losses which will occur downstream of the measurement plane due to further mixing. Since the mixing losses are unavoidable in cascade testing, they are included with the profile loss. This facilitates the comparison of results between studies. A study by Goobie (1989) showed that the fully mixed-out loss was the same regardless of the measurement plane.

This mixed-out loss is referred to as the mass average total pressure loss coefficient and is the term used in this work. It is defined as follows:

$$\bar{Y} = \frac{1}{\dot{m}} \int_0^s Y \, d\dot{m} + 1 - \frac{2 \int_0^1 v_1^2 d\left(\frac{y}{s}\right)}{\left(\int_0^1 v_1 d\left(\frac{y}{s}\right)\right)^2} + \frac{\int_0^1 v_1^3 d\left(\frac{y}{s}\right)}{\left(\int_0^1 v_1 d\left(\frac{y}{s}\right)\right)^3}$$

The first term is the mass average total pressure loss across a blade spacing, and the last terms are the losses due to the mixing process downstream of the trailing edge. The mixing loss is calculated using the velocity profile from the downstream traverse. The derivation of this expression can be found in Goobie (1989).

The loss Y at midspan for the design incidence and an $AVR=0.998$ at a location 40% axial chord length downstream of the trailing edge is composed of the following:

Y_{measured}	=	0.0208
$Y_{\text{additional mixing}}$	=	+ <u>0.0008</u>
$Y_m \text{ (midspan)}$	=	0.0216

These values were measured with the modified end-plates with leading edge slats. The additional mixing losses to be accounted for at this measurement plane account for only 4% of the total mixed-out loss. This is consistent with the results from Goobie (1989) who also calculated a 4% value for the mixing loss for the same measurement plane.

The value of midspan loss recorded by Tremblay (1989) for the original converging end-plates ($AVR>1$) at design is 0.021. The value reported for the midspan loss with no endplates (i.e. parallel solid end walls, $AVR>1$) is 0.0215. These values agree with the latest results found at $AVR=0.998$. The midspan loss is similar for all three configurations. Although these values seem to indicate that the profile loss is independent of the endwall boundary layer treatment, one must analyze the data with care and investigate the mechanisms which influence the losses. Factors which result from the convergence of the flow due to the endwall conditions, such as the thickening of the blade boundary layer, can

counteract other factors, like the free-stream flow acceleration which will tend to diminish the loss. The relative effect of these factors can change under different testing conditions.

The mass average total pressure loss values will be graphically presented in the following sections along with the measurements for different values of AVR and incidence.

6.3.2 Exit Flow Deviation

The exit flow deviation is another important parameter. This value is used to determine the geometry of the next downstream set of blades in a multistage turbine. The deviation, as mentioned earlier, is the difference between the exit flow angle and the trailing edge metal angle ($\alpha_2 - \beta_2$). The value of the deviation was determined from the downstream three-hole probe traverse. The probe was aligned to measure the pitch angle relative to the trailing edge blade metal angle. The final value reported was the flow exit angle α_2 . A mass average value of the deviation over one blade spacing was calculated in the following manner:

$$\bar{\delta} = \beta_2 - \frac{1}{m} \int_0^s \alpha_2 \, dm$$

At first, the integral was performed over one entire blade spacing, including the wake. As mentioned earlier, the finite width of the probe causes a slight distortion of the flow angle in the wake region. For this reason, the deviation values were recalculated without the values which fall inside the wake region and compared to the full spacing results. Both techniques yielded similar results and the latter calculation scheme was adopted to remove any sources of errors from the wake measurements.

The value of midspan deviation at design incidence for the different endwall configurations is summarized below:

New End-Plates with Slats AVR $\approx$ 1	:	$\delta = -4.9^\circ$
New End-Plates with Slats AVR $>$ 1	:	$\delta = -5.6^\circ$
Original End-Plates (AVR $\approx$ 1.07)	:	$\delta = -5.8^\circ$

The results show a 1° difference in deviation between the new end-plates with slats ($AVR=1$) and the original converging end-plates. This discrepancy can be attributed to the difference in the AVR and its effect on the flow behaviour. Further evidence of this will be presented in section 6.4.

6.3.3 Spanwise Profile Loss

The measure of the AVR is a good start in determining the two-dimensionality of the flow, but other tests must be performed in parallel to verify it. One such test is the measure of the losses in the spanwise direction. Secondary flows create regions of higher losses due to the higher turbulence and the unstructured flow patterns. Theoretically, two-dimensional flow would yield the same value of losses at any spanwise location, but practically this is still not achievable because of the presence of endwalls (porous or non-porous). However the degree of two-dimensionality can be enhanced by a relatively wide area of constant loss which is sometimes referred to as a loss "bucket".

The profile loss was measured at different spanwise locations and is presented in figure 6.5. The curves for the new end-plates, original end-plates and no end-plates condition reveal slightly different loss buckets. The addition of end-plates to control the endwall boundary layer is shown to work. The constant loss area is increased from about 20% to 40% of the span. It should be noted that the span H is the width of the passage in which the controlled flow occurs. It is the entire width of the cascade for the no end-plates condition (200mm) and is reduced by about 20% to ~ 160 mm in the cases with the end-plates. The new end-plates used to control the AVR seem to have slightly decreased the size of the loss bucket to just below 35% of the span. The extent of this region can also be observed from the oil flow visualization in figure 6.2a. The slight decrease can probably be explained by the fact that the new endplates have been extended by 50% axial chord length upstream to accommodate the addition of the knife-edge slats. This increased length in the endplates would thicken the boundary layer at the measurement plane, resulting in larger secondary flows. The difference in the flow convergence between the original endplates ($AVR>1$) and the new plates ($AVR=1$) can also influence the loss measurements at different spanwise locations. This will be shown in the following section.

Figure 6.5: Spanwise profile losses Y at design incidence for various end-plate configurations.

Although the area of constant loss varies for each case, it is interesting to notice that the midspan loss value remains fixed. One would expect the midspan loss to be influenced to a greater extent by a smaller bucket rather than a larger one. The presence of the nearby secondary flow in the smaller bucket could influence the midspan loss. The lack of difference in the midspan values can probably be attributed to the difference in flow acceleration and its effect on boundary layer behaviour. This acceleration would show up in the AVR value. That is why it is important to consider both the spanwise loss bucket and the AVR to determine the validity of the two-dimensional losses.

6.3.4 Blade Static Pressure Distribution

The development of the boundary layer is intimately related to the behaviour of the free-stream flow around the blade. The measure of the static pressure distribution (blade loading) on the blade can be used to analyze the response of a particular blade section under different operating conditions. These distributions are often used to shape or modify the profile of the airfoil. The pressure gradients are closely studied since they influence greatly the boundary layer and hence the profile loss.

The pressure coefficient C_p is used in a non-dimensional form and is defined as follows:

$$C_p = \frac{P_x - P_1}{q_1}$$

where: C_p = pressure coefficient
 P_x = Pressure at location x/C
 P_1 = Pressure at inlet
 q_1 = Dynamic pressure at inlet

This equation can be easily related to the flow acceleration through the passage by the use of continuity and Bernoulli's equation:

$$\frac{V_x}{V_1} = \frac{\cos\alpha_1}{\cos\alpha_x}$$

and

$$C_p = 1 - \left(\frac{V_x}{V_1}\right)^2$$

Figure 6.6: Blade static pressure distribution at design incidence for the original and new end-plates configurations.

This equation implies that a divergence in the flow would strengthen the adverse pressure gradient. The blade static pressure distribution was measured using the instrumented blade as described in section 5. The midspan blade loading for the original converging end-plates and the new end-plates at design incidence is shown in figure 6.6. Although the shapes are similar, the curve for the new end-plates ($AVR=0.998$) is shifted down to higher values of C_p (lower negative numbers). This phenomena could probably be explained by the difference in the AVR between the two cases. The above equations, relating the flow acceleration to the C_p value, show that an acceleration in the passage will decrease the value of C_p , or make it more negative, hence shifting the curve up for the converging end-plates. Another plausible explanation for the discrepancy, is the difference in the method used by Tremblay (1989) to relate the measured static pressure on the blade to the inlet conditions. The inlet conditions used by Tremblay were measured at 141% axial chord upstream of the blade leading edge and fell outside the passage of the endplates. The new end-plates were modified to allow the three-hole probe to be traversed inside the passage of the end-plates at a location 60% axial chord upstream of the blade leading edge. The two cases are compared in figure 6.7. The static and dynamic pressure are not expected to be the same in both cases. The measurements taken outside the end-plates should register a slightly lower dynamic pressure q_1 and possibly a higher static pressure P_1 than in the region of flow convergence. This difference in inlet conditions would lower the C_p values as defined above (or push the blade loading curve upwards). The influence of the AVR and incidence on the blade static pressure distribution will be presented and discussed in further detail in the following sections.

6.4 Effect of the Axial Velocity Ratio at Design

The main design objective of the new end-plates was to allow for the axial velocity ratio to be varied without drastically changing the boundary layer development on the end-plates. This flexibility provided the necessary tools to study the effect of the flow acceleration (or

Original converging end-plates with inlet probe location

New end-plates with inlet probe location

Figure 6.7: Inlet measurement planes for the original and new end-plates configuration

deceleration) on the two-dimensionality of the flow. The influence of the AVR on different aerodynamic parameters, such as profile loss, exit flow deviation, spanwise distribution loss and deviation, base pressure measurement, and blade loading will now be presented at the design incidence.

6.4.1 Profile Loss

The profile loss reported (from herein) is the fully mixed out losses as described in the earlier section. Figure 6.8 shows the effect of the AVR on the profile loss at midspan for the design incidence. The curve demonstrates that the profile loss seems to be more sensitive to a lower AVR (flow divergence) than to a higher AVR (flow convergence). The loss remains relatively constant at $Y_m=0.021$ from AVR=1 to 1.08, but increases rapidly when the AVR is reduced below unity. These observations were also noticed for a cascade of compressor blades. For example, in chapter 2, Fottner (1972), Iiyas and Norbury (1973), and Starken et al. (1975) all reported the same trends in profile loss. The values of loss are even similar at $Y=0.02$ to 0.03 with a level region of loss from AVR=1 to 1.05. The highest and lowest achievable AVR with the new end-plates was about 1.08 and 0.96 respectively at design incidence. The limitation was imposed by the tolerance of the blade hole cutouts in the end-plates. These holes were kept as small as possible and sealed to prevent flow leakage from the outer flow back into the end-plates passage. Flow visualization of the end-plates without seals, from Tremblay (1989), revealed some leakage of the flow into the controlled passage. The upper limit of the AVR imposed by the end-plates prevented the investigation of the losses past AVR=1.08. One would expect the profile loss to rise again at a higher AVR value since a strong convergence would affect the boundary layer. This rise in profile loss for high convergence was noticed for the compressor cascades reviewed in chapter 2.

The value of profile loss reported by Tremblay (1989) $Y_m=0.021$ was achieved for a quoted value of $V_{a2}/V_{a1}=1.15$ (probably not $V_{a_{10}}/V_{a_{1e}}$). This contraction coefficient (V_{a2}/V_{a1}) cannot be directly related to the AVR as measured in this work. The AVR calculations, as described in chapter 5, relate the mass average axial velocity at the trailing edge to the mass average axial velocity at the leading edge of the blades. These values are calculated with a linear interpolation from the measurement planes inside the end-plates

Figure 6.8: The effect of AVR on midspan profile loss Y_m at design incidence.

passage. The contraction coefficient from Tremblay used the axial component of velocity at the inlet and exit measurement plane. Although the outlet axial velocity V_{a2} would be similar to the present value measured, the inlet V_{a1} would be relatively lower due to the location of the measurement plane outside of the end-plates passage as described in the previous section. This would have the effect of increasing the value (V_{a2}/V_{a1}) relative to the AVR as defined in this work. An estimate of the AVR for Tremblay's results, based on his raw data, would be approximately 1.07. This is consistent with the profile loss curve of figure 6.8. The $Y_m=0.021$ and $AVR=1.07$ would fall directly on the curve. This AVR value is also compatible with the physical limitations of convergence achievable with the end-plates (i.e. same blade hole cutout for both end-plates).

6.4.2 Exit Flow Deviation

The mass average exit flow deviation at midspan was recorded for different AVR values at design incidence. The results are graphically presented in figure 6.9. The flow deviation from the blade metal exit angle increases linearly with an increase in the AVR. Testing of a turbine cascade with a typical value of $AVR=1.05$ in solid parallel end-walls could vary the exit angle by as much as 1° when compared to the 2-D conditions ($AVR=1$) for the design incidence. Although the profile loss showed some tolerance to a small convergence, the deviation angle did not.

This linearly increasing deviation trend is opposite to that for compressor cascades as shown in chapter 2 (table 2.1). All the cases reported for compressor cascades showed a linear (or close to linear) decrease in deviation. The discrepancy in the two blades can probably be explained by the differences in the trailing edge blade loading. The flow turning at the trailing edge is somewhat governed by the pressure differential between the suction surface and the pressure surface. The flow will leave the trailing edge at a angle dictated by conditions which will lead to a pressure equilibrium downstream. The turbine blade loading at the trailing edge seems to behave differently under the influence of the AVR when compared to the compressor blade. An increase in the AVR seems to slightly increase the pressure differential across the trailing edge, while the opposite is noticed (to a lesser extent) for the compressor blades.

Figure 6.9: The effect of AVR on midspan exit flow deviation δ at design incidence.

6.4.3 Spanwise Profile Loss and Deviation

The profile loss and flow exit deviation were measured in the spanwise direction at three different values of AVR, AVR=0.96 (flow diverging), AVR=0.998 ("two-dimensional"), and AVR=1.08 (flow converging). Both parameters are presented together graphically, for each AVR value, in figure 6.10, figure 6.11 and figure 6.12 respectively. The secondary flow from the endwalls should influence both the profile loss and the flow turning. This can clearly be seen for all three cases. The shape of the loss bucket is related to the shape of the deviation curve. Both the profile loss and the deviation have the same width of constant values for all AVR values. The region of secondary flow where the losses start to rise are matched by an underturning of the flow. This underturning is the result of the 3-D stagnation point vortices that are shed by the interaction of the endwall boundary layer and the leading edge of the blade. These vortices propagate downstream through the blade passage and cause a region of underturning near the interface of the 2-D region and the 3-D region, and also drastically increase the measured losses. An endwall flow model by Klein (1966) and Langston (1977) shows the mechanisms involved in the development of these vortices (figure 6.13).

As seen from the previous figures, the width of the constant profile loss and deviation region varies with the AVR. The spanwise profile loss for the three AVR values is shown again in figure 6.14 to facilitate the comparison. An AVR of 0.96 yields a 2-D region of about 30%. This region is increased to about 35% when the AVR is raised to near unity. Finally the "bucket" is increased to almost 40% when the AVR=1.08. It is interesting to notice that for the AVR=1 case, the deviation measured seems to lead to underturning of the flow (increased deviation) near the 2-D to 3-D interface at around $z/H=0.25$ and 0.75 and a subsequent reduction in deviation at $z/H=0.1$ and 0.9 . This would strengthen the argument of the stagnation point vortices, since the underturning would not be as severe in the eye of the vortex as compared to the edge.

These results seem to demonstrate that the spanwise deviation can be used in conjunction with the spanwise profile loss to determine the extent of the two-dimensional region for a cascade of turbine blades. These tests could be performed in conjunction with the measure of the AVR to determine the validity of the two-dimensional measurements.

Figure 6.10: Spanwise profile loss Y and exit flow deviation δ for $AVR=0.96$ at design incidence.

Figure 6.11: Spanwise profile loss Y and exit flow deviation δ for $AVR=1.0$ at design incidence.

Figure 6.12: Spanwise profile loss Y and exit flow deviation δ for $AVR=1.08$ at design incidence.

Figure 6.13: Endwall flow models by Klein and Langston, reproduced from Sieverding (1985).

Figure 6.14: Spanwise profile loss Y for AVR= 0.96, 1.0 and 1.08 at design incidence.

6.4.4 Base Pressure Measurements

The base pressure measurement is used to estimate the extent of the losses due to the finite thickness of the blade trailing edge. The base pressure is usually at a lower value than the average pressure in the flow passage. This pressure gradient leads to additional mixing losses. The base pressure itself is measured from the static pressure tap on the trailing edge of the instrumented blade (tap #43). The base pressure measurement is calculated in the following manner:

$$C_{P_b} = \frac{P_b - P_2}{q_2}$$

where:

C_{P_b}	= Coefficient of base pressure
P_b	= Trailing edge base pressure (tap #43)
P_2	= Mass average exit pressure
q_2	= Mass average exit dynamic pressure

The value of the base pressure is calculated from the blade static pressure distribution trailing edge value and the downstream three-hole probe traverse.

The effect of AVR on base pressure is shown in figure 6.15. The results show an increase in the base pressure to near $C_{P_b}=0$ for converging flow (AVR=1.04 and 1.08). The values remain relatively constant at $C_{P_b}=-0.05$ from AVR=0.96 to 1.01 with a small amount of scatter. This lower base pressure for the flow divergence could be one of the mechanisms increasing the profile loss in this AVR range. Tremblay (1989) reported a $C_{P_b}=-0.0153$, which would be consistent with the results shown for the design incidence and an AVR of ≈ 1.07 . Although the results for the base pressure C_{P_b} provide a relative relationship to the trailing edge losses for a variation in AVR, they do not provide the absolute value in terms of trailing edge loss. The decomposition of the total pressure loss into the boundary layer loss, the mixing loss, and the base pressure loss, is included in Tremblay (1989). These values were calculated using a boundary layer program. The results showed that the base pressure loss accounted for about 5% of the total pressure loss measured at design incidence, but was increased to 14% at a high positive incidence.

Figure 6.15: The effect of AVR on the base pressure coefficient C_{pb} at design incidence.

6.4.5 Blade Static Pressure Distribution

The blade static pressure distribution was measured for several AVR values at design incidence. To facilitate the analysis of the results, the blade loading for values of $AVR \leq 1$ are plotted in figure 6.16, and the blade loading for values of $AVR \geq 1$ are shown in figure 6.17. The static pressure distribution for the blade shows a smooth acceleration (favourable pressure gradient) of the flow on the suction surface from the leading edge to about $x/C=45\%$ followed by a smooth deceleration (adverse pressure gradient) to the trailing edge. The pressure surface of the blade reveals a stagnation point near the leading edge followed by a smooth decrease in pressure with a small acceleration region near the trailing edge.

The influence of a diverging flow, with AVR lower than unity seems to be very small. The curve is shifted slightly upwards to lower C_p values for the suction side of the blade with an increase in the AVR. The pressure side of the blade, on the other hand, shows virtually no movement. All of the curves are similar in shape except for the $AVR=0.96$. The lower AVR value seems to have slightly shifted the blade loading towards the leading edge. This shift is probably the result of one of the mechanisms influencing the rapid loss increase at lower values of AVR. The blade loading in figure 6.17 shows a greater sensitivity to AVR values above unity. A flow convergence leads to a relatively high decrease in the C_p values resulting in an upward shift of the curves. The shape of the curves (uniform loading) remains the same for all values of $AVR > 1$.

The blade loading distributions presented here do not appear to explain the previously discussed variations in losses. Denton and Cumpsty (1987), investigated the loss mechanisms in turbomachinery and reported that the loss production in the boundary layer should vary as the cube of the local free-stream velocity (approximately). This would be the opposite of what was observed and measured. The blade loading revealed a higher free-stream velocity on the suction surface at higher AVR values. In turn, a larger free-stream velocity would lead to a larger skin friction loss; one would therefore expect an increase in losses for an increase in the AVR. However figure 6.8 shows that the losses are

Figure 6.16: Midspan blade static pressure distribution at design incidence for AVR = 0.967 to 0.998.

Figure 6.17: Midspan blade static pressure distribution at design incidence for AVR = 0.998 to 1.078.

reduced with an increase in AVR. The explanation for this discrepancy appears to lie in the effect of the AVR on the base pressure losses as described in the previous sub-section.

6.5 Off-Design Measurements

The normal operation of an engine requires the turbine blades to perform under different conditions, e.g. idle, cruise and take-off power. These different regimes lead to a variation of the inlet flow angle to the blade relative to the designed value. Investigations on cascades of blades are often done to study the off-design performance. Tremblay (1989) investigated, in depth, the off-design behaviour of the blade section used in this work. For these reasons, the validity of the two-dimensional measurements is investigated at different values of incidence. The values of -20° and $+20^\circ$ of incidence were chosen to be compared with the performance of the blade at design incidence. These values were felt to be a broad enough range to encompass most of the values of incidence tested with cascades. The influence of the incidence on the different aerodynamic parameters, such as profile loss, exit flow deviation, spanwise distribution of loss and deviation, base pressure measurement, and blade loading will now be presented and compared.

6.5.1 Profile Loss

6.5.1.1 -20° Incidence

The midspan profile loss Y_m as a function of AVR for an incidence of -20° is shown in figure 6.18. The results show a slight linear decrease in the loss with an increase in the AVR. The loss measured is similar to the design conditions. Flow visualization and blade loading measurements showed that the boundary layer remained attached for both cases. Therefore, one would expect the loss values to be similar. On the other hand, the -20° incidence case shows only a modest increase in profile loss for a diverging flow, unlike the design incidence case.

Figure 6.18: The effect of AVR on midspan profile loss Y_m at -20° incidence.

6.5.1.2 +20° Incidence

Figure 6.19 presents the midspan profile loss Y_m as a function of AVR for an incidence of +20°. As mentioned earlier, the losses measured at +20° were considerably higher than expected. Flow visualization, the high values of losses and the deviation measurements all showed that a trailing edge separation was present. Figure 6.20a shows the suction surface flow visualization using oil dots. The dots are placed on the surface and are subjected to the local shear stress values resulting in vectors describing the direction and intensity of the skin friction. The extent of the trailing edge separation can clearly be seen. The oil dots are entrained from the leading edge to about $x/C=0.5$, where the oil is accumulated (i.e. no more shear stress). From $x/C=0.5$ to the trailing edge the dots are stationary, which would indicate a region with no shear stress or a trailing edge separation. Flow visualization with a regular layer of oil was also used (figure 6.20b) and the same phenomenon was observed. This trailing edge separation would indicate that the flow mechanisms involved would be somewhat different than those experienced by the two attached cases (-20° and design incidence).

6.5.1.3 Effect of Incidence

The effect of AVR on the profile loss for the above angles of incidence is included in figure 6.21 for the ease of comparison to the design conditions. The midspan profile loss for the three incidences is shown in figure 6.22 and compared to the off-design results from Tremblay (1989). The profile loss for the new end-plates configuration is shown at two values, $AVR \approx 1$ (2-D case) and at a value of $AVR > 1$ to facilitate the comparison with Tremblay's results at an $AVR \approx 1.07$. The results for the attached cases are in good agreement. The losses measured at +20° were four times greater than the original end-plates in tunnel # 1. This is believed to be the result of the trailing edge separation due to the differences in turbulence intensity in the two wind tunnels. The steep slope in losses at +25° incidence from the original end-plates appears to indicate that the blade is on the verge of experiencing high losses (edge of the bucket) at +20°, and that a small change in turbulence, Reynolds number, or incidence could increase the losses rapidly. The differences in the two wind tunnel flows, as revealed earlier, could be the cause of this increase.

Figure 6.19: The effect of AVR on midspan profile loss Y_m at +20° incidence.

a) Dot oil flow visualization at $+20^\circ$ incidence

b) Layer oil flow visualization at $+20^\circ$ incidence

Figure 6.20: Suction surface flow visualization at $+20^\circ$ incidence for
a) dot technique and b) layer technique.

Figure 6.21: The effect of AVR on midspan profile loss Y_m at -20° , design, and $+20^\circ$ incidence, reproduced from Rodger, Sjolander and Moustapha (1992).

Figure 6.22: Variation of midspan profile loss Y_m with incidence for the new end-plates and the original end-plates.

6.5.2 Exit Flow Deviation

6.5.2.1 -20° and +20° Incidence

The midspan exit flow deviation at -20° incidence is presented in figure 6.23 as a function of the AVR. The data reveals a small increase in the deviation with an increase in the AVR. The value of the deviation and its behaviour with the AVR is similar to the design case as expected. The results for the +20° incidence show, in figure 6.24, a large increase in the value of the deviation which is more than likely due to the trailing edge separation. A separated flow near the trailing edge would definitely be underturned resulting in a large deviation from the blade exit metal angle. The highly chaotic flow associated with such separation would also explain the scatter present in the data.

6.5.2.2 Effect of Incidence

The midspan deviation values for the three angles of incidence are shown in figure 6.25. The data includes the deviation for the new-end-plates at $AVR \approx 1$, $AVR > 1$, and the original end-plates. The off-design data shows some scatter between the measured values for $AVR > 1$ and the results from Tremblay (1989). The difference at the +20° incidence could probably be attributed to the trailing edge separation. The difference of about 2° in the flow deviation between the new end-plates $AVR > 1$ and the original end-plates, at -20° incidence, cannot be explained. It is interesting to note that the discrepancy is less for the $AVR \approx 1$ condition.

6.5.3 Spanwise Profile Loss and Deviation

6.5.3.1 -20° and +20° Incidence

A spanwise investigation was also performed at off-design conditions. The extent of the two-dimensional region can be seen in figure 6.26 for -20° incidence and figure 6.27 for +20° incidence. Once again the profile loss and deviation measured have the same shape and both show the same location of the 2-D to 3-D region interface, which reinforces the

Figure 6.23: The effect of AVR on midspan exit flow deviation δ at -20° incidence.

Figure 6.24: The effect of AVR on midspan exit flow deviation δ at $+20^\circ$ incidence.

Figure 6.25: Variation of the midspan deviation with incidence for the new end-plates and the original end-plates.

Figure 6.26: Spanwise profile loss Y and exit flow deviation δ at -20° incidence.

Figure 6.27: Spanwise profile loss Y and exit flow deviation δ at $+20^\circ$ incidence.

stagnation point vortex theory. Once more, the measure of the spanwise losses and deviation are closely related.

6.5.3.2 Effect of Incidence

The width of the profile loss bucket is compared with the design case in figure 6.28 for the operating conditions closest to $AVR=1$. The -20° case appears to have a slightly larger 2-D region than the design incidence. The width of the 2-D region has shrunk to slightly less than 30% of the effective span for the $+20^\circ$ incidence. This is also evident from the oil flow visualization in figure 6.20b. The results show that an adequate region of constant losses is obtained, even for the separated flow at $+20^\circ$ incidence. Although the spanwise investigation is informative, it cannot be used independently of other tests to confirm the validity of a two-dimensional measurement. It can be used, however, to determine if the midspan flow is three-dimensional (i.e. if no constant loss region is found).

6.5.4 Base Pressure Measurements

The base pressure measurements for the off-design conditions are shown in figure 6.29. They are compared to the base pressure measurements from Tremblay (original end-plates). The results show good agreement for the attached flow cases (0° and -20° incidence). The results for the $+20^\circ$ incidence are somewhat scattered. One would expect the base pressure to change considerably under trailing edge separation conditions. The base pressure for the new end-plates with the higher value of AVR is, however, close to the value found with the original end-plates. Some of the discrepancy can probably be attributed to the difference in the AVR as discussed in the previous section. Figure 6.30 presents C_F , as a function of AVR for the three values of incidence. The C_{pb} increases with the AVR for both the -20° and design incidence, but decreases at $+20^\circ$ incidence. As discussed previously, the base pressure is directly related to the trailing edge component of the midspan losses. An increase in C_{pb} would decrease the component of loss from the trailing edge thickness. Thus, it appears that the increase in the C_{pb} results in the reduction of the measured midspan losses for an increase in the AVR . This effect appears to be large enough to offset the increase in loss production in the blade boundary layer from the higher free-stream velocity due to a larger AVR value. The result for the profile loss at $+20^\circ$ incidence would,

Figure 6.28: Spanwise profile loss Y at an $AVR \approx 1$ for -20° , design and $+20^\circ$ incidence.

Figure 6.29: Variation of midspan base pressure C_{pb} with incidence for the new end-plates and the original end-plates.

Figure 6.30: The effect of AVR on midspan base pressure C_{p_b} at -20° , design, and $+20^\circ$ incidence, reproduced from Rodger, Sjolander and Moustapha (1992).

however, be dominated by the losses from the trailing edge separation and the mixing associated with it, rather than the conventional boundary layer loss and the trailing edge thickness loss.

6.5.5 Blade Static Pressure Distribution

The blade static pressure distributions for -20° incidence are presented in figure 6.31. Three curves are plotted for the -20° case. The curves relate to a diverging flow ($AVR=0.95$), a close approximation of a 2-D flow ($AVR=0.986$) and a converging flow ($AVR=1.05$) respectively. The shapes of the curves for the blade loading are similar to the design conditions with some exceptions. This is expected since both cases (-20° and design incidence) are operating with an attached boundary layer. Some of the differences noticed between the two cases include a shift of the stagnation point near the leading edge, steeper pressure gradients for the -20° case, and a large decrease in C_p values on the suction surface. The decrease in C_p values for an increase in AVR is the same trend as observed for the design incidence.

The blade loading distribution for $+20^\circ$ of incidence (figure 6.32) shows a vastly different picture than that of the previous cases. Since a value of AVR below unity was not achievable at $+20^\circ$ incidence, only the $AVR=1.01$ and $AVR=1.09$ cases were examined. The flow on the suction surface of the blade starts with an overspeed region near the leading edge, quickly followed by a steep adverse pressure gradient; all of this occurs within $x/C=0.1$. The data shows noticeably less pressure recovery on the rearward part of the blade $x/C>0.5$ than for the two attached flow cases. This would suggest the presence of a large separated zone leading to the trailing edge. The influence of the AVR on the blade loading is also considerably less. This is especially noticeable near the suction surface trailing edge. This difference could account for the relatively small change in deviation, with respect to AVR, noticed at this incidence. The blade loading for the $+20^\circ$ incidence appears to confirm the trailing edge separation observed from the suction surface oil flow visualization.

Figure 6.31: Midspan blade static pressure distribution at -20° incidence for AVR = 0.953, 0.986 and 1.047.

Figure 6.32: Midspan blade static pressure distribution at $+20^\circ$ incidence for $AVR = 1.008$ and 1.087 .

7 CONCLUSIONS

The literature survey demonstrated the lack of data on the influence of flow convergence and divergence on turbine cascades. The flow convergence and divergence can be controlled with the proper treatment of the endwall boundary layer and can be measured using the axial velocity ratio (AVR). The main conclusions from this work may be summarized as follows:

- 1- The flow quality from the new wind tunnel is very uniform with low levels of turbulence.
- 2- A good flow periodicity is achieved over the three middle blades with the use of the control surfaces.
- 3- The end-plates arrangement used in this work is an effective way of controlling the two-dimensionality of the flow in cascade tests and is more convenient than other measures, such as endwall boundary layer suction.
- 4- Flow convergence results in a modest reduction in losses compared to the two-dimensional value for the 0° and -20° incidence case, while a flow divergence is more sensitive resulting in a steeper rise in losses. These results are similar to the compressor cascade case.
- 5- Trailing edge separation, at $+20^\circ$ incidence, was observed with a drastic increase in the losses compared to the attached cases.

- 6- The flow deviation is affected by both the convergence and divergence of the flow. An increase in the AVR resulted in an increase in the deviation, which is opposite to the trend reported for compressor cascades.
- 7- Spanwise loss distribution can be used to identify severe three-dimensional flows, but cannot be used, on its own, to confirm two-dimensional flow.
- 8- The blade loading is influenced by the flow convergence. An increase in the AVR results in a reduction of the pressure coefficient to lower values.
- 9- The changes in the losses are apparently due to the increase in the base pressure coefficient with increasing AVR. A lower base pressure results in higher losses.

Hence, the results show that a small amount of convergence, which is usually present in cascade measurements, can be tolerated for loss measurements but not for deviation values. Therefore the values quoted in the literature should be treated with care, since most studies do not report the degree of convergence present in their tunnel.

The extent of the influence of the base pressure on the measured losses was also unexpected and can probably represent more severe repercussions for the transonic conditions where the base pressure losses are increasingly important.

8 RECOMMENDATIONS

Several recommendations are presented for the improvement of the experimental test section, testing technique and future work. They are as follows:

- 1- An additional control surface on the top portion of the test section would help to establish a better uniform inlet flow on all blades.
- 2- Static pressure taps could be installed on the end-plates to monitor true inlet and outlet conditions. These could also be used in the adjustments of the control surfaces and as a reference for the blade static pressure measurements.
- 3- A rack of pitot tubes connected to the scanivalve could be used to quickly determine the periodicity of the flow.
- 4- The range of AVR investigated in this work could be increased. It would be interesting to see if the losses would start to rise again at higher values of convergence.
- 5- The influence of the base pressure should be investigated in a closer manner. The relative value of the base pressure loss to the total measured loss should be estimated at different values of AVR.
- 6- The effect of the AVR on the base pressure and trailing edge losses could be investigated at transonic conditions.

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Appendix A

Blade and Cascade Geometry

Inlet flow angle α_1	28.35°
Exit flow angle α_2	57.65°
Inlet metal angle β_1	29.29°
Exit metal angle β_2	57.79°
Design incidence i_{dm}	-0.94°
Design deviation δ_{dm}	-0.14°
Stagger angle γ	23.06°
Camber angle θ	87.08°
Chord length C	162.81 mm
Axial Chord length C_x	150.00 mm
Pitch length (spacing) S	110.71 mm
Blade span H	200.00 mm
Pitch to chord ratio S/C	0.68
Aspect ratio (AR) H/C	1.23
Leading edge diameter	9.43 mm
Wedge angle (l.e.)	52.4°
Trailing edge thickness	4.20 mm
Maximum thickness	29.63 mm
t_{max}/C	0.182

Appendix B

Tables of Results

Table B.1a: Design Incidence, variation of Y_m , δ and C_{pb} with AVR

Table B.1b: Design Incidence, spanwise variation of Y and δ at AVR=1.0

Table B.1c: Design Incidence, spanwise variation of Y and δ at AVR=0.96

Table B.1d: Design Incidence, spanwise variation of Y and δ at AVR=1.08

Table B.1e: Design Incidence, spanwise variation of Y for the original and no end-plates

Table B.2a: -20° Incidence, variation of Y_m , δ and C_{pb} with AVR

Table B.2b: -20° Incidence, spanwise variation of Y and δ at AVR=0.99

Table B.3a: +20° Incidence, variation of Y_m , δ and C_{pb} with AVR

Table B.3b: +20° Incidence, spanwise variation of Y and δ at AVR=1.01

Table B.4a: Incidence comparison of Y , δ and C_{pb} for the original and new end-plates

Table B.5a: Design Incidence, variation of Y_m with Reynolds' number for the original and new end-plates

Table B.5b: +20° Incidence, variation of Y_m with Reynolds' number for the original and new end-plates

Design Incidence

Table B.1a: Design Incidence, variation of Y_m , δ and C_{pb} with AVR

AVR	Y_m	δ	C_{pbm}
0.9674	0.0305	-4.3°	-0.07001
0.977	0.0264	-4.5°	-0.06272
0.9866	0.0237	-4.6°	-0.04158
0.998	0.0216	-4.8°	-0.06736
1.01	0.0214	-4.9°	-0.05071
1.046	0.0191	-5.1°	-0.00778
1.078	0.0196	-5.6°	0.036622

Table B.1b: Design Incidence, spanwise variation of Y and δ at AVR=1.0

z/H	Y	δ
0.132	0.0687	-4.7°
0.232	0.0715	-5.6°
0.3	0.0297	-4.3°
0.366	0.0244	-4.6°
0.5	0.0229	-4.4°
0.634	0.0275	-4.0°
0.7	0.0376	-4.4°
0.768	0.0747	-6.1°
0.868	0.0653	-4.5°

Design Incidence

Table B.1c: Design Incidence, spanwise variation of Y and δ at $AVR=0.96$

z/H	Y	δ
0.232	0.0959	-6.2°
0.3	0.1049	-6.0°
0.366	0.04	-4.3°
0.5	0.0342	-3.8°
0.634	0.0364	-4.2°
0.7	0.0907	-5.6°
0.768	0.0859	-6.5°

Table B.1d: Design Incidence, spanwise variation of Y and δ at $AVR=1.08$

z/H	Y	δ
0.165	0.054	-6.9°
0.256	0.031	-6.2°
0.337	0.0202	-5.8°
0.5	0.0196	-5.5°
0.663	0.0198	-5.7°
0.744	0.0341	-5.9°
0.835	0.05	-7.5°

Design Incidence

Table B.1e: Design Incidence, spanwise variation of Y for the original and no end-plates

Original End-Plates, Y_{original}
No End-Plates, $Y_{\text{No E-P}}$

z/H	Y_{original}	$Y_{\text{No E-P}}$
0.165	0.069	
0.3	0.025	0.0415
0.4		0.024
0.5	0.0215	0.0215
0.6		0.026
0.7	0.0255	0.052
0.835	0.058	

-20° Incidence

Table B.2a: -20° Incidence, variation of Y_m , δ and C_{pb} with AVR

AVR	Y_m	δ	C_{pbm}
0.953	0.0265	-5.0°	-0.0673
0.986	0.025	-5.2°	-0.05573
1.022	0.0222	-5.8°	
1.047	0.0218	-6.6°	-0.03508

Table B.2b: -20° Incidence, spanwise variation of Y and δ at AVR=(0.99)

z/H	Y	δ
0.232	0.0408	-5.9°
0.299	0.0298	-5.3°
0.366	0.027	-5.6°
0.5	0.025	-5.2°
0.634	0.0246	-5.6°
0.701	0.0268	-5.3°
0.768	0.0368	-6.2°

+20° Incidence

Table B.3a: +20° Incidence, variation of Y_m , δ and C_{pb} with AVR

AVR	Y_m	δ	C_{pb}
1.008	0.148	-9.0°	0.035357
1.016	0.146	-9.7°	
1.022	0.1373	-9.6°	
1.035	0.1273	-9.4°	
1.041	0.1279	-9.2°	
1.087	0.1121	-9.8°	-0.06764

Table B.3b: +20° Incidence, spanwise variation of Y and δ at AVR=1.01

z/H	Y	δ
0.126	0.16	-4.8°
0.228	0.26	-7.0°
0.296	0.21	-7.1°
0.364	0.158	-8.9°
0.432	0.151	-8.0°
0.5	0.138	-8.0°
0.568	0.149	-8.6°
0.636	0.1515	-9.6°
0.704	0.26	-7.7°
0.772	0.2365	-6.7°
0.874	0.1471	-4.0°

Incidence Comparison

Table B.4a: Incidence comparison of Y , δ and C_{pb} for the original and new end-plates

	Incidence		
	-20°	Design	+20°
$Y_{m (AVR = 1)}$	0.0245	0.0216	0.1273
$Y_{m (AVR > 1)}$	0.0218	0.0196	0.1121
$Y_{m \text{ original}}$	0.0245	0.0210	0.036
$\delta_{(AVR = 1)}$	-5.3°	-4.9°	-9.5°
$\delta_{(AVR > 1)}$	-6.6°	-5.6°	-9.8°
δ_{original}	-7.5°	-5.8°	-8.7°
$C_{pbase (AVR = 1)}$	-0.056	-0.067	0.035
$C_{pbase (AVR > 1)}$	-0.0351	0.036	-0.061
$C_{pbase \text{ original}}$	-0.01	-0.015	-0.1

Critical Reynold's Number

Table B.5a: Design Incidence, variation of Y_m with Reynolds' number for the original and new end-plates

Re # ($\times 10^5$)	Y_m	Re # ($\times 10^5$)	$Y_{m \text{ original}}$
1.9	0.034	1.19	0.03
2.4	0.028	1.69	0.0255
2.7	0.024	2.19	0.023
2.9	0.0214	2.62	0.022
3.3	0.022	2.81	0.0215

Table B.5b: $+20^\circ$ Incidence, variation of Y_m with Reynolds' number for the original and new end-plates

Re # ($\times 10^5$)	Y_m	Re # ($\times 10^5$)	$Y_{m \text{ original}}$
1.3	0.1252	1	0.046
1.88	0.119	1.5	0.0395
2.36	0.1206	2.1	0.0375
2.79	0.1121	2.6	0.036
3.35	0.1038		

Appendix C

Static Pressure Tap Coordinates

Figure C.1: Graphical representation of the blade static pressure tap coordinates.

Table C.1 Instrumented blade static pressure tap coordinates

- Suction surface
- Pressure surface

Figure C.1: Graphical representation of the blade static pressure tap coordinates.

Static Pressure Tap Coordinates

Table C.1 Instrumented blade static pressure tap coordinates

Tap #	suction surface	
	x/C	y/C
25	0.0016	0.0244
24	0.0073	0.0565
23	0.0172	0.0837
22	0.0409	0.1279
21	0.0762	0.1731
20	0.1183	0.212
19	0.1657	0.2443
18	0.2038	0.264
17	0.2437	0.2796
16	0.2985	0.2941
15	0.3543	0.3013
14	0.4099	0.3017
13	0.4642	0.2959
12	0.5162	0.2852
11	0.5652	0.2708
10	0.6111	0.254
9	0.6539	0.2357
8	0.6937	0.2166
7	0.731	0.1971
6	0.766	0.1775
5	0.799	0.1578

	suction surface	
Tap #	x/C	y/C
4	0.8306	0.1381
3	0.8906	0.0989
2	0.9339	0.0693
1	0.9772	0.039
43	1.004	0.0085

	pressure surface	
Tap #	x/C	y/C
26	0.0284	0
27	0.0662	0.0144
28	0.0948	0.0277
29	0.1423	0.0489
30	0.1935	0.0698
31	0.242	0.0878
32	0.2884	0.1028
33	0.3544	0.1208
34	0.4374	0.1361
35	0.5153	0.1431
36	0.5889	0.1426
37	0.6586	0.1355
38	0.7248	0.1223
39	0.7878	0.1035
40	0.8482	0.0794
41	0.9207	0.0423
42	0.9633	0.0164
43	1.004	0.0085

Appendix D

Blade Static Pressure Distribution

Table D.1a: Design Incidence, variation of C_p with tap # for $AVR \leq 1$.

Table D.1b: Design Incidence, variation of C_p with tap # for $AVR \geq 1$.

Table D.2: -20° Incidence, variation of C_p with tap # for $AVR = 0.953, 0.986$ and 1.047 .

Table D.3: $+20^\circ$ Incidence, variation of C_p with tap # for $AVR = 1.008$ and 1.087 .

3 1 of/de 3

PM-1 3½"x4" PHOTOGRAPHIC MICROCOPY TARGET
NBS 1010a ANSI/ISO #2 EQUIVALENT

PRECISIONSM RESOLUTION TARGETS

PIONEERS IN METHYLENE BLUE TESTING SINCE 1974

10800 COUNTY ROAD 5, BURRISVILLE, MD 21817 USA
TEL: 410/465-7887 FAX: 410/465-7887 TUL: 8100000400

Blade Static Pressure Distribution

Table D.1a: Design Incidence, variation of C_p with tap # for $AVR \leq 1$.

	AVR			
Tap #	0.967	0.977	0.987	0.998
1	-1.527	-1.4257	-1.5433	-1.6108
2	-1.6153	-1.5103	-1.6323	-1.704
3	-1.7342	-1.6227	-1.7513	-1.8264
4	-1.8702	-1.7463	-1.8787	-1.9579
5	-1.9469	-1.8212	-1.9565	-2.0383
6	-1.9851	-1.8507	-1.9929	-2.0736
7	-2.0511	-1.9109	-2.0565	-2.1385
8	-2.1792	-2.0364	-2.1844	-2.2686
9	-2.2973	-2.1503	-2.2888	-2.376
10	-2.3923	-2.2443	-2.3664	-2.4504
11	-2.5409	-2.3487	-2.4651	-2.5522
12	-2.698	-2.4744	-2.5923	-2.6813
13	-2.8004	-2.5687	-2.683	-2.7683
14	-2.784	-2.5441	-2.6361	-2.7254
15	-2.7042	-2.4506	-2.5266	-2.6092
16	-2.7151	-2.4493	-2.5159	-2.5935
17	-2.6713	-2.3906	-2.4491	-2.5227
18	-2.626	-2.3436	-2.3918	-2.4632
19	-2.5684	-2.2843	-2.3316	-2.4014

	VR			
Tap #	0.967	0.977	0.987	0.998
20	-2.4299	-2.1554	-2.2007	-2.2677
21	-2.0548	-1.8207	-1.8697	-1.9328
22	-1.7743	-1.5726	-1.6353	-1.6946
23	-1.3306	-1.1855	-1.2626	-1.3199
24	-1.1633	-1.0519	-1.1488	-1.21
25	-0.3703	-0.3566	-0.4851	-0.5414
26	0.7425	0.8167	0.8402	0.8559
27	0.3076	0.4068	0.4218	0.427
28	0.2964	0.3861	0.3914	0.3942
29	0.2703	0.3546	0.3535	0.3524
30	0.2359	0.3176	0.3111	0.3077
31	0.1778	0.2611	0.251	0.2452
32	0.1735	0.2535	0.2409	0.2339
33	0.1839	0.2589	0.2427	0.2344
34	0.1682	0.2392	0.2185	0.2084
35	0.1457	0.2151	0.1904	0.1785
36	0.1481	0.2132	0.1857	0.1729
37	0.1239	0.1875	0.1575	0.1423
38	0.0752	0.1391	0.1044	0.0867
39	-0.041	0.0266	-0.0137	-0.0349
40	-0.1923	-0.119	-0.1655	-0.1912
41	-0.4871	-0.4028	-0.463	-0.4989
42	-0.9226	-0.816	-0.897	-0.9472
43	-1.288	-1.1819	-1.2803	-1.3484

Blade Static Pressure Distribution

Table D.1b: Design Incidence, variation of C_p with tap # for $AVR \geq 1$.

Tap #	AVR			
	0.998	1.01	1.046	1.078
1	-1.6108	-1.4257	-1.9769	-2.0832
2	-1.704	-1.5103	-2.0814	-2.189
3	-1.8264	-1.6227	-2.2235	-2.3364
4	-1.9579	-1.7463	-2.372	-2.4963
5	-2.0383	-1.8212	-2.466	-2.593
6	-2.0736	-1.8507	-2.5073	-2.6354
7	-2.1385	-1.9109	-2.5847	-2.7177
8	-2.2686	-2.0364	-2.714	-2.8476
9	-2.376	-2.1503	-2.809	-2.941
10	-2.4504	-2.2443	-2.8725	-3.0053
11	-2.5522	-2.3487	-2.9671	-3.1034
12	-2.6813	-2.4744	-3.1052	-3.2418
13	-2.7683	-2.5687	-3.1817	-3.3121
14	-2.7254	-2.5441	-3.1	-3.2291
15	-2.6092	-2.4506	-2.9295	-3.0611
16	-2.5935	-2.4493	-2.8892	-3.0098
17	-2.5227	-2.3906	-2.789	-2.901
18	-2.4632	-2.3436	-2.7062	-2.8196
19	-2.4014	-2.2843	-2.6255	-2.7353

	AVR			
Tap #	0.998	1.01	1.046	1.078
20	-2.2677	-2.1554	-2.485	-2.5849
21	-1.9328	-1.8207	-2.1301	-2.2182
22	-1.6946	-1.5726	-1.8939	-1.9648
23	-1.3199	-1.1855	-1.5204	-1.576
24	-1.21	-1.0519	-1.4366	-1.4823
25	-0.5414	-0.3566	-0.7835	-0.7966
26	0.8559	0.8167	0.8983	0.8705
27	0.427	0.4068	0.4466	0.4059
28	0.3942	0.3861	0.3976	0.3595
29	0.3524	0.3546	0.343	0.3065
30	0.3077	0.3176	0.2881	0.2506
31	0.2452	0.2611	0.2167	0.1779
32	0.2339	0.2535	0.1986	0.1611
33	0.2344	0.2589	0.1927	0.1546
34	0.2084	0.2392	0.1567	0.1177
35	0.1785	0.2151	0.1172	0.0772
36	0.1729	0.2132	0.1045	0.0644
37	0.1423	0.1875	0.0654	0.0243
38	0.0867	0.1391	-0.0019	-0.0448
39	-0.0349	0.0266	-0.141	-0.1901
40	-0.1912	-0.119	-0.3225	-0.3745
41	-0.4989	-0.4028	-0.677	-0.748
42	-0.9472	-0.816	-1.1961	-1.2711
43	-1.3484	-1.1819	-1.6537	-1.7415

Blade Static Pressure Distribution

Table D.2: -20° Incidence, variation of C_p with tap # for AVR= 0.953, 0.986 and 1.047.

	AVR		
Tap #	0.953	0.986	1.047
1	-1.8081	-2.3061	-2.5821
2	-1.9211	-2.4407	-2.7182
3	-2.0686	-2.6176	-2.9124
4	-2.2365	-2.8123	-3.1159
5	-2.3371	-2.9376	-3.2306
6	-2.3827	-2.9842	-3.2839
7	-2.4688	-3.0818	-3.424
8	-2.6246	-3.259	-3.5435
9	-2.7852	-3.4254	-3.6405
10	-2.9099	-3.5387	-3.7095
11	-3.0341	-3.6589	-3.8111
12	-3.2018	-3.8468	-3.9777
13	-3.3249	-3.9646	-4.0353
14	-3.286	-3.8885	-3.8969
15	-3.1262	-3.6724	-3.6195
16	-3.0731	-3.5876	-3.4806
17	-2.914	-3.3817	-3.2284
18	-2.7477	-3.1821	-3.0036
19	-2.548	-2.9428	-2.7518

	AVR		
Tap #	0.953	0.986	1.047
20	-2.1587	-2.5098	-2.3121
21	-1.5116	-1.7825	-1.6276
22	-0.9257	-1.1357	-1.0228
23	-0.2437	-0.3854	-0.3199
24	0.173	0.0668	0.1009
25	0.9103	0.8896	0.8799
26	-0.8104	-0.9034	-0.7325
27	-0.739	-0.8516	-0.6924
28	-0.2046	-0.2649	-0.1798
29	-0.0302	-0.1075	-0.0353
30	-0.0078	-0.0901	-0.0279
31	-0.023	-0.1121	-0.0565
32	0.0021	-0.0886	-0.0401
33	0.044	-0.0472	-0.0098
34	0.0435	-0.0544	-0.0299
35	0.0324	-0.0738	-0.0597
36	0.0393	-0.0712	-0.069
37	0.0144	-0.1047	-0.1142
38	-0.0384	-0.1709	-0.1915
39	-0.1589	-0.3138	-0.3514
40	-0.315	-0.5007	-0.5582
41	-0.6303	-0.8733	-0.9716
42	-1.0943	-1.4209	-1.5753
43	-1.4672	-1.9034	-2.1362

Blade Static Pressure Distribution

Table D.3: +20° Incidence, variation of C_p with tap # for AVR= 1.008 and 1.087.

	AVR	
Tap #	1.008	1.087
1	-0.4116	-0.5909
2	-0.4185	-0.6244
3	-0.4292	-0.6692
4	-0.4604	-0.7531
5	-0.4766	-0.8042
6	-0.4971	-0.8496
7	-0.5243	-0.9041
8	-0.5658	-0.9753
9	-0.6207	-1.046
10	-0.6879	-1.1055
11	-0.7714	-1.1753
12	-0.8893	-1.2733
13	-0.9903	-1.3294
14	-1.0413	-1.3174
15	-1.0539	-1.2718
16	-1.1242	-1.2925
17	-1.1922	-1.3183
18	-1.256	-1.3635
19	-1.3457	-1.4391

	AVR	
Tap #	1.008	1.087
20	-1.4854	-1.5799
21	-1.5359	-1.6646
22	-1.6942	-1.8972
23	-2.4177	-2.5555
24	-2.497	-3.0963
25	-3.1342	-3.8549
26	0.5849	0.3129
27	0.9469	0.9785
28	0.8732	0.9158
29	0.7913	0.8325
30	0.7249	0.7595
31	0.6668	0.6944
32	0.6379	0.6583
33	0.6175	0.6268
34	0.5866	0.582
35	0.5613	0.5449
36	0.5522	0.5252
37	0.5313	0.4936
38	0.4984	0.4487
39	0.4317	0.3656
40	0.3447	0.2561
41	0.1754	0.0358
42	-0.1008	-0.2817
43	-0.3471	-0.5799

Appendix E

Control Surfaces Influence on Inlet and Outlet Flow Behaviour

Figure E.1 Control surfaces adjustment nomenclature.

Table E.1 Periodicity settings for 0° , -20° and $+20^\circ$ incidence at $AVR=1$.

Comments on the control surfaces influence.

Figure E.1: Control surfaces adjustment nomenclature.

Table E.1: Periodicity settings for the control surfaces

Design incidence AVR = 0.998						
Top Tailboard ttb (°) 83	Bottom Tailboard btb (°) 79	Top Flap tf (°) 53	Bottom Flap bf (°) 35.4	Bypass #1 bp1 (mm) 23	Bypass #2 bp2 (mm) 17	Floating wall fw (°) 2 (+)
End-plates adjustments, AVR = 0.998 (see figure 4.7)						
k1 (mm) 28	l1 (mm) 144	k2 (mm) 25	l2 (mm) 150	ε (mm) 9		

-20° incidence AVR = 0.986						
Top Tailboard ttb (°) 61	Bottom Tailboard btb (°) 56	Top Flap tf (°) 21	Bottom Flap bf (°) 18	Bypass #1 bp1 (mm) 15	Bypass #2 bp2 (mm) 0	Floating wall fw (°) 1.6 (+)
End-plates adjustments, AVR = 0.986 (see figure 4.7)						
k1 (mm) 32	l1 (mm) 144	k2 (mm) 29	l2 (mm) 150	ε (mm) 9		

+20° incidence, AVR = 1.008						
Top Tailboard ttb (°) 98.5	Bottom Tailboard btb (°) 96	Top Flap tf (°) 75	Bottom Flap bf (°) 58	Bypass #1 bp1 (mm) 77	Bypass #2 bp2 (mm) 82	Floating wall fw (°) 3.5 (+)
End-plates adjustments, AVR = 1.008 (see figure 4.7)						
k1 (mm) 44	l1 (mm) 120	k2 (mm) 29	l2 (mm) 150	ε (mm) 12		

Comments on the Control Surfaces Influence

The adjustment of the floating wall should be made first during the inlet measurement tests. This can be done by monitoring the pressure differential with the two pitot tubes located near the leading edge of the wall. These tubes can be connected to a inclined micromanometer. The ideal positioning would have the wall lined-up with the flow. However, it was found that a small positive angle of attack was needed to improve the flow uniformity. The next control surface to be adjusted is the top flap, since there is only one control surface at the top of the passage. It was found that an alignment of the top flap to give a 0° angle into the top blade would create a "dip" in the mid-height velocity. The bottom flap and bypass flaps can be used to trim the flow to desired values of velocity and inlet angle over the three middle blades. A uniform inlet angle over the five blades was, however, not achievable. A constant difference of 2° between the midblade value and the top blade was present regardless of the control surface settings.

The tailboard settings were more easily determined. A good starting point can be determined by lightly loosening the bolts retaining the tailboards (to allow play) and running the tunnel to normal operating speed. The bolts can then be re-tighten after the tailboards have settled into position while the tunnel is running. It was found that only small adjustments from this point were required. Little or no coupling effects between the tailboard and inlet flow quality were observed.

A summary of the effects of the individual control surfaces is included below:

Floating Wall:	Affects inlet flow angularity and mass flow rate through passage.
Top Flap:	Affects both the inlet angle and velocity for the top half of the passage. Some coupling with the lower control surfaces.

- Bottom Flap:** Affects both the inlet angle and velocity for the bottom half of the passage. Some coupling with the top flap.
- Bypass #1:** Affects the inlet velocity for the lower part of the passage. Some mild influence on the flow angle.
- Bypass #2:** Affects the mass flow rate of the passage. Bleeds off the floor boundary layer. Influences the sensitivity of the other control surfaces.
- Top Tailboard:** Affects the exit angle and velocity for the upper part of the passage. No coupling to inlet conditions (except at high positive incidence).
- Bottom Tailboard:** Affects the exit angle and velocity for the lower part of the passage. No coupling to inlet conditions (except at high positive incidence).
- Perforated Screen:** Increases the back pressure in the passage, which provides more sensitivity to the inlet control surfaces. (Note: only needed at high positive incidence).

Appendix F

Data Reduction Programs

Program 1: Three-Hole Probe Data Reduction

Program 2: Blade Static Pressure Distribution Data Reduction

Program 3: Profile Loss Calculation

Program 4: Mixing Loss Calculation

Program 1: Three-Hole Probe Data Reduction

File name: THPRED2.FOR

```
PROGRAM REDUCE
  INTRINSIC sqrt
  INTEGER chline
  REAL yct(300),cp1(300),cp2(300),cp3(300)
  REAL beta(300),vel(300),q(300),ps(300),ptloss(300)
  REAL k1,k12,k13,k23,kps2,kps3,array(2,246)
  CHARACTER*12 infil,inarray,outv,outdat,junk2*2,dte*25
C
  nold = 0
C
  DO 5 K = 1,8
C
C    Reading formatted file
C
    write(*,*) ' Enter input file name : '
    read(*,40) infil
    write(*,2) infil
    open(5,file=infil,status='old')
    read(5,90) dte
    write(*,3) dte
2   format(iX,A12)
3   format(1X,A25)
    read(5,91) tipclr
    write(*,80) tipclr
    read(5,41) junk2
    read(5,41) junk2
    read(5,41) junk2
    read(5,41) junk2
    read(5,41) junk2
    read(5,41) junk2
    read(5,41) junk2
    read(5,92) alpha
    write(*,80) alpha
    read(5,41) junk2
```

```

      read(5,93) zinit
      write(*,80) zinit
      read(5,94) yinit
      write(*,80) yinit
      read(5,41) junk2
      read(5,41) junk2
      read(5,41) junk2
80   format(2X,F9.3)
91   format(45X,F7.5)
92   format(46X,F7.2)
93   format(38X,F6.2)
94   format(55X,F8.3)
C
      read(5,56) stepy
      write(*,80) stepy
      read(5,41) junk2
      read(5,57) nstaty
      write(*,*) nstaty
56   format(27X,F7.4)
57   format(34X,I3)
C
      read(5,41) junk2
      read(5,41) junk2
      read(5,41) junk2
      read(5,41) junk2
      read(5,41) junk2
40   format(A12)
41   format(A2)
C
      ns = nstaty
      nstaty = nold + ns
      nstart = nold + 1
C
C      Reading Data
C
      DO 10 I = nstart,nstaty
        read(5,*) chline,ny,nz,qin,pin,q1,p1,q2,p2,q3,p3
        read(5,*) chline,ny,nz,press,temp,cp1(I)
        *,cp2(I),cp3(I)
C
        yct(I) = (yinit - ((ny-1)*stepy))/111
C

```

```

10  CONTINUE
C
    nold = nstaty
C
    write(*,*) ' other input files (north and down)'
    write(*,*) ' no(0)  yes(1)'
    read(*,*) incheck
    IF (incheck.EQ.0) GOTO 7
C
5   CONTINUE
C
7   write(*,*) ' end after ',K,' input files'
C
    write(*,*) 'axial location x in mm (upstream +)'
    read(*,*) xloc
    xc = xloc/162.8
    zc = zinit/200
C
C   Preparing file for data reduction from THP_DATR3.pas
C
    write(*,*) ' Do you want to prepare file for input'
    write(*,*) ' into THPDATR3.pas ?. yes(1), no(0)'
    read(*,*) mout
C
    IF (mout.eq.1) then
        write(*,*) ' Output file for ver data:'
        read(*,40) outv
        open(8,file=outv,status='new')
        write(8,98) dte
        write(8,99) nstaty,tipclr
        DO 70 I = nstaty,1,-1
            write(8,101) xc,yct(I),zc,alpha,cp1(I)
            *,cp2(I),cp3(I)
70   CONTINUE
        END IF
C
C   Input of calibration array
C
    write(*,*) ' Input file of calibration array.'
    write(*,*) ' (array10.dat)'
    read(*,40) inaray
    OPEN(6,file=inaray,status='old')

```

```

C      DO 20 I = 1,246
        read(6,102) array(1,I),array(2,I)
20    CONTINUE
C
C      DO 60 I = 1,nstaty
C
C        k23 = cp2(I) - cp3(I)
        kps2 = (cp1(I)-cp2(I))/(cp1(I)-cp3(I))
        kps3 = 1/kps2
        write(*,140) k23,kps2,kps3
C
C      For positive angles using KPS3 and K13 curves
C
C      IF (k23.LE.0) then
        CALL INTERP(kps3,array,6,angle)
        CALL INTERP(angle,array,3,k13)
        CALL INTERP(angle,array,1,k1)
        write(*,141) angle,k13,k1
        beta(I) = (0-alpha) - angle
        q(I) = (cp1(I)-cp3(I))/k13
        ps(I) = cp1(I) - q(I)*k1
        ptloss(I) = q(I) + ps(I) - 1
        IF (q(I).GE.0) then
            vel(I) = sqrt(q(I))
        ELSE
            write(*,*) ' Negative Q ??? error'
        END IF
    END IF
C
C      For negative angles using KPS2 and K12 curves
C
C      IF (k23.GT.0) then
        CALL INTERP(kps2,array,5,angle)
        CALL INTERP(angle,array,2,k12)
        CALL INTERP(angle,array,1,k1)
        write(*,142) angle,k12,k1
        beta(I) = (0-alpha) - angle
        q(I) = (cp1(I)-cp2(I))/k12
        ps(I) = cp1(I) - q(I)*k1
        ptloss(I) = q(I) + ps(I) - 1
        IF (q(I).GE.0) then

```

```

        vel(I) = sqrt(q(I))
        ELSE
            write(*,*) ' Negative Q ??? error'
        END IF
    END IF
C
60    CONTINUE
C
    write(*,*) ' Output file for data:'
    read(*,40) outdat
    open(9,file=outdat,status='new')
    write(9,98) dte
    write(9,97)
        write(9,111)
    write(9,97)
        write(9,112)
        write(9,113)
    DO 84 I = nstaty,1,-1
        nbr = nstaty - (I-1)
        write(9,130) nbr,xc,yct(I),zc
84    CONTINUE
    write(9,97)
        write(9,114)
        write(9,115)
        write(9,116)
        write(9,117)
        write(9,118)
        write(9,119)
        write(9,120)
        write(9,121)
        write(9,122)
        write(9,123)
        write(9,124)
        write(9,97)
        write(9,125)
        write(9,126)
    DO 85 I = nstaty,1,-1
        nbr = nstaty - (I-1)
        write(9,131) nbr,q(I),vel(I),ps(I),beta(I),ptloss(I)
85    CONTINUE
C
90    format(5X,A20)

```



```

97      format(' ')
98      format(A20)
99      format(I3,1X,F6.4)
101     format(3(F7.4,1X),F7.2,1X,3(F8.5,1X))
102     format(1X,F9.4,1X,F9.4)
C
111     format(' x, y and z location of three-hole probe')
112     format(' #  X/C  Y/S  Z/H ')
113     format('-----')
114     format(' Beta : Flow direction relative to the')
115     format('      horizontal axis. Positive air ')
116     format('      flows upward.')
117     format(' Q : Dynamic pressure at the data point')
118     format('      nondimensionalised by Qin')
119     format(' V : Velocity at the data point ')
120     format('      nondimensionalised by centerline Vin')
121     format(' Ps : Static pressure at the data point.')
122     format('      (Ps - Psin)/Qin')
123     format(' PtLoss : Total pressure loss coef. at the')
124     format('      data point (Ptot - Ptot(in))/Qin')
125     format(' #    Q    V    Ps    Beta  PtLoss')
126     format('-----')
130     format(1X,I3,2X,3(F6.3,2X))
131     format(1X,I3,2X,5(F8.4,1X))
140     format(' K23,Kps2,Kps3 ',3(F6.3,1X))
141     format(' yaw,K13,K1 ',F6.1,1X,2(F6.3,1X))
142     format(' yaw,K12,K1 ',F6.1,1X,2(F6.3,1X))

```

```

C
  STOP
  END

```

```

C
C *****
C

```

```

C Subroutine INTERP : Linear interpolation between two sets of
C      numbers (A,B) from array "L"
C

```

```

C      SUBROUTINE INTERP(A,array,L,B)
C      real array(2,246)

```

```

C
  B = 999.
  J = 41*(L-1) + 1
  DO 11 I = J,(J+39)

```

```

      IF (((A.GE.array(1,I)).AND.(A.LT.array(1,I+1))).OR.
*      ((A.LE.array(1,I)).AND.(A.GT.array(1,I+1)))) then
      pct = (A-array(1,I))/(array(1,I+1)-array(1,I))
      B = array(2,I) + pct*(array(2,I+1)-array(2,I))
      END IF
11  CONTINUE
C
C  Checking for out of range...
C
      IF (B.EQ.999.) then
      write(*,*) ' Out of range ? in interpolation routine.'
      write(*,*) ' at array L = ',L
      END IF
C
      RETURN
      END

```

Program 2: Blade Static Pressure Distribution Data Reduction

File name: BLDRED.FOR

```
C*****
C
C   This program reduces the data from BLADE3.PAS for
C   blade static pressure distribution. The input file
C   should be the processed data. The tap # and the
C   x-coordinates of the blade are in file TAPXCOORD.DAT
C   and must be present to run program. The output gives
C   the tap #, x-coord and Cp respectively.
C
C*****
C
C   PROGRAM BLDRED
C   DIMENSION CP(43)
C   INTEGER NB(43),ANSWER
C   CHARACTER*14 INPFIL,OUTFIL,JUNK*2,DTE*25

5    CONTINUE
    WRITE(*,10)
10   FORMAT(' INPUT FILE NAME?')
    READ(*,20) INPFIL
20   FORMAT(A)
    OPEN(4,FILE=INPFIL,STATUS='OLD')
    WRITE(*,30)
30   FORMAT(' OUTPUT FILE NAME?')
    READ(*,20) OUTFIL
    OPEN(5,FILE=OUTFIL,STATUS='UNKNOWN')

    WRITE(*,41)
41   FORMAT(' WHAT IS THE RATIO Q1/Qref ?')
    READ(*,*) RAT1

    WRITE(*,42)
42   FORMAT(' WHAT IS THE RATIO (Ps1 - Pref)/Qref ?')
    READ(*,*) RAT2
```

```

      READ(4,43) DTE
43  FORMAT(A25)

      DO 44 I=1,15
      READ(4,*) JUNK
44  CONTINUE

      DO 45 I=1,43
      READ(4,*) NTAP,CP(I)
45  CONTINUE

      DO 46 I=1,43
      CP(I) = (CP(I) - RAT2)/RAT1
46  CONTINUE

      OPEN(6,FILE='TAPXCOOR.DAT',STATUS='OLD')
      DO 47 I=1,43
      IF ((I.EQ.1).OR.(I.EQ.26)) READ(6,*) JUNK
      READ(6,*) NTAP,XC(NTAP)
47  CONTINUE

      WRITE(5,48) DTE
      WRITE(5,54)
48  FORMAT(2X,A25)

      DO 50 I=1,43
      K=I
      IF (K.EQ.41) THEN
      WRITE(5,55) 41,XC(K),CP(42)
      ELSE
      IF (K.EQ.42) THEN
      WRITE(5,55) 42,XC(K),CP(41)
      ELSE
      WRITE(5,55) K,XC(K),CP(K)
      ENDIF
      ENDIF
50  CONTINUE

54  FORMAT(' TAP X/C Cp')
55  FORMAT(2X,I2,2X,F6.4,2X,F7.4)
      CLOSE(5,FILE=OUTFIL,STATUS='OLD')
      WRITE(*,60)

```

```
60  FORMAT (' RUN PROGRAM AGAIN ? YES(1), NO(0)')  
    READ(*,*) ANSWER  
    IF (ANSWER.EQ.1) THEN  
      GOTO 5  
    ENDIF  
    STOP  
    END
```

Program 3: Profile Loss Calculation

File name: PLC2.FOR (modified from PL.FOR from Tremblay (1989))

```
DIMENSION Q(200),V(200),PS(200),BETA(200)
DIMENSION CP(200),X(200),VA(200),STPSIZ(10),SLOPE(200)
INTEGER EXPNB(200),NBPTS(10),TOTPTS
CHARACTER*12 INFILE,JUNK*2

PI=4.0*ATAN(1.0)
PITCH=111.36

40  CONTINUE

    WRITE(*,50)
50  FORMAT(' *****')
    WRITE(*,60)
60  FORMAT(' TOTAL LOSS COEFFICIENT ;          (1)')
    WRITE(*,70)
70  FORMAT(' MASS AVERAGE TOTAL LOSS COEFFICIENT ; (2)')
    WRITE(*,80)
80  FORMAT(' EXIT THE PROGRAM ;          (0)')
    WRITE(*,90)
90  FORMAT(' *****')
    WRITE(*,100)
100 FORMAT(' OPTION=')
    READ(*,*) OPT

    IF (OPT.EQ.0) THEN
        GO TO 220
    ENDIF

    WRITE(*,10)
10  FORMAT(' UNIT 4: INPUT FILE'//)
    READ(*,15) INFILE
15  FORMAT(A)

    OPEN(4,FILE=INFILE,STATUS='OLD')
```

```

WRITE(*,*) ' Total number of stations ?'
READ(*,*) NTOT

WRITE(*,*) ' Begining and End station for one passage ?'
READ(*,*) NBEG,NEND

TOTPTS = NEND-NBEG+1

DO 16 I=1,(5+NBEG)
  READ(4,21) JUNK
16 CONTINUE
DO 17 I=1,TOTPTS
  READ(4,*) NB,XT,Y,ZT
  X(I)=Y*PITCH
  WRITE(*,*) NB
17 CONTINUE
  XPITCH=(X(TOTPTS)-X(1))/PITCH
  WRITE(*,300) XPITCH
300 FORMAT(' PASSAGE = 'F5.2,' OF PITCH')
DO 18 I=1,(14+NTOT-NEND+NBEG)
  READ(4,21) JUNK
18 CONTINUE
DO 19 I=1,TOTPTS
  READ(4,*) NB,Q(I),V(I),PS(I),BETA(I),CP(I)
  WRITE(*,*) NB
19 CONTINUE

21 FORMAT(A2)

DO 37 K=1,TOTPTS
  IF(K.NE.1) THEN
    SLOPE(K)= (CP(K) - CP(K-1))/(X(K)-X(K-1))
  ENDIF
37 CONTINUE

WRITE(*,35) (I+NBEG-1,CP(I),SLOPE(I),I=1,TOTPTS)
35 FORMAT(' STAT #',I3,5X,'CP=',F10.4,3X,'SLOPE=',F10.4)

C {CALCULATES THE LINE OF ZERO TOTAL PRESSURE LOSS}

CALL LINFIT(X,CP,TOTPTS,NBEG,BCOEFF,SCOEFF)

```

```

WRITE(*,76) SCOEFF,BCOEFF
76  FORMAT( ' m=',F10.4, ' b=',F10.4)

```

C {RESCALING OF THE 'CP' WITH RESPECT TO THIS ZERO PRESSURE LOSS LINE}

```

DO 140 K=1,TOTPTS
  CORREC=SCOEFF*X(K)+BCOEFF
  CP(K)=CP(K)-CORREC
140 CONTINUE

```

IF(OPT.EQ.1) THEN

CPAVG=0.0

```

DO 160 K=1,TOTPTS-1
  COUNT2=(CP(K+1)+CP(K))*(X(K+1)-X(K))/PITCH/2
  CPAVG=CPAVG+COUNT2
  WRITE(*,150)CPAVG
150  FORMAT( ' Cp(avg) =',F10.4)
160  CONTINUE

```

ELSE

IF(OPT.EQ.2) THEN

```

WRITE(*,170)
170  FORMAT( ' INLET FLOW ANGLE ?')
  READ(*,*) ALPHA1

```

VAAVG=0.0

VAVG=0.0

```

DO 180 K=1,TOTPTS-1
  ANGLE1=(BETA(K)-ALPHA1)*PI/180.
  VA(K)=V(K)*COS(ANGLE1)
  ANGLE2=(BETA(K+1)-ALPHA1)*PI/180.
  VA(K+1)=V(K+1)*COS(ANGLE2)
  WRITE(*,*) ANGLE1,ANGLE2
  COUNT1=(VA(K+1)+VA(K))*(X(K+1)-X(K))/2./PITCH

```



```

      VAAVG=COUNT1+VAAVG
      COUNT2=(V(K+1)+V(K))*(X(K+1)-X(K))/2./PITCH
      VAVG=COUNT2+VAVG
180   CONTINUE

      WRITE(*,190)VAAVG,VAVG
190   FORMAT( ' VA(avg) =',F10.4,2X,' V(avg) =',F10.4)

      CPAVG=0.0
      BETAVG=0.0

      DO 210 K=1,TOTPTS-1
        COUNT3=(CP(K+1)+CP(K))*(X(K+1)-X(K))/PITCH/
&      2.*(VA(K+1)+VA(K))/VAAVG/2.0
        CPAVG=CPAVG+COUNT3/VAVG**2
        COUNT4=(BETA(K+1)+BETA(K))*(X(K+1)-X(K))/PITCH/2.
        BETAVG=BETAVG+COUNT4
        WRITE(*,200)CPAVG,BETAVG
200   FORMAT(' Cp(avg) =',F10.4,' BETA(avg) =',F6.2)
210   CONTINUE

      ENDIF

      ENDIF

      GOTO 40

220  CONTINUE

      STOP
      END

```

C *****

```

      SUBROUTINE LINFIT(A,B,N,NBEG,BCOEFF,SCOEFF)
      DIMENSION A(200),B(200),W(200)

      SUM1=0.0
      SUM2=0.0
      SUM3=0.0
      SUM4=0.0
      SUM5=0.0

```

```

WRITE(*,3)
WRITE(*,1)
WRITE(*,2)
1  FORMAT(' STATIONS BETWEEN WHICH THE WEIGHT FUNCTION')
2  FORMAT(' IS ZERO, i.e. BORDER STATIONS AT EDGE OF WAKE')
3  FORMAT(' ')
   READ(*,*) MINSTA,MAXSTA

   MIN=MINSTA-NBEG+1
   MAX=MAXSTA-NBEG+1

   DO 5 I=1,N
     IF((I.GT.MIN).AND.(I.LT.MAX)) THEN
       W(I)=0.0
     ELSE
       W(I)=1.0
     ENDIF
5    CONTINUE

   DO 10 I=1,N
     SUM1=SUM1+W(I)*A(I)
     SUM2=SUM2+W(I)*B(I)
     SUM3=SUM3+W(I)*A(I)*B(I)
     SUM4=SUM4+W(I)*A(I)*A(I)
     SUM5=SUM5+W(I)
10   CONTINUE

   SCOEFF=(-SUM5*SUM3+SUM2*SUM1)/(ABS(SUM1)**2.0-SUM5*SUM4)
   BCOEFF=(SUM2-SUM1*SCOEFF)/SUM5

   RETURN
   END

```

Program 4: Mixing Loss Calculation

File name: **MIXLOSS2.FOR** (modified from Tremblay (1989))

```

    DIMENSION Q(200),V(200),PS(200),BETA(200)
    DIMENSION CP(200),X(200),VA(200),STPSIZ(10)
    REAL MEALOS,MIXLOS
    INTEGER EXP,NB(200),NBPTS(10),TOTPTS,CHOICE
    CHARACTER*12 INFILE,JUNK*2

    PITCH=111.36

5   CONTINUE

    WRITE(*,10)
10  FORMAT( ' UNIT 4: INPUT FILE'//)
    READ(*,15) INFILE
15  FORMAT(A)

    OPEN(4,FILE=INFILE,STATUS='OLD')

    WRITE(*,*) ' Total number of stations ?'
    READ(*,*) NTOT

    WRITE(*,*) ' Begining and End station for one passage ?'
    READ(*,*) NBEG,NEND

    TOTPTS = NEND-NBEG+1

    DO 16 I=1,(5+NBEG)
        READ(4,21) JUNK
16  CONTINUE
    DO 17 I=1,TOTPTS
        READ(4,*) NB,XT,Y,ZT
        WRITE(*,*) NB
        X(I)=Y*PITCH
17  CONTINUE
    DO 18 I=1,(14+NTOT-NEND+NBEG)
        READ(4,21) JUNK
```

```

18  CONTINUE
    DO 19 I=1,TOTPTS
      READ(4,*) NB,Q(I),V(I),PS(I),BETA(I),CP(I)
      WRITE(*,*) NB
19  CONTINUE

21  FORMAT(A2)

```

C { CALCULATION OF AVERAGE VELOCITY V1 }

```

    V1AVG=0.0

    DO 70 I=1,TOTPTS-1
      SUM1=(V(I+1)+V(I))*(X(I+1)-X(I))/PITCH/2.0
      V1AVG=V1AVG+SUM1
70  CONTINUE

    WRITE(*,75)V1AVG
75  FORMAT(' THE AVERAGE VELOCITY V1(avg) =',F10.4)

```

C { CALCULATION OF AVERAGE VELOCITY SQUARED V1**2 }

```

    V1AVG2=0.0

    DO 80 I=1,TOTPTS-1
      SUM2=(V(I+1)**2+V(I)**2)*(X(I+1)-X(I))/PITCH/2.0
      V1AVG2=V1AVG2+SUM2
80  CONTINUE

    WRITE(*,85)V1AVG2
85  FORMAT(' THE AVERAGE VELOCITY V1**2(avg) =',F10.4)

```

C { CALCULATION OF AVERAGE VELOCITY CUBED V1**3 }

```

    V1AVG3=0.0

    DO 90 I=1,TOTPTS-1
      SUM3=(V(I+1)**3+V(I)**3)*(X(I+1)-X(I))/PITCH/2.0
      V1AVG3=V1AVG3+SUM3
90  CONTINUE

    WRITE(*,95)V1AVG3

```

```

95  FORMAT(' THE AVERAGE VELOCITY V1**3(avg) =',F10.4)
    WRITE(*,100)
100  FORMAT(' MEASURED LOSSES=')
    READ(*,*) MEALOS

    MIXLOS=1-2*V1AVG2/V1AVG**2+V1AVG3/V1AVG**3

    WRITE(*,105)MIXLOS
105  FORMAT(' THE MIXING LOSS=',F10.4)
    TOTLOS=MEALOS+MIXLOS
    WRITE(*,110)TOTLOS
110  FORMAT(' THE TOTAL MIXED OUT LOSSES =',F10.4)

    WRITE(*,120)
120  FORMAT(' DO YOU WANT TO RUN THE PROGRAM AGAIN? Y(1), N(0)')
    READ(*,*) CHOICE

    IF (CHOICE.EQ.1) THEN
        GOTO 5
    ENDIF

    STOP
    END

```

Appendix F

Apparatus Troubleshooting

Comments on the maintenance and some of the problems encountered with the new wind tunnel and the data acquisition equipment.

Apparatus Troubleshooting

The motor and fan belt connected to the fan must be greased and checked periodically. The tension in the belt must be checked and adjusted to give 1/2" deflection at midheight with a 30 to 35 lbs force as shown in the figure below.

The screens in the diffuser and settling chamber must be cleaned periodically along with the honeycomb section. A small flashlight inserted through the fan blades can be used to inspect the honeycomb section. To remove the screens in the settling chamber, the whole test section must first be removed, then the wood inserts must be taken out to free the screens.

The foam insulation on the diffuser door and all over the test section must be periodically changed to prevent leakage from wear and tear. Cleaning of the blades is also essential for good, dependable results.

The computer should always be turned on before the traverse gear motor controller and all other amplifiers are turned on. If a loud humming noise is heard from the motor, turn the controller off, then restart computer before turning on the power. Sending a signal to the motor controller can also stabilize the motors. The wiring diagrams for all of the equipment are located in the laboratory (room 126ME). Some loose wires have been found in the past, it is therefore recommended to move the equipment as little as possible. If the equipment must be moved, it should be moved as a complete unit, without disconnecting cables and wires.

The backlash in the traverse gear should also be checked periodically as the teflon gear mechanism wears down. The accuracy can be improved by changing the teflon collar, a new one can be manufactured at the Mechanical Engineering Technical Centre. The instrument slots in the test section would sometimes bind and prevent motion of the traverse gear. The slot and the traverse gear mechanism should be kept well lubricated.

A frequent problem observed with the equipment, is the non-operation of the scanivalve unit via the computer, while still operational in manual mode. This will show up as constant values of pressure for all pressure ports during a run controlled by the computer. It was found that the chip in the amplifier box (octal buffer line driver 3-state non inverter (20 pins)) would often fail for no apparent reason.

END

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FIN