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**A TEST SECTION FOR INVESTIGATING THE OFF-DESIGN
PERFORMANCE OF PLANAR CASCADES**

by

Scott Malcolm Goobie

B.Sc. (Mechanical Engineering)

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

Ottawa Carleton Institute for Mechanical and Aeronautical Engineering
Department of Mechanical and Aeronautical Engineering
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Ottawa, Ontario

March 1989

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"A TEST SECTION FOR INVESTIGATING THE OFF-DESIGN
PERFORMANCE OF PLANAR CASCADES"

Submitted by Scott M. Goobie, B.Sc. (Mech.Eng.)
in partial fulfilment of the requirements for
the degree of Master of Engineering

Thesis Supervisor

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March 23, 1989

ABSTRACT

Discussions with industry and a review of the literature indicated that there is a lack of data on the losses in axial turbines operating at off-design conditions, particularly for blades of recent design. Accordingly, a wind tunnel test section has been designed and constructed for testing rectilinear cascades over a wide range of incidences.

Initial studies have focused on the two dimensional losses in a turbine cascade. The measurements were intended mainly to verify the capabilities of the test section and were limited to incidences of -15° , 0° and $+15^\circ$. Data are presented which show that excellent flow periodicity can be achieved in the cascade and that a large span of two dimensional flow can be produced about the blades. The effects of varying the Reynolds number and the downstream location of the loss measurements were also investigated. It was found that the additional loss due to the diffusion, or "mixing out", of the two dimensional blade wake was relatively small. Thus, the precise downstream location of two dimensional loss measurements does not appear to be particularly important. It was also found that the widely used Ainley and Mathieson loss correlation considerably overestimates the losses for the present cascade; on the other hand, the measurements were in good agreement with the correlation recently proposed by Moustapha, Kacker, and Tremblay (1989).

The preliminary measurements have demonstrated the flexibility of the test section and its ability to produce loss data of very high quality. It should be a very satisfactory tool in the planned study of off-design losses in turbines.

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NOMENCLATURE

A	=	area
C	=	blade chord length
C_x	=	blade axial chord length
C_p	=	$(P - P_{cl})/(1/2\rho V_{cl}^2)$ = static pressure coefficient
P	=	static pressure
P_o	=	total pressure
q	=	$1/2\rho V_{cl}^2$ = dynamic pressure
Re	=	Reynolds number
S	=	blade spacing
t_{max}	=	blade maximum thickness
u,v,w	=	components of velocity in the x,y and z directions
u'	=	root mean square velocity fluctuation in the x direction
x,y,z	=	co-ordinates in the axial, tangential and spanwise directions
Y_{2D}	=	$(P_o - P_{ocl})/(1/2\rho V_{cl}^2)$ = two dimensional total pressure loss coefficient
Y_m	=	mixing loss coefficient
α	=	flow angle, measured from the axial direction
β	=	blade metal angle
γ	=	blade stagger angle
ρ	=	density
ξ	=	energy loss coefficient
$\Delta\phi^2$	=	velocity loss coefficient

Subscripts

CL	=	undisturbed centreline value
m	=	mixing
x,y,z	=	component in x,y and z direction
$1,2$	=	cascade inlet and outlet

1.0 INTRODUCTION

In a turbine work is extracted from the working fluid by changing its angular momentum as it passes through successive rows of turbine blades. This process is highly efficient for well designed blades when operating at design conditions. The main result of operating at an off-design condition is that the direction of the oncoming flow does not match the optimum incidence or angle of attack for the leading edge of the blades. In many applications a gas turbine engine is required to spend a considerable portion of its operating time at conditions away from the design point. The behaviour under these conditions is referred to as the off-design performance.

Turbine designers must strive to produce highly efficient blades and blade configurations for a wide range of operating conditions. It is therefore necessary to predict the off-design performance and to identify the blade and flow parameters which influence the aerodynamic losses at off-design conditions. These predictions are usually based on empirical correlations of experimental data taken from rectilinear or annular cascade test sections or engine test rigs. The most well known and widely used correlation method is due to Ainley and Mathieson (1951).

With continuing improvements in turbine design and performance it is necessary to update the existing correlations and to formulate new correlations with more recent data to reflect both the changes in turbine performance and also the increase in understanding of the flow. A comprehensive review of the literature indicated that there is a definite lack of recent cascade performance data for off-design conditions. There have also been several studies which have provided important insights into the nature of the flow mechanisms which determine the level of the aerodynamic losses in a turbine passage. However, there is still much to be understood about the effects of many blade and flow parameters on the turbine efficiency.

As a result of these findings this work was initiated to provide valuable off-design performance data and to investigate various aspects of the flow at off-design conditions. A variable incidence rectilinear cascade test section was designed and constructed. Two of the main features of the test section are the wide range of possible incidence settings and an innovative flow control system. A full description of the test section and instrumentation and the test section commissioning results are contained herein. Design point loss data and blade static pressure distributions are presented. The results of an

investigation of the effect of traverse location on the losses is also included. Off-design data was taken at incidence settings of $+15^\circ$ and -15° respectively. These results are presented along with comparisons of the measured data with the predictions of two correlation methods. A comparison of measured off-design blade static pressure distributions with the results of a cascade panel method prediction are also given. Finally, conclusions and recommendations for further work are put forward.

2.0 LITERATURE REVIEW

Turbomachines are designed to operate efficiently at the design point and, depending on the application, the off-design performance is often of the utmost importance. Aero engines in particular encounter a wide range of operating points between take off and landing of a given aircraft. This requires that a number of compromises must be made in engine design in order to achieve a wide operating range within which the aerodynamic losses are kept to a minimum.

According to Denton and Cumpsty (1987) the main types of losses in a turbomachine may be categorized as follows:

- 1) Profile loss which is the loss arising from the growth of blade surface boundary layers.
- 2) Tip leakage loss which is due to leakage of flow around the ends of shrouded or unshrouded blades.
- 3) Trailing edge loss which is due to the finite thickness of the blade trailing edges.
- 4) Annulus loss which is due to skin friction on the hub and casing of the machine outside of the blade rows.
- 5) Secondary loss which is due to viscous effects on the endwalls within the blade rows.
- 6) Leaving loss which is due to the kinetic energy of the flow leaving the machine.
- 7) Shock loss which is due to the formation of shock waves within or at the trailing edge of blade rows.
- 8) Windage loss which is due to the drag on the rotor disks and shaft rotating in the working fluid.

Mixing is another source of loss which is not a separate loss mechanism but rather a combination of viscous and heat transfer effects arising from the non uniformity of the flow field within the machine passages.

Denton and Cumpsty (1987) also pointed out that "entropy generation" is a more correct term than "loss" since the loss of efficiency in a turbomachine due to irreversible

effects may be directly related to the entropy created. For the purpose of this review both terms will be used since most authors still refer to the irreversibilities as "losses"

This work is directed towards studying two dimensional losses and the effects of incidence on these losses. However, before considering the effect on the losses of operating at an off-design point it is important to understand the various loss mechanisms at play. The following sections will discuss in some detail the flow mechanisms associated with aerodynamic losses in turbomachines and the various methods used to predict the losses

2.1 LOSS MECHANISMS IN TURBOMACHINES

This section presents an overview of the current understanding of both two dimensional and three dimensional losses in turbomachinery as presented in the literature. More emphasis is focused on the two dimensional losses because the experimental portion of this work is concentrated solely on two dimensional losses. However, future experimental work on the same apparatus will include the investigation of three dimensional losses associated with secondary and tip clearance flows.

2.1.1 TWO DIMENSIONAL LOSSES

In a cascade with purely two dimensional flow, losses may be separated as follows

1) PROFILE LOSSES - the energy loss associated with the loss of momentum due to friction between adjacent layers of fluid near the surface of the blade. The shear forces between the layers cause a thin boundary layer of low momentum fluid to form. The velocity of the fluid in the layer varies from zero at the blade surface to the free stream value at the edge of the boundary layer. The layer subsequently produces blockage and thus influences the pressure downstream of that point.

2) TRAILING EDGE LOSS - The separation of the fluid from the blade surface at the trailing edge of a blade with finite trailing edge thickness. This creates additional blockage which influences the downstream pressure distribution in the wake. This loss is not easily measured and therefore in experimental results it may not be separated from the loss due to the boundary layer blockage and these are sometimes lumped together under the term "midspan loss". For example, the midspan results of Hodson and Dominy (1987) include the trailing edge loss.

3) MIXING LOSS - The momentum loss due to mixing in the blade wake which may be calculated by using the inviscid, adiabatic equations for conservation of mass, energy and momentum. This mixing-out calculation was used by Hodson and Dominy (1987). They found the difference between the mixed-out values and the simple mass averaged loss values to range from 7 to 15 percent of the total measured loss when traversing 42% axial chord downstream of the trailing edge of the blades.

For the purpose of this work the total loss at midspan is referred to as "two-dimensional loss".

2.1.2 PROFILE LOSS

The profile loss is an important value to quantify as it comprises up to 25% of the total losses in a turbine stage. These losses may be predicted by 1) direct measurement in a turbine cascade wind tunnel test section or engine test rig, 2) correlations based on test data or 3) two-dimensional boundary layer calculation methods. Denton (1973) noted that there are many situations where only the second method is practical. This is true because either the detailed blade profiles are not known or the time required in data preparation would be excessive. This point stresses the need for cascade tests on turbine blades of more recent design in order to update existing correlations with new data.

2.1.2.1 CASCADE WIND TUNNEL TESTS

As previously mentioned, cascade test results for turbines of recent design are necessary for development of new correlations which represent the current levels of turbine performance. A cascade model is also very useful in developing a better understanding of the various flow mechanisms within a turbomachine. Recent turbine cascade tests of note have been performed by Sjolander (1975), Aronov et al. (1975), Langston (1977), Hodson (1984), Moore (1984), Patterson and Hoeger (1986), Hodson and Dominy (1987), Yaras (1987), Vijayaraghavan and Kavanagh (1988), Hodson and Addison (1988) and Yamamoto (1988). This appears to be a substantial amount of available data but actually each of these authors focused on a different aspect of the flow so there is still a great need for continued wind tunnel testing. In terms of need it might be compared to external aerodynamic model tests of aircraft where literally hundreds of different investigations are necessary to provide a sufficient data base for current design methods.

Dixon (1978) gives a good description of two dimensional cascade testing. He explains that performance data are obtained from measurements of pressure, flow angle, and velocity taken across one or more pitches at entry to and at exit from the cascade. A typical traverse result across two blade pitches is shown in Figure 2.1.1 from Todd (1947) as reproduced by Dixon (1978). From these results, mass averaged total pressure loss and fluid outlet angles may be determined. Similar tests performed for a large number of incidences at the same inlet Mach number and Reynolds number will define the complete performance of the cascade (at that M_i and Re). Cascade performance results may be easily represented on a single graph of total pressure loss and fluid deflection versus incidence as shown in Figure 2.1.2 from Ainley (1948).

A complete review of cascade design and a more detailed description of cascade testing techniques is given in section 2.3

2.1.2.2 PROFILE LOSS CORRELATIONS

According to Dixon (1978), the purpose of correlating the data of various cascade tests is to provide a method which enables engineers to predict the first-order behaviour of the blade losses and limiting fluid deflection with sufficient accuracy.

Most correlations are based on an empirical approximation of a given turbine loss coefficient. This empirical formula is derived from cascade test results. The purpose is to develop an easy-to-use method for predicting solutions to complex thermodynamic relationships. Various assumptions and approximations are used to ensure that the correlation is as closely related to the loss coefficient in question as is possible.

One of the most widely used loss correlations methods is that by Ainley and Mathieson (1951). This method predicts the total pressure loss and fluid deflection for each row of a turbine stage at a mean reference diameter. The method is graphical and includes a comprehensive means for calculating the effect of incidence on profile losses. It also includes a correlation for the effect of trailing edge thickness on the losses. The method is based on a Reynolds number of 2×10^5 and a rough estimate of performance at other Reynolds numbers is given by a simple power law. Profile loss at zero incidence is assumed to be a function of gas and blade angles, pitch to chord ratio, and thickness to chord ratio as described by the following:

$$Y_{p(i=0)} = \left\{ Y_{p(\beta_1=0)} + \left(\frac{\beta_1}{\alpha_2} \right)^2 \left[Y_{p(\beta_1=\alpha_2)} - Y_{p(\beta_1=0)} \right] \right\} \left(\frac{t_{max}/C}{0.2} \right)^{\beta_1/\alpha_2} \quad (2.2)$$

Figure 2.1.1 Sample Plot of Total Pressure Loss and Fluid Outlet Angle by Todd (1947)
(Reproduced from Dixon (1978))

Figure 2.1.2 Sample Plot of Variation in Profile Loss with Incidence by Ainley (1948)
(Reproduced from Dixon (1978))

The accuracy of the method is limited by the fact that it is based on turbine design of the 1950's.

Denton (1973) reviewed and compared seven different correlation methods for predicting the profile loss of turbine blades. His study included methods by Ainley and Mathieson (1951), Balje and Binsley (1968), Bolte (1964), Craig and Cox (1970), Deich (1953), Denton (1971) and Traupel (1966). The losses predicted by the different correlations were found to vary over a fairly wide range. Figures 2.1.3 and 2.1.4 from Denton's paper (1973) show a variation of ± 35 per cent of the mean value of the losses at $\beta_1 = 90^\circ$ and ± 50 per cent at $\beta_1 = 150^\circ$ (measured from the tangential direction).

Denton also compared the known results from 79 cascade tests to the losses predicted by these seven methods. He found that all the methods give poor agreement with experimental results. Balje and Binsley's and Ainley and Mathieson's methods gave the least scatter but the mean predicted value of the latter is much too high. On this basis Denton concluded that Balje and Binsley's method was the best. This method is based on a calculation of the momentum thickness of the boundary layers at the trailing edge by assuming a linear variation of surface velocity from inlet to outlet on both surfaces and integrating Truckenbrodt's (see Schlichting (1962)) turbulent boundary layer equation along the surfaces. Denton (1973) pointed out that use of the method is rather limited by the small scale of the graphs provided but it is formulated so that it is easily programmed on a digital computer. Unfortunately, Balje and Binsley's method does not include a correlation for the incidence losses.

Sieverding (1985) also gives a comprehensive review of axial turbine performance prediction methods. Both Sieverding's and Denton's study emphasize once more the need for updating correlation methods. An integral part of this process is continued cascade testing of turbine blades of modern design.

Denton discovered one other shortcoming of the correlation methods he studied. The basic philosophy of the methods is that the loss is mainly dependent upon the inlet and outlet blade angles and is not greatly influenced by the detailed blade shape. The large scatter of the predictions of the different methods and the poor agreement with experimental results suggest that this may not be true. Figure 2.1.5 indicates a scatter of roughly $\pm 25\%$ in the losses predicted by a boundary layer calculation method for blades with similar inlet and outlet angles but different profile shapes. New correlations should account for the effect of blade shape parameters, such as leading edge diameter.

Figure 2.1.3 Predicted Loss of Blades With $\beta_1 = 90^\circ$
(Reproduced from Denton (1973))

Figure 2.1.4 Predicted Loss of Blades With $\beta_1 = 150^\circ$
(Reproduced from Denton (1973))

Figure 2.1.5 Calculated Losses of Blades With Similar Inlet and Outlet Metal Angles but Different Shape (Reproduced from Panton (1973))

wedge angle and loading distribution on the losses. This is particularly true for off-design conditions where the blade performance becomes increasingly dependent on the blade loading which is directly influenced by blade shape parameters. An incidence loss correlation by Mukhtarov and Krichakin (1969) which has been significantly improved by Moustapha, Kacker and Tremblay (1989) does account for the effect of blade shape on both two dimensional and secondary incidence losses.

2.1.2.3 BOUNDARY LAYER CALCULATION METHODS

In order to calculate the losses based on boundary layer parameters it is necessary to predict the development of the boundary layer on the blade surfaces. The first step is to obtain the blade surface velocity distribution, second, to compute the boundary layer momentum thickness at the trailing edge and third, to calculate the resulting entropy generated. According to Denton (1973), there are several boundary layer calculation methods which can predict the momentum thickness to within $\pm 5\%$ provided the boundary layers are either completely laminar or completely turbulent. The difficulty arises in predicting the location of the boundary layer transition point. For the case where laminar separation occurs, the point of separation can be accurately predicted by Thwaites (1949) method, however, the length of the resulting separation bubble is very difficult to predict.

Figure 2.1.6, from Walker (1968), shows the stages of boundary layer development on a typical blade suction surface. Walker concluded that: 1) Extensive regions of laminar flow may exist on turbomachine blading in spite of the high level of disturbance from the wakes of upstream blade rows. 2) The laminar boundary layer development on the blade surface is predicted well by Thwaite's method. 3) Transition is dominated by the streamwise pressure gradient. It should therefore be possible to obtain a reasonable estimate of the transition location on a turbomachine blade provided the surface pressure distribution is known. 4) Head's entrainment method (1960) predicts the turbulent boundary layer development to within tolerable accuracy but appreciable errors arise due to uncertainty about starting values for this calculation. 5) It should be possible to calculate blade profile losses with an accuracy of $\pm 10\%$, using present methods.

Since 1968, with the advances in understanding of basic boundary layer transition mechanisms and greater speed and capacity for numerical calculations several boundary layer calculation codes have been developed. One such code has been developed at Carleton University. It employs Thwaites's method for prediction of the laminar boundary

Figure 2.1.6 Blade Surface Boundary Layer Development
(Reproduced from Walker (1968))

layer development and Head's entrainment method for the turbulent region of the boundary layer. The accuracy of the current methods in predicting the losses in a turbomachine is still hampered by the limits of complexity of the methods.

Sharma and Graziani (1983) proposed that in a real machine the endwall flow would have a significant effect on the airfoil midheight boundary layer development. They cite Langston's (1977) conclusions: 1) viscous flow from the regions near the endwall on the suction surface migrated toward the midheight region of the cascade, 2) installation of a fence on the airfoil reduced this migration and 3) the importance of this effect must be considered whenever cascade data is used to obtain information at midheight locations. Sharma and Graziani proposed that the magnitude of this effect could be incorporated into an existing boundary layer calculation method by Crawford and Kays (1976). Their results show a substantial increase in accuracy for predicting the midspan losses in a low aspect ratio cascade via boundary layer calculation.

The conclusions of Denton and Cumpsty (1987) are instructive in understanding the state of the art in both two dimensional and three dimensional loss prediction by calculation methods. They state that efficiency prediction will continue to be based on correlations of experimental data and that the methods will continue to improve based on a qualitative understanding of the flow rather than quantitative predictions.

2.1.3 TRAILING EDGE LOSSES

Trailing edge losses comprise a very significant proportion of the total losses in many turbomachines. At the same time they are very difficult to measure or to predict. Surprisingly, in many cascade test results presented in the open literature the trailing edge thickness, which is the most appropriate scaling factor for trailing edge loss, is not even documented.

Trailing edge loss is related to the form drag experienced by any blunt body in a free stream. The flow separates from the surface of the body when the surface curves sharply away from the mean direction of the flow. This creates a region of low pressure and low momentum which must subsequently mix with the freestream and thus cause a loss in total pressure. Denton and Cumpsty (1987) define the base pressure coefficient as:

$$C_{pb} = (P_b - P_2)/(P_{02} - P_2) \quad (2.3)$$

The reason for the extreme difficulty in predicting the magnitude of the base pressure coefficient (C_{pb}) is that the base pressure (P_b) is strongly influenced by 1) the trailing edge geometry, 2) the mainstream flow field and 3) the state of the blade surface boundary layers at the trailing edge.

Measurement of the base pressure is possible in a cascade model with a sufficiently large scale to allow for a static pressure tap to be placed in the trailing edge of the blade. Denton (1973) refers to such a test being performed by Bammert and Fiedler (1966) in which the base pressure coefficient was measured to be 0.2.

A systematic investigation into the effect of trailing edge thickness on blade loss and stage efficiency was carried out by Johnston et. al. (1967). They predicted the total losses would be increased by up to 2 times due to an increase in trailing edge thickness of 400%. Their experimental results showed that their predictions were accurate for a single stage rotor but the stator loss coefficient increased by 1.8 times greater than they predicted (see Figure 2.1.7). These results do not explain the nature of the loss mechanisms at work but they do emphasize the magnitude of the effect of the trailing edge thickness on the machine efficiency.

Kacker and Okapuu (1982) developed a correlation for trailing edge loss based on published and in-house cascade results that were available at that time. The graphical representation of the correlation is shown in Figure 2.1.8. An investigation of the spanwise variation in the base pressure coefficient would be useful in understanding the flow mechanisms associated with trailing edge loss.

2.1.4 MIXING LOSSES

While most losses in a turbomachine are caused by the skin friction and pressure drag on the nozzles and rotors within the turbine and/or compressor stages of the machine, a significant proportion of the entropy generation actually takes place downstream of the blade rows. The non-uniform flow field which exits each blade row must mix in the flow passages downstream of the blades. In a multi-stage machine much of this mixing process will take place within the downstream blade rows. The conditions at the trailing edge plane and the area at the downstream mixed out plane must be specified in order to predict the magnitude of this loss. Creating a model of a non-uniform flow field entering each blade row and then predicting the exit flow field of that blade row to a point sufficiently far downstream that the losses are completely mixed out is extremely difficult. This point

Figure 2.1.7 Effect of Trailing Edge Thickness on Two Dimensional Loss Coefficients
(Reproduced from Johnson et. al. (1967))

Figure 2.1.8 Trailing Edge Loss Correlation
(Reproduced from Kacker and Okapuu (1982))

illustrates the complexity of the three dimensional flows within turbomachines.

Denton and Cumpsty (1987) showed that the losses created during mixing are highly dependent on the velocity at the point of mixing (see Figure 2.1.9). This suggests that the mixing losses may be reduced by accelerating the downstream flow as in a multi stage turbine. However, this model is only known to be valid for steady, two dimensional flow fields.

It can be seen that mixing loss may not be accurately predicted based on a single blade row cascade test. However, the mixing loss may not be ignored in cascade testing. It is important that the distance downstream of the blade row at which the total pressure is measured be documented so that the mixing losses can be estimated. A study of the variation of loss with downstream distance is included in the present work.

2.1.5 ENDWALL LOSSES

The two sources of endwall losses, namely annulus loss and secondary loss, are very closely related. Both arise from boundary layer growth on the endwalls of the machine and the interaction of the low momentum boundary layer fluid with the blade rows. The mechanisms which cause these losses to arise are illustrated in Figure 2.1.10 from Denton and Cumpsty (1987). Yaras (1987) gave a good description of the passage flow picture based on various secondary flow studies. He pointed out that the endwall boundary layer is rolled up into a vortex inside the passage which is located at the blade suction surface-endwall corner. Also, because of the interaction of the endwall boundary layer flow with the blade leading edge, a horseshoe vortex is formed which interacts with the passage vortex. This description of the secondary flow is illustrated in Figure 2.1.11 by Klein (1966) and Langston (1977) as reproduced by Sieverding (1985).

The three dimensionality of the endwall flow makes the secondary and annulus losses the most difficult to understand and predict. Until recently secondary losses were predicted based on cascade measurement results with little attention paid to the actual physics of the flow. Denton and Cumpsty (1987) cite two recent attempts to improve the physical basis of endwall loss prediction by Gregory-Smith (1982) and Sharma and Butler (1986).

Figure 2.1.9 Variation of Mixing Loss With Velocity at Point of Mixing and with Wake Depth
(Reproduced from Denton and Cumptsy (1987))

Figure 2.1.10 Loss Mechanisms on the Endwall of a Turbine Cascade (Reproduced from Denton and Cumptsy (1987))

**Figure 2.1.11 Endwall Flow Models by Klein and Langston
(Reproduced from Sieverding (1985))**

2.1.6 TIP LEAKAGE LOSSES

A gap must be maintained between the tip endwall and the blade tip within rotating blade rows of turbomachines. A small portion of the working fluid will leak through this gap rather than follow the blade passage. This tip leakage flow creates a total pressure loss when it mixes with the main stream fluid. In addition, the leakage flow has a strong radial component which increases the three dimensionality of the flow field near the blade tip. The magnitude of the losses due to tip leakage should be closely proportional to the flow rate of the leaking fluid. It is therefore common to employ such schemes as active clearance control or blade shrouds to reduce the leakage flow rate where feasible. Yaras (1987) gives a comprehensive review of tip leakage studies as well as extensive cascade test results which clarify the complex flow mechanisms involved.

2.2 OFF-DESIGN PERFORMANCE OF TURBOMACHINES

A given turbomachine is likely to spend some of its operating time at some point other than the design point around which the machine efficiency was optimized. According to Cohen et al. (1987) the requirement for an engine to perform at various operating points will occur for one of two reasons: 1) a change in the power requirement or 2) a change in the ambient conditions. The authors point out that both of these factors weigh heavily on the final design of an engine for a given application. The first reason for operating at off-design conditions determines the variation of specific fuel consumption with load requirement and may be referred to as part load performance. The second reason is particularly critical for aero engines which encounter a wide range of ambient conditions. Both factors will have a critical effect on both safety and economics. Accordingly, accurate predictions of the off-design performance of a turbomachine during the preliminary stages of design is crucial. A better understanding of the effects of blade and cascade geometry on the off-design losses is also valuable so that improved turbine performance at off-design conditions can be achieved by engine designers.

2.2.1 INCIDENCE EFFECTS

The most significant effect of operating at off-design conditions is the altering of the velocity triangles within the turbine and compressor stages of the engine. As pointed out by Moustapha, Kacker, and Tremblay (1989), this will cause a mismatch in the velocity vectors and leading edge angles of the blades. This results in a thickening of the blade

boundary layers which causes a direct increase in the profile losses and an indirect increase in the secondary losses.

Moustapha (1987) illustrated the effect of engine speed changes on the velocity triangles in a turbine. The losses increase more rapidly with positive incidence due to the increased blade loading as shown in Figure 2.2.1 from Hodson and Dominy (1987). The heightening of the suction peak causes a strong adverse pressure gradient so that the fluid near to the blade suffers a greater loss of momentum, or in other words, a thicker boundary layer arises.

The subsequent increase in losses due to the thickening of the blade boundary layers may be referred to as incidence losses. These are usually predicted by incidence loss correlations based on experimental cascade tests. As can be seen from the following review of incidence loss cascade tests and incidence loss correlation methods, substantial improvements in both the prediction of incidence losses and understanding of off-design loss mechanisms are needed.

2.2.2 CASCADE TESTS FOR INCIDENCE LOSSES

Cascade testing is recognized as a very reliable method for generating loss data at a relatively low cost. It is possible to design a cascade test section in which the incidence angle of the inlet flow can be systematically varied to find the variation of loss with incidence.

Moustapha, Kacker and Tremblay (1989) point out the need for an incidence loss cascade data base in the development of an incidence loss correlation. They cite eleven references from which data was compiled and subsequently used to propose an improved prediction method. Of these eleven only five other than this work were published within the last decade. Moustapha, Kacker and Tremblay (1989) showed the distribution of the available data for the variation of profile loss with incidence. There is a definite sparsity of data at higher positive and negative incidence angles. The authors also found a lack of incidence data related to the effects of Mach number, Reynolds number, turbulence level, leading edge geometry, exit flow angle and axial loading distribution. A short description of the work of each of the five most recent incidence loss cascade studies is given to indicate the amount of existing data.

Hodson and Dominy (1987) measured the total pressure loss for both midspan and

Figure 2.2.1 Variation of Blade Surface Mach Number Distribution with Incidence
(Reproduced from Hodson and Dominy (1987))

secondary flow at three Reynolds numbers, three incidences and three pitch to chord ratios. Their cascade had a design inlet angle of 38.8° and exit angle of -53.9° from axial. Incidence loss tests were performed at design incidence, $+8.6^\circ$ and -20.3° .

Patterson and Hoeger (1986) performed loss measurements on three blades with different axial loading distributions generated using a prescribed velocity distribution technique. The tests were performed at three Reynolds numbers and over a wide range of incidence and Mach number. The incidence loss data is not presented. The main focus of the study is the effects of Reynolds number and velocity distribution on two dimensional losses in an LP turbine cascade.

Yamamoto (1988) carried out total pressure loss measurements for a cascade with design inlet angle of 43.6° and outlet angle of -63.5° from axial. The measurements were taken at eight different incidences ranging from $+7.2^\circ$ to -53.3° . No separate measurement of the two dimensional losses was performed, however, a rough separation of the overall loss into the various loss components was presented.

Vijayaraghavan and Kavanagh (1988) also performed total pressure loss measurements on a cascade with inlet metal angle of -38.0° and exit angle of -23.7° from tangential. Testing was done at three incidences including design incidence, -7° and -14° . Their study concentrated on the measurement of profile losses. They employed endwall suction to achieve two dimensionality at midspan.

Finally, Hodson and Addison (1988) performed total pressure loss measurements on a single stage rotating test section with rotor blade inlet angle of 0.0° and exit angle of -65.0° from axial. The incidence on the rotor was varied by controlling the flow coefficient. Profile loss data was collected for eight different incidences ranging from $+60^\circ$ to -40° .

2.2.3 OFF-DESIGN LOSS CORRELATION METHODS

Of the seven different correlation methods investigated by Denton (1973) only two include a correlation of the losses vs. incidence. These are Ainley and Mathieson's (1951) and Craig and Cox's (1970) methods. This is indicative of the difficulty in predicting the off-design losses based on the inlet and outlet flow angles. As discussed above the incidence losses are a strong function of the detailed blade shape as well as the flow angles.

Ainley and Mathieson (1951) define a stalling incidence which is a function of blade inlet and exit angles and pitch to chord ratio. The incidence losses are then presented graphically as a function of an incidence to stalling incidence ratio. The correlation is based on turbine design of the 1950's. It also neglects the effects of blade and cascade geometry. Figure 2.2.2 and 2.2.3 show the relationship between Y_p and incidence according to Ainley and Mathieson.

Mukhtarov and Krichakin (1969) were the first to introduce the effects of blade geometry in an incidence loss correlation. Martelli and Boretti (1987) also considered the effect of leading edge geometry on cascade performance. Moustapha, Kacker and Tremblay (1989) reviewed both Ainley and Mathieson's and Mukhtarov and Krichakin's methods in detail. They subsequently proposed a new correlation which is based on data from 18 different cascade tests. Their system includes the effects of airfoil leading edge diameter and the degree of acceleration in the blade channel.

Further cascade studies are necessary in order to verify and update current correlation methods. Research into the effect of incidence on trailing edge loss, deviation, and other parameters are also needed.

2.3 CASCADE TEST SECTION DESIGN

Gostelow (1984) defines a cascade as an infinite row of equidistant similar bodies or blades which, in engineering practice, are used to divert a stream of fluid. In this work we are particularly interested in turbomachinery flows in which work is extracted from the working fluid by changing its angular momentum as it crosses a row of turbine blades. The flow in a turbomachine is modeled by considering a finite row of blades scaled to some appropriate size so that the detailed flow mechanisms may be more easily measured. The complexity of the test section will be discussed in the following section on versatility. The ability of the test section to produce data which is representative of the actual turbomachinery flow mechanisms will be discussed in the section on flow quality.

2.3.1 TEST SECTION VERSATILITY

Test section versatility is really a measure of the number of different types of data which may be extracted from a given cascade. Often the main purpose in cascade testing is to vary one geometric or aerodynamic parameter and study its effect on the associated

Figure 2.2.2 Determination of Stalling Incidence for Turbine Blades after Ainley and Mathieson (Reproduced from Moustapha et al. (1988))

Figure 2.2.3 Variation of Profile Loss With Incidence
After Ainley and Mathieson
(Reproduced from Moustapha et al. (1988))

losses. In other words, the most versatile cascades are those in which the loss mechanisms can be easily separated and studied individually for a given configuration while allowing for investigation of many different configurations. The parameters which are important to consider in this context are:

- 1) Reynolds Number
- 2) Mach number
- 3) Incidence
- 4) Cascade Geometry

The Reynolds Number is the governing parameter of the viscous losses in a turbomachine. In general the total loss decreases with increasing Reynolds Number. At very low Reynolds Number ($Re < 10^5$) the blade boundary layers are completely laminar. It is in this regime that the total loss is a strong function of the Reynolds Number. At high Reynolds Numbers ($Re > 10^6$) when the boundary layers are completely turbulent the total loss is nearly independent of the Reynolds Number.

Balje (1964) proposed that in many operating regimes the Reynolds Number can be neglected. He defined a viscous - nonviscous loss ratio where viscous losses may be interpreted as those losses which are dependent on Reynolds number and nonviscous losses as those which are independent of Reynolds number. Balje concluded that it is possible to estimate the influence of the Reynolds Number on the losses only after the factors which affect the above mentioned ratio are established.

Before 1964 the effect of Reynolds Number on the efficiency of a turbomachine was assumed to be adequately represented by a simple equation which expressed the losses as a function of the Reynolds number raised to some power multiplied by a constant. Bullock (1964), Holeski and Stewart (1964) and Smith (1964) all refuted the idea that a simple power law could be used to determine the effect of the Reynolds Number on the total losses in a turbomachine.

More recent loss correlation methods such as Denton (1973) and Kacker and Okapuu (1982) agree that the Reynolds Number has a significant effect on only one component of the total loss, namely the profile loss. Also, the effect of the Reynolds Number depends on the nature of the boundary layer. The critical Reynolds Number may be defined as the beginning of the transition regime for the blade boundary layer. This value will depend on the free stream turbulence intensity, free stream pressure gradient,

blade surface roughness and Mach number. In a typical cascade with low turbulence intensity ($Tu < 1.5\%$) the critical Reynolds Number is 2×10^5 . According to Sieverding (1985) most modern loss correlations neglect Reynolds Number effects above this value. It is good practice to identify the critical Reynolds Number for a given cascade to determine whether Reynolds Number effects are important. Figure 2.3.1 shows some results from Blight and Howard (1952) as referred to by Gostelow (1984) for the variation of loss with Reynolds Number.

A detailed description of the effects of Reynolds Number on the primary and secondary flow in a low pressure turbine rotor cascade is given by Hodson and Dominy (1987).

The effects of Mach number are beyond the scope of this work. The consideration of the Mach number is nevertheless extremely important since in most modern gas turbine engines some portion of the flow will experience compressibility effects. High speed cascade tunnels have therefore played an important role in understanding the effect of Mach Number on blade performance. Gostelow (1984) points out six uses for high speed cascade testing. These include:

- 1) to gain a physical understanding of shock-boundary layer interaction and base pressure modelling.
- 2) to provide data for comparison with flow computation methods for use in blade design.

The effects of incidence are discussed in the section on off-design performance of turbomachines. Cascade geometry includes the effects of thickness to chord ratio, leading edge radius, trailing edge thickness, camber and stagger angles, pitch to chord ratio, camber line shape and thickness distribution. A comprehensive review of the design application of these parameters is given by Gostelow (1984).

2.3.2 FLOW QUALITY

The main purpose of cascade testing is to model the actual flow in a turbomachine and measure the various loss coefficients which are pertinent to engine design. It is therefore critical that the cascade flow be representative of the actual flow mechanisms present in the blade passages of a real machine. There are various methods used by the cascade designer to achieve this goal. The parameters which need to be considered are:

Figure 2.3.1 Variation of Loss Coefficient with Reynolds Number According to Blight and Howard (1952)
(Reproduced from Gostelcw (1984))

- 1) Periodicity
- 2) Free stream turbulence
- 3) Inlet flow uniformity
- 4) Two dimensionality

Each of these are important to consider during design of a cascade test section. The tools employed to control these parameters are as follows:

Periodicity;

- 1) flaps to throttle the flow which passes through the outboard passage of the end blades to ensure that the stagnation point on the end blades coincides with the location of the stagnation point on the center blades.
- 2) tailboards to direct the outlet flow of the outboard blades parallel to the mean outlet flow direction.

Free stream turbulence;

- 1) turbulence generating grids such as employed by Vijayaraghavan and Kavanagh (1988).

Inlet flow uniformity;

- 1) settling chambers and screens to evenly distribute the mass flow and reduce peak total pressure regions in the flow field.

Two dimensionality;

- 1) boundary layer bleed slots or porous walls with suction boxes to reduce endwall boundary layer thickness.
- 2) endplates to provide a bypass space for the endwall boundary layers in the blade passages.
- 3) variable aspect ratio to increase the width of the region at midspan which is unaffected by the secondary flow.

Results of periodicity, free stream turbulence, inlet flow uniformity, and two dimensionality for the current work are presented in chapter 5.

3.0 APPARATUS DESCRIPTION

3.1 WIND TUNNEL

The cascade test section has been designed for use in the turbomachinery wind tunnel at Carleton University. It is an open jet circular tunnel with a centrifugal fan driven by a variable speed DC motor. Ambient air enters the fan through a bell mouth and is blown into a large settling chamber with three screens in series. A fiberglass circular to rectangular transition piece is used to connect the test section to the outlet of the settling chamber. The fan delivers up to 140 ft³/s at a total pressure of 4.7" of water. The maximum test section inlet velocity is 100 ft/s which, for the chosen blade chord, provides a maximum Reynolds number of 3.5×10^5 .

For a more complete description of the wind tunnel see previous work on the same tunnel by Amrud (1985) or Yaras (1987).

3.2 TEST SECTION

The new rectilinear cascade test section has been designed and constructed to meet the need for reliable turbine blade performance data. The major goal during design of the test section was to produce a relatively low cost tool for gaining a better understanding of the effects of various parameters on the flow mechanisms and associated losses in a turbine blade passage. In accordance with this goal the following geometric parameters have been chosen as variables:

- 1) Incidence
- 2) Pitch/chord Ratio
- 3) Tip Clearance
- 4) Aspect Ratio

In the following text each of these capabilities will be described in detail along with the innovative methods used to vary the parameters.

3.2.1 GENERAL DESCRIPTION AND PRINCIPLE DIMENSIONS

Table 3.2.1 and Figures 3.2.1 - 3.2.3 give the principle dimensions and layout of the test section. It is built on an angle iron frame with heavy duty casters to allow for

Figure 3.2.1 Variable Incidence Cascade Test Section

Figure 3.2.2.2 Portable Test Section Window

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Figure 3.2.7 Instrumented Blade Photograph

portability. The test section is built around the blade root endwall or backplate. The backplate is constructed of 0.75" thick aluminum plate. The blade tip endwall is a sheet of 0.75" thick plexiglass mounted on it's own portable frame to allow for easy access to the test section control surfaces.

Overall Length	75"
Overall Height	75"
Overall Width	75"
Inlet Passage Height	30"
Inlet Passage Width	8"
Incidence Variation	$\pm 45^\circ$

Table 3.1 Test Section Principal Dimensions

The flow through the test section is controlled by fine tuning of the flaps and tailboards shown in Figure 3.2.1. These control surfaces and the test section sidewalls are constructed from 0.5" thick aluminum plate. The inlet boundary layer may be bled off via the adjustable bleed slots on both endwalls downstream of the cascade inlet.

3.2.2 CASCADE GEOMETRY AND BLADE CONSTRUCTION

The blade which was chosen for this work is the midspan section of a power turbine blade from a medium sized turboprop engine. The characteristics of the cascade are shown in Figure 3.2.4. Main blade and cascade geometric parameters are given in table 3.2.2 and the blades are shown in Figure 3.2.5. The blade profile coordinates are contained in Appendix A. Past experience at Carleton University has shown that it is advantageous to use a large chord length (6.0"-10.0") for the model so that the various flow mechanisms are easily measurable. A chord length of 6.5" was chosen to achieve this goal without compromising the aspect ratio. The first series of tests concentrated on the profile losses. An aspect ratio sufficiently large to ensure 2-D flow at midspan was therefore necessary. For the chosen chord length and cascade passage height 5 blades were needed for the given pitch/chord ratio.

The cascade is designed so that different blade profiles may be tested. The blades are bolted to a separate 0.5" thick aluminum plate (see Figure 3.2.5) which is inserted into a precisely machined space in the dural disk. Various blades of virtually any profile shape

Figure 3.2.4 Cascade Geometry

INLET FLOW ANGLE	=	28.35°
EXIT FLOW ANGLE	=	57.65°
INLET METAL ANGLE	=	29.29°
EXIT METAL ANGLE	=	57.51°
DESIGN INCIDENCE	=	-0.94
DESIGN DEVIATION	=	-2.14
PITCH / CHORD RATIO	=	0.63
LEADING EDGE DIAMETER	=	9.43 mm
TRAILING EDGE THICKNESS	=	4.20 mm
STAGGER ANGLE	=	23.06°
MAXIMUM THICKNESS	=	29.63 mm
CHORD LENGTH	=	162.31 mm
AXIAL CHORD LENGTH	=	150.00 mm
ASPECT RATIO	=	1.23

Table 3.2.2 Blade and Cascade Geometry

Figure 3.2.5 Blade Assembly Photograph

and stagger angle up to a maximum chord length of 10.0 may be mounted in the test section. The separate mounting plate for which a replacement may be manufactured and put in place in a few hours also allows for variation in the pitch/chord ratio. This feature of the test section makes it possible to test the effect of virtually any geometric variable on the associated losses.

Each blade was constructed from a 2"x7"x8" block of aluminum. Aluminum was chosen because of its machinability. First the given coordinates for the profile were rotated to a more convenient axis for machining. Then a 1.0" wide template was machined on a vertical milling machine to precisely match the coordinates. The template was bolted to one of the aluminum blocks and the block and template were set up on the vertical milling machine with a pin through the horizontal axis. The blade was then rotated about this axis while successive passes were made with the cutting tool to follow the template (see Figure 3.2.6). This procedure combined with several hours of hand finishing for each blade produced a high quality finish to within 0.005" of the specified coordinates. This method was very cost effective for such a small number of blades. If a larger number of blades were required a numerically controlled milling machine would be more appropriate.

The center blade of the cascade was instrumented with 11 spanwise rows of 43 static pressure taps as shown in Figure 3.2.7. The rows are spaced across half the span in order to allow for investigation of spanwise variation in the blade loading distribution. A static tap is located in the base of the trailing edge for each of the spanwise rows. The base pressure data will be used in calculating the trailing edge loss.

3.2.3 VARIABLE INCIDENCE MECHANISMS

As shown in Figure 3.2.1 the incidence is varied by rotating the large dural disk. The disk is made from the same 0.75" thick aluminum as the rest of the backplate. The disk is fastened to a 2.0" diameter steel shaft which is supported by two pillow block bearings. The bearings are bolted to a support structure which is welded directly to the frame. The disk also bears on the fixed portion of the backplate. The bearing surface is a 0.5" thick strip of plastic which is fastened to the fixed backplate. A large protractor fastened to the back of the fixed backplate allows for angle adjustment to within 0.1 degrees. Once the disk is adjusted to the desired inlet flow angle a small screw type clamp holds it in place.

As the inlet flow angle varies the flaps and tailboards must be manipulated in order

1. Cutting Tool
2. Aluminum Block
3. Template
4. Support Bracket
5. Milling Machine Table

Figure 3.2.6 Blade Production Method

Figure 3.2.7 Instrumented Blade Photograph

to direct the flow through the cascade parallel to the sidewalls and endwalls and without accelerating or decelerating the flow upstream of the blade row. Seven different adjustable control surfaces are used to accomplish this task. These are described in detail in section 3.2.4.

3.2.4 WALLS, FLAPS AND TAILBOARDS

The seven control surfaces and sidewalls are shown in plan view in Figure 3.2.8. This arrangement was chosen for maximum flow control so that inlet and outlet flow periodicity were sure to be achieved.

The sidewall flaps are designed to control the flow over the top and bottom of the outboard blades. By throttling the flow over the pressure surface of the upper blade the dividing streamline which stagnates on the leading edge of that blade can be made to coincide with the location of the stagnation point on the center blades. Similar control is achieved by throttling the flow over the suction surface of the bottom blade. The sidewall flaps thus allow for simulation of an infinite cascade with a very small number of blades

There is a 20" long tailboard directly downstream of each of the outboard blades. These surfaces complement the sidewall flaps in simulating an infinite cascade. The tailboards help in achieving periodic flow by directing the streamlines parallel to the exit flow angle. The tailboards diverge a small amount to allow for boundary layer growth on the surface.

When the dural disk is rotated to change the incidence, it is necessary that the lower sidewall and flap be raised. In the past, variable incidence cascades have often employed parallelogram walls to achieve this. One of the more innovative features of this test section is the moveable sidewall which is attached directly to the turntable. The moveable sidewall pivots on a 1/2" diameter steel pin. As the turntable is rotated the sidewall is adjusted to remain parallel to the inlet flow. A simple claw probe on the leading edge of the moveable sidewall is used to ensure that there is no flow angularity which might cause separation near the leading edge. The flow angularity at the leading edge is controlled by a simple pivoting throttle flap located underneath the moveable sidewall. This arrangement allows for up to 18" of vertical displacement of the lower sidewall. This is sufficient to accommodate ± 45 degrees of incidence variation.

The final control surface is a simple pivoting bleed valve located in the lower

Figure 3.2.8 Test Section Flow Control Surfaces

sidewall. This valve remains closed except at high values of negative incidence when a large portion of the inlet flow must be vented through the bottom of the test section.

3.2.5 VARIABLE ASPECT RATIO

The test section is designed for investigation of two-dimensional profile losses as well as three-dimensional secondary and total losses. In order to accommodate blades of various pitch/chord ratios and also to be able to separate the different loss mechanisms the aspect ratio may be varied by increasing the blade height.

The blade height is increased by adding a new section to the existing blades. The blade extension is supported by a 3/4" steel tube which extends from the backplate through the center of both blade sections. The tube is then bolted directly to the backplate. Two 1/4" locating pins are used to ensure an accurate alignment of the blade sections.

The passage height is varied by extending the sidewalls, flaps and tailboards to the required height. The extensions are fastened directly to the existing framework. The sidewall framework extends to twice the original blade height to allow for modification of the aspect ratio.

When the blade height is increased the inlet flow is diffused by a variable plane wall diffuser located directly upstream of the test section. Preliminary design for the diffuser was carried out according to performance characteristics from Reid (1953) and Engineering Science Data Unit 74015 (1974).

4.0 INSTRUMENTATION AND ERROR ANALYSIS

4.1 THREE HOLE PROBE

In order to evaluate the performance of the cascade in terms of aerodynamic two dimensional losses it is necessary to measure the mass averaged total pressure loss coefficient across the cascade. This is carried out according to the following procedure:

- 1) Measure the inlet total pressure with a pitot probe.
- 2) Measure the outlet total, static and dynamic pressure and flow angle by traversing in a pitchwise direction with a three hole probe across one blade pitch.
- 3) Mass average the total pressure loss at each point and nondimensionalize it with respect to the outlet dynamic pressure.
- 4) Integrate the total pressure loss coefficients across one blade pitch to find the mass averaged total pressure loss coefficient.

The three hole probe is the primary measuring device in this procedure. The probe is as shown in Figure 4.1.1. Pressure coefficients are formed from pressure measurements of each of the three holes, two of which will always be oblique to the oncoming flow. By calibrating the probe according to the procedure outlined in the following section the total, static and dynamic pressure and the flow angle can be extracted from the measured data.

The probe is mounted in a holder which is attached to a 0.25" diameter steel rod which is perpendicular to the probe. The traverse gear moves the rod in either a pitchwise or spanwise direction. The probe is mounted so that the probe tip is 69 millimeters upstream of the center of the rod to insure that the probe holder and rod do not influence the flow being measured.

The probe is small enough (0.7 mm x 2.0 mm) so that detailed measurements of the blade wakes can be taken with good resolution. The effect of a highly sheared flow on the accuracy of the probe measurements is addressed in section 4.1.2. The probe was ideally suited to two dimensional loss measurements where the angles of the flow vectors measured only vary by less than $\pm 5^\circ$.

Figure 4.1.1 Three Hole Probe

4.1.1 THREE HOLE PROBE CALIBRATION

The three hole probe is calibrated in a flow for which the total pressure, velocity and flow direction are known and at the same Reynolds number as the probe will encounter in the wind tunnel. The calibration apparatus is shown in Figure 4.1.2. The probe which was used for this work was previously calibrated by Yaras (1987). During the commissioning process of the new test section the probe was recalibrated using the same procedure as Yaras. Measurements were taken over a range of $\pm 70^\circ$ at one degree intervals. The data was then used to form the following coefficients as defined in a method for three hole probe data reduction by Lewis (1966) :

$$K_1 = (P_1 - P_s)/q \quad (4.1.1)$$

$$K_{12} = (P_1 - P_2)/q \quad (4.1.2)$$

$$K_{13} = (P_1 - P_3)/q \quad (4.1.3)$$

$$K_{23} = (P_2 - P_3)/q \quad (4.1.4)$$

$$K_2 = (P_1 - P_2)/(P_1 - P_3) \quad (4.1.5)$$

$$K_3 = (P_1 - P_3)/(P_1 - P_2) \quad (4.1.6)$$

These coefficients are subsequently used to calculate the flow angle, dynamic pressure and static pressure as described in section 4.2.1.

4.1.2 THREE HOLE PROBE MEASUREMENT ACCURACY

The three hole probe provides very accurate data for the following reasons:

1) In determining the flow angle from the raw data there are two coefficients which may be used. Only one of the curves need to be used. It is therefore possible to choose the curve whose slope is closest to 1.0 at that particular point. Each data point is therefore unique and independent of all other points which insures good resolution and less possibility of error in the data reduction process.

2) In determining the dynamic pressure a similar choice between coefficients is possible in order to ensure that a number close to zero does not appear in the denominator.

Figure 4.1.2 Calibration Apparatus Photograph

This reduces the possibility of instability or excessive errors in certain flow angle ranges.

3) The three hole probe is relatively sturdy and insensitive to hours of use or handling in comparison to other anemometry devices. The probe calibration was checked after 40+ hours of use and it was found that there was no detectable shift in the calibration curves. It is, of course, true that care must be taken in handling to ensure that the probe tip is not damaged or altered in any way. Occasional cleaning of the probe is also necessary to remove any dirt particles which could alter the calibration curves.

As previously mentioned the probe used for this work was recalibrated during the commissioning of the test section. The calibration curves which are shown in Appendix A are curve fitted by the method of least squares. The order of the polynomials formed which describe these curves varies from 6 to 8. Each section of each curve shown in Appendix A corresponds to a separate polynomial. All of the coefficients for each polynomial form an array which is used by the data reduction program to determine the value of the flow angle, dynamic pressure and static pressure for any given data point. The source code for the data reduction program can be found in Appendix C of Yaras (1987).

4.1.2.1 FLOW ANGLE

The errors in the flow angle measurements were investigated by Yaras (1987) for the same probe used in this work. He found that the calculated flow angle agreed to within $\pm 0.5^\circ$ of the known angle for pitch angles within a $\pm 30^\circ$ range and $\pm 1.0^\circ$ outside this range. It is important to note that this error is not in the form of scatter but is a systematic shift in the results. The effect of such a shift in the calculated flow angle on the mass averaged total pressure loss coefficient was evaluated and was found to vary by less than 0.02 %.

For the purpose of this work where only two dimensional losses are being measured all of the flow angles measured are within a range of $\pm 5.0^\circ$.

One other possible source of error in measuring the flow angle across a blade wake may arise due to the nature of the flow. In a highly sheared flow because of the finite width of the probe there will be a difference in the dynamic pressure across the face of the probe. This could be misinterpreted by the probe as flow angularity. Since the flow angle is expected to vary across the blade wake a simple check on the pitchwise variation of the flow angle would not reveal the source of the apparent flow angularity. However, we would not expect the static pressure to vary in the pitchwise direction across the wake since

the flow angles only vary by $\pm 2.0^\circ$ depending on the distance downstream of the trailing edge at which the measurements are taken. By checking the pitchwise variation in the static pressure measured by the probe we can determine if there is any error due to the sheared flow. It was found that when traversing very close to the trailing edge there is some variation in the static pressure across the wake but it disappears when the probe is moved out to 40% C_x downstream of the trailing edge where some mixing has already taken place. An investigation into the effect of downstream distance on the measured losses was carried out and the results are presented in section 6.1.2. In summary it was found that the total loss which includes the measured loss and the calculated mixing loss did not vary with downstream distance. It can be concluded from this that the nature of the flow did not cause errors to arise in the flow angle measurements.

4.1.2.2 DYNAMIC PRESSURE

Yaras (1987) measured the magnitude of the error in the measured dynamic pressure. The errors were found to be proportional to the error in flow angle measurement. For the range of $\pm 30^\circ$ the maximum error in the dynamic pressure was less than 1.0%. This would result in a similar error in the measured total pressure loss coefficient since they are normalized on the dynamic pressure.

It is believed that the improved data acquisition system and data sampling procedures have resulted in increased accuracy in the three hole probe measurements. All of the measured values are very accurate with little scatter as shown by the results in chapters 6 and 7.

4.2 DATA ACQUISITION SYSTEM

The layout of the data acquisition system and probe traverse system are shown in Figure 4.2.1. Both systems are integrated and controlled by the computer so that the data acquisition process is highly automated and efficient. The data acquisition system consists of:

- 1) an electronic measurement device which receives analog DC voltage signals from the pressure transducers and converts them to a digital signal.
- 2) a CPU module which controls the measurement device and communicates the digital signals from the measuring device to the serial port of the microcomputer.
- 3) a microcomputer which controls the CPU module, receives the digital signals from the measuring device, calculates the pressures from the transducer calibration factors and

Figure 4.2.1 Data Acquisition System for Three Hole Probe Traverse Measurements

averages them over a given sampling period. The computer then stores the data in a file according to a preset format.

The electronic measurement device is a Sciometric Instruments Model 81. It is a general purpose measuring system capable of direct measurement of a wide range of sensor outputs with no extra hardware or signal conditioning. The A/D converter is a 12 bit-plus sign, dual slope integrator. The 12 bits result in an integration resolution of 1 in 4096 or 0.024% of full scale. The measurement resolution is 0.00122 Volts divided by the gain value. The gain value settings are discussed in section 4.2.1.

The CPU module / Serial Interface is a Sciometric Instruments Model 901. It provides an ASCII based command set (52 different commands) for operating the model 81 measurement system. The module may be controlled through the serial port on a personal computer.

4.2.1 THREE HOLE PROBE SAMPLING AND DATA REDUCTION

The three hole probe sampling and data reduction procedure was originally designed by Yaras (1987) and has been slightly modified for the purpose of this work. As shown in Figure 4.2.1 each of the three ports on the probe is connected to a separate pressure transducer. Tygon tubing was used throughout with metal or plastic connectors to eliminate the possibility of leaks. The reference pressure for each of these ports is the test section inlet static. The fourth transducer is connected to the inlet total pressure pitot tube and is also referenced on the inlet static therefore giving a direct measurement of inlet dynamic pressure. Each of these pressure differences is sampled to determine the appropriate gain for that particular transducer during that sampling period. The gain is programmable and can be used to eliminate the possibility of poor resolution if one of the probe pressures is very close to the inlet static pressure. Once the gain is set, the sampling period of 45 seconds begins. At the end of the sampling period the data is sent to the computer where it is averaged and stored. The computer then automatically steps the traverse gear to the next measurement location.

Following the data acquisition the data must be reduced in order to find values of loss coefficients, flow angles, Reynolds number, etc. The data reduction program forms the coefficients listed in section 4.1.2 and the calibration curves shown in Appendix A are used to find values of total, dynamic and static pressure and flow angularity at the test section inlet and outlet. The corresponding non-dimensional coefficients are then

calculated. The output of the data reduction program consists of several files containing both raw data and reduced data. The file containing the total pressure loss coefficients is used as input to a subsequent program which mass averages and integrates across one blade pitch to find the overall mass averaged total pressure loss coefficient. Another similar program is used to determine the mass averaged deviation. The code for these programs is shown in Appendix C.

4.2.2 BLADE STATIC PRESSURE MEASUREMENTS

The blade static pressure data acquisition arrangement is shown in Figure 4.2.2. The blade static pressure ports are connected to the Scanivalve which is connected to one pressure transducer. The reference pressure and inlet total and dynamic pressures are the same as for the three hole probe measurements. The data sampling is done as for the three hole probe except instead of stepping the traverse gear after each measurement the computer steps the Scanivalve to the next pressure port. There are 11 rows of 43 static pressure taps on the blade surface. One of the taps in each row is located in the trailing edge so that the base pressure may be measured. Repeatability tests by Yaras (1987) showed that blade static pressure measurements were repeatable to within ± 0.015 %.

Figure 4.2.2.2 Data Acquisition System for Blade Static Pressure Measurements

5.0 TEST SECTION COMMISSIONING

5.1 TUNNEL OPERATING POINT

The wind tunnel was originally instrumented so that the settling chamber contraction pressure difference could be used to define the tunnel operating point. The test section was calibrated so that the inlet dynamic pressure could be inferred from the pressure difference in the contraction as described by Yaras (1987). However, with the new test section, due to the changes in acceleration through the blade row as the incidence is changed, in addition to the boundary layer bleed slots being located at the test section inlet, the test section could not be calibrated in this way. For this reason, the inlet total pressure was measured directly with a small pitot tube placed downstream of the boundary layer bleed slots and directly across from the inlet static pressure tap. By assuming uniform inlet flow it was possible to measure the total, dynamic, and static pressure at inlet with these two instruments. The uniformity of the test section inlet flow was measured and is discussed in section 5.3.

The tunnel operating point is specified in terms of the Reynolds number based on the test section inlet velocity, blade chord, atmospheric pressure, and settling chamber stagnation temperature. The actual design point Reynolds number in the engine from which this blade was taken varies from 1.5×10^5 at sea level to 0.7×10^5 at an altitude of 25,000 feet. It should be remembered that the critical Reynolds number is significantly lower in the engine than in the cascade due to a much higher freestream turbulence level within the actual engine passages. Reynolds number tests were performed to determine the critical Reynolds number for the cascade as described in Chapter 6. The operating point is above the critical Reynolds number so that the total pressure loss across the cascade is only slightly dependent on the Reynolds number. The critical Reynolds number is 2.0×10^5 and the operating point is $3.0 \times 10^5 \pm 0.1 \times 10^5$.

5.2 FREESTREAM TURBULENCE INTENSITY

The freestream turbulence intensity is one of the important parameters in determining the location of the boundary layer transition point on the blade suction surface. For this reason it is an important parameter to consider when measuring the two dimensional losses. The turbulence intensity is defined as the root mean square of the fluctuating velocity component in the longitudinal direction divided by the center line

velocity (u'/V_∞) and is usually expressed as a percentage. Yaras (1987) measured the freestream turbulence intensity for the present test section configuration and obtained a value of 1.5%.

5.3 INLET FLOW UNIFORMITY AND BOUNDARY LAYER THICKNESS

Inlet flow velocity profiles which indicate the uniformity of the flow as well as the thickness of the inlet boundary layer for various bleed slot settings are shown in Figure 5.3.1. The exact thickness and nature of the test section inlet boundary layer is not extremely important due to the use of endplates in taking total pressure loss data as described in section 5.4.3. The purpose of bleeding the boundary layer at inlet was to minimize the layer thickness without severely lessening the mass flow so that the endplates could be placed as close to the endwall as possible. For this reason bleed slot setting number 2 represented in Figure 5.3.1 was chosen.

Figure 5.3.1. indicates that there is some spanwise variation in the total pressure at inlet. This variation has been substantially removed by the time the flow reaches the cascade inlet which is approximately 5 chord lengths downstream of the test section inlet. These results are shown in Figure 5.3.2.

Pitchwise uniformity is also very important. Figures 5.3.3 and 5.3.4 shows midspan pitchwise traverse results taken at 142 % upstream of the leading edge of the cascade. The flow uniformity at inlet to the cascade is closely related to the test section periodicity. As mentioned in the apparatus description, the flow periodicity is controlled by the seven control surfaces surrounding the cascade. Inlet flow periodicity test results for the effects of each of the control surfaces on the inlet flow are presented in the following section.

5.4 PERIODICITY

Perhaps the most time consuming and difficult task in cascade commissioning is establishing periodic flow. It is particularly important when measuring the two dimensional losses in a test section with only 5 blades to ensure that the flow is similar in each passage or the resulting data would be of little value.

Before setting the periodicity it was necessary to investigate the effect of each of the seven control surfaces on the periodicity. This was done by systematically stepping each control surface and recording the corresponding change in periodicity.

Figure 5.3.1 Boundary Layer Bleed Spanwise Inlet
Flow Velocity Profiles

Figure 5.3.2 Spanwise Inlet Flow Velocity Profiles

Figure 5.3.3 Pitchwise Inlet Flow Velocity Profile

Figure 5.3.4 Pitchwise Inlet Flow Angle Profile

The periodicity was set according to the following procedure and criteria:

- 1) The control surfaces were adjusted until the blade passage endwall static taps (two taps in each passage, one close to the suction surface and one close to the pressure surface, in the exact same location in each passage) indicated that the pressure difference across each blade was similar. The taps were connected to a bank of manometers and the pressure differences were measured in inches of water. In some cascade tests such as Amrud (1985) this was the only criteria for periodicity.
- 2) The three hole probe was used to traverse in the pitchwise direction downstream of the blade row at midspan with a large step size to determine which control surface requires fine adjustment. This was repeated until the flow angle from midpassage to midpassage varied by less than $\pm 0.5^\circ$.
- 3) The three hole probe was used to traverse upstream of the blade row at midspan in the pitchwise direction to ensure that the oncoming flow angle was within the range of $0.0 \pm 0.5^\circ$. The upper and lower flaps were fine tuned and this step was repeated until this criteria was met. Results from this test are shown in Figure 5.3.4.
- 4) Finally the three hole probe was used to traverse in the pitchwise direction downstream of the blade row at midspan with a small step size. The total pressure loss coefficient for each blade was determined. If the flow angles were still within the acceptable range and the loss coefficients were similar then periodicity was assumed acceptable for taking loss data. Results for this final step at design incidence are shown in Figure 5.4.1.

This procedure was carried out rigorously for the design incidence setting until periodicity was achieved. With a change in the incidence setting it was expected that changes in deviation and acceleration through the cascade would require fine tuning of the periodicity setting. Fine tuning was carried out for incidence settings of $+15.0^\circ$ and -15.0° . The results are shown in Figures 5.4.2 and 5.4.3.

5.5 TWO DIMENSIONALITY OF FLOW

The main emphasis of this work is the effect of incidence on the two dimensional

Figure 5.4.1 Design Incidence Pitchwise Total Pressure
Distribution at 41° Axial Chord Downstream
of Trailing Edge

Figure 5.4.2 Positive Incidence Pitchwise Total
Pressure Distribution at 41% Axial Chord

Figure 5.4.3 Negative Incidence Pitchwise Total
Pressure Distribution at 41½ Axial Chord

Figure 5.5.1 Spanwise Two Dimensional Flow Region
Without End Plates

total pressure losses. It is therefore imperative that there be a wide region of flow at midspan which is uninfluenced by the secondary flow. The criteria for two dimensionality is that the losses within the region at midspan are constant in the spanwise direction. The width of this region is determined by traversing at several spanwise locations. Blade suction surface oil flow visualization is also used to demonstrate the width of the two dimensional region.

As was mentioned in the literature review, Sharma and Graziani (1983) showed that even if the losses at midspan are constant in the spanwise direction there will still be some migration of fluid toward midspan within the boundary layer and therefore the secondary flow will still influence the losses. This means that the two dimensional region at midspan must be a significant proportion of the span. From the results given by Sharma and Graziani 40% of the span was chosen as the criteria for two dimensionality.

Another point to consider is that as the incidence is changed, particularly in the positive direction, the blade loading will increase as will the strength of the secondary flow. The two dimensionality must therefore be checked when the incidence is increased to determine the reduction in the width of the two dimensional region.

The test section was designed for doing secondary flow and tip clearance studies as well as two dimensional loss measurement. For this reason, porous endwalls were not used as are sometimes employed in profile loss measurement to remove the endwall boundary layer and reduce the strength of the secondary flow. The width of the two dimensional region was determined for the test section with no mechanism for reducing the influence of the secondary flow other than the boundary layer bleed slots as mentioned in section 5.2. The two dimensional region was approximately 20% of the span as shown in Figure 5.5.1.

It was decided that some mechanism for increasing the width of the two dimensional region of the flow was necessary. A very innovative solution was arrived at to accomplish this with a minimum of modification to the test section. Adjustable endplates were designed which would fit over the blades and be fixed close to each endwall. The endplates allow the endwall boundary layers to pass in the space between the plates and the endwalls. Since the plates are adjustable they can easily be moved to correspond with the thickness of the boundary layer on that wall. The plates are shown in Figure 5.5.2.

A comparison of the spanwise two dimensional pressure loss variation with the endplates and without is shown in Figure 5.5.3. The reduction in the influence of the

Figure 5.5.2 End Plate Photographs

Figure 5.5.3 Comparison of Spanwise Two Dimensional Flow Region With and Without End Plates

secondary flow is also shown in the flow visualization pictures of the middle blade suction surface both with and without the endplates (see Figure 5.5.4).

Figure 5.5.4 Suction Surface Flow Visualization With and Without End Plates

6.0 PRESENTATION AND DISCUSSION OF DESIGN POINT MEASUREMENTS

6.1 TWO DIMENSIONAL LOSSES

The mass averaged total pressure loss was measured at midspan. The two dimensionality of the flow at midspan was established as described in the preceding chapter. The following values that are presented therefore represent the two dimensional mass averaged total pressure loss which includes the profile loss and trailing edge loss. Figure 6.1.1 shows the wake of the center blade and gives the value of the total pressure loss at 42% Axial Chord downstream of the trailing edge, at a Reynolds number of 3.0×10^5 . The effects of Reynolds number and downstream traverse distance are presented in the following sections.

6.1.1 EFFECT OF REYNOLDS NUMBER ON TWO DIMENSIONAL LOSSES

The effect of Reynolds number on the losses was determined by systematically varying the inlet velocity and measuring the corresponding loss coefficients. The experimental results are shown in Figure 6.1.2. These results are quite similar to those by Blight and Howard (1952) which are included in the literature review. They indicate a critical Reynolds number, based on the inlet velocity, of approximately 2×10^5 for this cascade at design conditions. As a result, most measurements were performed at a Reynolds number of 3.0×10^5 to ensure that the loss coefficients are only slightly dependent on the Reynolds number. The Reynolds number for this blade in its actual application varies from 0.7×10^5 to 1.5×10^5 at a much higher freestream turbulence intensity than in the cascade test section as discussed in Chapter 5. The present loss measurements should therefore be reasonably representative of the actual engine losses.

6.1.2 EFFECT OF DOWNSTREAM TRAVERSE LOCATION

The effect of the downstream traverse location on the measured loss is really a measure of the mixing that has taken place between the trailing edge and the point of measurement. At the trailing edge the entropy generated is a result of skin friction on the blade surfaces only. The boundary layer and finite trailing edge thickness cause blockage which produces a region of low momentum fluid downstream of the blades. This region is called the wake. As the flow moves downstream of the trailing edge the freestream fluid

Figure 6.1.1.1 Design Incidence Total Pressure Loss at
Midspan at 41% Axial Chord Downstream of
Leading Edge

Figure 6.1.2 Variation of Losses With Reynolds Number

mixes with the fluid in the wake until some point far downstream where the wake is no longer evident and the flow is said to be completely mixed out. This process of mixing causes a further increase in entropy. The process can be analyzed using the control volume shown in Figure 6.1.3.

A mixing loss coefficient, Y_m , may be evaluated which may then be added to the measured loss coefficient to obtain a total two dimensional mixed out pressure loss coefficient. The following derivation shows how the mixing loss coefficient is determined.

First, consider the control volume shown in Figure 6.1.3. Assume incompressible flow and that the mixing occurs at constant area. Apply the linear momentum equation in the x-direction:

$$\Sigma F_x = \int_{out} u d\dot{m} - \int_{in} u d\dot{m}$$

where:

$$d\dot{m} = \rho u dA \text{ and}$$

$$dA = dy \text{ (for unit depth in } z \text{ direction).}$$

Also assume the static pressure is constant across planes 1 and 2 indicated in Figure 6.1.3. We can therefore write:

$$P_1 s - P_2 s = \int_0^s \rho u_2^2 dy - \int_0^s \rho u_1^2 dy$$

which may be simplified to:

$$P_1 s - P_2 s = \rho u_2^2 s - \rho s \int_0^1 u_1^2 d(y/s).$$

This may be shortened by replacing the integral in the equation with the following variable:

$$I_n = \int_0^1 u_1^n d(y/s).$$

We may then write equation 6.1:

$$P_1 - P_2 = \rho u_2^2 - \rho I_2 \quad (6.1)$$

Figure 6.1.3 Two Dimensional Blade Wake Control Volume

From the continuity equation we can write:

$$\dot{m}_1 = \dot{m}_2$$

and for the assumptions mentioned above this may also be written:

$$\int_0^s u_1 dy = \int_0^s u_2 dy$$

which may be simplified to form equation 6.2:

$$\int_0^1 u_1 d(y/s) = u_2 \quad (6.2)$$

By combining (6.1) and (6.2) we may write:

$$P_1 - P_2 = \rho I_1 - \rho I_2$$

or, by normalizing on the downstream dynamic pressure:

$$\left(\frac{1}{2} \rho u_2^2 = \frac{1}{2} \rho I_1^2 \right)$$

we can write an expression for the static pressure loss coefficient between planes 1 and 2:

$$C_{p12} = \frac{P_1 - P_2}{\frac{1}{2} \rho I_1^2} = \frac{\rho I_1^2 - \rho I_2^2}{\frac{1}{2} \rho I_1^2}$$

which may be simplified to form equation 6.3:

$$C_{p12} = 2 \left(1 - \frac{I_2^2}{I_1^2} \right) \quad (6.3)$$

We can now write an expression for the mass averaged total pressure at plane 1:

$$\bar{P}_{01} = \frac{1}{\dot{m}} \int P_{01} d\dot{m}$$

which can be rewritten as:

$$\bar{P}_{01} = \frac{1}{\bar{m}} \int_0^s (P_1 + \frac{1}{2} \rho u_1^2) \rho u_1 dy.$$

Then, by separating the integral and rearranging we get:

$$\bar{P}_{01} = \frac{1}{\bar{m}} (\rho P_1 s \int_0^1 u_1 d(y/s) + \frac{1}{2} \rho^2 s \int_0^1 u_1^3 d(y/s)).$$

The integrals may then be replaced by the variable defined above in order to obtain equation 6.4:

$$\bar{P}_{01} = P_1 + \frac{1}{2} \rho \frac{I_3}{I_1} \quad (6.4).$$

Finally we can define the mixing loss coefficient as:

$$Y_m = \frac{\bar{P}_{01} - \bar{P}_{02}}{\frac{1}{2} \rho u_2^2}.$$

Then by substituting in equation 6.4 and the components of the total pressure at plane 2 as defined by the energy equation we obtain:

$$Y_m = \frac{P_1 + \frac{1}{2} \rho \frac{I_3}{I_1} - (P_2 + \frac{1}{2} \rho I_1^2)}{\frac{1}{2} \rho I_1^2}$$

which can be rearranged and combined with equation 6.3 to obtain equation 6.5:

$$Y_m = 1 - 2 \frac{I_2}{I_1^2} + \frac{I_3}{I_1^3} \quad (6.5).$$

This integration is performed numerically with measured values of u_1 . Measurements were taken at four locations downstream of the trailing edge and the corresponding mixing loss coefficients were calculated. The results are shown in Figure 6.1.4. As is expected, the sum of the measured loss and the mixing loss is nearly a constant and is therefore independent of the traverse location. The error in the data point at 20% C_x downstream of the trailing edge is explained in section 4.1.2.1. The mixing loss is found to be approximately 4% of the total loss when traversing at 41% C_x downstream of the trailing edge. This indicates that traverse results for two-dimensional losses for which the distance downstream of the trailing edge is greater than about 40% of the axial chord include a large enough portion of the mixed out loss to be deemed acceptable without further calculation. Hodson and Dominy (1987) traversed at nearly the same location and found that the mixing loss accounted for approximately 15% of the total (profile and secondary) mixed out loss. In this case the mixing calculation is needed in order to find the total mixed out loss. This calculation is necessary so that cascade measurements from different cascades which may have been performed at different downstream locations can be properly compared and combined to both test and improve existing loss correlation methods.

There is a definite need for some criteria which would clearly show if the mixing loss calculation is necessary based on the relative magnitude of the mixing loss and the measured loss for a given set of data. An analysis of the mixing out of a simplified blade wake produced useful conclusions about the variation of two dimensional loss and mixing loss with wake width and velocity deficit. The complete analysis is contained in Appendix D. It was found that, in general, the static pressure will always rise during the mixing process provided it occurs at a constant area. It was also found that the ratio of mixing loss to total loss actually decreases with increased wake width. This is true because the magnitude of the mixing loss is nearly independent of the wake width even though a wider wake signifies increased two - dimensional loss at the trailing edge plane.

This analysis could be used to determine whether the mixing loss calculation described above is necessary for a given set of data. For the present work the measured data at three incidences was compared with the results of the analysis for the appropriate velocity deficits. The analysis confirmed the conclusion that at all three incidences the mixing loss is a small enough percentage of the total loss to be assumed negligible. These results are also shown in appendix D.

2 of/de 2

MicroD

Figure 6.1.4 Design Incidence Two Dimensional Loss Variation With Traverse Location

6.2 BLADE PRESSURE DISTRIBUTION

The blade pressure distribution is very useful in determining the relative magnitude of the losses generated at a given point on the blade surface. Denton and Cumptsy (1987) showed that the entropy generation in the blade boundary layer is approximately proportional to the cube of the velocity at that point.

The blade pressure distribution at midspan for the design condition is shown in Figure 6.2.1. It is apparent that this blade has a very uniform loading distribution with no strong suction peak or adverse pressure gradient at design conditions. This would cause the suction surface boundary layer to remain relatively thin and we would therefore expect the two - dimensional losses at design conditions to be relatively low. This is shown to be true in the comparison of the loss measurements with existing correlations presented in chapter 8. This relationship between the suction surface pressure distribution and the boundary layer thickness is discussed in more detail in section 7.2.

6.2.1 REYNOLDS NUMBER EFFECTS ON BLADE LOADING

The effect of the Reynolds Number on the blade loading distribution is shown in Figure 6.2.2. The Reynolds number has little effect on the blade surface velocity distribution at design conditions. We do know, however, that the critical Reynolds number is 2×10^5 . We can conclude that the onset of transition near the leading edge which occurs at the critical Reynolds number does not significantly affect the loading distribution.

6.3 EXIT FLOW ANGLE AND DEVIATION

The exit flow angle traverse results are shown in Figure 6.3.1. The measured exit flow angle was subtracted from the exit metal angle and the mass averaged deviation was calculated. According to a correlation by Carter and Hughes (1950) as quoted to by Horlock (1966) this deviation is quite close to the predicted value. The correlation states that the unstalled 'low speed' deviation for turbine cascades may be written as a function of the camber angle. According to the correlation the mass averaged deviation for this cascade should be 6.4° which agrees quite well with the measured deviation of 5.3° .

Figure 6.2.1 Design Incidence Midspan Static Pressure Distribution

Figure 6.2.2 Design Incidence Blade Loading Variation
With Reynolds Number

Figure 6.3.1.1 Design Incidence Pitchwise Exit Flow Angle and Deviation

7.0 PRESENTATION AND DISCUSSION OF OFF-DESIGN MEASUREMENTS

7.1 VARIATION OF TWO DIMENSIONAL LOSS WITH INCIDENCE

In order to define the complete performance of the cascade at a given Reynolds number and inlet Mach number it is necessary to perform loss measurements at various inlet flow angles. A plot showing the variation of loss with incidence defines the off-design performance of the cascade.

The present measurements were of a preliminary nature and do not cover the full range of incidence for which the test section was designed. The purpose was to perform measurements at one positive and one negative incidence to:

- 1) demonstrate the capability of the test section to produce off-design loss data.
- 2) establish the required control surface adjustments for achieving periodicity and two dimensional flow at off-design conditions.
- 3) to provide preliminary data on the incidence characteristics of this cascade and draw some preliminary conclusions about its off-design performance.

Total pressure loss measurements were performed at $+15^\circ$ and -15° incidence settings. The results are shown in Figure 7.1.1. There was a large increase in the two dimensional losses at positive incidence and a relatively smaller increase at negative incidence. The reason for this trend is discussed in detail in the following section on blade surface pressure distribution at off-design conditions. The mixed out losses as predicted by the calculation presented in chapter 6 are also shown. These results are compared to the Ainley and Mathieson (1951) off-design loss correlation in the following chapter.

7.2 VARIATION OF PRESSURE DISTRIBUTION WITH INCIDENCE

The effect of incidence on the blade surface static pressure distribution is important for observing the severity of the leading edge overspeed and the resulting adverse pressure gradient and possible separation bubble, particularly at positive incidence. The results are shown in Figure 7.2.1. Other points of interest which are observable in the figure are discussed below.

There is a large overall increase in acceleration at positive incidence as contrasted by

Figure 7.1.1.1 Two Dimensional Total Pressure Loss Variation With Incidence

Figure 7.2.1 Midspan Static Pressure Distribution
Variation with Incidence

the small decrease at negative incidence. This is shown by the shift in the static pressure coefficient at the trailing edge. This shift is not apparent in the results of Hodson and Dominy (1987) as shown in Figure 2.2.2. This is because Hodson and Dominy used the exit dynamic pressure as a reference whereas the inlet dynamic pressure is used in this work.

There is a movement of the stagnation point around the leading edge due to the change in inlet flow angle. This in part explains the severe suction peak at positive incidence. The suction surface flow experiences a fairly large change in direction around a large proportion of the leading edge radius. This means a rapid acceleration of the flow will occur which results in a sharp suction peak and a strong adverse pressure gradient. The opposite is true for the negative incidence case where the stagnation point moves around the leading edge so that there is less of the leading edge radius for the flow to encounter. This causes the acceleration to be very gradual as shown in the figure.

Figure 7.2.1 indicates the likely reason for the large increase in two dimensional losses at positive incidence. The suction surface boundary layer will thicken very rapidly after it passes the region of high velocity near the leading edge due to the strong adverse pressure gradient. As mentioned in the previous chapter, the generation of entropy or losses in a blade boundary layer at a given point on the blade surface is very strongly influenced by the velocity at that point.

Conclusions about the qualitative performance of the cascade at off-design conditions and design considerations for improving the off-design performance are included in chapter 9.

7.3 VARIATION OF DEVIATION WITH INCIDENCE

Another important consideration when investigating the off-design performance of a turbine cascade is the variation of flow deviation with incidence. Deviation is the difference between the exit flow angle and the blade exit metal angle. As expected, the deviation increases with increased flow turning as shown in Figure 7.3.1. This is particularly important in a multi stage turbine where the increased deviation on the first stage rotor will alter the inlet flow conditions for the second stage. The design point deviation predicted by Carter and Hughes (1950) as presented in chapter 6 is also shown in the figure.

Figure 7.3.1 Midspan Deviation Variation With Incidence

A correlation of deviation at off-design conditions based on recent cascade data is necessary. At the present time the effect of incidence on the deviation in a turbine cascade is not well documented and a comprehensive study of the same would prove valuable

8.0 COMPARISON OF RESULTS WITH NUMERICAL CALCULATION AND CORRELATION METHODS

This chapter presents comparisons of the measured blade static pressure distributions with panel method calculation results as well as comparisons of the measured loss coefficients with off-design correlation methods.

8.1 COMPARISON WITH PANEL METHOD PREDICTIONS

The purpose of comparing the measured results with an inviscid panel method calculation scheme is to investigate the ability of the method to predict the off-design pressure distributions around the blades. The likely role of the blade pressure distribution was briefly discussed in the last chapter. If the panel method predicts pressure distributions reasonably reliably, then it could be a useful tool for investigating the off-design behaviour of alternative blade shapes. Promising shapes identified with the panel method might then be investigated experimentally. It could also be a useful tool for helping to interpret off-design loss measurements for blades where loading measurements might not be available. This particular panel method uses vortex sheets of linearly varying strength. The program was developed by Rahman (1986) as a fourth year project.

The panel method results are compared to the measured results for incidence settings of 0° , $+15^\circ$ and -15° in Figures 8.1.1, 8.1.2 and 8.1.3 respectively. The method does not precisely duplicate the measured results but the incidence effects are modelled quite well. The predicted trends in the leading edge acceleration and the resulting suction surface pressure gradient are correct, however, the leading edge overspeed is over-estimated, particularly on the blade pressure surface. This suggests that there may be a laminar separation bubble developing on the pressure side of the blade at negative incidence. The panel method could be a useful tool in determining the effects of design modifications on the cascade off-design performance.

8.2 COMPARISON WITH EXISTING CORRELATION METHODS

The measured loss coefficients have been compared with Ainley and Mathieson's (1951) and Moustapha, Kacker and Tremblay's (1989) off-design loss correlation methods. There are two reasons for the comparison. First, to show the difference between the performance of early turbine cascades and more modern designs and, secondly, to show the difference between the measured loss and the loss predicted by

Figure 8.1.1 Measured Results Compared to Panel Method Results For Design Incidence

Figure 8.1.2 Measured Results Compared to Panel Method
Results for +15° Incidence

Figure 8.1.3 Measured Results Compared to Panel Method
Results for -15° Incidence

the most widely used and the most recent off-design loss systems.

Figure 8.2.1 and Figure 8.2.2 show comparisons between the measured data and the two loss systems mentioned. In the first figure the losses are presented as total pressure loss coefficients. The measured and predicted design point losses as used in the figure are shown in table 8.2.1. It should be pointed out that prediction of the two-dimensional losses with the Ainley and Mathieson (1951) system is difficult because the trailing edge loss component of the total loss is multiplied by all of the other loss components. The second figure shows the losses represented as the change in the kinetic energy loss coefficient at off-design conditions. The first figure is a comparison of the complete loss systems and the second is a comparison of the incidence loss systems only.

	Y_{2D}
MEASURED LOSS	0.0213
PREDICTED LOSS:	
AINLEY AND MATHIESON	0.0417
AMDCKO*	0.0254

* AINLEY AND MATHIESON, DUNHAM AND CAME, KACKER AND OKAPUU

TABLE 8.2.1 DESIGN POINT TWO-DIMENSIONAL LOSS COEFFICIENTS

The design losses predicted by Ainley and Mathieson are significantly higher than the measured loss. This was expected due to the advances in blade performance since the 1950's when Ainley and Mathieson formulated their correlation. The measured values agree quite closely with Moustapha, Kacker and Tremblay's correlation.

Figure 8.2.1.1 Comparison of Total Measured Pressure Loss
With Ainley and Mathieson (1951)
Correlation and Moustapha, Kacker and
Tremblay (1989) Correlation

Figure 8.2.2 Comparison of Measured Incidence Loss With
Ainley and Mathieson (1951) and Moustapha,
Kacker and Tremblay (1989)

9.0 CONCLUSIONS AND RECOMMENDATIONS

9.1 CONCLUSIONS

The main conclusion drawn from the literature review which led to this work was the lack of recent off-design turbine cascade performance data. Accordingly, an extensive investigation into the off-design performance of axial flow turbines was initiated. The present work represents the first stage of that investigation. The main aim of this work was to design, construct and test a variable incidence cascade test section.

The test section commissioning demonstrated that a useful tool for cascade performance measurement has been successfully designed and constructed. The following specific conclusions are drawn:

- 1) A two dimensional region of approximately 40% of the span can be established with the use of endplates. This is sufficient to ensure that the endwall flow did not influence the two-dimensional loss measurements.
- 2) Excellent periodicity was achieved with the use of the seven flow control surfaces.
- 3) The modular design of the test section and an incidence range of 90° make the test section very versatile. It is possible to systematically vary the incidence, pitch/chord ratio, aspect ratio and tip clearance or to install new turbine or compressor blades with little modification to the test section.

The following conclusions can be drawn from the tests conducted:

- 1) For the present cascade, the critical Reynolds number is about 2.0×10^5 . Since Reynolds numbers of greater than 3.0×10^5 are available in the test section, it is possible to obtain results which are representative of the actual Reynolds number behaviour of the present blades.
- 2) The two-dimensional total pressure loss coefficient was measured and the corresponding mixing loss was calculated for four downstream locations. The results show good agreement in the total mixed out loss coefficient (the sum of the measured and the calculated mixing loss) independent of the traverse location. This calculation was found to

be unnecessary for two-dimensional loss measurement but has been shown to be important for total loss measurement.

3) The design incidence blade surface static pressure distribution shows a well designed blade with a very mild suction peak and adverse pressure gradient. The Reynolds number had no significant influence on the static pressure distribution.

4) The static pressure distributions show a sharp suction peak and strong adverse pressure gradient at positive incidence. These trends are accurately predicted by the panel method precise.

5) It was shown that the measured losses are in good agreement with the Moustapha, Kacker and Tremblay (1989) correlation predictions but are markedly lower than Ainley and Mathieson (1951).

6) A trend of higher deviation at positive incidence and lower deviation at negative incidence was found.

9.2 RECOMMENDATIONS

It is recommended that testing be continued with this test section to build a strong data base and understanding of off-design flow mechanisms as follows:

1) The testing should first complete the systematic measurement of the variation of two-dimensional loss with incidence for the full range of the test section.

2) Next, the secondary and tip clearance flow should be investigated in order to complete the overall picture of the off-design performance for this particular cascade configuration.

3) The different cascade parameters such as pitch/chord ratio and aspect ratio should be investigated to determine their effect on the performance.

4) In conjunction with the experimental investigation a computational analysis of the performance of this particular cascade should also be carried out. The three dimensional, Navier-Stokes program of Dawes (1987), which is available in the department, would be appropriate for such a study.

5) These results should also be combined with a more extensive flow visualization study in order to provide a greater understanding of the flow mechanisms.

6) Finally, the data should be used to determine which parameters are the most appropriate for inclusion in an off-design loss correlation system. The data can then be used in the formulation of new correlations or in the updating of existing ones.

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APPENDIX A

TURBINE BLADE PROFILE COORDINATES

THE FOLLOWING COORDINATES REPRESENT THE
PROFILE OF THE BLADE TO BE USED IN THE WIND TUNNEL
TEST SECTION. THE COORDINATES ARE GIVEN HERE IN
INCHES.

	XU	YU	XL	YL
	0.0753	0.0372	0.0753	0.0372
	0.0489	0.0612	0.1057	0.0186
	0.0276	0.0897	0.1392	0.0060
	0.0119	0.1218	0.1743	0.0000
DATUM	0.0026	0.1563	0.2100	0.0008
→	0.0000	0.1918	0.2449	0.0082
	0.0041	0.2273	0.2778	0.0220
	0.0091	0.2500	0.3007	0.0342
	0.0201	0.2954	0.3470	0.0579
	0.0392	0.3624	0.4166	0.0921
	0.0685	0.4503	0.5098	0.1352
	0.1025	0.5363	0.6000	0.1775
	0.1615	0.6613	0.7333	0.2374
	0.2542	0.8201	0.9044	0.3132
	0.3618	0.9693	1.0711	0.3929
	0.4808	1.1094	1.2326	0.4477
	0.6110	1.2393	1.3895	0.5079
	0.7509	1.3588	1.5433	0.5628
	0.8992	1.4676	1.6932	0.6136
	1.0542	1.5659	1.8406	0.6589
	1.2988	1.6919	2.0549	0.7214
	1.5541	1.7923	2.2637	0.7740
	1.9056	1.8852	2.5344	0.8300
	2.2634	1.9315	2.7958	0.8722
	2.6197	1.9339	3.0495	0.9008
	3.0676	1.8965	3.2955	0.9170
	3.5008	1.8279	3.5347	0.9211
	3.9154	1.7355	3.7672	0.9142
	4.3096	1.6278	3.9935	0.8956
	4.6838	1.5105	4.2138	0.8634
	4.9331	1.3835	4.4285	0.8305
	4.9777	1.2636	4.6380	0.7839
	4.9020	1.1377	4.8424	0.7277
	5.1141	1.0114	5.0422	0.6634
	5.3165	0.8854	5.2376	0.5902
	5.5112	0.7596	5.4290	0.5090
	5.7006	0.6341	5.6171	0.4198
	5.8401	0.5392	5.7582	0.3478
	5.9637	0.4439	5.8939	0.2713
	6.0710	0.3798	5.9850	0.2177
	6.1630	0.3149	6.0760	0.1625
	6.2563	0.2501	6.1670	0.1054
	6.3281	0.2002	6.2347	0.0609
	6.3614	0.1754	6.2684	0.0380
	6.3852	0.1583	6.2909	0.0226
	6.3971	0.1506	6.3021	0.0147
	6.4131	0.1360	6.3213	0.0047
	6.4248	0.1177	6.3424	0.0000
	6.4314	0.0971	6.3641	0.0009
	6.4324	0.0754	6.3819	0.0074
	6.4277	0.0543	6.4031	0.0191
	6.4177	0.0350	6.4177	0.0350

APPENDIX B

THREE HOLE PROBE CALIBRATION CURVES

THREE HOLE PROBE CALIBRATION

THREE HOLE PROBE CALIBRATION

THREE HOLE PROBE CALIBRATION

THREE HOLE PROBE CALIBRATION

THREE HOLE PROBE CALIBRATION

THREE HOLE PROBE CALIBRATION

Three hole probe calibration
Curve fitted by least-square method

Three hole probe calibration

Curve fitted by least-square method

Three hole probe calibration

Curve fitted by least-square method

Three hole probe calibration

Curve fitted by least-square method

Three hole probe calibration

Curve fitted by least-square method

Three hole probe calibration
Curve fitted by least-square method

Three hole probe calibration

Curve fitted by least-square method

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Three hole probe calibration
Curve fitted by least-square method

Three hole probe calibration

Curve fitted by least-square method

Three hole probe calibration
Curve fitted by least-square method

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Three hole probe calibration

Curve fitted by least-square method

Three hole probe calibration

Curve fitted by least-square method

Three hole probe calibration

Curve fitted by least-square method

Three hole probe calibration

Curve fitted by least-square method

APPENDIX C

DATA REDUCTION SOURCE CODES

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APPENDIX D

ANALYSIS OF A SIMPLIFIED BLADE WAKE

MIXING OUT OF A SIMPLIFIED BLADE WAKE

First, consider a simplified blade wake with a constant velocity deficit:

The wake integrals may then be defined in terms of the velocity deficit and the relative wake width as follows:

$$\begin{aligned}
 I_n &= \int_0^1 u^n d(y/s) \\
 &= \int_0^{0.5 - w/2} v^n d(y/s) + \int_{0.5 - w/2}^{0.5 + w/2} (v - \Delta V)^n d(y/s) + \int_{0.5 + w/2}^1 v^n d(y/s) \\
 &= v^n (0.5 - \frac{w}{2}) + (v - \Delta V)^n \left[(0.5 + \frac{w}{2}) - (0.5 - \frac{w}{2}) \right] + v^n (1 - (0.5 + \frac{w}{2})) \\
 &= v^n (1 - w) + (v - \Delta V)^n w \quad (1)
 \end{aligned}$$

I_1 , I_2 , and I_3 may therefore be written:

$$\begin{aligned}
 I_1 &= v (1 - \Delta V/v) w \\
 I_2 &= v^2 (1 - w) + (1 - \Delta V/v)^2 w \\
 I_3 &= v^3 (1 - w) + (1 - \Delta V/v)^3 w
 \end{aligned}$$

From these equations we can derive expressions for the static pressure change from plane 1 to 2, the total two-dimensional loss at plane 1 and the mixing loss from plane 1 to 2.

(1) Static Pressure Change

First consider the static pressure change during the mixing process. From the analysis given in chapter 6 we may write:

$$\begin{aligned}
 C_{p,2} &= \frac{P_1 - P_2}{\frac{1}{2} \rho U_1^2} \\
 &= 2 \left(1 - \frac{I_2}{I_1} \right) \\
 &= 2 \left(1 - \frac{V^2 \left((1-w) + (1 - \Delta V/V)^2 w \right)}{V^2 (1 - \Delta V/V w)^2} \right) \\
 &= 2 \left(1 - \frac{(1-w) + (1 - \Delta V/V)^2 w}{(1 - \Delta V/V w)^2} \right) \quad (2)
 \end{aligned}$$

It can be seen from equation (2) that $C_{p,2} < 0$

if

$$\frac{(1-w) + (1 - \Delta V/V)^2 w}{(1 - \Delta V/V w)^2} > 1$$

or, in other words:

$$(1-w) + (1 - \Delta V/V)^2 w > (1 - \Delta V/V w)^2$$

which may be simplified to:

$$w < 1$$

Therefore, since $w < 1$ for all cases of interest, in general the static pressure rises during the mixing process.

(2) Total Pressure Losses

a. Two Dimensional Losses to plane 1

First, assume there is no loss outside the wake;

$$\bar{P}_{o\infty} = P_1 + \frac{1}{2} \rho V^2$$

and from the analysis in chapter 6, we can write:

$$\bar{P}_{0,1} = P_1 + \frac{1}{2} \rho \frac{I_3}{I_1}$$

$$\bar{P}_{0,\infty} - \bar{P}_{0,1} = \frac{1}{2} \rho (V^2 - \frac{I_3}{I_1})$$

The two-dimensional losses may therefore be written:

$$\begin{aligned} Y_{2D} &= \frac{\bar{P}_{0,\infty} - \bar{P}_{0,1}}{\frac{1}{2} \rho u_1^2} \\ &= \frac{\frac{1}{2} \rho \left(V^2 - \frac{V^3((1-W) + (1 - \frac{\Delta V}{V})^3 W)}{V(1 - \frac{\Delta V}{V} W)} \right)}{\frac{1}{2} \rho V^2 (1 - \frac{\Delta V}{V} W)^2} \\ &= \frac{W \left(1 - \frac{\Delta V}{V} \right) - \left(1 - \frac{\Delta V}{V} \right)^3}{\left(1 - \frac{\Delta V}{V} W \right)^3} \end{aligned}$$

There is no obvious way to simplify this, but we plot the relation for values of velocity deficit and wake width which are of interest. The figure is shown at the end of the analysis.

b. Mixing Loss

Again, from the analysis given in chapter 6, we know:

$$Y_m = 1 - 2 \frac{I_2}{I_1^2} + \frac{I_3}{I_1^3}$$

which may be rewritten as:

$$Y_m = 1 - 2 \left(\frac{V^2((1-W) + (1 - \frac{\Delta V}{V})^2 W)}{V^2(1 - \frac{\Delta V}{V} W)^2} \right) + \left(\frac{V^3((1-W) + (1 - \frac{\Delta V}{V})^3 W)}{V^3(1 - \frac{\Delta V}{V} W)^3} \right)$$

$$= 1 - 2 \left(\frac{((1-W) + (1 - \frac{\Delta V}{V})^2 W)}{V^2(1 - \frac{\Delta V}{V} W)^2} \right) + \left(\frac{((1-W) + (1 - \frac{\Delta V}{V})^3 W)}{(1 - \frac{\Delta V}{V} W)^3} \right)$$

As before this relation may be presented on a graph which would show the relative magnitude of the mixing and total pressure losses for cases of interest.

RESULTS

Table D1 shows the measured two-dimensional losses and the calculated mixing losses as well as the velocity deficit values for the three incidences at which data was taken. The variation of loss with wake width for each of these cases is shown in figure D1

INCIDENCE	Y_{2D}	Y_m	Y_{TOTAL}	$\Delta V/V$
DESIGN	0.0213	0.0008	0.0221	0.0863
+15	0.0339	0.0014	0.0353	0.1418
-15	0.0241	0.0005	0.0246	0.0897

As shown in the figure, the mixing losses tend to dominate the loss picture for large values of wake width and velocity deficit. However, as mentioned in the discussion contained in chapter 6, the relative magnitude of the mixing losses actually decreases with increasing wake width. It is apparent from the analysis that the mixing losses are very small compared to the two-dimensional losses for these particular measurements.

Figure D1 Mixing Out of a Simplified Blade Wake -
Variation of Y_{2D} with Wake Width for Specified
Wake Velocity Deficits

END

1|8.0|2.90

FIN