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**TIP LEAKAGE IN A PLANAR CASCADE
OF
TURBINE BLADES**

Kim K. Amrad

*Thesis submitted in partial fulfillment of
the requirements for the degree of
Master of Engineering*

**DEPARTMENT OF MECHANICAL AND AERONAUTICAL ENGINEERING
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Abstract

An extensive survey of the published literature on tip leakage in turbomachines was carried out. It was found that there is not a full understanding of the fluid dynamics of the flow field in the tip region of rotor blades. Consequently, a rectilinear cascade test section was designed and fabricated to further pursue this subject. Versatility, so that the facility could be used for other flow related studies, was a prime consideration in the design. Calibration experiments confirmed that the flow quality in the test section was acceptable for its intended purposes.

In order to gain a better insight into the tip region flow, flow visualization experiments were carried out using surface oil flow and smoke. The results showed the formation of a core of highly rotational fluid near the blade tip and indicated that the leakage flow is primarily pressure driven for normal values of clearance. The location and size of the core was dependent on the gap size. In addition, the apparent occurrence of a second distinct tip leakage vortex was noticeable for the larger gaps.

Detailed blade unloading experiments as a function of clearance height/chord ratio were carried out. The blade loading distribution curves were strongly dependent on the clearance height. Computed blade lift coefficients show a continuous reduction in the midspan value for increasing gap size. However, there was a peak in the lift and normal blade force coefficients very close to the blade tip before the eventual decline right at the tip. In the case of the larger gaps, this peak was significantly higher than the midspan value.

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Nomenclature

A/D	-	Analog to Digital Converter
B.L.	-	Boundary Layer
C	-	Nominal Blade Chord
Cl	-	Clearance Height (mm)
C _A	-	Axial Force Coefficient
C _N	-	Normal Force Coefficient
C _P	-	Pressure Coefficient ($\frac{P - P_{static}}{(P_{static})_{DYN}}$)
H	-	Shape Factor (Boundary Layer Parameter)
h	-	Blade Height (mm)
L.E.	-	Leading Edge of Blade
P	-	Static Pressure (Pa)
P _{DYN}	-	Dynamic Pressure (Pa)
P _{PS}	-	Bottom Wall Static Pressure near Pressure Surface of Blade (Pa)
P _{SC1}	-	Static Pressure at End of Settling Chamber Contraction (Pa)
P _{SC2}	-	Static Pressure at start of Settling Chamber Contraction (Pa)
P _{SS}	-	Bottom Wall Static Pressure near Suction Surface of Blade (Pa)
Re	-	Reynolds Number
T.E.	-	Trailing Edge of Blade
U	-	Mean Axial Velocity (m/sec)
u'	-	Fluctuating Component of Axial Velocity (m/sec)
V	-	Velocity (m/sec)
W	-	Cascade Width (mm)

X	-	Streamwise Direction (mm)
Y	-	Channel-wise Direction (mm)
Z	-	Spanwise Direction (mm)
δ	-	Boundary Layer Thickness (mm)
δ^*	-	Displacement Thickness (mm)
θ	-	Momentum Thickness (mm)
η	-	Efficiency

Subscripts

bl	-	Boundary Layer
cline	-	Centre line at Test Section Inlet
CP	-	Centre of Pressure
DYN	-	Dynamic Pressure
Inlet	-	Inlet to Test Section
loc	-	Local
m	-	Mean
midspan	-	Midspan of Test Section
o	-	Value at Zero Clearance
PS	-	Pressure Surface
SC	-	Settling Chamber
SS	-	Suction Surface
TS	-	Test Section
w	-	Wall
wall	-	Topwall of Test Section
2D	-	Midspan (two-dimensional) value at zero clearance

Chapter 1

INTRODUCTION

The continuing pressure to produce even more efficient gas turbine engines has resulted in a resurgence of interest in tip leakage flows. Some of the classical and most enlightening studies in this field were done in the 1950's and 1960's, notably those of *Rains* (1954), and *Lakshminarayana* and *Horlock* (1963, 1967). There was a lull in activity until recently in this area, mainly because more urgent and important flow related problems, such as secondary flows, had to be dealt with. However, with significant progress having been made on many of these problems, there is renewed interest in tip leakage flows.

In keeping with the current trend and with the intention of making a meaningful contribution in resolving the tip leakage problem, a two-dimensional rectilinear cascade test section was designed and constructed. The design was centred around the study of tip leakage flows, but an attempt was also made to make the test section as versatile as possible. An investigation into the effect of clearance gap size on blade loading distribution was carried out for a row of turbine blades. In addition to this, a qualitative study of the flow field in the tip region was done using flow visualization.

Flow in the rotor tip region of a turbomachine accounts for a large percentage of the overall losses of a stage. Not only does the leakage of a particular stage affect the performance of that stage; but it also leads to off-design inflow conditions in later stages of a multi-stage machine. This results from the interaction of the leakage flow through the clearance gap and the mainstream flow. Experiments have also shown that the gap size affects the stall margin in compressors (*Greitzer*, 1980), as well as the loading distribution along the blade span (*Lakshminarayana* and *Horlock*, 1963).

The flow field at the tip of a rotor blade involves the interaction of many different

flow phenomena. The mechanics of the flow that results in a performance loss are not understood in detail, but it may be assumed that the contributing factors include:

- (1) The pressure difference between the the pressure and suction surfaces of the blade;
- (2) The presence of the endwall boundary layer;
- (3) The relative motion between the blade and casing;
- (4) The clearance gap geometry;
- (5) The existence of radial pressure gradients in the blade passage.

A detailed discussion of the basic flow mechanisms listed above, as well as an overview of the relevant published material is contained in Chapter 2.

Tip losses tend to be higher in turbines than in compressors due to the increased gap size at high operating temperatures. This results from the engine casing expanding more than the rotor blades. However, temperatures do reach considerable levels at the back end of high pressure ratio compressors and tip losses can become relatively significant. Active clearance control is employed in some engines, especially the larger ones with sophisticated control systems, to counteract this problem. However, in smaller engines it is not possible to justify the cost of employing such technique. In any case, there will be tip leakage regardless of the gap size and an understanding of the flow is necessary for design purposes.

To determine the status of research into the tip leakage problem, an extensive survey of the published literature was conducted. It was found that there is a lack of knowledge and full understanding of the actual physics of the leakage flow. Many of the experiments that have been done were of the *black-box* type, with tip clearance height being the input and overall stage performance the output. Consequently, it was decided to construct a rectilinear cascade test section which could be used to further pursue this problem. This section was designed and built for use with the existing open jet wind tunnel in the Hydraulics Laboratory at Carleton University.

The apparatus, which is described in detail in Chapter 3, consists of a single row

of turbine vanes of 250 mm chord and 200 mm span. The geometric specifications from which the blade shapes were generated were provided by Pratt & Whitney Canada Inc. Based on a chord of 250 mm, realistic Reynolds numbers of $4-5 \times 10^5$ using air at room temperature are achievable. It must be noted that the compressibility effects are not simulated since the section inlet velocity is limited to about 30 m/sec. Other idealized features of the current arrangement include the absence of relative motion between the blade and endwall, and the centrifugal effects that exist in actual machines.

As mentioned earlier, the overall leakage flow in a machine is a very complex issue. Therefore, it would be advantageous to isolate the flow phenomena individually, starting with seemingly important ones, before considering the resultant flow.

Various experiments were run to show the acceptability of the flow quality in the test section. The description of these experiments as well as the results of these experiments are documented in Chapter 4. Flow visualization was conducted in the blade tip region using smoke and oil. The effect of clearance height on blade unloading was also investigated. This was done by using a blade instrumented with static taps which allowed detailed blade pressure measurements. The tip leakage related experiments are described in Chapter 5, while their results are presented and discussed in Chapter 6.

Chapter 2

REVIEW OF PREVIOUS WORK

Leakage flow between the rotor tip and the endwall in a turbomachine is influenced by both viscous and inertia forces. The driving force of the flow is the pressure difference between the pressure and the suction surfaces of the blade. However, there are other flow mechanisms that act concurrently which affect the overall leakage flow. There is the secondary flow which results from the cross-channel pressure gradient caused by the turning of the flow by the blades. This pressure gradient causes the lower momentum fluid in the endwall boundary layer to be turned more than the mainstream flow, resulting in the endwall fluid being swept over to the suction surface of the adjacent blade. This swept fluid rolls up into a vortex along the suction surface of the adjacent blade. As will be evident later in this chapter, previous studies have suggested that the tip leakage flow does eventually roll up into a vortex with a streamwise component of vorticity. One question which then arises is whether the leakage flow vorticity is influenced or caused by the secondary flow.

The relative motion between the rotor and the casing endwall results in a layer of fluid being transported across the rotor tip. In a blade-tip reference frame, the boundary layer on the endwall is dragged over the rotor tip. In the case of a compressor, the scraping flow augments the leakage flow due to the pressure difference between the pressure and suction surfaces. The reverse occurs in a turbine. The work of Phillips and Head (1982) suggests the formation of a vortex on the pressure surface of the rotor blade in a compressor.

Resistance to the leakage flow in the clearance gap must be considered when looking at the leakage flow through the clearance gap. Naturally, a passage that offers high resistance will result in a lower leakage flow. Blade tip and endwall treatments have been and are currently being investigated. These treatments consist of shrouds, slots and grooves of

various configurations. The idea here is to minimize the leakage mass flow by maximizing the resistance to the leakage flow. To maximize the resistance to the flow, its direction in the clearance gap must be known. In addition to being used to reduce tip losses in turbines, casing treatments are also employed in compressors for improving stall margins. A graphical summary of the different flows in the tip region of a compressor rotor is given in figure 2.1.

As mentioned in Chapter 1, a survey of the published literature was carried out to determine the present status of tip leakage research. It was found that the traditional approach for determining tip clearance losses in turbomachines has been based upon measuring the effect of clearance height on the overall machine performance. These studies provided, at best, a semi-empirical relation between gap size and machine losses.

The early works of *Lakshminarayana* and *Horlock* (1962, 1967) are two of the classic studies done in this field. The latter investigation was carried out in a compressor cascade test section. There are many geometric similarities between compressors and turbines in the region of the tip gap. Therefore, in cascade studies with no relative movement between vanes and endwall, trends and patterns should be similar.

In the review of previous works which follow, some of the more enlightening studies of tip leakage flows in both compressors and turbines are discussed. Emphasis is placed on works which are directly relevant to the present study. During the course of the review a large number of references were examined. Those which appear to contain useful results have been included in the list of references, even though they are not explicitly cited in the present text. The list of references can therefore be used as a fairly comprehensive bibliography on tip clearance flows.

2.1 Compressor Cascade Studies

In an attempt to isolate the individual flow phenomena that occur in the tip leakage problem, *Lakshminarayana* and *Horlock* (1967) carried out a study using a two-dimensional compressor cascade. The vanes in the cascade were split at the midheight

Trailing vortices = Shed vortices + trailing filament vortices

Fig. 2.1 Secondary flows and vortices in an axial flow compressor rotor (Lakshminarayana & Horlock, 1967)

section, effectively producing a gap between blade ends. Ignoring the viscous growth on the surface of the blade, the flow may be assumed to be inviscid. The centre line of the gap, therefore, becomes equivalent to a wall.

Experiments measuring the spanwise lift distribution were done for a wide range of gap/chord ratios. The blade experienced a reduction in lift, the spanwise extent and the overall lift reduction being progressive with gap size. In terms of normal force coefficients, for a particular gap size there was a peak in the distribution very close to the blade tip before the eventual rapid decline. These results are shown in figure 2.1.1.

Lakshminarayana and Horlock (1967) also measured a progressive increase in induced drag with gap size. This change with gap/chord ratio indicated a change in shed vorticity at the tip. The bound vorticity towards the blade leading edge bridged the clearance gap in accordance with Helmholtz's second theorem on vorticity. However, this contradicts *Dean's (1954)* supposition that, in an inviscid flow field, as soon as the tip clearance becomes finite, no vortex lines could cross the clearance gap. Dean concluded that all of the bound vorticity must then trail off as a vortex sheet of varying strength behind the blade in the manner of a wing.

Lakshminarayana (1970) later developed a model for small gap/chord ratios based on observations made by himself and *Horlock (1967)*. The shed vorticity from the tip was $(1 - k)\Gamma_m$, where k , the measured lift coefficient retained at the tip as a proportion of its two-dimensional value, was derived from experiments. Γ_m was defined as the mean local value of circulation. For large gap/chord ratios the flow situation resembled more that of a wing with little flow interaction between the two blade tips. The shed vorticity thus tended to Γ_m .

Inlet shear was simulated in *Lakshminarayana and Horlock's (1967)* study by adding a midspan splitter in the clearance gap. Results show the formation of a secondary flow within the channel. This flow was opposite to the direction of the leakage flow observed in the tests without the splitter. The secondary and leakage flows combined to form a core of high losses in the exit plane at mid-passage. With no gap, this core was confined to the interface between the blade suction surface and the endwall. The effect of the secondary

Fig. 2.1.1 Spanwise distribution of normal force coefficient for various gap/chord ratios (Lakshminarayana & Horlock, 1967)

flow on the spanwise distribution of the loss coefficient was to spread it over a larger part of the blade span. The magnitude of the losses increased with gap size. Over much of the blade span, the normal force coefficients based on inlet condition, was lower than that of the no gap case, except in the region of the tip where it was enhanced. The secondary flow also reduced the underturning created by the leakage flow through the tip gap.

At a tip gap ratio of 0.04, *Lakshminarayana* and *Horlock* (1967) noted that the leakage flow tended to displace the corner separation zone at the intersection of the blade and the endwall surfaces; also, the secondary flow in opposing the leakage flow, prevented the leakage flow from moving away from the suction surface along the blade passage. Thus the leakage flow was confined to a region near the suction surface.

Studies carried out by *Phillips* and *Head* (1982), and *Dean* (1954) replaced the endwall over the blade tips with a moving belt. However, these investigations were mostly of a qualitative nature. As expected, their results indicated an increase in leakage mass flow and leakage flow velocity at the tip. This led to the core of high losses being swept further across the passage.

2.2 Experiments in Compressors

Due to instrumentation problems, the flow measurements made in actual rotating machinery are not as extensive as those done in cascades. *Lakshminarayana* and colleagues (1981, 1982a-d) have presented data from flow field measurements in a compressor rotor stage, using a tri-axial hot-wire probe rotating with the rotor. The measurements indicated that the leakage flow began downstream of the quarter chord position and tended to roll up farther away from the suction surface than that observed in cascades.

In general, the effect of tip gap size on overall performance parameters such as work coefficient and efficiency, has been examined. These measurements are sometimes supplemented by visual observations. *Rains* (1954) investigated a three stage axial water pump at its design point. Tip clearance height was varied up to 24% of the blade maximum thickness or 1.4% of the chord. Absolute efficiency fell 3.5% over this range (*figure 2.2.1*).

Fig. 2.2.1 *Variation of the efficiency with rotor tip clearance* (Rains 1954)

Fig. 2.2.2 *Variation of the work coefficient with tip clearance* (Rains, 1954)

The work coefficient, defined as the work input divided by the dynamic pressure based on the tip speed, fell by 0.007 (*figure 2.2.2*). *Rains* noted that for large tip clearances, the geometry of the leakage vortex was similar for both rotating and stationary blades with tip clearance. For small tip gaps in stationary blade rows, the origin of the vortex was just behind the quarter-chord position, as also observed by *Lakshminarayana* and *Horlock* (1967). With rotating blades, however, *Rains* observed that the vortex was attached to the leading edge of the blade. He also noted that the tip clearance flow was significantly increased as a result of the rotor relative velocity, confirming the cascade work done by *Dean* (1954).

Hutton (1956) observed that the surge margin was affected by the tip gap size. He found that, in addition to surge occurring at higher mass flow rates with increasing tip gap height, the hysteresis effect was removed. Examinations by *Greitzer* (1980) confirm that the change was the result of changes in the rotating stall pattern.

Spencer (1956) used a four bladed axial pump for his investigation which included clearances from 0.6 to 2.4% of the blade height. He found that the tip gap effects were most significant at low mainstream flowrates when pressure differences at the tip were the greatest. There was a slight initial improvement in efficiency as the clearance was opened. At the time, *Spencer* thought this effect to be spurious. However, succeeding work by *Lakshminarayana* and *Horlock* (1967), and others (*see Peacock (1983)*) indicate the possibility of an optimum gap size. *Lakshminarayana* (1970) postulated that since the secondary and the leakage flows opposed each other, an optimum should exist when the sum of the two are equal. He found qualitative agreement when he applied this principle to *Dean's* (1954) results.

2.3 Turbine Studies

Unlike the case of compressors, there have been few detailed investigations into the tip leakage aerodynamics of turbine stages. However, experiments by *Lakshminarayana* and *Horlock* (1963), using a single blade with a gap at the mid-span in uniform inviscid

flow, do simulate some of the conditions in a turbine. In addition to this, trends observed in compressor cascade studies are generally applicable to turbine cascades. Most studies in turbines have concentrated on measuring the effect of clearance height on overall performance. Generally, turbines show a linear drop in stage efficiency with clearance height. A drop in stage efficiency of 1.0 to 1.5% for every percent increase of gap/blade height ratio is commonly used in the turbomachinery industry for providing a rough estimate of clearance losses.

Hass and Kofskey (1979) measured the effect of clearance height on stage losses for a single stage turbine. Different tip and casing geometries, as well as stage reactions, were investigated. They included shrouded rotors, unshrouded rotors with recessed casing and reduced blade height configurations. Their results are shown in graphical form in figure 2.3.1. An extrapolated value for stage efficiency at zero tip clearance was used as the datum for each curve.

Using a water-table and a series of simple geometries, *Booth et al* (1982a—b) examined the flow across a tip gap without any relative wall movement. They identified that:

- (1) At low clearance levels, shear forces directly balanced pressure forces;
- (2) At normal clearance levels, pressure and inertia forces were dominant over viscous forces;
- (3) At high clearances, the pressure difference across the tip became small because of unloading.

Their results showed that the leakage flow was primarily inviscid with a high discharge coefficient and insensitive to flow on the suction surface.

2.4 Concluding Remarks

It is relatively easy to obtain adequate tip clearance loss correlations for turbines and compressors from the published literature. A good body of detailed work has been done

Fig. 2.3.1 Effect of rotor tip clearance on performance for various turbines.

on diffusing cascades, however, information for accelerating cascades is both sparse and lacking in details.

The tip leakage flow mechanism and its effect on the overall flow field in a turbomachine are far from being fully understood. Because of the complexity in an actual machine, it would seem advantageous to first of all gain an insight into the flow in turbine cascade without relative motion between the blades and the endwall, and then proceed to investigate the situation with relative movement.

Chapter 3

DESCRIPTION OF APPARATUS

As one can conclude from the previous chapters, there are many unresolved problems surrounding tip leakage in turbomachines. Since there seems to be a lack of the full understanding of the flow dynamics in the blade tip region and its effect on the overall performance of the entire blade, it was decided to further pursue this subject experimentally. Subsequently, a two-dimensional cascade section was designed and fabricated. The test section was designed to be used with the annular wind tunnel located in the Hydraulics Laboratory at Carleton University.

An attempt was made to incorporate as many features as possible into the design since the tip leakage flow appears to be affected or influenced by many variables. At the same time simplicity and ease of manufacture were prime considerations. In the sections that follow, the annular wind tunnel, the transition section, the test section and its components, and other equipment used in this study are described in detail. Figure 3.1 shows the assembled test facility.

3.1 Annular Wind Tunnel

The annular wind tunnel situated in the Hydraulics Laboratory in the Faculty of Engineering was used as the flow source for the test cascade. In the paragraphs that follow, the existing features and characteristics of this wind tunnel are described.

This open-circuit wind tunnel consists of a variable speed DC motor driven Budworth two-stage axial flow compressor, capable of supplying flowrates of up to 4 Kg/sec at a total pressure rise of about 500 mm of water. A cross-sectional drawing of the wind tunnel arrangement is given in figure 3.1.1. The compressor and settling chamber (*item 2, figure*

Fig. 3.1 Test facility

① Compressor	④ Annular Diffuser	⑦ Bleed Ring
② Settling Chamber	⑤ Honeycomb	
③ Settling Chamber Outlet	⑥ Stainless Steel Screens	

Fig. 3.1.1 Annular Wind Tunnel
(Sjolander, 1975)

3.1.1) are enclosed in a compartment lined with acoustic tiles to reduce the high noise level of the compressor.

A straight annular diffuser (item 4, figure 3.1.1) is used to achieve the rise in static pressure. This diffuser has an area ratio of 2.72 and an equivalent cone angle of 14° . Hexagonal honeycomb (item 5, figure 3.1.1) with 9.5 mm wide by 76 mm deep cells is used to remove any residual swirl from the compressor outflow. Seamless stainless steel screens, 7.9 mesh/cm and an open area ratio of 0.58, are situated in the settling chamber to reduce the turbulence intensity in the flow and improve the mean flow uniformity. The first screen is located approximately 125 mm from the diffuser outlet and uses three layers of screens at the centre decreasing to one layer at the settling chamber wall. The layered screens were intended to further diffuse the flow and to spread it more uniformly over the settling chamber cross-section. Two other single-layered screens spaced 140 mm apart are intended to further reduce the mean flow non-uniformities and turbulence. The last screen is located approximately 500 mm from the settling chamber outlet.

At the downstream end of the settling chamber there is a ring of thirty-six 51 mm diameter holes (item 7, figure 3.1.1) which enables some of the tunnel flow to be bled off. A sliding stainless steel band is used to block the bleed holes by any desired amount. Opening the bleed ring reduces the throttling that the compressor experiences during start-up, thereby avoiding run-up stall. The downstream end of the settling chamber is reduced to a 406 mm inside diameter circular outlet via a radial contraction.

The above gives a brief description of the wind tunnel facility as it presently stands. Any further details on this facility can be had from system manuals and measured operating characteristics available at the University.

3.2 Transition Section

The test section used in this study is a two-dimensional rectilinear cascade test section. However, as described in the last section, the outlet of the wind tunnel is a 406 mm diameter circle. Thus a transition from circular to rectilinear cross-section was required.

The inlet of the test section is 675 mm wide and 200 mm deep. This results in an area ratio of approximately 1.2 between the transition section inlet and outlet. Therefore, the flow will be decelerating throughout the section, resulting in an adverse pressure gradient. This raises the possibility of boundary layer separation problems. Thus the need for a uniform gradual increase in cross-sectional area.

A form corresponding to the transition shape was made from styrofoam. A nichrome 0.81 mm diameter toaster wire heated electrically was used to cut the styrofoam to the required shape. The procedure used for cutting the final shape was determined by trial and error using templates corresponding to the endshapes of the transition section. This shape was chosen to be the one that gave the smoothest, most uniform transition in cross-section throughout the length. The length of the section was chosen to be 875 mm. This length resulted in mild changes in the wall slopes at the inlet and outlet.

Metal flanges were fitted over the ends of the form and seven layers of fibreglass cloth and polyester resin were then laid onto the form such that they overlapped and bonded to the flanges. This gave substantial strength. The inner surface required some hand finishing once the form was removed. The transition section is shown in, figure 3.2.1 (item 1), an isometric drawing of the assembled test rig. Also illustrated in this figure is the coordinate system and direction nomenclature used throughout this report.

3.3 Cascade Test Section

The cascade test section, shown in side view in figure 3.3.1, was specifically designed for detailed investigations into tip leakage flows in turbomachines. However, an attempt was also made to make the section as versatile as possible so that it could be used for other flow studies. Tip leakage involves many variables such as clearance gap height, blade loading and inlet endwall boundary layer thickness. Hence, the resulting design attempted to allow as many of these quantities as possible to be varied.

The inside dimensions of the test section are 200 mm deep and 675 mm wide. The mainstay of this structure is a 7.9 mm thick steel back-plate (item 2, figure 3.2.1). All

Fig. 3.3.1 Test Section

other components are in some way supported by this component. At 275 mm downstream of the test section inlet is a line of 9.5 mm diameter holes on 13 mm centres (*item 3, figure 3.2.1*), drilled at 30° to the flow direction. These holes are intended for boundary layer blow-off. A line of holes was chosen over a milled slot so as to retain strength in the back-plate. As it turned out, no boundary layer bleeds were used during the present study, but will undoubtedly be used in future work.

To provide access for a traversing gear, six 20 mm wide, 3.2 mm deep grooves (*item 4, figure 3.2.1*) are located at appropriate streamwise locations in the back-plate. Within these grooves are 7.9 mm wide slots through the thickness of the plate. These extend in the lateral direction along the grooves and are intended to be used by probes to access the blade channels, allowing traverse in the spanwise and lateral directions. Aluminium sliders that move with the probe stem are used to fill the grooves.

The other walls of the test section are made from 12.7 mm thick plexiglass. This facilitates flow visualization in addition to reducing fabrication cost. The topwall (*item 5, figure 3.2.1*) consists of four sections, the window (*item 6, figure 3.2.1*) and three separate plexiglass strips. The centre strip is the easily removable ramp strip (*part 1, figure 3.3.1*). The aft section comprises the endwall boundary layer bleed, a 10 mm wide slot (*part 2, figure 3.3.1*) inclined at 30° to the streamwise direction. The ramp piece allows for a smooth inclined transition in the flow for different tip clearance settings. Tip gap height is varied by placing metal shims between the topwall and the sidewalls (*item 7a and 7b, figure 3.2.1*).

The hinged window, which allows for quick and easy access to the blade passages, is made up of three parts. The centre piece which extends beyond the blade leading and trailing edges is intended to be replaceable. This is to enable the investigation of the effects of different endwall treatments on the leakage flow. An ambitious project would be to replace this section with a moving belt to simulate the relative motion between the blade and endwall.

The side-flaps (*parts 3a and 3b, figure 3.3.1*) are used to regulate the amount of flow being blown off laterally. Periodicity of flow in the blade channels is achieved by

appropriate side and tail-flap settings. This is determined from bottom-wall (*back-plate*) static pressures measured at positions close to the pressure and suction surfaces of the blades in the four blade passages. The tail-flaps (*parts 4a and 4b, figure 3.3.1*) are added to guide the outflow from the channels in the direction of the exit flow angle and thus prevent an uncontrolled expansion. The tail-flaps must be set at an angle compensating for flow deviation at the blade trailing edges. However, it was observed that an incorrect setting of either tail-flap had a significant effect on the pressure on the suction side of the adjacent blade. Therefore, periodicity is attained by iterative settings of side and tail-flaps.

Flow turning in the cascade is restricted to a range of flow angles from approximately 30° to 70° . The upper limit is set by the aerodynamic performance of the Budworth compressor. This two stage fan can only produce the required head rise and flowrate for a flow turning of about 70° from the axial direction. The absence of pre-swirl in the test section thus limits the overall flow turning to about 70° . Convenient location of the outer walls of the test section at the exit plane determined the lower limit of 30° .

The Budworth compressor is capable of producing air flow at 25–35 m/sec in the test section for a moderate (60°) degree of flow turning. In order to operate the test section at typical machine Reynolds numbers, a blade chord length of 250 mm was chosen for the test cascade. This resulted in Reynolds numbers of $4\text{--}5 \times 10^5$ using air at room conditions.

Based on the chosen chord of 250 mm, the cascade solidity (*blade spacing divided by blade chord*), was chosen to be 0.6, a value somewhat higher than that of the rotor row from which the cascade blades were derived. This then fixed the blade spacing at 150 mm. Because of the way the blades are held in place in the cascade, the blade spacing becomes a fixed parameter. Changes to the test cascade solidity can therefore only be achieved by changing the chord length. Also note that the height of the test section was chosen to be 200 mm to obtain the necessary flow velocities in the test section. Hence, changes in aspect ratio (*blade height/blade chord*) can only be achieved by varying the chord length. As a result of this, the aspect ratio and solidity of the cascade become directly related

through the blade chord.

Blades are held in place in the cascade by a stem-collar arrangement. The 19 mm diameter steel stems are permanently attached to the base of the blades at approximately $\frac{1}{3}C$ from the leading edges. These stems fit through 19.2 mm diameter holes in the back-plate. The collars, steel tubing 19.5 mm inside diameter and 39 mm outside diameter, fit onto the blade stems on the back side of the back-plate. The collars lock onto the blade stems via set-screws. Tension which holds each blade in place is supplied by four set-screws pressing on the outer surface of the back-plate. This arrangement enables the blades to be set at any angle.

The angle, α , between a straight edge running from the leading to the trailing edges of the blade and the vertical (y) axis is used to accurately set the angle of attack of the blades. For a required angle of attack, α is easily measurable or as was the case in this study, a two-legged fixture that incorporates this angle was used to set the angle. One leg rests against the leading and trailing edges of the blade while the other sits in a traversing groove (item 4, figure 3.2.1). This ensures good consistency in all the blade settings.

One short-coming of the present arrangement is that a change in the blade incidence can only be achieved by restaggering the cascade. This changes the geometry of the blade channels. However, there is a possibility of replacing the part of the back-plate where the blades are attached with a turn-table.

3.4 Test Cascade

The geometric specifications of the turbine rotor blade and row provided by Pratt & Whitney Canada Inc. are listed in Appendix A. As discussed in the previous sections, the chord length for the cascade blades was chosen to be 250 mm. This meant that all geometric parameters of the Pratt & Whitney blade had to be scaled up by a factor of 12.29. The cavities at the tip of the cascade blades differ slightly in geometric shape from the one used in the actual blade. This cavity is used for stress reduction purposes in the machine. A cross-section of the blade at the tip along with relevant dimensions and blade

setting of the cascade are listed in Appendix B. Appendix C contains the data used for generating the shape of the cavity used in the model.

The blade shape was provided in terms of 121 coordinate points. It was decided to mill the shapes using a CNC point-to-point milling machine. Thus, there was a need for many more coordinate points in order to obtain a smooth contour. A cubic spline routine was used to provide another 14 interpolated points between each pair of the 121 points. The CNC machine also requires the contour of the tool centre to be specified, rather than the path of the cutting edge. To simplify orientation of the block to be milled on the machine turntable, blade coordinates were rotated so that the block centre lines corresponded to the x , y directions of the machine, with the z direction being vertical. A listing of the interpolation, tool radius compensation and coordinate rotation routines can be found in Appendix D.

Four of the five 200 mm span blades were milled from plexiglass, while the main static-tapped blade (*blade 3, figure 3.3.1*) was milled from aluminium. Plexiglass was chosen over other materials because of its workability. The main blade depicted in figure 3.4.1 was made in two halves. The top half has sixty-seven 3.2 mm diameter channels drilled through its height. The channels are connected from the blade pressure and suction surfaces by fourteen rows of 0.51 mm diameter static taps around the contour of the blade. The rows consist of thirty-seven and thirty-six taps on the pressure and suction surfaces, respectively. The number of static taps is greater than the number of channels because the last six channels towards the trailing edge are connected by taps from both the pressure and suction surfaces of the blade. The tap on one of these surfaces for each of the six channels has to be taped over during testing.

The first row of static taps is located 2 mm from the tip, the next eight rows are spaced 5 mm apart and the remaining five, 10 mm apart. The location of the taps is within 0.05 mm and 2° drill-angle of specifications. The static channels are connected to the flexible polyethylene tubings via stainless steel hypodermic tubings at the base of the channels. The flexible tubes are then passed through the core of the bottom half of the blade and out through the hollow blade stem. With this arrangement only one row

Fig. 3.4.1 Static-tapped Blade

of static taps can be used at any one time. Monocote, 0.04 mm thick, is used for sealing the other rows of static taps. Details, such as row and tap location on the main blade are summarized in Appendix E.

3.5 Traverse Gears and Probes

A previously existing 250 mm long L-shaped pitot probe was used to traverse the inlet flow to the test section. The details of the probe are given in figure 3.5.1 (a). This size proved to be sufficiently small for the boundary layer measurements in the test section. The traverse gear used for mounting this probe is shown in figure 3.5.1 (b). It allows rotation in the $x-y$ plane and spanwise translation in the z -direction to within ± 0.05 mm.

A seven-hole pressure probe (item 8, figure 3.2.1) was also designed and fabricated for three-dimensional flow field measurements in the test section. This probe was chosen over the more conventional five-hole probe because of the ease of manufacture and the extended flow incidence that can be tolerated. This probe is easier to fabricate because seven tubes whose individual outside diameters are exactly one-third of the inside diameter of an enclosing tube, naturally orient themselves in a symmetric fashion when placed within the larger tube. The calibration procedure used for this probe is outlined in detail by *Everette et al* (1982).

A traverse gear for use with the seven-hole probe was also manufactured. It allows for translation in all three directions, as well as rotation in the $x-y$ plane. Mounting holes and slots are located in the back-plate at appropriate locations for the traverse gear. Figure 3.5.2 (a and b) depicts the assembled traverse gear and shows the details of the probe tip, respectively.

3.6 Smoke Generator

Oil flow visualization can be revealing, but this procedure does have its limitations

Fig. 3.5.1(a) Pitot Probe

Fig. 3.5.1(b) Pitot Probe and Traverse Gear

Fig. 3.5.2(a) Seven-hole Probe and Traverse Mechanism

— SOLID —

Scale 20:1

All dimensions in mm

Fig. 3.5.2(b) Details of Seven-hole Probe Tip

in that it may alter the no slip boundary condition and it indicates the direction of the wall shear stress vectors rather than the flow direction and thus only indicates the limiting flow direction at the boundary surface. Consequently, a smoke generator developed for flow studies at Carleton University was used. This unit was designed and fabricated by J. Jarvo, an undergraduate engineering student, under the supervision of Professor R.J. Kind.

The smoke generator generates a narrow stream of clean non-toxic concentrated smoke. The tip of the smoke generator is a 0.8 mm outside diameter stainless steel tube. Thus the disturbances to mainstream flow due to the presence of the smoke probe is minimal. Smoke is generated by burning pressurized Bay oil (*a mineral oil used in the cosmetic industry*) using filament heating. Further details on this smoke generator can be obtained from Professor R.J. Kind at the Department of Mechanical and Aeronautical Engineering.

3.7 Pressure Measurement System

All pressures were measured using two Rosemount model 831A3 capacitive type differential pressure transducers. One of these is built into a forty-eight port Scanivalve, while the other was a single unit. These transducers will be referred to as the *Scanivalve* and *Single-input* transducers, respectively, for the remainder of this report. The Single-input transducer was used to constantly monitor the operating point of the wind tunnel as indicated by the contraction pressure difference at the settling chamber exit. This pressure is approximately the dynamic pressure at the chamber exit and is directly related to the dynamic pressure at the test section inlet. The Scanivalve transducer was used to measure all other pressures from the test section. Output signals from both transducers were monitored visually using digital voltmeters in addition to being recorded by the computerized data acquisition system.

The pressure transducers were calibrated using a Bets micrometer. The calibration constants for the Single-input and the Scanivalve transducers were determined to be 1.408 Pa and 1.392 Pa per mV, respectively. These were measured to an accuracy of better than

1% At the end of the blade unloading experiments, which spanned a period of approximately four weeks, both transducers were recalibrated to check for any drift. Results show that there were negligible changes in both calibration curves. These calibration curves are presented in Appendix F.

A Hewlett-Packard model 3054A Automatic Data Acquisition/Control System was used throughout the test section calibration and blade unloading with tip clearance experiments. An A/D converter converts an input DC voltage to a digital signal using a multi-slope conversion technique (*Multi-Slope II*). The high speed automated HP system is composed of a System maincontroller, the HP 3497A Data Acquisition/Control unit, two voltmeters, the HP 3456A and the HP 3437A, and the HP 85A desktop computer. The HP 3456A is a $6\frac{1}{2}$ -digit resolution voltmeter, while the HP 3437A is a high speed DC voltmeter. It was decided to use the HP 3456A for all measurements because of its high digital resolution. These voltmeters are interfaced with the HP 85A through the 82937A HP Interface Bus I/O card. The HP 85A is a self-sustained unit that uses the *BASIC* programming language. The main program that incorporate these routines are written and discussed in Chapter 4.

Chapter 4

DESCRIPTION OF TEST SECTION CALIBRATION EXPERIMENTS AND RESULTS

The first experiments conducted were the calibration measurements for the new test section. Since the results are relevant to any future experiments conducted in the test section, they are presented separately from the results of the tip leakage study. The calibration measurements included the determination of the contraction calibration, the mean flow uniformity and the freestream turbulence intensity. The averaging times needed to obtain reliable values of mean static and total pressures were also determined at this stage.

4.1 Wind Tunnel Operating Point

The operating point of the compressor was monitored from static pressure readings at the settling chamber contraction. These two static taps are located at the beginning and at the end of the contraction and hence, the pressure difference, $(P_{sc1} - P_{sc2})$, is proportional to the test section dynamic pressure. This pressure difference was measured and displayed on a Disa digital voltmeter. A filter with a three second time constant was used to damp out the high frequency fluctuations. This arrangement enabled the damped $(P_{sc1} - P_{sc2})$ to be held within ± 0.002 volts, translating to about $\pm 0.4\%$ maximum variation in the test section dynamic pressure. The operating point was chosen to give a Reynolds number of 4.3×10^5 . This value is based on the nominal blade chord, the properties of air at the settling chamber total temperature and atmospheric pressure, and the mean velocity at the centre of the test section 160 mm downstream of the inlet. Typical values of the variables were 25°C , 0.76 m of Hg and 30 m/sec.

Once the *steady-state* (after a 15-minute warm-up) settling chamber total tempera-

ture and barometric pressure were measured, the required test section centre line dynamic pressure was computed. From this, the corresponding value of $(P_{ss1} - P_{ss2})$ was calculated using the calibration factor of 0.610 for the ratio of the test section centre line to settling chamber contraction dynamic pressure. Based on these air conditions the operating Reynolds number was kept to within $\pm 0.1 \times 10^5$ of the desired value by occasionally adjusting the operating point of the tunnel, compensating for any tunnel drift.

Changes in the settling chamber total temperature and the barometric pressure were not accounted for during any particular run. However, variations in these parameters were small (*less than 1% of their absolute value*) even during an extended run (5 hours).

4.2 Data Acquisition Technique

Flow through the test section is of a fluctuating nature, consequently, measurements involved averaging a number of samples from the flow field to obtain a representative mean flow value. The operating point of the tunnel drifted with time and thus all test section measurements were normalized using the contraction pressure difference $(P_{ss1} - P_{ss2})$. Experiments were carried out to determine the optimum sampling time and technique for obtaining reliable flow field measurements.

The HP system can only monitor one signal at a time. As a result of this, one cycle of data acquisition involved sampling the test section working pressure as well as the contraction pressure $(P_{ss1} - P_{ss2})$, individually. To further reduce errors caused by drift in the running point of the compressor one cycle was divided into three phases:

- (1) Sampling the contraction pressure $(P_{ss1} - P_{ss2})$ for T_1 seconds;
- (2) Acquiring pressure data from the test section for T_2 seconds;
- (3) repeating (1).

The variables T_1 and T_2 and the number of cycles necessary to obtain a representative mean were optimized by trial and error. Embedded in T_1 and T_2 is the number of samples (*window openings*) to be taken for the specified time T_1 or T_2 . Thus the output after

sampling for T_1 or T_2 seconds is the average of a preset number of shorter samples.

It was found that a combination of $\frac{1}{2}$ - 1 - $\frac{1}{2}$ second with 30 window openings per $\frac{1}{2}$ second was best for both total and surface pressure measurements. In the case of freestream velocity measurements, 40 data cycles were collected. The cumulative averages were computed for the first through the fortieth cycle. The mean of the thirtieth to the fortieth cumulative means was then chosen to be a representative value of the mean value of $(P_{TS}/(P_{se1} - P_{se2}))$. This yielded a value which was within 0.2% of the mean value obtained after taking 500 data cycles. The variation of this quantity after 500 samples was negligible. Boundary layer and blade surface mean pressures were based on the fortieth to fiftieth and tenth to fifteenth samples, respectively. This resulted in the same level of accuracy as the above.

Appendix G contains data justifying the above numbers. The *BASIC* computer programs used for data acquisition at the section inlet and on the blade surface are listed in Appendix H.

4.3 Test Section Inlet Flow

Velocity profile measurements were taken at seven inline lateral locations spanning the width of the test section. These stations are located 160 mm downstream of the test section inlet and are equally spaced 82 mm apart with station number 4 being at midwidth. Total pressures were measured with the L-shaped pitot static probe described in section 3.5. Static pressures were obtained from wall taps which were located at the same streamwise position as the probe mouth. The location of the probe tip away from the topwall was measured to an accuracy of ± 0.05 mm using the traverse mechanism described in section 3.5. Pneumatic signals were converted to analog signals using a Rosemount pressure transducer connected to a Scanivalve. This signal was displayed on an HP voltmeter while being fed to the HP Data Acquisition System. A predetermined number of flow field samples were taken to give a reliable mean value (section 4.2).

Before any data were collected, the flaps were set so that there was periodicity in the

cascade flow. Table 4.3.1 contains periodicity data for each blade passage. Passage 1 is defined by blades 1 and 2 (*figure 3.3.2*). All pressures were referenced to the inlet wall static pressure and non-dimensionalized with respect to the centre line dynamic. As can be seen, there was good symmetry in the channels, however channel 4 was a slight exception. As it turned out, the side-flap next to blade 5 was kept fully closed, and yet there seems to be a slight flow deficiency in channel 4. This implies that the bottom sidewall of the test section is slightly within the would-be streamtube for passage 4 for the current cascade setting. To correct this minor fault, the bottom wall would have to be lowered, increasing the width of the test section. However, the indicated pressure difference within channel 4 is only slightly different than those of the other three channels. In addition to this, the static pressure measured at the periodicity tap near the suction surface of blade 4 differs only slightly from its counterparts in the other channels. Therefore, the slight lack of symmetry in passage 4 is concentrated towards the pressure surface of blade 5.

Emphasis was placed on carefully measuring the boundary layer on the endwall rather than the layer on the back-plate. This is because the flow in the tip region of the cascade is of primary interest in this study. Figure 4.3.1(a & b) shows the measured spanwise profiles at the seven traversing stations along the width of the test section. Station 1 corresponds to the one next to the top sidewall (*figure 3.2.1*). All velocity measurements are referred to the centre line velocity, V_{line} recorded at midspan of the centre traversing station. The relevant boundary layer parameters δ^* , θ and H were computed for each profile and are tabulated in table 4.3.2. The parameters were computed by two methods. Firstly, by fitting a least-squares curve through the measured data points and integrating this curve over the boundary layer thickness to determine the appropriate quantities. As a check on the procedure, the measured boundary layer data were plotted on an enlarged scale and a best curve fitted using a french curve. Points were read from this curve and the profile integrated using Simpson's Rule to determine the boundary layer parameters. Since a slight error in the location of the outer edge of the boundary layer does not significantly affect the parameters, the boundary layer thickness was chosen to be the point where the velocity was approximately ninety-nine percent of the maximum velocity in a particular

Passage #	$\frac{P_{st} - P_{inlet}}{(P_{st,ave})_{DYN}} (1)$	$\frac{P_{st} - P_{inlet}}{(P_{st,ave})_{DYN}} (2)$	(2)-(1)
1	-1.311	-0.102	1.209
2	-1.309	-0.101	1.208
3	-1.314	-0.100	1.214
4	-1.319	-0.092	1.227

Table 4.3.1: Cascade Periodicity Data

Station #	Lateral Position (Y/W)	$\frac{\delta^*}{k}$	$\frac{\theta}{k}$	H
Top Sidewall	-0.50			
1	-0.36	0.0083	0.0050	1.66
2	-0.24	0.0118	0.0075	1.57
3	-0.12	0.0087	0.0054	1.61
4	0	0.0084	0.0052	1.62
5	0.12	0.0091	0.0059	1.59
6	0.24	0.0126	0.0084	1.50
7	0.36	0.0148	0.0106	1.39
Bottom Sidewall	0.50			

Table 4.3.2: Inlet Boundary Layer Parameters

Fig. 4.3.1(a) Inlet Velocity Profiles at Outer Stations

Fig. 4.3.1(b) Velocity Profiles at Test Section Inlet

profile. Of the two procedures used for computing the parameters, δ^* , θ and H , the latter one proved to be more reliable because the least-squares curve required for a good fit to the measured data tended to vary significantly with the shape of the velocity profile. A listing of the Simpson's integration routine can be found in Appendix I.

From table 4.3.2, the H values of the topwall boundary layer indicate that it is fully turbulent since they are close to the approximate zero pressure gradient values for turbulent layers. The two outer profiles taken at stations 1 and 7 differ slightly from the inner ones. This is probably due to the effects of the sidewalls, the outflow characteristics from the transition section, and/or the slightly different streamwise pressure gradient that the flow experiences in these regions.

As can be seen from the velocity profile plots in figure 4.3.1(b), there is good flow uniformity at the middle section of the cascade width. This is especially true for the region spanned by traversing stations 3, 4 and 5, i.e., the middle one-third of the test section width. The velocity profiles are very similar in this region. The outer profiles differ somewhat from these profiles. This is mainly due to the sidewall effects as mentioned earlier, and non-uniformity in the outflow from the settling chamber. This non-uniformity probably originates at the layered first screen in the settling chamber.

The inner five velocity profiles show a slight deficiency in velocity towards the midspan of the test section, especially for profiles at stations 3, 4 and 5. A close look at these graphs reveal that there is also a deficiency in the midspan velocities across the width of the test section. The midspan velocities at each traversing station are listed in table 4.3.3.

A spanwise static pressure profile (figure 4.3.2) taken at midwidth of the test section (station 4) shows that the static pressure variation is less than one-half percent of the centre line dynamic pressure. The inplane wall pressure was used as the reference. The slight variations which were localized near the top and bottom walls were probably due to the misalignment of the pitot static probe used for the static pressure measurements. It is conceivable that a misalignment of up to 10° may have occurred since the probe was set visually. Therefore, for all intents and purposes, the static pressure across the span of

Station #	Lateral Position (Y/W)	$\frac{V_{midspan}}{V_{center}}$
Top sidewall	-0.50	
1	-0.36	1.014
2	-0.24	1.009
3	-0.12	1.000
4	0	1.000
5	0.12	1.000
6	0.24	1.011
7	0.36	1.017
Bottom Sidewall	0.50	

Table 4.3.3: Midspan Lateral Velocity Variation

Fig. 4.3.2 Spanwise Static Pressure at Midwidth of Inlet

the test section at the midwidth station was quite uniform.

Results of a lateral static wall pressure survey across the width of the test section at inlet are tabulated in table 4.3.4. All measurements were referred to the centre station. The static pressure was quite uniform in the centre half of the test section width, increasing towards the sidewalls. This results from the boundary layer growth on these walls, causing a lateral curvature of the main stream streamlines towards the centre line of the test section.

From the above, it can be concluded that the spanwise and lateral deficiencies in the velocity profile towards the centre line of the test section really reflect a deficiency in total pressure. The settling chamber, located at the back end of the Budworth compressor, has a concentration of metal screens at its centre with a gradual decrease towards the outer wall. This is described in section 3.1. The objective of having this high inner radius concentration was to aid in the radial diffusion of the flow in the settling chamber. It appears that these screens have done more than was desired and the high concentration at the centre resulted in higher total pressure losses in this region. The outer stations at the test section inlet do not show as much of a total pressure deficiency spanwise because the flow in these regions originate in the outer section of the settling chamber. The same argument can be applied to explain the lateral variation in midspan dynamic pressure. However, in this case there is the additional effect of the increased static pressure towards the sidewalls, to further accentuate the deficiency in total pressure in the midwidth region.

Up to this point there has been no discussion of scatter in the velocity profile data caused by errors. The possible sources of error were probe misalignment, probe geometry, turbulence in the flow, voltmeter A/D conversion, transducer calibration and the finite number of data samples from the flow field. It is rather difficult to quantify the first three sources, however, they are usually relatively insignificant. For the range used on the HP 3456A voltmeter, the manufacturer, Hewlett Packard, quotes an accuracy of 0.003 percent of the measured value. The error resulting from the transducer calibration is naturally combined with the error resulting from taking a finite number of data samples. Note that the scatter in the calibration data is minimal (*less than 0.1%*) as shown in Appendix F.

Station #	Lateral Location (Y/W)	$\frac{P_W - P_{INLET}}{(P_{static})_{DYN}}$
Top sidewall	-0.50	
1	-0.36	0.049
2	-0.24	0.021
3	-0.12	0.003
4	0	0
5	0.12	0.005
6	0.24	0.015
7	0.36	0.038
Bottom Sidewall	0.50	

Table 4.3.4: Lateral Wall Static Pressure across Width of Test Section

As can be seen from the results in Appendix G, the variation of the cumulative mean dynamic pressure at the inlet as a function of number of samples leads to an error of approximately 0.2% of the centre line dynamic pressure for both boundary layer and freestream measurements. Therefore, this error dominates over the A/D conversion error of the HP 3456A which can then be neglected according to *Moffat* (1982).

In any case, it must be pointed out that the objective of measuring the inlet velocity profiles was to check for flow uniformity and not to obtain precise boundary layer measurements. The results shown in figure 4.3.1 (a, b) confirm that the technique and apparatus used were certainly adequate for this purpose.

4.4 Turbulence Measurements

The freestream turbulence intensity was measured at the midspan of the midwidth traversing station at the test section inlet (station 4). A single straight hot-wire probe was used for measuring the axial fluctuation of velocity, u' . The bridge output from this probe was linearized for flow speeds ranging up to the maximum speed encountered in the test section. The mean voltages were displayed on a Disa Digital Voltmeter, type 55 D 31 with a time constant setting of three seconds. A Brüel and Kjaer type 2425 RMS voltmeter indicated the root-mean-square of the fluctuating component of velocity.

The turbulence intensity, the ratio of the root-mean-square of the axial flow fluctuation to mean axial flow velocity, u'/U , was determined from the mean and fluctuating voltages, mean flow velocity and the hot-wire linearizing constants using King's law. This quantity was measured to be approximately 0.4%. This is quite low compared to typical turbomachinery flows where values of up to 10% are common. At low turbulence levels, boundary layers on blade surfaces and endwalls are more susceptible to flow separation and represent a worst case situation. In any case, turbulence generators can be added at the test section inlet to increase the intensity level if so desired.

Chapter 5

DESCRIPTION OF TIP LEAKAGE EXPERIMENTS

Since a large part of the present work consisted of the design, manufacture and calibration of the new test section, only a fairly modest program of tip leakage experiments was actually undertaken. Nevertheless, a number of interesting results were obtained. The experiments are divided into two parts:

- (1) The effect of the gap size on the unloading of the blade;
- (2) Flow visualization of the flow field in the blade tip region.

The following sections deal with these subjects individually. Note that the exact setting of the cascade blades is listed in Appendix B.

5.1 Blade Unloading Experiments

The effect of gap size on the blade loading was measured by taking pressure measurements around the blade profile at various spanwise locations for different clearances. Five gap/chord ratios were investigated; 0, 0.0095, 0.0158, 0.0222 and 0.0286. In the actual machine from which the blades were taken, the ratio lies between 0.0158 and 0.0222. Clearance heights between the blade tips and the endwall were measured to an accuracy of ± 0.2 mm using feeler gauges. This figure was obtained with the wind on.

Since the entire topwall of the test section is moved outwards downstream of the ramp section to increase the gap size, the compressor mass flow has to be increased to maintain a constant Reynolds number at the test cascade. This was done by increasing the mass flow at the test section inlet by an amount proportional to the change in flow area downstream of the ramp section. To check whether the ramp section succeeded in providing a smooth transition in the flow, a velocity profile was measured at a point $\frac{2}{3}C$

upstream of the centre blade leading edge for each gap setting.

As described in section 4.2, preliminary tests were conducted to determine the minimum data acquisition time necessary to obtain representative mean flow field measurements. To further reduce the data acquisition time, pressure distribution around the blade contour utilizing all 67 static taps were plotted, initially. From these plots, the minimum number of static taps around the blade profile required to give a smooth pressure distribution was determined, eliminating the *redundant* taps.

5.2 Flow Visualisation Experiments

Flow visualization experiments were carried out to gain a better insight into the tip leakage flow and its effects on the mainstream flow. Tests were done for gap/chord ratios of 0, 0.0095 and 0.0286. Both smoke and surface oil film were used. In the case of the smoke studies, the Reynolds number had to be reduced to approximately one-third of its full value. This was necessary to facilitate photography. However, no significant differences in the flow field were noticeable for the two operating conditions.

Smoke was injected through the topwall at a point $\frac{1}{2}C$ upstream of the leading edge of the second blade (*blade 2, figure 3.3.1*) from the top of the cascade. This blade was chosen over the centre static-tapped blade to simplify photography and to avoid fouling of the static taps on the main blade. As discussed earlier, the flow in each channel should be the same due to the periodicity of the flow field. The plexiglass blade was covered with a sheet of navy-blue monocoat and the back-plate painted flat black to aid in the photography.

The objectives of the smoke studies were to qualitatively establish the amount of upstream flow being swept over the blade tip, to observe the streamlines through the tip gap and note their point of origin, and to establish the interaction of the leakage flow and the rest of the flow field. In order to achieve these goals, three different smoke probe tips were made. Two were L-shaped with tips 10 and 80 mm long, while the other was a multi-hole 60 mm long tip. The latter was used to generate a plane of smoke in the

spanwise direction.

A mixture of SAE-50 motor oil and white titanium oxide pigment was used for surface flow visualization on the blade surfaces and the endwall. Patterns were allowed to develop for about 15 minutes. The streaks obtained represent the surface shear stress trajectories, and are most convenient for identifying converging and diverging shear stress vectors, and therefore three-dimensional separation and reattachment lines. The oil film is also useful for identifying regions of high and low wall shear stress.

Chapter 6

RESULTS AND DISCUSSION OF TIP LEAKAGE EXPERIMENTS

As described in Chapter 5, flow visualization experiments using both smoke and oil were carried out. The objective was to gain a better insight into the tip flow dynamics. In the sections that follow, results from the blade unloading with clearance are preceded by the results from the flow visualization investigations. This is not the actual order in which these experiments were conducted, however, observations made from the flow visualization experiments will aid in the explanation of the results from the blade unloading experiments. For convenience, the presentation and discussion of the results have been combined.

One point which the reader must bear in mind is that the interpretation of the results from the flow visualization experiments are those of the author. Therefore, these interpretations are somewhat subjective and in some cases speculative. An attempt was made to present as many pictures of the actual flow patterns so that the reader can draw his own conclusions.

6.1 Smoke Flow Visualization

Unlike oil, which can only be used to study flows in surface regions, smoke can be used to indicate mean flow characteristics at any point in the flow field. To enable the smoke to be visible photographically, the Reynolds number of the test section was reduced to 1.27×10^5 , approximately one-third of its full operating value of 4.3×10^5 . To verify that the operation at this reduced Reynolds number did not result in any significant changes in the flow field, such as flow separation on the blades, blade loading measurements were compared for both operating conditions. The results from static tap rows 1 and 13 are presented in figure 6.1.1. The details of row locations and parameters plotted are discussed

Fig. 6.1.1 Blade Loading Distribution at Full and Reduced Reynolds Numbers

in Section 6.3. As can be seen, the loading distributions are quite similar indicating that the flow field dynamics were basically the same. The slight difference may have been due to the different inflow conditions to the cascade, namely, the topwall boundary layer profile. Figure 6.1.2 compares the velocity profiles to midspan at $\frac{2}{3}C$ upstream of the centre blade's leading edge for both Reynolds numbers. The velocities are non-dimensionalized with respect to the centre line inlet velocity (V_{oline}). As expected, the lower Reynolds number produced a thicker profile.

In addition to the above mentioned verification, a topwall surface flow visualization experiment was conducted for a $\frac{Cl}{C}$ ratio of 0.0095 using a diluted oil-pigment mixture. Figure 6.1.3 shows the reduced Reynolds number (1.27×10^5) case and figure 6.2.1 shows the full Reynolds number (4.3×10^5) picture. The shear stress vector patterns over the blade tips are quite similar. The only noticeable difference is the location of the low shear stress area in relation to the suction surface. For the full Reynolds case, this region is farther away from the suction surface than at the reduced Reynolds number. This may have resulted from using an oil with too high a viscosity at the low Reynolds number, and not allowing the flow pattern to reach steady state. In addition to this, varying the operating point of the wind tunnel over a range where the smoke remained visible did not seem to result in any change in the smoke pattern. However, it is also conceivable that there could have been small differences in the flow fields. Nevertheless, it does appear that the reduced Reynolds number case does in fact represent a very similar aerodynamic situation.

One of the main objectives of the smoke study was to establish the mean leakage flow path within the clearance gap for different clearances. This was done for three clearance/chord ratios, 0, 0.0095 and 0.0286. An overall streamline pattern over the blade tip at different chordwise locations was obtained for each clearance by superimposing individual patterns similar to the one shown in figure 6.1.4. The patterns were obtained by injecting smoke close to the blade tip using the appropriate smoke probe tip. However, it was quite noticeable that the upstream flow further away from the endwall was being swept over the blade tip for the larger gap compared to the smaller clearance.

Fig. 6.1.2 Velocity Profile at $(2/3)C$ Upstream of blade L.E.

Fig. 6.1.3 Oil traces on endwall at $Re=1.27 \times 10^5$
for $(Cl/C)=0.0095$

Fig. 6.1.4 Streamline pattern in tip gap
(C_l/C)=0.0286.

The leakage flow paths over the blade tip are summarized in figure 6.1.5. The vectors show the direction of the leakage flow and their lengths are proportional to the pressure difference across the blade for row 1, the one closest to the blade tip. No attempt was made to quantify the actual leakage volume flowrate at different chordwise locations from the pictures and observations in the laboratory. This would require flow field measurements using at least a three-hole probe. However, it was distinctly noticed that for both clearances there was a concentration of flow in the fore $\frac{1}{2}C$ of the blade, especially around the $\frac{1}{4}C$ location. Thus indicating that the leakage flow is primarily driven by the pressure difference between the pressure and suction surfaces of the blade, since this is the region of highest loading as shown by the vectors in figure 6.1.5. Also indicated in this figure are the apparent locations of the starting point of the leakage flow vortex on the suction surface of the blade. From these, it appears that this starting point moves towards the trailing edge of the blade as the gap size is increased.

For the larger gap, the leakage flow vectors for the fore $\frac{1}{4}C$ of the blade are oriented more towards the streamwise direction than the vectors for the smaller gap, as shown in figure 6.1.5. This indicates that inertia forces are more significant in this region for the larger gap. At the larger clearance, the blade tip was located approximately at the edge of the upstream endwall boundary layer. For the $\frac{C}{C}$ ratio of 0.0095, the tip was located at a point where the velocity was only 70% of the maximum velocity in the upstream profile. Therefore, the streamwise momentum of fluid entering the tip gap at the leading edge was higher for the larger gap than the smaller gap. Also note that for the smaller gap setting, viscous forces within the tip gap itself were probably becoming important. From $\frac{1}{4}C$ to $\frac{2}{3}C$ from the leading edge, the flow direction is practically perpendicular to the camber line for both clearances. This implies that the leakage flow is primarily pressure driven in this region. In the latter $\frac{1}{3}C$, where the loading decreased significantly, the leakage paths became increasingly streamwise, implying that inertia forces had become important again. This was particularly true for the larger gap.

Smoke flow visualization was also used to show the amount of upstream flow that gets swept over the blade tip. A spanwise plane of smoke 40 mm deep was injected

↓ Origin of Leakage Vortex

Fig. 6.1.5 Summary of Leakage Streamlines and Corresponding Blade Loading Across Blade Tip

at approximately $\frac{1}{2}C$ upstream of the leading edge of the blade used for smoke flow investigations. The resulting pictures for both clearances are presented in figure 6.1.6. The entire plane of smoke is swept over the blade tip for the larger gap, while approximately 50% of the smoke stream was swept for the reduced clearance.

Tests using both smoke and oil were conducted to determine the effect of the blade tip cavity on the mean leakage streamlines. Flow patterns from experiments with the cavity opened and closed were quite similar and no differences were noticeable.

Figure 6.1.7(a) shows the flow field at the exit plane of the cascade for the zero clearance setting. From similar observations at $\frac{Cl}{C}$ ratios of 0.0095 and 0.0286, figure 6.1.7(b) was derived. This figure summarizes the size and location of the loss core comprising the horse-shoe and the passage vortices, and the leakage vortex in the non-zero clearance case. Qualitatively, the core extended to approximately $\frac{1}{10}$, $\frac{1}{8}$ and $\frac{2}{8}$ of the blade height for $\frac{Cl}{C}$ ratios of 0, 0.0095 and 0.0286, respectively. As shown, the centre of the core was displaced farther away from the suction surface and the endwall as the gap size was increased. Note that the term *loss core* is used to describe the combined induced flows in the tip region. Observations made through the cross-section of this core using a stroboscope showed that the flow in the core was highly rotational for the three gap settings. In addition to this, flow field measurements made by *Lakshminarayana* and *Horlock* (1967) showed a significant total pressure deficiency within this core for both zero and non-zero clearances. One short-coming of the present study is that it was unable to determine whether the different vortices within this core did combine or remained isolated. In any case they were in close proximity of each other and, therefore, will be considered as a single entity for the remainder of this report.

A comparison of each channelwise (*blade-to-blade plane*) interaction between the leakage flow and the mainstream flow for $\frac{Cl}{C}$ ratios of 0.0286 and 0.0095 is provided by figure 6.1.8. For the larger gap, the loss core flow extended to about one-half of the channel width at the cascade exit plane. This region was reduced by about 50% and 75% for the smaller clearance and zero clearance cases, respectively. Thus the leakage flow not only increased the size of the loss core flow, but also displaced the centre of the core away

(a) $(Cl/C)=0.0095$

(b) $(Cl/C)=0.0286$

Fig. 6.1.6 Swept flow through tip gap

Fig. 6.1.7(a) Smoke flow at exit of cascade for $(C1/C)=0$

$(C1/C)=0$

$(C1/C)=0.0095$

$(C1/C)=0.0286$

Fig. 6.1.7(b) Schematic of overall loss core at exit plane of cascade.

(a) $(Cl/C)=0.0095$

(b) $(Cl/C)=0.0286$

Fig. 6.1.8 Channelwise (blade-to-blade) interaction
between loss core and mainstream flow.

from the suction surface for the no gap situation.

In concluding it can be said that the leakage flow direction through the tip gap is dependent on the gap size. At small clearances the leakage flow path tends to be perpendicular to the camber line for most of the blade chord, indicating that the flow is primarily pressure driven. For the larger clearances, the streamwise momentum of the fluid entering the tip gap is higher, plus there is more pressure unloading at the blade tip. This results in the leakage streamlines being oriented more towards the streamwise direction, especially in the fore and aft regions of the blade chord.

A final note to anyone intending to further pursue tip flow visualization studies. It was found that photography utilizing flash lighting produced the best results for smoke visualization. This is based on the extensive trial and error experiments conducted during this study. Natural lighting was found to be better suited for oil film pictures.

6.2 Surface Flow Visualization

One of the main disadvantages of oil flow visualization is that it only indicates the limiting flow condition at the boundary surface. In regions where there are strong cross-flow pressure gradients, the direction of the low momentum boundary layer fluid would be strongly influenced by the pressure field, while the mainstream fluid would not be significantly affected because of its higher inertia forces. However, the direction of the shear stress vectors over the blade tip in figure 6.2.1 are quite similar to those of the mean leakage flow vectors in figure 6.1.5. This is especially true for the smaller gap. Therefore, it appears that the pressure forces are dominant over the inertia forces in the tip gap. In addition to this, one of the strong points of surface flow visualization is that it readily shows regions of low and high shear stress, i.e., the location of vortices.

Oil flow experiments were done for gap/chord ratios of 0, 0.0095 and 0.0286. Figure 6.2.1 shows the patterns established on the endwall for the three clearances. The zero clearance setting isolated the secondary flow in the blade passage. This is a combination of the two legs of the horse-shoe vortex formed at the leading edge of the blade and the

passage vortex. The horse-shoe vortex results from the three-dimensional separation of the incoming endwall boundary layer. One leg of this vortex rolls up around the suction surface of the blade (at about $\frac{1}{8}C$ from the leading edge) and quickly gets confined to a narrow region at the intersection of the blade suction surface and the endwall. The other leg of this vortex is swept across the channel to the suction surface of the adjacent blade. The passage vortex which strengthens with streamwise distance from the leading edge flows across the passage and gets confined in the same region as the leg of the horse-shoe vortex that flows around the suction surface. Figure 6.2.2 shows the effect of the vortices on the suction surface at the exit plane. As can be seen, the vortices scoured off the oil from the blade surface for a region spanning approximately one-fifth of the blade height.

At a $\frac{Cl}{C}$ ratio of 0.0095, the situation at the endwall is dramatically changed from the no gap setting. The leakage flow displaces the region of low shear stress (*separation lines*) away from the suction surface of the blade. The low shear stress region originates at approximately $\frac{1}{10}C$ away from the leading edge and thus corresponds to the origin of the leakage vortex, as shown by the smoke studies (*figure 6.1.5*). This low shear stress area is much wider in the blade-to-blade plane at any chordwise location compared to the no gap case. This probably results from the leakage vortex strengthening the loss core and inducing smaller vortices as shown in figure 6.2.3(a). It is also conceivable that the leakage and secondary vortices did not combine and remained side by side as shown in figure 6.2.3(b). Of course, a combination of both of the above is also possible. Flow field pressure measurements and detailed smaller flow visualization using different coloured smoke concurrently in an attempt to isolate the vortices is needed to clarify the situation. Also noticeable at this gap setting is the apparent weakening of the horse-shoe vortex. This can be seen from the weak separation line that flows across the passage to the adjacent suction surface from the leading edge of the blade where it originates.

The flow situation at the endwall is further altered at a $\frac{Cl}{C}$ ratio of 0.0286, the largest gap investigated. The low stress area is displaced farther away from the suction surface and the horse-shoe vortex is weakened to the point of being barely noticeable. In addition to this, two high shear stress areas are located in the vicinity of the tip gap. These along with

Fig. 6.2.2 Oil film traces on suction surface at the exit plane for $(C_l/C)=0$.

Fig. 6.2.3 Vortices and separation lines(S_1, S_2) in tip region.

the observed separation lines indicate the formation of two distinct tip leakage vortices. The origin of these apparent vortices are $\frac{1}{4}C$ and $\frac{3}{4}C$ away from the blade's leading edge. As will be seen in Section 6.3, these locations correspond to suction peaks in the pressure distributions of the upper rows of static taps, confirming the existence of high energy fluid in these regions. From the oil and smoke experiments, it was not possible to determine if these vortices combined or remained distinct. Note that this second leakage vortex was not observed in the smoke flow visualization. This was probably because only a narrow stream of smoke was injected into the flow field at any instance and the smoke flow in the tip gap only spanned a narrow chordwise region. The factor causing the two tip leakage vortices cannot be determined from the observations made and further investigations similar to the ones mentioned in the previous paragraph are needed.

Figure 6.2.4 shows oil traces on the pressure and suction surfaces of the blade at the largest gap setting. The induced spanwise flow on the pressure surface is almost perpendicular to the endwall for the fore $\frac{1}{4}C$ of the blade and extends spanwise for about 20 mm from the blade tip. This corresponds to the region of high blade loading. The stress vectors become increasingly streamwise towards the trailing edge of the blade. The suction surface shows the location of the two tip leakage vortices discussed earlier, as well as the induced spanwise flow. Also noticeable on the suction surface is the region of very low shear stress that is located at about $\frac{1}{4}C$ from the leading edge and spans almost the entire blade height. This appears to be a region of near flow separation, however, the static pressure gave no evidence of a separated region on the suction surface.

Similar investigations to the above were carried out for $\frac{G}{C}$ ratios of 0 and 0.0095. There was no trace of spanwise flow on the pressure surface for the no gap setting. This indicates that the presence of a tip gap significantly affects the flow on the pressure surface of the blade in the tip region. Surprisingly, as will be seen in Section 6.3, this does not result in substantial changes in the static pressures on the pressure surface. At the smaller gap setting, the second leakage vortex was non-existent, however, the rest of the flow field was quite similar with the spanwise effect due to the tip leakage reduced by about 50% when compared to the largest gap tested.

Fig. 6.2.4 Oil film traces on suction and pressure surfaces
for $(C_l/C)=0.0286$.

6.3 Blade Unloading With Clearance

The effect of clearance height between the blade tips and the endwall on the loading distribution of the blades was measured for five different clearances. The clearance height/blade chord ratios were 0, 0.0093, 0.0158, 0.0222 and 0.0286. For each clearance setting the pressure distribution around the main blade was measured from nine rows of static taps; rows 1 to 6, 8, 10 and 13. The coordinates of the static taps are given in Appendix E. For convenience the location of the above rows with respect to the blade is also given in table 6.3.1. The non-dimensionalizing parameter, h , is the height of the test section at inlet, which also corresponds to the blade height.

At each clearance setting the velocity profile at $\frac{2}{3}C$ upstream of the leading edge of the main blade was measured to midspan. This was done to check whether the transition at the ramp section produced a smooth velocity profile at the entry to the cascade. The reference non-dimensionalizing velocity used for each profile is the midspan value for that setting. As can be seen from figure 6.3.1, the profiles are very similar (*note: the curves are plotted on a staggered scale*). This indicates that the inflow to the cascade for each clearance setting is practically the same.

Plots of pressure coefficients based on the undisturbed freestream dynamic and static pressures are presented in the following sections. On the graphs the measured points are connected by straight lines. At the end of the blade unloading experiments it was discovered that static channel number 37 had a leak and thus had been giving erroneous pressure measurements. Channel 37 corresponds to the third point from the trailing edge on the pressure surface of the blade. This point is included in all pressure distribution plots and force coefficient calculations. Computations of force coefficient with and without this point included yielded almost identical numbers (to within 0.5%) and it was decided not to omit it from the pressure distribution data.

The blade pressure distributions for zero clearance are shown in figure 6.3.2 (a, b). At zero clearance the actual blade metal tip was 0.67 mm away from the endwall. A rubber seal, cut to the shape of the blade, was used to seal the gap between the endwall

Row #	Z (mm)	Z/h
1	-2	-0.01
2	-7	-0.035
3	-12	-0.060
4	-17	-0.085
5	-22	-0.110
6	-27	-0.135
8	-37	-0.185
10	-52	-0.260
13	-82	-0.410

Table 6.3.1: Location of Static Rows from
away from the Metal Tip of the Main Blade

Fig. 6.3.1 Velocity Profile at $(2/3)C$ Upstream of blade L.E.

Fig. 6.3.2(b) Blade Pressure Distribution (C_l/C)=0

and the blade tip. This means that static tap row 1 was located 2.67 mm away from the endwall, a point where the velocity in the upstream boundary layer is about 75% of its maximum value in the profile (*figure 6.3.1*). The boundary layer just upstream of the cascade is about 15 mm thick and the profile is quite full. Therefore, row 3 was situated at approximately the edge of the boundary layer. However, the location of the upper rows away from the endwall with respect to the upstream boundary layer thickness is not too significant in determining the blade static pressure because of the formation of the horse-shoe and passage vortices in this region. *Figure 6.3.2 (a)* shows that the suction peak of row 13 is higher than that of row 3. Because of the total pressure deficiency towards the centre of the test section, row 3 should have experienced higher dynamic pressures than row 13 if the upstream boundary layer did not separate.

Figure 6.3.2(b) shows that row 8, which was at 37.67 mm away from the endwall, still experienced a slightly lower suction pressure on the suction surface than row 13. This further confirms the spanwise extent of the influence of the endwall flow along the suction surface. Force coefficients which were computed from the pressure distributions are presented later in this section. They show the effect of the induced flows in the blade tip region on the blade forces at different spanwise positions

Figure 6.3.3(a, b) shows the pressure distributions for rows 1 to 3 and 13 at a $\frac{C_l}{C}$ ratio of 0.0095. There has been a streamwise shift of the suction peak for row 1 compared to both row 1 at the no gap setting and row 13 at this $\frac{C_l}{C}$ ratio. This probably resulted from the leakage flow displacing the mainstream flow away from the blade surface near the tip, thus altering the detailed pressure distribution since the apparent blade shape seen by the freestream is changed. This displacement effect results in two pressure minima in the pressure profile of row 2. The mainstream flow is displaced further downstream from the leading edge for row 2 than for row 1. This is because of the streamwise movement of the leakage as it leaves the gap. As a result there is a suction peak, the actual blade profile suction peak, very close to the leading of the blade for row 2. The second pressure minimum appears to be located where the leakage vortex is in contact with the blade suction surface. The pressure distributions for the lower rows are not as significantly

Fig. 6.3.3(b)

affected as the upper rows because of the weakening of the leakage vortex.

At the above clearance setting the suction peak for row 1 is higher than that of row 13. This is presumably the result of the formation of the tip leakage vortex discussed earlier in Section 6.2, especially since the location of this peak seems to correspond to the region of the tip leakage vortex. There is also substantial unloading on the pressure surface of row 1. This extends for the entire chordwise length and is a direct result of the induced spanwise flow on the pressure surface due to the tip gap. Oil film traces at the setting showed that the spanwise flow was pronounced in the top 10 mm of the blade. This pressure surface unloading is significantly reduced by row 2. Figure 6.3.3(b) shows the spanwise extent of the effect of the leakage vortex on the pressure distribution of the blade. It is evident that spanwise movement of this vortex as it moves streamwise has the effect of maintaining a higher suction pressure at a much later chordwise location. A summary of the location of the suction pressure peaks for the different clearances is presented later.

The trend in the pressure loading distribution started at a C_l ratio of 0.0095 is further established at a ratio of 0.0158. This is shown in figure 6.3.4(a, b). The streamwise shift and actual magnitude of the leakage suction peaks of the upper rows have increased. This is a result of increased mass flow through the clearance gap and the location of the leakage vortex. The pressure unloading on the pressure surface has also increased compared to the smaller gap size. In addition to the above, a third pressure minimum seems to occur on the suction surface for row 1 at approximately $0.6C$ downstream of the leading edge. This peak is somewhat insignificant compared to the other two. However, as will be seen in the following paragraphs, this third suction peak increases in relative magnitude as the gap is opened and also becomes noticeable in the lower static tap rows.

At a clearance height/chord ratio of 0.0222, the third pressure minimum in the pressure distribution on the suction surface of the blade is very distinct in the upper rows of static pressure taps (rows 1 to 3), as shown in figure 6.3.5(a, b). This third suction peak located just downstream of the midchord location is still relatively insignificant compared to the profile and the first tip leakage peaks. However, it appears to be increasing in

Fig.6.3.4(a) Blade Pressure Distribution ($C_l/C=0.0158$)

Fig. 6.3.4(b) Blade Pressure Distribution $(C1/C)=0.0158$

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Fig. 6.3.1(b) Blade Pressure Distribution ($C1/C2=0.0222$)

Fig. 6.3.(b) Blade Pressure Distribution ($C/C=0.0222$)

magnitude as the gap is opened. No flow visualization experiments were conducted at this clearance, nevertheless, trends established up to this point became more pronounced at the largest clearance tested.

The pressure distributions for the maximum $\frac{Q_l}{Q}$ ratio of 0.0286 are shown in figure 6.3.6(a, b). The three pressure minima are very pronounced in the upper static tap rows. The locations of the second and third suction peaks away from the leading edge seem to correspond to the locations of the two tip leakage vortices discussed in Section 6.2. For the remainder of this report, these peaks will be referred to as the *first* and *second* tip leakage peaks, respectively. The second leakage peak is relatively significant at this gap setting. This is probably the result of a stronger second tip leakage vortex due to the increased leakage flow. Also noticeable in figure 6.3.6(a) is the suction pressure minimum between the two leakage peaks. For row 1 this pressure is about 10% higher than the corresponding pressure at row 13. This may be an indication of relatively low kinetic energy fluid in this region. Figure 6.3.6(b) shows that the spanwise effect of the second leakage vortex extends beyond row 6 and succeeds in maintaining a higher suction pressure than row 13 all the way to the trailing edge.

Figure 6.3.7(a to c) summarizes the effect of clearance height on the pressure distribution of rows 1, 2 and 13, respectively. It should be noted that a given row moves away from the endwall as the clearance is increased and will thus be influenced less and less by the endwall boundary layer. It is in fact unclear from the oil flow to what extent a horse-shoe vortex continues to be formed as the clearance increases. In any event, it is reasonable to assume that as the clearance increases the blade pressure distribution near the tip will be influenced mainly by the tip leakage flow field. At smaller values of clearance, a fairly complex interaction between the tip leakage flow and the endwall boundary layer is likely to be occurring.

Because of the fullness of the upstream velocity profiles (figure 6.3.2), only row 1 is significantly affected by changes in the position in the profile. For convenience the location of rows 1, 2 and 13 with respect to the endwall are listed in table 6.3.1. Figure 6.3.7(a) shows the continuous streamwise shift in the leakage suction pressure peaks and increased

Fig. 6.3.7(a) Blade Pressure Distribution ($CI/C=0.0286$)

Fig. 6.3.3(b) Blade Pressure Distribution ($C_I/C=0.0286$)

Fig. 6.3.7(a) Blade Pressure Distribution for Row 1

Fig. -6.3.7(b) Blade Pressure Distribution for Row 2

Fig. 6.3.7(c) Blade Pressure Distribution for Row 13

Row #		Z/h			
Cl/c	0	0.0095	0.0158	0.0222	0.0286
1	-0.013	-0.022	-0.030	-0.038	-0.046
2	-0.038	-0.047	-0.055	-0.063	-0.071
13	-0.413	-0.422	-0.430	-0.438	-0.446

Table 6.3.2: Location of Static Tap Rows away from the Endwall

magnitude of these peaks with clearance height for row 1. The increase in the suction peak is quite distinct, especially for the lower clearances. Increased tip leakage flow seems to be the main reason for the increase.

The shift in the leakage suction peaks seems to imply a chordwise movement away from the leading edge of the origin of the leakage vortices as the clearance height is increased. A graphical summary of the location of the suction peaks is provided in figure 6.2.8. From the loci of the measured peaks, the location of the peak at the blade tip is extrapolated. As expected, there is little change in the location of the profile peaks for the rows at different clearances.

The pressure distribution for row 2, figure 6.3.7(b), show similar trends as those observed for row 1. The profile suction peak is more noticeable for this row than for row 1. Figure 6.3.7(c) shows the pressure distribution for row 13 at the five gap settings. The unloading on the suction surface is quite substantial as the gap is opened. At the maximum clearance height there is approximately 20% increase in the suction peak compared to the no gap case. The upstream dynamic pressure is quite uniform towards the midspan region, thus this row should have experienced very similar dynamic pressures at the different clearances. However, increasing the gap size effectively reduces the aspect ratio of the blade and the suction surface unloading results from the induced spanwise flow towards the blade tip.

One striking feature from figure 6.3.7(a) is the amount of flow diffusion that occurs on the suction surface of the blade. The worse case occurred for row 1 at maximum clearance. Based on a diffusion factor defined as the ratio of the difference between maximum and minimum velocity on the suction surface to the maximum suction surface velocity, a value of 0.40 was calculated. This compares quite well to the two-dimensional value (*midspan at zero clearance*) of about 0.35, and is within the accepted range of 0 to 0.45 (*Horlock, 1966*). However, the diffusion for row 1 at maximum clearance occurs over a chordwise length stretching from approximately $\frac{1}{4}C$ to $\frac{1}{2}C$, while in the two-dimensional case, this diffusion practically spans the entire chordwise length of the blade. Therefore, the adverse pressure gradient is much more severe for row 1 at maximum clearance. Diffusion factors

Fig. 6.3.2 Location of Suction Peaks for Different Clearances

from two-dimensional cascade studies which included the rate of diffusion with chordwise distance would probably indicate flow separation for this case. However, it is evident that there was no flow separation on the blade suction surface. This appears to result from the three-dimensional nature of the flow field in the tip region.

From the pressure distributions presented thus far, it is rather difficult to determine the effect of clearance height on the overall performance of the blade. It is evident that the loading distribution is altered from the zero clearance case, increased in some areas and decreased in others. For example, the suction peak pressure for row 1 is higher for the maximum clearance than any other row at any clearance. In order to quantify the effect of clearance on the blade loading, force coefficients were computed from the pressure distributions.

The blade pressure distribution for each row of static taps was integrated to determine the resultant force (*lift*) per unit span generated by that row. This was done by connecting each pair of points on the pressure profile with a straight line, creating a panel. The resultant force normal to each panel is approximated from the mean of the two end-point pressures and the length of the panel. By repeating this around the entire blade contour, the lift, and the normal and axial forces were computed. In addition, the lines of action of these forces were determined by summing the moments caused by each panel. The routine used for calculating these performance parameters is listed in Appendix J. Note that this procedure is somewhat approximate since a linear variation of C_p is assumed along each panel. However, the panel lengths are short enough that the resulting error should be small. In any case, the parameters computed are used for comparison with parameters obtained by the same technique. Therefore, the trends resulting from using this procedure would be the same as those of a more accurate method employing a more elaborate integrating scheme.

The ratios of the lift coefficients for different clearances to midspan the lift coefficient of row 13 at zero clearance (*effectively, the two-dimensional value for the blade profile*) are plotted in figure 6.3.9. At zero clearance the unloading due to the presence of the upstream endwall boundary layer is evident for the upper three rows. From row 4 to row

Fig. 6.3.9 Spanwise Variation of Lift

13, the loading is nearly constant with a slight increase towards midspan.

The situation is quite different for the non-zero clearances. As the gap size is increased the lift generated at row 13 is gradually reduced. At the maximum $\frac{Cl}{C}$ ratio, there is actually about 20% reduction in the lift generated at row 13 compared to the zero clearance case. From figure 6.3.7(c) it is evident that the main factor causing the reduction in blade loading is the reduction in suction pressure on the suction surface of the blade as the gap size is increased. This confirms that the spanwise effect of the loss core (*mainly leakage for large gaps*) on the suction surface flow increased with gap size, and this effect extended far away from the tip even for the smaller gaps.

Another noticeable development for the non-zero clearance case is the peak in the lift coefficient close to the blade tip, before the eventual decrease at the tip. In fact for the maximum clearance tested, the lift coefficient at row 2 is approximately 15% higher than it is for row 13 at that clearance. This indicates that the high energy leakage fluid increased the overall suction pressure along the suction surface for the upper rows. This would also have been the case for row 1 if the significant pressure surface unloading had not occurred for this row. Note that the lift retained at row 1 for the different clearances is 80–90% of the two-dimensional value, indicating that a high level of loading is retained very close to the tip.

Figure 6.3.10(a, b) divides the lift force into force components normal and parallel to the undisturbed streamwise directions. Again the two-dimensional value is used as the normalizing factor. The trends in the normal force coefficient are similar to those observed for the lift coefficient. However, the peaks that occur at row 2 are slightly less significant compared to the midspan value. Actually, for the smaller clearances there is a gradual decrease before the eventual peak and decline towards the tip. Thus the increase observed in the lift coefficient at row 2 was mainly due to an increased axial force component, as is evident by comparing figure 6.3.10(a) and 6.3.10(b). This implies a rotation of the lift vector for this row towards the streamwise direction with increasing clearance. Note that it is the normal force that creates the torque in a turbine. Therefore, this is usually the component of interest.

Fig. 6.3.10(a) Spanwise Variation of Normal Force Coeff.

Fig. 6.3.10(b) Spanwise Variation of

The trends observed in the variation of the blade pressure distributions and blade forces for the different clearances appear to agree with those of the flow models by *Lakshminarayana* and *Horlock* (1967) and *Lakshminarayana* (1970). These models also claim to predict the flow direction at the exit plane of the cascade quite well. One short-coming of these models is that they require k , the fraction of lift retained at the tip for a particular clearance to be known. However, for typical turbomachinery clearances, this parameter can be estimated from the cascade geometry (see *Lakshminarayana*, 1970). These models could be an invaluable tool for the the designer once their validity and diversity have been further investigated.

Figure 6.3.11 shows the axial and normal variation of the location of the centre of pressure for each row at the different clearances. The coordinate system used is as defined in figure 3.2.1, with the leading edge of the blade being the origin of the axes. The orientation of the blade is as defined in Appendix B. As expected there is a streamwise shift in the location of the centre of pressure as the tip is approached. However, this trend only extends to row 2. At row 1 the X_{CP} is slightly closer to the leading edge than row 2. Also indicated is the streamwise movement of this point for a particular row at different clearances. The location of the centre of pressure with respect to the y -axis mainly shows the effect of the unloading that occurs on the pressure surface of the blade for non-zero clearances. This is most pronounced for the larger gaps where the pressure surface unloading shifts the centre of pressure towards the suction surface.

The effect of the cavity at the blade tips on the pressure distribution for a particular static tap row and clearance setting is shown in figure 6.3.12. As shown, the measured points with and without the cavity open are quite similar. At the end of the blade unloading experiments a repeatability check was done. This included resetting the clearance height. The results shown in figure 6.3.13 are practically identical.

Up until now there has been no mention of possible errors in the pressure distributions. The possible sources for blade surface pressure measurement errors are:

- (1) Static tap geometry;

Fig. 6.3.11 Location of Centre of Pressure

Fig. 6.3.12 Blade Pressure Distribution ($C_{l,C} = 0.0286$)

Fig. 6.3.13 Blade Pressure Distribution (C_l/C)=0.0286

- (2) Turbulence in the flow;
- (3) Static tap location;
- (4) The HP A/D converter;
- (5) Static hole error;
- (6) Transducer calibration;
- (7) The error due to taking a finite number of samples from the flow field.

It is difficult to quantify (1) to (3), however, they are expected to be negligible. As discussed previously, the error due to the A/D converter is negligible. Estimating the static hole error based on experimental data from *Shaw* (1959), it was found that its contribution would be less than 0.01% of the actual blade surface pressure. When compared to the estimated 0.2% uncertainty (Section 4.2) due to the combined effect of taking a finite number of flow-field samples and the transducer calibration error, this source can be neglected. Thus it can be concluded that the measured blade surface pressures are within an accuracy of 0.2%. When the inaccuracy of the integration method for the blade force coefficients are taken into account, it is expected that the blade force coefficients should be within 1% of the actual values.

Chapter 7

CONCLUSIONS AND RECOMMENDATION

Since the design and fabrication of the cascade test section constituted a major portion of the present work, the test section calibration experiments are dealt with before dealing with the tip leakage experiments.

7.1 Test Section Calibration Experiments

Simplicity and versatility were prime considerations in the design of the test section. In general the performance of the facility was found to be satisfactory. The flow quality in the test section was adequate for the experiments conducted, however, improvements can be made.

The non-uniformity in the total pressure profile towards the centre line of the test section can probably be reduced by increasing the porosity of the screens in the centre region of the settling chamber. The high concentration of screens in this region does not appear to be necessary to improve the uniformity and turbulence level of the outflow from the settling chamber. This is not a flaw in the test section itself, but rather in the flow source. However, since the test section design was centred around this existing piece of hardware, the two more or less become one entity.

For this particular test cascade, the lower sidewall of the test section was slightly within the would be streamtube through the blade passage between blades 4 and 5 in the test cascade. This did not turn out to be a problem in this study, however, for other test cascades this problem can be eliminated by lowering the sidewall or more simply by moving the blade stem towards the trailing edge. The location of the blade stem is important since for a particular blade stagger the position of the leading edge of the blade

is determined by the point of rotation of the blades.

The turbulence intensity of the freestream flow in the test section flow was measured to be approximately 0.4%. This is somewhat lower than the levels experienced in actual machines (*about 10%*). Turbulence generators could be added at the test section inlet to obtain a higher level of turbulence.

7.2 Tip Leakage Experiments

Despite only carrying out a modest amount of tip leakage experiments the results were revealing and in some cases, surprising. Note that the simplest configuration and situation, a two-dimensional cascade with no relative motion and a plain endwall, was used throughout. Despite this the results indicate a complex flow field in the tip region of the blade.

The flow visualization experiments show that for the zero clearance, the loss core formed by the horse-shoe vortex and the passage vortex is confined to a narrow region at the intersection of the blade suction surface and the endwall. The channelwise (*blade-to-blade plane*), spanwise and cross-sectional area of this rotational core increases as the gap size is opened. For the larger clearances the loss core is mainly comprised of the tip leakage flow since the horse-shoe and passage vortices seem to decrease in strength. Also noticed at the larger gap settings is the apparent emergence of a second distinct tip leakage vortex. Further investigations into the occurrence of this second vortex is necessary since the source of it is unclear from this study. Actual three-dimensional flow field measurements with a pressure probe in the tip region would go a long way in clarifying this situation. Flow visualization experiments conducted with parameters such as the endwall boundary layer thickness varied, may shed some light in this area. In addition to this, smoke studies using different coloured smoke injected concurrently at different points in the flow field may be able to isolate the different vortices in the loss core.

The smoke flow experiments qualitatively showed that the flow through the tip gap is primarily driven by the pressure difference between the pressure and suction surfaces

of the blade. This is especially true for the smaller clearances, for the larger gaps the inertia forces become more significant. The exact leakage flowrate as well as flow direction measurements using at least a three-hole pressure probe should be carried out in the future to quantify the above observations.

The measured blade pressure distributions show a continuous streamwise shift in the location of the main suction pressure peak for regions close to the blade tip as the gap size is increased. This implies a shift in the point where the tip leakage vortex is in contact with the suction surface. The magnitude of this peak also increases with increasing clearance height. This appears to be the result of increased leakage flow at the larger gap settings. In addition to the above mentioned pressure minimum, there are two other pressure minima. These correspond to the profile suction peaks and a relatively small peak apparently resulting from the second tip leakage vortex.

The effect of the leakage flow is to decrease the pressure on the pressure and suction surfaces of the blade for regions close to the blade tips. For the range of gap/chord ratios tested the decrease in the pressure surface pressure is almost compensated for by an increase in the suction pressure on the suction surface at a particular clearance. In fact there are peaks in the lift and normal blade force coefficients a little inward from the blade tip. For the larger gaps these peaks are significantly (10%) higher than the midspan value. This seems to indicate that only part of the bound vorticity is shed at the blade tip.

Tip retained loading implies that vortex lines must cross the tip gap. Endwall static pressures measured from static taps located around the profile of the blade tip should indicate whether this does in fact happen.

The performance prediction models that were developed by *Lakshminarayana* and *Horlock* (1967) and *Lakshminarayana* (1970) appear to predict trends observed in this study and other similar works, such as *Lakshminarayana* and *Horlock* (1967). These could be very useful to the designer who in the past did not have any way of incorporating the aerodynamics of the tip region flow into his design calculations. However, these models need to be further tested to check their generality. The data obtained in this study can

As mentioned at the beginning of this report, the tip region flow field appears to be affected by many variables such as endwall boundary layer thickness, tip gap geometry, blade loading and relative motion between the endwall and blade tip, to mention a few. The design of the cascade test section allows many of these parameters to be varied. Therefore, many more experiments remain to be carried out, but firstly, the simplest case, the one initiated here, needs to be thoroughly investigated and fully comprehended.

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APPENDIX A

Geometric specifications of the Pratt & Whitney turbine blade :

Leading edge wedge angle = 53.99°

Trailing edge wedge angle = 1.96°

Maximum thickness = 0.0788"

Blade stagger = 49.44°

Incidence (design) = -7.48°

Coordinate points on blade profile based on the coordinate system shown in Fig. A.1 (i.e. Pratt's) :

Fig. A.1

Pt. No.	X (in.)	Y (in.)
1.000	-.26085	.23003
2.000	-.26072	.23271
3.000	-.26014	.23515
4.000	-.25909	.23759
5.000	-.25832	.23921
6.000	-.25777	.24003
7.000	-.25722	.24085
8.000	-.25663	.24166
9.000	-.25607	.24229
10.000	-.25386	.24434
11.000	-.25104	.24622
12.000	-.24609	.24859
13.000	-.24102	.25050
14.000	-.23585	.25213
15.000	-.23073	.25337
16.000	-.22570	.25430
17.000	-.22064	.25505
18.000	-.21543	.25540
19.000	-.21047	.25545
20.000	-.19765	.25444
21.000	-.18487	.25166
22.000	-.17217	.24737
23.000	-.15933	.24180
24.000	-.14648	.23494
25.000	-.13362	.22702

26.000	-.12086	.21781
27.000	-.10796	.20751
28.000	-.09507	.19615
29.000	-.08217	.18361
30.000	-.06935	.16987
31.000	-.05642	.15502
32.000	-.04343	.13899
33.000	-.03053	.12193
34.000	-.01766	.10406
35.000	-.00468	.08533
36.000	.00830	.06620
37.000	.02129	.04647
38.000	.03419	.02629
39.000	.04719	.00571

Pt. No.	X (in.)	Y (in.)
40.000	.06018	-.01548
41.000	.07313	-.03709
42.000	.08608	-.05920
43.000	.09907	-.08184
44.000	.11205	-.10488
45.000	.12503	-.12831
46.000	.13789	-.15226
47.000	.15085	-.17661
48.000	.16379	-.20135
49.000	.17671	-.22636
50.000	.18951	-.25176
51.000	.20239	-.27745
52.000	.21524	-.30340
53.000	.22807	-.32951
54.000	.23822	-.35059
55.000	.24846	-.37175
56.000	.25244	-.38039
57.000	.25315	-.38242
58.000	.25335	-.38395
59.000	.25326	-.38480
60.000	.25297	-.38625
61.000	.25260	-.38731
62.000	.25216	-.38824
63.000	.25096	-.38974
64.000	.24988	-.39074
65.000	.24909	-.39121
66.000	.24776	-.39182
67.000	.24614	-.39219
68.000	.24450	-.39233
69.000	.24402	-.39227
70.000	.24288	-.39207
71.000	.24166	-.39158
72.000	.24030	-.39089
73.000	.23897	-.38975
74.000	.23800	-.38849
75.000	.23550	-.38430
76.000	.23046	-.37476
77.000	.21765	-.35121
78.000	.20480	-.32843
79.000	.19193	-.30624
80.000	.17915	-.28462

Pt. No.	X (in.)	Y (in.)
81.000	.16623	-.26333
82.000	.15330	-.24273
83.000	.14036	-.22235
84.000	.12750	-.20225
85.000	.11453	-.18254
86.000	.10155	-.16302
87.000	.08856	-.14379
88.000	.07567	-.12475
89.000	.06267	-.10591
90.000	.04966	-.08737
91.000	.03665	-.06903
92.000	.02374	-.05100
93.000	.01072	-.03328
94.000	-.00230	-.01597
95.000	-.01531	.00082
96.000	-.02823	.01721
97.000	-.04125	.03308
98.000	-.05426	.04845
99.000	-.06728	.06331
100.000	-.08019	.07757
101.000	-.09320	.09123
102.000	-.10621	.10430
103.000	-.11921	.11678
104.000	-.13211	.12866
105.000	-.14511	.13996
106.000	-.15810	.15067
107.000	-.17109	.16089
108.000	-.18398	.17054
109.000	-.19696	.17969
110.000	-.21509	.19185
111.000	-.22549	.19842
112.000	-.23579	.20459
113.000	-.24618	.21063
114.000	-.25122	.21349
115.000	-.25623	.21745
116.000	-.25740	.21888
117.000	-.25844	.22053
118.000	-.25944	.22295
119.000	-.26006	.22498
120.000	-.26062	.22742
121.000	-.26085	.23003

APPENDIX B

Cascade specifications:

Scale 1:2.25

Fig. A.2 Blade Setting

Scale (model/P&W prototype)	=	1:12.29
Nominal blade chord	=	250 mm
Blade height	=	200 mm
Blade spacing	=	150 mm
Blade maximum thickness	=	24.6 mm
Inlet flow angle	=	0° (axial)
Stagger	=	35.6°
Metal turning	=	59.2°
Flow incidence	=	-12.89°

APPENDIX C

Data used to generate the shape of the cavity. These coordinates are provided with the blade oriented in Pratt's coordinate system, Fig. C.1. However, the blade has been scaled up to full scale (i.e., by a factor of 12.29). These coordinates are those to be contoured by the centre of a 3.2-mm diameter end-mill. The cavity depth is 12.7 mm.

Fig. C.1

Pt. No.	X (in.)	Y (in.)
1.000	-2.100	2.520
2.000	-1.915	2.480
3.000	-1.675	2.315
4.000	-1.440	2.125
5.000	-1.215	1.900
6.000	-1.000	1.650
7.000	-0.800	1.400
8.000	-0.580	1.100
9.000	-0.330	0.740
10.000	-0.125	0.500
11.000	-0.330	0.740
12.000	-0.625	1.085
13.000	-0.840	1.365
14.000	-1.050	1.570
15.000	-1.310	1.810
16.000	-1.530	2.020
17.000	-1.760	2.200
18.000	-1.965	2.360
19.000	-2.100	2.520

APPENDIX .D

Listing of the Fortran program used to generate the full scale blade shape and perform the required radius compensation using the provided data points from the actual (P&W) blade shape. The coordinate system rotation and translation routine is also included.

```

C      PROGRAM GENERATES BLADE PROFILE USING GIVEN BLADE COORDINATES
C      USING THE CUBIC SPLINE FITTING TECHNIQUE
C      AUTHOR: KIM AMRUD
C      DATE: FEB 29, 1984
      PROGRAM BLABEP
      REAL X(500),Y(500),S(500),B1(500),C1(500),D1(500),B2(500),
*        C2(500),D2(500),XOUT,YOUT,SIN,MAG,RAD,DSIN1,DSIN2,
*        XOUT1,XOUT2,YOUT1,YOUT2,DX,DY,XX(500),YY(500),BB1(500),
*        CC1(500),DD1(500),BB2(500),CC2(500),DD2(500)
      COMMON /RADIUS/ RAD
      INTEGER I,M,NODIV,J,NNP,NEL

C
C      INPUT NO. OF COORDINATE POINTS, MAGNIFICATION FACTOR AND
C      COORDINATE POINTS
      READ(5,*) M,MAG,RAD
      READ(5,*) (X(I),Y(I),I=1,M)

C
C      READ NO. OF SUBDIVISIONS IN S FOR EACH SEGMENT
      READ(5,*) NODIV

C
C      EXPRESSING X AND Y AS A FUNCTION OF S AND MAGNIFYING BLADE
      DO 10 I=1,M
        S(I)=I
        X(I)=X(I)*MAG
        Y(I)=Y(I)*MAG
10    CONTINUE

C
C      DETERMINING SPLINE COEFFICIENTS USING X AND Y AS F(S)
      CALL SPLINE(M,S,X,B1,C1,D1)
      CALL SPLINE(M,S,Y,B2,C2,D2)

C
C      WRITE(6) FOR ASGARD "TOPLOT" ROUTINE - CP6
      NNP=(M-1)*NODIV+1
      NEL=NNP-1

```

```

WRITE(6,*) '0',NNP,NEL,'2',' 0'
DO 5 I=1,NEL
  WRITE(6,*) I,' 21 ',' 1 ',I,(I+1)
5 CONTINUE
C
C WRITING THE COORDINATES OF THE BLADE AT SMALLER INCREMENTS
XOUT=X(1)-RAD
YOUT=Y(1)
WRITE(6,*) '1',XOUT,YOUT,'0'
XX(1)=XOUT
YY(1)=YOUT
K=2
DO 30 I=2,(M-1)
  DSIN1=S(I)-0.05
  DSIN2=S(I)+0.05
  XOUT1=SPLINT(M,S,X,B1,C1,D1,DSIN1)
  YOUT1=SPLINT(M,S,Y,B2,C2,D2,DSIN1)
  XOUT2=SPLINT(M,S,X,B1,C1,D1,DSIN2)
  YOUT2=SPLINT(M,S,Y,B2,C2,D2,DSIN2)
  CALL NEWCO(XOUT1,XOUT2,YOUT1,YOUT2,DX,DY)
  XOUT=X(I)+DX
  YOUT=Y(I)+DY
  WRITE(6,*) K,XOUT,YOUT,'0'
  K=K+1
  XX(I)=XOUT
  YY(I)=YOUT
30 CONTINUE
WRITE(6,*) NNP,(X(1)-RAD),Y(1),'0'
XX(NNP)=X(1)-RAD
YY(NNP)=Y(1)
CALL SPLINE(M,S,XX,BB1,CC1,DD1)
CALL SPLINE(M,S,YY,BB2,CC2,DD2)
K=1
DO 40 I=1,(M-1)
  WRITE(8,100) XX(I),YY(I)
  K=K+1
  IF(NODIV .GT. 1) THEN
    DO 20 J=1,(NODIV-1)
      SIN=S(I)+((S(I+1)-S(I))/NODIV)*J
      YOUT=SPLINT(M,S,YY,BB2,CC2,DD2,SIN)
      XOUT=SPLINT(M,S,XX,BB1,CC1,DD1,SIN)
      WRITE(8,100) XOUT,YOUT
      K=K+1
20 CONTINUE
    ELSE
    ENDIF
40 CONTINUE
WRITE(8,100) XX(NNP),YY(NNP)
100 FORMAT(10X,F9.4,5X,F9.4)
STOP
END
C
C*****

```

```

C
C SUBROUTINE FOR RADIUS COMPENSATION
SUBROUTINE NEWCO(XOUT1,XOUT2,YOUT1,YOUT2,DX,DY)
REAL XOUT1,XOUT2,YOUT1,YOUT2,DX,DY,THETA,RAD
COMMON /RADIUS/ RAD
IF((XOUT2 .GT. XOUT1) .AND. (YOUT2 .GE. YOUT1)) THEN
    THETA=ATAN((YOUT2-YOUT1)/(XOUT2-XOUT1))
    DX=-RAD*SIN(THETA)
    DY=RAD*COS(THETA)
ELSE
ENDIF
IF((XOUT2 .GT. XOUT1) .AND. (YOUT2 .LE. YOUT1)) THEN
    THETA=ATAN((YOUT1-YOUT2)/(XOUT2-XOUT1))
    DX=RAD*SIN(THETA)
    DY=RAD*COS(THETA)
ELSE
ENDIF
IF((XOUT2 .LT. XOUT1) .AND. (YOUT2 .LE. YOUT1)) THEN
    THETA=ATAN((YOUT1-YOUT2)/(XOUT1-XOUT2))
    DX=RAD*SIN(THETA)
    DY=-RAD*COS(THETA)
ELSE
ENDIF
IF((XOUT2 .LT. XOUT1) .AND. (YOUT2 .GE. YOUT1)) THEN
    THETA=ATAN((YOUT2-YOUT1)/(XOUT1-XOUT2))
    DX=-RAD*SIN(THETA)
    DY=-RAD*COS(THETA)
ELSE
ENDIF
IF((ABS(XOUT1-XOUT2) .LT. 0.01E-9) .AND. (YOUT2 .LE. YOUT1))
* THEN
    DX=RAD
    DY=-RAD
ELSE
ENDIF
IF((ABS(XOUT2-XOUT1) .LT. 0.01E-9) .AND. (YOUT2 .GE. YOUT1))
* THEN
    DX=-RAD
    DY=0
ELSE
ENDIF
RETURN
END

```

C*****A.

```

C
C      S P L I N E   M O D U L E
C
C      AUTHOR:   J E NEILSON
C
C      DATE: SEPT 1978.
C
C*****
C
C*****
C
C      S P L I N E   C A L C U L A T I O N   R O U T I N E
C
C*****
C
C      SUBROUTINE SPLINE(N,X,Y,B,C,D)
C      INTEGER I,ISTOP,J,N
C      REAL X(N),Y(N),B(N),C(N),D(N),DELTA(500)
C
C*****
C
C      DEFINITIONS -
C
C      ARGUMENTS-
C
C      N - NUMBER OF DATA PAIRS
C      X,Y - VECTORS OF DATA PAIRS
C      B,C,D - VECTORS OF SPLINE COEFFICIENTS
C
C      LOCAL VARIABLES-
C
C      DELTA - WORKING VECTOR
C      I,J - DO LOOP INDICES
C      ISTOP - DO LOOP PARAMETER
C
C*****
C
C      DO 99 I=1,N
C          B(I)=0
C          C(I)=0
C          D(I)=0
C
C      99  CONTINUE
C          IF(N.LT.4.OR.N.GT.500) GOTO 60
C          ISTOP = N - 1
C          DO 10 I = 1,ISTOP
C              D(I) = X(I+1)-X(I)
C              IF(D(I).LE.0.) GOTO 60
C              DELTA(I) = (Y(I+1)-Y(I))/D(I)
C
C      10  CONTINUE

```

```

      B(1) = -D(1)
      DO 20 I = 2,ISTOP
        B(I) = 2.*(D(I-1)+D(I))-D(I-1)**2/B(I-1)
20    CONTINUE
      B(N) = -D(N-1)-D(N-1)**2/B(N-1)
      C(1) = D(1)**2*((DELTA(3)-DELTA(2))/(X(4)-X(2))-
+ (DELTA(2)-DELTA(1))/(X(3)-X(1)))/(X(4)-X(1))
      DO 30 I = 2,ISTOP
        C(I) = (DELTA(I)-DELTA(I-1))-D(I-1)*C(I-1)/B(I-1)
30    CONTINUE
      C(N) = -D(N-1)**2*((DELTA(N-1)-DELTA(N-2))/(X(N)-X(N-2))-
+ (DELTA(N-2)-DELTA(N-3))/(X(N-1)-X(N-3)))/(X(N)-X(N-3))-
+ D(N-1)*C(N-1)/B(N-1)
      C(N) = C(N)/B(N)
      DO 40 J = 1,ISTOP
        I = N - J
        C(I) = (C(I)-D(I)*C(I+1))/B(I)
40    CONTINUE
      DO 50 I = 1,ISTOP
        B(I) = DELTA(I)-D(I)*(C(I+1)+2.*C(I))
        D(I) = (C(I+1)-C(I))/D(I)
        C(I) = 3.*C(I)
50    CONTINUE
      RETURN
60    CONTINUE
      N = -IABS(N)
      RETURN
      END

```

```

C
C
C
C
C*****
C
C          E V A L U A T I O N   R O U T I N E
C
C*****
C
C      REAL FUNCTION SPLINT(N,X,Y,B,C,D,XIN)
C      INTEGER N,LEFT,RIGHT,MID
C      REAL X(N),Y(N),B(N),C(N),D(N),XIN,DX

```

```

C*****
C
C  DEFINITIONS -
C
C  ARGUMENTS -
C
C    N - NUMBER OF DATA PAIRS
C    X,Y - VECTORS OF DATA PAIRS
C    B,C,D - VECTORS OF SPLINE COEFFICIENTS
C    XIN - VALUE OF INDEPENDENT VARIABLE TO BE INTERPOLATED
C
C  LOCAL VARIABLES -
C
C    DX - DISPLACEMENT FROM LEFT BOUNDARY
C    LEFT - LEFT CURSOR FOR BINARY SEARCH
C    MID - MID POINT PROBE
C    RIGHT - RIGHT CURSOR FOR BINARY SEARCH
C*****
C
C    LEFT = 1
C    RIGHT = N
10  MID = (LEFT+RIGHT)/2
C    IF(MID.EQ.LEFT) GOTO 30
C    IF(XIN.LT.X(MID)) GOTO 20
C    LEFT = MID
C    GOTO 10
20  RIGHT = MID
C    GOTO 10
30  DX = XIN-X(LEFT)
C    SPLINT = Y(LEFT)+DX*(B(LEFT)+DX*(C(LEFT)+DX*D(LEFT)))
C    RETURN
C    END

```

```
PROGRAM ROTATE
* THIS PROGRAM READS A SHAPE FILE AND ROTATES IT PHETA DEGREES
  REAL S, C, XR, YR
  INTEGER J
  REAL PHETA, X, Y
* ROTATE COORDINATES 51 DEGREES
  PHETA=0.89011792
  S=SIN(PHETA)
  C=COS(PHETA)
  DO 20 J=1,121
    READ (5,*) X,Y
    XR=(C*X)-(S*Y)
    YR=(S*X)+(C*Y)
    XR=XR+5.500
    YR=YR+1.925
    WRITE(6,*) XR,YR
20  CONTINUE
40  STOP
END
```

APPENDIX E

Details of main (static-tapped) blade:

Fig. E.1 Location of Static Tap Rows away from the tip of the blade.

Fig. E.2

A listing of the coordinates of the static taps and the slope of the blade surface at those points, based on the coordinate system in Fig. E.2. :

Tap No.	X (in.)	Y (in.)	θ (degrees)
1.000	-3.2052	2.8267	-91.19
2.000	-2.9803	3.0725	-19.63
3.000	-2.7554	3.1290	-9.51
4.000	-2.5317	3.1364	3.62
5.000	-2.2872	3.0971	15.60
6.000	-2.0844	3.0283	21.96
7.000	-1.8472	2.9140	28.91
8.000	-1.6112	2.7689	34.72
9.000	-1.4220	2.6276	38.31
10.000	-1.2155	2.4543	41.95
11.000	-1.0410	2.2884	45.05
12.000	-.8677	2.1053	47.87
13.000	-.6932	1.9049	49.93
14.000	-.5334	1.7083	52.01
15.000	-.3748	1.4982	53.64

Tap No.	X (in.)	Y (in.)	θ (degrees)
16.000	-.2175	1.2794	54.83
17.000	-.0578	1.0483	55.58
18.000	.1020	.8136	56.19
19.000	.2618	.5715	57.10
20.000	.4203	.3232	57.55
21.000	.5801	.0701	58.07
22.000	.7399	-.1905	58.81
23.000	.8984	-.4560	59.35
24.000	1.0582	-.7276	59.91
25.000	1.2179	-1.0053	60.40
26.000	1.3765	-1.2892	60.76
27.000	1.5363	-1.5768	61.42
28.000	1.6948	-1.8718	61.91
29.000	1.8533	-2.1704	62.17
30.000	2.0131	-2.4752	62.51
31.000	2.1716	-2.7825	62.99
32.000	2.3290	-3.0946	63.35
33.000	2.4875	-3.4105	63.50
34.000	2.6448	-3.7288	63.73
35.000	2.8033	-4.0496	64.08
36.000	2.9275	-4.3089	64.24
37.000	3.0541	-4.5682	64.43
38.000	2.8734	-4.6825	242.42
39.000	2.7272	-4.4109	241.33
40.000	2.5784	-4.1442	240.45
41.000	2.4064	-3.8455	239.79
42.000	2.2331	-3.5506	239.25
43.000	2.0426	-3.2360	238.28
44.000	1.7648	-2.7947	237.57
45.000	1.5670	-2.4850	237.02
46.000	1.4072	-2.2429	236.48
47.000	1.2474	-2.0033	236.15
48.000	1.0889	-1.7673	235.93
49.000	.9304	-1.5338	235.70
50.000	.7706	-1.3015	235.13
51.000	.6108	-1.0741	234.76
52.000	.4510	-.8480	234.57
53.000	.2913	-.6268	234.08
54.000	.1315	-.4093	233.38
55.000	-.0283	-.1966	232.61
56.000	-.1880	.0098	232.02
57.000	-.3466	.2114	231.24
58.000	-.5063	.4068	230.16
59.000	-.6673	.5961	229.26
60.000	-.8271	.7780	228.36
61.000	-.9857	.9537	227.15
62.000	-1.1454	1.1208	225.74
63.000	-1.3052	1.2818	224.47
64.000	-1.4809	1.4502	223.17
65.000	-1.6555	1.6100	221.50

<u>Tap No.</u>	<u>X (in.)</u>	<u>Y (in.)</u>	<u>θ (degrees)</u>
66.000	-1.8312	1.7599	219.76
67.000	-2.0070	1.9025	218.34
68.000	-2.1999	2.0512	216.67
69.000	-2.3830	2.1827	214.64
70.000	-2.5809	2.3167	213.67
71.000	-2.7394	2.4187	211.62
72.000	-2.8980	2.5145	210.69
73.000	-3.0664	2.6104	209.22

APPENDIX F

Calibration curves for Pressure Transducers:

1) Single Input Transducer.

Conversion formula:

$$\text{Pressure difference (Pa)} = (\text{Transducer output for no flow (mV)} + \text{Transducer output (mV)}) \times 1.408$$

2) Scanivalve Transducer (Fig. F.2).

Conversion formula:

$$\text{Pressure difference (Pa)} = (\text{Transducer output (mV)} \times 1.392)$$

APPENDIX G

Determination of sampling time necessary to obtain a representative mean flow field quantity.

- 1) Freestream Inlet Velocity. Measurements were done at midspan of the midheight traversing station described in section 4.3. Fig. G.1 shows the cumulative average of $(P_{TS}/P_{SC})_{dyn}$ as a function of the number of sampling cycles.

- 2) Inlet Boundary Layer. Samples were taken at 2.5 mm away from the topwall at the same station as in (1). See Fig. G.2 for results.

- 3) Blade Surface Pressures. Fig. G.3(a-d) show the variation for Different static taps for rows 1 and 13 at the no gap setting. Row 1 was at 2.67 mm away from the endwall, while row 8 was 37.67 mm away.

APPENDIX H

Listing of the HP Basic programs used for:

1) flowfield measurements at the test section inlet.

```

10 ! INLET FLOW PROFILE
20 ! OPTION BASE 1
30 DIM L(100), P1(100), P2(100)
40 DIM A(200), B(200)
45 DIM D1(200), D2(200)
50 P1=3456
60 GOSUB 440 ! CALL Init
70 FOR J=1 TO 100
75 BEEP 50,500
80 DISP "Enter 0 to stop"
90 INPUT T
100 IF T=0 THEN GOTO 180
102 DISP "Input no. of calls to
    Avdcv - Avdcv samples for 1
    2 sec for each call"
103 INPUT N1
104 DISP "Input starting point o
    f averaging averages"
105 INPUT N2
106 DISP "Input single input tra
    nsducer output for zero pres
    sure difference - in mV"
108 INPUT C3
109 C3=C3/1000
110 DISP "Input Y coordinate"
120 INPUT L(J)
130 BEEP 50,500
140 DISP "Press continue when re
    ady"
150 PAUSE
160 GOSUB 240
170 NEXT J
180 N=J-1
190 FOR I=1 TO N
200 P3=P1(I)/P2(I)
210 PRINT I, "Y=", L(I)
212 PRINT "P test sect. =", P1(I)
    , "Pa"
213 PRINT "P set. cham. =", P2(I)
    , "Pa"
216 PRINT "P test / P set. =", P3
217 P4=ABS(P3)^.5
218 PRINT "V test / V set. =", P
    4
220 NEXT I
230 END
240 FOR K=1 TO N1
250 P1=2
260 GOSUB 1250 ! CALL Avdcv
270 E1=P
280 P1=1
290 GOSUB 1250 ! CALL Avdcv
292 F1=F
294 GOSUB 1250 ! CALL Avdcv
296 F2=F
298 A(K)=(F1+F2)/2
300 P1=2
302 GOSUB 1250 ! CALL Avdcv
304 E2=F
306 B(K)=(E1+E2)/2
310 NEXT K
320 C1=0
330 C2=0
340 FOR M=1 TO N1
350 D1(M)=(C1+A(M))/M
360 C1=C1+A(M)
370 D2(M)=(C2+B(M))/M
380 C2=C2+B(M)
385 G1=0
390 G2=0
400 NEXT M
410 FOR M1=N2 TO N1
415 G1=D1(M1)+G1
420 G2=D2(M1)+G2
424 NEXT M1
426 P1(J)=G1/(N1-N2+1)*171.5
427 P1(J)=P1(J)*8.0931
428 P2(J)=(G2/(N1-N2+1)+C3)*174
429 P2(J)=P2(J)*8.0931
430 RETURN
440 ! SUBPROGRAM Init hp85
450 ! QA 7/08/80 GAC
460 ! P1=Dvmeter
470 DIM Q7$(23) ! DEVICE NAMES
480 !
490 ! INITIALIZE DEVICE ADDRESS
    ES
500 !
510 R0=709 ! 3497A SCANNER ADDRE
    SS
520 R1=722 ! 3456A DIGITAL VOLT
    METER ADDRESS
530 R2=724 ! 3437A SYSTEM VOLTM
    ETER ADDRESS
540 R3=0 ! ERROR FLAG
550 R4=0 ! USED IN SUBROUTINE "S
    trip"
560 O3=R0 DIV 100 ! INTEGER DIV
    IDE, i.e. Bus=IP(Scn/100)
570 O4=R1
580 R1$="0000000000000000"

```

```

590 ON ERROR GOTO 1020 ! IF P1 H
    AS NOT BEEN DEFINED
600 IF P1=3497 THEN R1=R0 ! PASS
    PARAMETER USED HERE
610 IF P1<>3456 AND P1<>3497 THE
    N 1020 ! Abort
620 OFF ERROR
630 !
640 !
650 ! VERIFY THAT INTERFACE IS H
    P-IB
660 !
670 ON ERROR GOTO 740
680 STATUS 03,0 : 06
690 OFF ERROR
700 !
710 ! PRINT ERROR MESSAGE TO CRT
    IF NO INTERFACE OR NOT HP-IB
    INTERFACE.
720 !
730 IF 06=1 THEN 840 ! OK
740 OFF ERROR
750 CLEAR
760 DISP USING "/,K" : "CAUTION
    HP-IB INTERFACE #"&VAL$(03
    )&" IS NOT PRESENT."
770 DISP USING "/,K" : "PROGRAM
    EXECUTION TERMINATED"
780 BEEP
790 STOP
800 !
810 ! INITIALIZE 82937A HP-IB IN
    TERFACE, RESET INTERFACE
820 !
830 !
840 ABORTIO 03 ! INITIALIZES IN
    TERFACE AND BUS
850 RESET 03 ! SENDS DEVICE CLE
    AR (DCL)
860 !
870 !
880 ! CHECK FOR EQUIPMENT ON BUS
    AT ALL ADDRESSES AND PRINT
    DEVICE NAMES
890 !
900 !
910 CLEAR @ DISP USING "/,2X,K,D
    ,/,/" : "EQUIPMENT PRESENT
    ON BUS " ; 03
920 FOR 02=0 TO 30
930 OUTPUT 03*100+02 USING "#" :
    ! ADDRESS DEVICE TO LISTEN
940 RESUME 03 ! DROP ATN
950 STATUS 03,2 : 05 ! SEE IF IT
    LISTENED
960 IF BINAND(05,3)=0 THEN 990 !
    NOT THERE
970 GOSUB 1000 ! CALL DEV_NAME
980 DISP USING 1000 : Q7$,02
990 NEXT 02
1000 IMAGE 37A," at address " 22
1010 GOTO 1060 ! EXIT
1020 OFF ERROR
1030 R3=2
1040 R$="Init"
1050 GOSUB 1600 ! CALL Warn
1060 RETURN
1070 ! SUBEND
1080 ! SUB DEV_NAME
1090 ! Q7$=DEVICE NAME
1100 Q7$=""
1110 IF 02<>04 MOD 100 THEN 1130
1120 Q7$="3456A Digital Voltmete
    r"
1130 IF 02<>R2 MOD 100 THEN 1150
1140 Q7$="3437A System Voltmeter
    "
1150 IF 02<>21 THEN 1170
1160 Q7$="hp85A System Computer"
1170 IF 02<>R0 MOD 100 THEN 1200
1180 Q7$="3497A Scanner,Timer"
1190 IF R1=R0 THEN Q7$=Q7$&"DVM
    "
1200 IF Q7$<>" THEN 1220
1210 Q7$="Device Unknown" ! UNKN
    OWN DEVICE=ANY DEVICE WITH
    NO LABEL
1220 !
1230 RETURN
1240 ! SUBEND
1250 ! SUBPROGRAM Avdcv hp85
1260 ! QA 7/11/80 GAC
1270 ! P1=Channel
1280 OUTPUT R1 USING "#" ! TEST
    FOR DVM
1290 RESUME R1 DIV 100 ! DROP AT
    N
1300 STATUS R1 DIV 100,2 : 02
1310 IF BINAND(02,3)=0 THEN 02=1
    ELSE 02=0
1320 03=0 ! FLAG FOR SCANNER PRE
    SENCE
1330 IF R1=R2 OR P1=-1 THEN 1380
1340 OUTPUT R0 USING "#" ! TEST
    FOR SCN
1350 RESUME R0 DIV 100 ! DROP AT
    N
1360 STATUS R0 DIV 100,2 : 03
1370 IF BINAND(03,3)=0 THEN 03=1
    ELSE 03=0
1380 R3=2*((P1<0 OR P1>999)*(P1<
    >-1))+16*(03 OR 02 AND R1=R
    0)
1390 R3=R3+32*(02 AND R1<>R0)
1400 IF R3 THEN 1550
1410 IF P1<>-1 THEN OUTPUT R0 : "
    AC";INT(P1)
1420 IF R1=R0 THEN 1460
1430 OUTPUT R1 : "H6STGT3"

```

```

1440 ENTER P1 . P
1450 GOTO 1580 ! RETURN
1460 OUTPUT P1 ."VR5WA1WF1WD5VCO
VS0VW0S01VN10VT3"
1470 P=0
1480 FOR O4=1 TO 30
1490 ENTER R1 USING "#,K" . 05
1500 P=P+05
1510 NEXT O4
1520 OUTPUT P1 ."500" ! EXIT SYS
TEM OUTPUT MODE
1530 P=P/30
1540 GOTO 1580 ! RETURN
1550 R$="Avdcv"
1560 GOSUB 1600 ! CALL Warn
1570 P=9.E19
1580 RETURN
1590 ! SUBEND
1600 ! SUBPROGRAM Warn hp85
1610 ! QA 7/30/80 GAC
1620 ! R3=ERROR WORD
1630 ! R$=NAME OF CALLING ROUTIN
E
1640 DIM Q8$(100),Q6$(15)
1650 O5=2 ! MAXIMUM NUMBER OF WA
RNINGS ON EACH ERROR ( <=9
)
1660 O2=0 ! ERROR FLAG
1670 ON ERROR GOTO 1710
1680 R1$=R1$
1690 OFF ERROR
1700 GOTO 1730
1710 OFF ERROR
1720 R1$="0000000000000000" ! RES
ET ERRORS
1730 !
1740 FOR O4=0 TO 14 ! ANY ERRORS
THAT NEED WARNINGS?
1750 IF BIT(R3,O4) AND VAL(R1$(O
4+1,O4+1))<O5 THEN O2=O2+2^
O4
1760 NEXT O4 ! DUMMY INDEX
1770 IF O2=0 THEN 2290 ! NO ERRO
RS...RETURN
1780 !
1790 O6=0
1800 ON ERROR GOTO 1950
1810 IF R0=0 THEN 1950 ! Nodate;
PRINT HEADING
1820 ON TIMEOUT R0 DIV 100 GOTO
1950
1830 SET TIMEOUT (R0 DIV 100);10
00
1840 OUTPUT R0 USING "#"
1850 RESUME R0 DIV 100 ! DROP AT
N
1860 STATUS R0 DIV 100,2 ; 03 !
FLAG FOR PRESENCE OF SCANNE
R
1870 OFF ERROR
1880 IF BINAND(03,3)=0 THEN 1950
! Nodate
1890 OUTPUT R0 USING "2A" . "TD"
1900 ENTER R0 . Q6$ ! Q6$=Date
1910 IF Q6$="01:01 00 00:00" THE
N 1950 ! Nodate
1920 PRINT USING "/,K" , "*****
*****"
1930 PRINT USING " 3(K)" ; " "
;"Date: "&Q6$(1,2)&"/"&Q6$(
4,5)&" Time: "&Q6$(7,14)&"
**"
1940 GOTO 1970
1950 OFF ERROR
1960 PRINT USING "/,K" ; "*****
*****"
1970 PRINT USING "/,K,/,K,/,/" ;
"WARNING: SUBPROGRAM "&R$&
" WAS", " NOT EXECUT
ED"
1980 ON ERROR GOTO 2000 ! IN CAS
E ISC IS UNDEFINED
1990 OFF TIMEOUT R0 DIV 100
2000 OFF ERROR
2010 RESTORE 2140
2020 FOR O4=0 TO 14 ! IDENTIFY T
HE INDIVIDUAL ERRORS
2030 READ Q8$
2040 IF NOT BIT(O2,O4) THEN 2090
! Next_indx
2050 R1$(O4+1,O4+1)=VAL$(VAL(R1$
(O4+1,O4+1))+1) ! INCREMENT
2060 O6=O6+1 ! NUMBER OF ERRORS
2070 PRINT Q8$ ! ERROR MESSAGE
2080 IF VAL(R1$(O4+1,O4+1))=O5 T
HEN PRINT USING "K,/,/" ; "
-LAST WARNING FOR THIS ERR
OR-"
2090 NEXT O4
2100 IF O6>=1 THEN PRINT USING "
/,K" ; "SEE THE "&R$&" DOCU
MENTATION TO"
2110 IF O6=1 THEN PRINT "INTERPR
ET THIS ERROR"
2120 IF O6>1 THEN PRINT "INTERPR
ET THESE ERRORS"
2130 PRINT "*****
*****"
2140 DATA "ERROR 0: Passed arra
y is dimen- sioned
improperly"
2150 DATA "ERROR 1: First pass
parameter out of
range"
2160 DATA "ERROR 2: Second pass
parameter out of
range"

```

```
2170 DATA "ERROR 3: Third pass  
parameter out of  
range"  
2180 DATA "ERROR 4: 3497 scanner  
not found on bus"  
2190 DATA "ERROR 5: 3456 DVM no  
t found on bus"  
2200 DATA "ERROR 6: 3437 system  
voltmeter not fo  
und on bus"  
2210 DATA "ERROR 7: Thermocoupl  
e 'volt' to 'temp'  
conversion out o  
f range"  
2220 DATA "ERROR 8: Thermocoupl  
e 'temp' to 'volt'  
conversion out o  
f range"  
2230 DATA "ERROR 9: Xmin=Xmax o  
r Ymin=Ymax"  
2240 DATA "ERROR 10: Scaling val  
ue<=0 on a log ax  
is"  
2250 DATA "ERROR 11: DataPoint<=  
0 on a log axis"  
2260 DATA "ERROR 12: Level cross  
ing not found"  
2270 DATA "ERROR 13: User define  
d error"  
2280 DATA "ERROR 14: User define  
d error"  
2290 RETURN  
2300 ! SUBEND
```

2) blade surface pressure measurements:

```

10  ! INLET FLOW PROFILE
20  ! OPTION BASE 1
30  DIM L(100),P1(100),P2(100)
40  DIM A(200),B(200)
45  DIM D1(200),D2(200)
50  P1=3456
52  N1=15
54  N2=10
56  DISP "Input single input tra
    nsducer output for zero Pres
    sure difference - in mV"
58  INPUT C3
59  C3=C3/1000
60  GOSUB 440 ! CALL Init
70  FOR J=1 TO 67
75  BEEP 50,500
110 DISP "Input Y coordinate"
120 INPUT L(J)
125 IF L(J)=99 THEN GOTO 180
130 BEEP 50,500
140 DISP "Press continue when re
    ady"
150 PAUSE
160 GOSUB 240
170 NEXT J
180 N=J-1
190 FOR I=1 TO N
200 P3=P1(I)/P2(I)
210 PRINT I,"Y=",L(I)
212 PRINT "P test sect. =",P1(I)
    ,"Pa"
213 PRINT "P set. cham. =",P2(I)
    ,"Pa"
216 PRINT "P test / P set. =",P3
217 P4=ABS(P3)^.5
218 PRINT "V test / V set = ",P
    4
220 NEXT I
230 END
240 FOR K=1 TO N1
250 P1=2
260 GOSUB 1250 ! CALL Avdcv
270 E1=P
280 P1=1
290 GOSUB 1250 ! CALL Avdcv
292 F1=P
294 GOSUB 1250 ! CALL Avdcv
296 F2=P
298 A(K)=(F1+F2)/2
300 P1=2
302 GOSUB 1250 ! CALL Avdcv
304 E2=P
306 B(K)=(E1+E2)/2
310 NEXT K
320 C1=0
330 C2=0
340 FOR M=1 TO N1
350 D1(M)=(C1+A(M))/M
360 C1=C1+A(M)
370 D2(M)=(C2+B(M))/M
380 C2=C2+B(M)
385 G1=0
390 G2=0
400 NEXT M
410 FOR M1=N2 TO N1
415 G1=D1(M1)+G1
420 G2=D2(M1)+G2
424 NEXT M1
426 P1(J)=G1/(N1-N2+1)*171.5
427 P1(J)=P1(J)*8.0931
428 P2(J)=(G2/(N1-N2+1)+C3)*174
429 P2(J)=P2(J)*8.0931
430 RETURN
440 ! SUBPROGRAM Init hp85
450 ! QA 7/08/80 GAC
460 ! P1=DvMeter
470 DIM Q7$(23) ! DEVICE NAMES
480 !
490 ! INITIALIZE DEVICE ADDRESS
    ES
500 !
510 R0=709 ! 3497A SCANNER ADDRE
    SS
520 R1=722 ! 3456A DIGITAL VOLT
    METER ADDRESS
530 R2=724 ! 3437A SYSTEM VOLTM
    ETER ADDRESS
540 R3=0 ! ERROR FLAG
550 R4=0 ! USED IN SUBROUTINE "S
    trip"
560 O3=R0 DIV 100 ! INTEGER DIV
    IDE,i.e. Bus=IP(Scn/100)
570 O4=R1
580 R1$="0000000000000000"
590 ON ERROR GOTO 1020 ! IF P1 H
    AS NOT BEEN DEFINED
600 IF P1=3497 THEN R1=R0 ! PASS
    PARAMETER USED HERE
610 IF P1<>3456 AND P1<>3497 THE
    N 1020 ! Abort
620 OFF ERROR
630 !
640 !
650 ! VERIFY THAT INTERFACE IS H
    P-IB
660 !
670 ON ERROR GOTO 740
680 STATUS O3,0 : O6
690 OFF ERROR
700 !
710 ! PRINT ERROR MESSAGE TO CRT
    IF NO INTERFACE OR NOT HP-I
    B INTERFACE.
720 !
730 IF O6=1 THEN 840 ! OK
740 OFF ERROR
750 CLEAR
760 DISP USING "/",K" ; "CAUTION:
    HP-IB INTERFACE *"&VAL$(O3
    )&" IS NOT PRESENT."

```

```

770 DISP USING "/,K" , * PROGRAM.
    EXECUTION TERMINATED"
780 BEEP
790 STOP
800 !
810 ! INITIALIZE 82937A HP-IB IN
    INTERFACE, RESET INTERFACE
820 !
830 !
840 ABORTIO 03 ! INITIALIZES IN
    TERFACE AND BUS
850 RESET 03 ! SENDS DEVICE CLE
    AR (DCL)
860 !
870 !
880 ! CHECK FOR EQUIPMENT ON BUS
    AT ALL ADDRESSES AND PRINT
    DEVICE NAMES .
890 !
900 !
910 CLEAR @ DISP USING "/,2X,K,D
    D,/,/" , "EQUIPMENT PRESENT
    ON BUS ",03
920 FOR 02=0 TO 30
930 OUTPUT 03*100+02 USING "*" ;
    ! ADDRESS DEVICE TO LISTEN
940 RESUME 03 ! DROP ATM
950 STATUS 03,2 , 05 ! SEE IF IT
    LISTENED
960 IF BINAND(05,3)=0 THEN 990 !
    NOT THERE
970 GOSUB 1000 ! CALL DEV_NAME
980 DISP USING 1000 ; Q7$,02
990 NEXT 02
1000 IMAGE 37A," at address ",22
1010 GOTO 1060 ! EXIT
1020 OFF ERROR
1030 R3=2
1040 R$="Init"
1050 GOSUB 1600 ! CALL Warn
1060 RETURN
1070 ! SUBEND
1080 ! SUB DEV_NAME
1090 ! Q7$=DEVICE NAME
1100 Q7$=""
1110 IF Q2<>04 MOD 100 THEN 1130
1120 Q7$="3456A Digital Voltmete
    r"
1130 IF Q2<>R2 MOD 100 THEN 1150
1140 Q7$="3437A System Voltmeter
    "
1150 IF Q2<>21 THEN 1170
1160 Q7$="h=85A System Computer"
1170 IF Q2<>R0 MOD 100 THEN 1200
1180 Q7$="3497A Scanner,Timer"
1190 IF R1=R0 THEN Q7$=Q7$&" ,DVM
    "
1200 IF Q7$<>"" THEN 1220
1210 Q7$="Device Unknown" ! UNKN
    OWN DEVICE=ANY DEVICE WITH
    NO LABEL
1220 !
1230 RETURN
1240 ! SUBEND
1250 ! SUBPROGRAM Avdcv hp85
1260 ! QA 7/11/80 GAC
1270 ! P1=Channel
1280 OUTPUT R1 USING "*" ! TEST
    FOR DVM
1290 RESUME R1 DIV 100 ! DROP AT
    N
1300 STATUS R1 DIV 100,2 ; 02
1310 IF BINAND(02,3)=0 THEN 02=1
    ELSE 02=0
1320 03=0 ! FLAG FOR SCANNER PRE
    SENCE
1330 IF R1=R2 OR P1=-1 THEN 1380
1340 OUTPUT R0 USING "*" ! TEST
    FOR SCH
1350 RESUME R0 DIV 100 ! DROP AT
    N
1360 STATUS R0 DIV 100,2 ; 03
1370 IF BINAND(03,3)=0 THEN 03=1
    ELSE 03=0
1380 R3=2*((P1<0 OR P1>999)*(P1<
    >-1))+16*(03 OR 02 AND R1=R
    0)
1390 R3=R3+32*(02 AND R1<>R0)
1400 IF R3 THEN 1550
1410 IF P1<>R1 THEN OUTPUT R0 ; "
    AC",INT(R1)
1420 IF R1=R0 THEN 1460
1430 OUTPUT R1 ; "H6STGT3"
1440 ENTER R1 ; P
1450 GOTO 1580 ! RETURN
1460 OUTPUT R1 ; "VR5VA1VF1VD5VC0
    VS0VW0S01VH10VT3"
1470 P=0
1480 FOR 04=1 TO 30
1490 ENTER R1 USING "/,K" ; 05
1500 P=P+05
1510 NEXT 04
1520 OUTPUT R1 ; "S00" ! EXIT SYS
    TEM OUTPUT MODE
1530 P=P/30
1540 GOTO 1580 ! RETURN
1550 R$="Avdcv"
1560 GOSUB 1600 ! CALL Warn
1570 P=9.E19
1580 RETURN
1590 ! SUBEND
1600 ! SUBPROGRAM Warn hp85
1610 ! QA 7/30/80 GAC
1620 ! R3=ERROR WORD
1630 ! R$=NAME OF CALLING ROUTIN
    E
1640 DIM Q8$(100),Q6$(15)
1650 05=2 ! MAXIMUM NUMBER OF WA
    RNINGS ON EACH ERROR ( <=9
    )
1660 02=0 ! ERROR FLAG

```



```

1670 ON ERROR GOTO 1710
1680 R1$=P1$
1690 OFF ERROR
1700 GOTO 1730
1710 OFF ERROR
1720 R1$="0000000000000000" ! RES
    ET ERRORS
1730 !
1740 FOR O4=0 TO 14 ! ANY ERRORS
    THAT NEED WARNINGS?
1750 IF BIT(R3,O4) AND VAL(R1$C0
    4+1,O4+1)<05 THEN O2=O2+2^
    O4
1760 NEXT O4 ! DUMMY INDEX
1770 IF O2=0 THEN 2290 ! NO ERRO
    RS RETURN
1780 !
1790 O6=0
1800 ON ERROR GOTO 1950
1810 IF R0=0 THEN 1950 ! Nodate,
    PRINT HEADING
1820 ON TIMEOUT R0 DIV 100 GOTO
    1950
1830 SET TIMEOUT (R0 DIV 100);10
    00
1840 OUTPUT R0 USING "# "
1850 RESUME R0 DIV 100 ! DROP AT
    N
1860 STATUS R0 DIV 100,2 ; O3 !
    FLAG FOR PRESENCE OF SCANNE
    R
1870 OFF ERROR
1880 IF BINAND(O3,3)=0 THEN 1950
    ! Nodate
1890 OUTPUT R0 USING "2A" ; "TD"
1900 ENTER R0 ; Q6$ ! Q6$=Date
1910 IF Q6$="01:01:00:00:00" THE
    N 1950 ! Nodate
1920 PRINT USING "/,K" ; "*****
    *****"
1930 PRINT USING " 3(K)" ; "**
    " ; "Date: "&Q6$[1,2]&"/"&Q6$
    [4,5]&" Time: "&Q6$[7,14]&"
    **"
1940 GOTO 1970
1950 OFF ERROR
1960 PRINT USING "/,K" ; "*****
    *****"
1970 PRINT USING "/,K,/K,/," ;
    "WARNING: SUBPROGRAM "&R$&
    " WAS";" NOT EXECUT
    ED"
1980 ON ERROR GOTO 2000 ! IN CAS
    E ISC IS UNDEFINED
1990 OFF TIMEOUT R0 DIV 100
2000 OFF ERROR
2010 RESTORE 2140
2020 FOR O4=0 TO 14 ! IDENTIFY T
    HE INDIVIDUAL ERRORS
2030 READ Q8$
2040 IF NOT BIT(O2,O4) THEN 2090
    ! Next_Lindy
2050 R1$C04+1,O4+1)=VAL$(VAL(R1$
    C04+1,O4+1))+1 ! INCREMENT
2060 O6=O6+1 ! NUMBER OF ERRORS
2070 PRINT Q8$ ! ERROR MESSAGE
2080 IF VAL(R1$C04+1,O4+1)=05 T
    HEN PRINT USING "K,/," ; "
    -LAST WARNING FOR THIS ERR
    OR-"
2090 NEXT O4
2100 IF O6>=1 THEN PRINT USING "
    /,K" ; "SEE THE "&R$&" DOCU
    MENTATION TO"
2110 IF O6=1 THEN PRINT "INTERPR
    ET THIS ERROR"
2120 IF O6>1 THEN PRINT "INTERPR
    ET THESE ERRORS"
2130 PRINT "*****
    *****"
2140 DATA "ERROR 0: Passed arra
    y is dimen- sioned
    improperly"
2150 DATA "ERROR 1: First pass
    Parameter out of
    range"
2160 DATA "ERROR 2: Second pass
    Parameter out of
    range"
2170 DATA "ERROR 3: Third pass
    Parameter out of
    range"
2180 DATA "ERROR 4: 3497 scanne
    r not found on bus"
2190 DATA "ERROR 5: 3456 DVM no
    t found on bus"
2200 DATA "ERROR 6: 3437 system
    voltmeter not fo
    und on bus"
2210 DATA "ERROR 7: Thermocoupl
    e 'volt. to
    conversion out
    of range"
2220 DATA "ERROR 8: Thermocoupl
    e 'temp. to
    conversion out
    of range"
2230 DATA "ERROR 9: Xmin=Xmax o
    r Ymin=Ymax"
2240 DATA "ERROR 10: Scaling val
    ue<=0 on a log ax
    is"
2250 DATA "ERROR 11: Data point<=
    0 on a log axis"
2260 DATA "ERROR 12: Level cross
    ing not found"
2270 DATA "ERROR 13: User define
    d error"
2280 DATA "ERROR 14: User define
    d error"
2290 RETURN
2300 ! SUBEND

```

APPENDIX I

Listing of Fortran program used for computing boundary layer parameters:

```

C      PROGRAM THAT COMPUTES BOUNDARY LAYER PARAMETERS
C      USING SIMPSON'S INTEGRATION TECHNIQUE
C
C      PROGRAM SIMPSONS
C      REAL Y(100),V(100,7),DELTA,DELTAS,THETA,H,UE(7)
C      INTEGER N,NPROS,NN
C
C      INPUT NO. OF PROFILES, NO. OF POINTS , BOUNDARY LAYER
C      THICKNESS AND FREESTREAM VELOCITY
C      READ(5,*) NPROS,N,DELTA
C      READ(5,*) (UE(I),I=1,NPROS)
C      NN=N/2
C
C      INPUT COORDINATE AND CORRESPONDING VELOCITIES
C      DO 10 I=1,N
C          READ(5,*) Y(I),(V(I,J),J=1,NPROS)
10    CONTINUE
C
C      DO 100 II=1,NPROS
C          DELTAS=0
C          THETA=0
C          DO 20 I=1,N
C              V(I,II)=V(I,II)/UE(II)
20    CONTINUE
C          DO 30 I=1,NN
C              DELTAS=(Y(2*I+1)-Y(2*I-1))/6.0*(V(2*I-1,II)+
C              *      4*V(2*I,II)+V(2*I+1,II))+DELTAS
C              THETA=THETA+(Y(2*I+1)-Y(2*I-1))/6.0*(V(2*I-1,II)**2+
C              *      4*(V(2*I,II)**2)+V(2*I+1,II)**2)
30    CONTINUE
C          DELTAS=DELTA-DELTAS
C          THETA=DELTA-DELTAS-THETA
C          H=DELTAS/THETA
C          WRITE(6,*) 'DELTA=',DELTA,'DELTA STAR=',DELTAS
C          WRITE(6,*) 'THETA=',THETA,'H=',H
100  CONTINUE
C      STOP
C      END

```

```

C      PROGRAM TO DETERMINE THE FORCES AND POINT OF ACTION
C      DEVELOPED BY BLADE
C
C      AUTHOR : KIM AMRUD
C
C      DATE : OCTOBER 12,84
C
C      PROGRAM FCOEFF
C      REAL X(55),Y(55),CP(5,55),L(54),THETA,FX(54),FY(54),F,
C      *   MX(54),MY(54),LX,LY,FXR,FYR,MXR,MYR,XR,YR
C      INTEGER I,J,ROWS
C
C      INPUT COORDINATES OF STATIC TAPS ON BLADE
C      READ(5,*) (X(I),Y(I),I=1,55)
C
C      INPUT # OF STATIC TAP ROWS TO BE ANALYSED
C      READ(6,*) ROWS
C
C      INPUT PRESSURE COEFF. AT EACH STATIC TAP
C      DO 10 I=1,55
C          READ(6,*) (CP(J,I),J=1,ROWS)
10      CONTINUE
C
C      CALCULATING LENGTH OF EACH PANEL
C      DO 20 I=1,54
C          L(I)=((X(I+1)-X(I))**2.0+(Y(I+1)-Y(I))**2.0)**0.5
20      CONTINUE
C
C      CALCULATING THE ANGLE OF EACH PANEL TO THE HORIZONTAL,
C      FORCE AND MOMENT COEFFS. OFR EACH PANEL
C      DO 50 J=1,ROWS
C          DO 30 I=1,54
C              F=(CP(J,I)+CP(J,(I+1)))/2.0*L(I)
C              LX=ABS((X(I)+X(I+1))/2.0)
C              LY=ABS((Y(I)+Y(I+1))/2.0)
C              IF((X(I+1).GE.X(I)).AND.(Y(I+1).GE.Y(I))) THEN
C                  THETA=ATAN((Y(I+1)-Y(I))/(X(I+1)-X(I)))
C                  FX(I)=F*SIN(THETA)
C                  FY(I)=-F*COS(THETA)
C              ELSE
C                  THETA=PI-ATAN((Y(I+1)-Y(I))/(X(I+1)-X(I)))
C                  FX(I)=F*SIN(THETA)
C                  FY(I)=F*COS(THETA)
C              ENDIF
C          END DO
C      END DO
C      END

```

```

      IF((X(I+1).GE.X(I)).AND.(Y(I+1).LE.Y(I))) THEN
        THETA=ATAN((Y(I)-Y(I+1))/(X(I+1)-X(I)))
        FX(I)=-F*SIN(THETA)
        FY(I)=-F*COS(THETA)
      ELSE
      ENDIF
      IF((X(I+1).LE.X(I)).AND.(Y(I+1).LE.Y(I))) THEN
        THETA=ATAN((Y(I)-Y(I+1))/(X(I)-X(I+1)))
        FX(I)=-F*SIN(THETA)
        FY(I)=F*COS(THETA)
      ELSE
      ENDIF
      IF((X(I+1).LE.X(I)).AND.(Y(I+1).GE.Y(I))) THEN
        THETA=ATAN((Y(I+1)-Y(I))/(X(I)-X(I+1)))
        FX(I)=F*SIN(THETA)
        FY(I)=F*COS(THETA)
      ELSE
      ENDIF
C
      MX(I)=LX*FY(I)
      MY(I)=LY*FX(I)
      THETA=THETA*57.296
30  CONTINUE
C
C  CALCULATING THE TOTAL FORCE COMPONENTS AND POINT OF ACTION
      FXR=-FX(33)
      FYR=-FY(33)
      MXR=-MX(33)
      MYR=-MY(33)
      DO 40 II=1,54
        FXR=FXR+FX(II)
        FYR=FYR+FY(II)
        MXR=MXR+MX(II)
        MYR=MYR+MY(II)
40  CONTINUE
      XR=MXR/FYR
      YR=MYR/FXR
      XRR=XR-.618
      YRR=YR-.719
      WRITE(7,*) 'LIFT COEFF = ',FYR
      WRITE(7,*) 'DRAG COEFF = ',FXR,
& 'FORCE WRT BLADE L.E. '
      WRITE(7,*) 'X = ',XRR,'Y = ',YRR
50  CONTINUE
      STOP
      END

```

END

2 4 0 2 8 6

FIN