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**MEASUREMENT AND PREDICTION OF TIP-CLEARANCE EFFECTS
IN A LINEAR TURBINE CASCADE**

by

Metin Ilbay Yaras

B. Sci. (Aeronautical), M. Eng. (Mechanical)

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

Doctor of Philosophy in Aeronautical Engineering

**Ottawa-Carleton Institute for Mechanical
and Aeronautical Engineering**

**Department of
Mechanical and Aerospace Engineering
Carleton University
Ottawa, Ontario
Canada
December, 1990**

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**Measurement and Prediction of Tip-Clearance Effects
in a Linear Turbine Cascade**

submitted by

M.I. Yaras, B.Sci, M.Eng.
in partial fulfilment of the requirements
for the degree of Doctor of Philosophy

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December, 1990

ABSTRACT

Measurements are performed in a stationary linear cascade of turbine blades for a range of clearances varying from 0.0 to 5.5 percent blade chord. The downstream flow field is measured with a seven-hole pressure probe at several axial locations. The results are analyzed both in terms of vorticity and total pressure loss. The former quantity provides insight into several aspects of the leakage flow, including size and strength of the leakage vortex in relation to the tip-clearance size and the bound circulation of the blade; the diffusion rate of the tip-leakage vortex and its interaction with the passage flow. In this context, a simple model of the vortex based on the diffusion of a line vortex is found to be quite accurate. Evaluation of the losses in the downstream measurement planes in relation to the losses measured inside the tip gap help clarify the dominant tip-leakage loss mechanisms. Based on experimental observations, full verification of existing turbomachinery tip-leakage loss prediction schemes is performed. An improved tip-leakage loss model is developed using one of the existing models as the starting point.

Following the stationary linear cascade tests, relative tipwall motion is simulated by a moving belt. Three clearance sizes of 1.5, 2.4 and 3.8 percent blade chord are examined. The tip-gap and downstream flow together with blade pressures are measured for several wall speeds. As the relative wall speed is increased, the mass flow rate through the gap is significantly reduced. The dominant mechanism behind this trend is identified. The relative wall motion alters the tip vortex and passage vortex structures substantially. The trends in the blade loading are found to be in agreement with the observed changes in the vortex structures. Based on the experimental results, the tip-vortex and tip-gap flow models are extended to the case of relative tipwall motion.

Succeeding the experimental and theoretical work on tip-leakage flow for plain tip geometry, blade tip treatments to reduce tip-leakage losses are tested with stationary and moving tipwall for clearance sizes of 2.4 and 3.7 percent blade chord. Three types of winglets consisting of pressure-side, suction-side and a combination of the two are considered in comparison to the regular blade tip. Losses extracted from measurements

downstream of the blades suggest some reduction of tip-leakage losses with the application of winglets. The possible physical mechanisms by which winglets reduce the losses are discussed on the basis of downstream flow field data and tipwall static pressures.

to my parents

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NOMENCLATURE

a_s	= Squire's eddy viscosity constant (Eqn. 5.5.2)
A	= area
c	= blade chord length
C_D	= discharge coefficient for tip gap
C_L	= lift coefficient
c_x	= blade axial chord length
C_P	= $\frac{P - P_{CL}}{\frac{1}{2}\rho V_{CL}^2}$ = static pressure coefficient
C_{P_t}	= $\frac{P_o - P_{o_a}}{\frac{1}{2}\rho V_{CL}^2}$ = total pressure loss coefficient
C'_{P_t}	= local mass averaged total pressure loss coefficient (Eqn. 6.2.2)
C''_{P_t}	= total pressure loss coefficient averaged over reference mass flow rate (Eqn. 6.2.1)
C_q	= $\frac{\frac{1}{2}\rho V^2}{\frac{1}{2}\rho V_{CL}^2}$ = kinetic energy coefficient
C''_{q_s}	= mass-averaged secondary kinetic energy coefficient for downstream flow
C''_{q_a}	= mass-averaged coefficient of gap kinetic energy normal to chord line
d	= tip leakage vortex diameter
E	= kinetic energy
h	= span
H	= $\frac{\delta^*}{\theta}$ = boundary layer shape factor
k_s	= fraction of zero-clearance secondary loss still present with clearance (Eqn. 6.8.3.13)
K_E, K_G	= constants related to the blade loading distribution (Eqns. 6.8.3.12 and 6.8.3.15)
L	= blade lift force

$\dot{m}_g$	= mass flow rate through the tip gap
$\dot{m}_p$	= mass flow rate through the blade passage
$\dot{m}_{REF}$	= reference mass flow rate for half passage at inlet
P	= static pressure
P_o	= total pressure
P_{ps}	= blade pressure-surface pressure
P_{ss}	= blade suction-surface pressure
q	= $\frac{1}{2}\rho V^2$ = dynamic pressure
r	= radial co-ordinate
Re	= $\frac{\rho V_{cl} c}{\mu}$ = Reynolds number based on blade chord
s	= streamwise co-ordinate
S	= blade spacing
t	= time
t_{MAX}	= blade maximum thickness
U	= blade tip speed
u, v, w	= components of velocity in x, y and z directions, respectively
u'	= root mean square velocity fluctuation in the x direction
v_θ	= tangential component of velocity
V	= resultant velocity
V_e	= velocity at the boundary layer edge
V_m	= mean velocity through the cascade
V_n	= component of velocity normal to area of interest
V_r	= relative velocity
V_s	= velocity component in the tip-vortex centerline direction
V_{wall}	= relative tipwall speed
x, y, z	= co-ordinates in axial, tangential and spanwise directions
x'	= co-ordinate in chordwise direction
y'	= local pitchwise co-ordinate (Figure 5.2.1)
Y	= total pressure loss coefficient based on outlet dynamic pressure

Y'	= loss coefficient averaged over local mass flow rate
α	= pitch flow angle, measured from the axial direction
α_m	$= \tan^{-1} \left(\frac{\tan(\alpha_1) + \tan(\alpha_2)}{2} \right)$ = mean flow angle
β	= blade metal angle
γ	= blade stagger angle
Γ_b	= bound circulation
Γ_{TV}	= tip-vortex circulation
Γ'	= non-dimensional circulation
δ	= boundary layer thickness
δ_x^*	$= \int_0^\delta \left(\frac{V_x}{V_e} - \frac{V_x}{V_e} \right) dz$ = axial boundary layer displacement thickness
δ_y^*	$= \int_0^\delta \left(\frac{V_y}{V_e} - \frac{V_y}{V_e} \right) dz$ = transverse boundary layer displacement thickness
θ_x	$= \int_0^\delta \left(\frac{V_x}{V_e} - \frac{V_x}{V_e} \right) \frac{V_x}{V_e} dz$ = axial boundary layer momentum thickness
θ_y	$= \int_0^\delta \left(\frac{V_y}{V_e} - \frac{V_y}{V_e} \right) \frac{V_y}{V_e} dz$ = transverse boundary layer momentum thickness
θ_{xy}	$= \int_0^\delta \left(\frac{V_x}{V_e} - \frac{V_x}{V_e} \right) \frac{V_y}{V_e} dz$ = axial interaction boundary layer momentum thickness
θ_{yx}	$= \int_0^\delta \left(\frac{V_y}{V_e} - \frac{V_y}{V_e} \right) \frac{V_x}{V_e} dz$ = transverse interaction boundary layer momentum thickness
Λ	= Owen's eddy viscosity constant (Eqn. 5.5.3)
μ	= dynamic viscosity
ν	= kinematic viscosity
ν_e	= eddy viscosity
η	= efficiency
ρ	= density

σ	= blade row solidity (c/S)
τ	= tip gap height
ω	= vorticity
ω'	= non-dimensional vorticity

SUBSCRIPTS

<i>B</i>	= bound
<i>CL</i>	= centerline value at inlet
<i>e</i>	= boundary layer edge value
<i>g</i>	= gap
<i>m</i>	= mean
<i>min,max</i>	= minimum and maximum values
<i>n</i>	= component of vector quantity normal to the area of interest
<i>o</i>	= stagnation (total) quantity
<i>p</i>	= passage
<i>PS</i>	= blade pressure surface
<i>r</i>	= relative to blades
<i>REF</i>	= reference
<i>SS</i>	= blade suction surface
<i>TV</i>	= tip vortex
<i>x,y,z,s</i>	= component in x, y, z and streamwise direction
<i>1,2</i>	= cascade inlet and outlet

1. INTRODUCTION

In rotating turbomachine blade rows a clearance is kept between the blade tip and the annulus wall to avoid rubbing of the blades against the casing. It is well known that leakage through the tip clearance introduces significant detrimental effects to the aerodynamic performance of the turbomachine stage. In some designs, tip leakage is reduced by employing an inner casing (shroud) that rotates with the blades. However, this configuration has its own disadvantages such as extra weight and increased centrifugal stresses in the rotor blades. There are examples where the rotor blades are allowed to rub against an abradable casing. This, however, does not eliminate the tip leakage all together because the wide range of temperatures (in the case of turbines) and rotational speeds experienced during operation result in variable clearance sizes. In this relation, active control of the tip clearance is an alternative that has been used on some engines. The main disadvantage of this design approach is the relatively high cost and complexity associated with it.

Due to the drawbacks of the above noted design configurations employed to reduce tip-leakage effects, a considerable number of recent designs still incorporate rotors with plain tip clearances. In such designs being able to predict the magnitude of the effect of tip leakage on performance is essential. Because of the relative complexity of the tip-leakage flow, only a few tip-leakage loss correlations have been proposed which are based on assumptions regarding the flow physics as opposed to being simple correlations of experimental data. Even then, very few attempts have been made to justify those assumptions based on experimental work. Within the last decade, however, intense competition to achieve higher turbomachinery stage efficiencies has initiated considerable research aiming towards a sound understanding of the tip-leakage flows. One such research project, funded by Pratt and Whitney Canada, was initiated seven years ago at Carleton University.

A linear turbine cascade has been used in the experimental part of the research. Due to the incompressible flow conditions, low level of free-stream turbulence, absence of heat transfer and relative tipwall motion, the experiments are somewhat idealized. However, in view of the apparent complexity of the tip-leakage flow, it seems only appropriate to begin with an idealized configuration and tackle the more realistic configurations as understanding of the flow is advanced. Initially, Amrud (1985) acquired blade loading data in the cascade to identify the effects of tip leakage on loading near the tip. The results suggested an increased loading near the tip. This was due to the presence of the tip vortex formed by the tip-leakage flow as it merged with the passage stream. It was also found that the increased loading was accompanied by a rotation of the load vector towards the axial direction. Thus, the majority of the rise occurred in the axial component of the blade force.

Subsequent to Amrud's work, Yaras (1987) studied the flow inside the tip gap. Measurements showed a separation bubble on the blade tip. Particularly towards the rear of the blade the separation bubble took the form of a vortex. The portion of the gap occupied by this separation bubble was found to contribute little to the total leakage through the gap. At a given chordwise location, outside the separation bubble the velocity distribution in the gap was quite uniform. Even though Amrud's results showed considerable distortion of the blade pressures near the tip, the leakage flow was found to be accelerated into the gap by driving pressure differences basically equal to the "undisturbed" blade pressure differences. The flow was found to have essentially completed its acceleration before it enters the gap. Little change in the chordwise momentum was observed as the fluid went through the gap. Finally, the bulk of the leakage flow was found to be free-stream fluid that had entered the gap with essentially no loss. Based on these observations, a simple scheme for predicting the tip-gap velocity distribution outside the separation bubble, and thus the mass flow rate through the gap, was proposed.

The natural follow-up of this work is to study the tip leakage as it merges with the passage flow and evolves downstream. While the roll-up of the tip-leakage flow into a vortex has been known for some time, little work has been done to explore the details of the merging process and the characteristics of the vortex. In this context, a quantity of particular interest is the losses caused by the mixing of the tip leakage with the passage stream and the subsequent loss generation due to diffusion of the tip vortex. This is one of the issues tackled within the present work and is documented in Chapter 6. The experimental observations enabled the evaluation of various models proposed for predicting the tip-leakage losses. One of the existing models was used as a starting point to derive an improved tip-leakage loss model consistent with the actual physics of the flow. This model is documented in Chapter 6 as well.

One of the parameters that is commonly used in tip-leakage loss prediction schemes is the tip-vortex circulation. This quantity and its relation to the tip-clearance size has been examined in several studies. However, the present work raised doubts with regard to the accuracy of the tip-vortex circulations quoted in several studies and the physical explanations provided for the observed trends. Thus, the downstream vorticity field was studied in detail for the present cascade. These results are discussed in Chapter 5. In addition to being a parameter that has been related to tip leakage losses, the tip-vortex strength is also significant in terms of the distortion it causes at the inlet to a subsequent blade row. In this context, the cross-flow velocities associated with the tip vortex and their variation with downstream distance are examined in the same chapter. A simple model is proposed that is capable of predicting these cross-flow velocities given the vortex strength.

While stationary cascade measurements provide useful insights into various aspects of the tip-leakage flow, it is essential to establish the applicability of the conclusions drawn to more realistic configurations. Experimental results with relative tipwall motion are discussed in Chapter 8, and provide a step forward in this context. The effects of wall motion on both the tip-gap flow and the downstream flow field are examined. In

addition, the validity of the models developed on the basis of stationary tests are evaluated for this more realistic configuration and modifications are suggested if necessary.

The final issue addressed in the present work is blade tip treatment to reduce the detrimental effects of tip leakage. As previously noted, the relatively common approaches for reducing tip-leakage losses have rather significant detrimental effects from other points of view. Having developed a good understanding of the tip-leakage flow for plain tip clearance, the likelihood for devising a blade tip and/or tipwall treatment that is effective in reducing tip-leakage losses has improved. In this context, in Chapter 9 mini-shrouds (winglets) are evaluated in terms of their effectiveness in reducing tip-leakage losses. The physical mechanisms by which this blade tip treatment reduces the tip-leakage losses are explored for several winglet shapes.

2. LITERATURE REVIEW

A comprehensive review of tip-leakage flow studies in axial turbomachinery was undertaken by Yaras (1987) as part of the master's thesis preceding the work presented herein. This was succeeded by an updated survey by Yaras and Sjolander (1989). There have been a number of valuable studies on tip leakage since this last review. The original intention was to present an updated review in this chapter. On the other hand, the present work is quite comprehensive in that a wide range of aspects associated with tip-leakage flow are dealt with. Thus, it seems more suitable to discuss the literature in relation to the present research within appropriate sections of the thesis. This provides the grounds for effective discussion of the recent work in connection with the work that has already been done.

With the literature review being presented in the main text as described above, it becomes difficult to acquire an overall perception of the research that has been done on tip-leakage flows. Therefore, an updated version of the aforementioned review report (Yaras and Sjolander, 1989) is appended to the thesis (Appendix A).

3. DESCRIPTION OF EXPERIMENTAL RIG AND INSTRUMENTATION

3.1 Annular Wind Tunnel

The annular wind tunnel employs a centrifugal fan driven by a variable speed 40 HP DC motor as the wind source. The centrifugal fan can deliver a maximum flow rate of 4 kg/sec at a maximum total pressure rise of 48 cm of water. At the fan exit the flow enters a straight annular diffuser with an area ratio of 2.72 and an equivalent cone angle of 14 degrees. Any possible swirl at the diffuser exit is removed by a honeycomb screen with cells of 9.5 mm width and 76 mm depth. A settling chamber following the diffuser accommodates three screens. The contraction at the settling chamber outlet is preceded by a transition section that provides conversion from a circular to a rectangular cross section. The area increase factor of 1.2 in this transition section results in some diffusion before the flow enters the test section.

3.2 Test Section

A linear cascade is used as the test section (Figure 3.2.1). It is 675 mm wide, 202.3 mm deep and accommodates a row of five turbine blades of 250 mm chord length. The blades are mounted on a 7.9 mm thick steel backwall which is essentially the main support of the test section. The blades are scaled versions of the tip section of a power turbine blade of recent design by Pratt and Whitney Canada. A clearance size range from 0.0 to 5.5 percent of the blade chord (0.0 to 6.8 percent of the blade span) is investigated. The clearance size is adjusted by inserting shims between the sidewalls and the plexiglass tipwall.

Periodic flow in blade passages is obtained by adjusting the tail-flaps downstream of the trailing edge plane and the side-gates at the blade row inlet. Periodicity checks are performed by three-hole probe traverses at midspan downstream of the blades. Details on periodicity measurements are given by Yaras (1987). Main cascade dimensions are

Figure 3.2.1 Linear turbine cascade.

provided in Appendix B and more details on the components and dimensions of the cascade are again given by Yaras (1987).

3.3 Moving Belt

A moving belt is used to simulate the relative blade motion. It replaces the plexiglass window of the test section in Fig. 3.2.1. The moving belt rig shown in Figure 3.3.1 was designed by Danias (1987). The belt is 61 cm wide, 276.3 cm long, and is made out of SAB-8E synthetic rubber with a mechanical friction coefficient of 0.15.

Figure 3.3.2 shows the rig together with the linear cascade. The belt extends 120 mm upstream of the blade leading edge plane and about one chord length (255 mm) downstream of the trailing edge plane. The length of the belt upstream of the blades is crucial since it will determine the thickness of the transverse tipwall boundary layer at the leading edge plane. In addition, the gap between the belt leading edge and the plexiglass belt guide (Figure 3.3.3) is likely to alter the axial boundary layer thickness. These issues will be elaborated upon in Chapter 7.

The top drum pulley on the moving belt rig (Figure 3.3.1) is crowned to prevent the belt from sliding sideways. Proper tension on the belt is achieved by moving the top pulley vertically using tensioning bolts. A back plate covered with a teflon sheet provides support for the face of the belt. While tensioning the belt prevented flapping of the belt during experiments, it did not avoid formation of an air film between the belt and the teflon covered back plate. The thickness of this air film was found to depend on the speed of the belt and the static gauge pressure inside the test section. This would result in substantial uncertainty in the tip clearance size and its uniformity along the blade chord. Therefore, a suction box was attached to the back plate (Figure 3.3.1) and the air film was removed through holes opened on the back plate. Measurements with a gooseneck dial gauge indicated that with suction and proper longitudinal tensioning the tip clearance size can be kept within ± 0.2 mm regardless of the wall speed.

Figure 3.3.1 Moving belt rig.

Figure 3.3.2 Cascade test section and moving belt.

Figure 3.3.3 Moving belt leading edge configuration.

A 3 HP three phase motor with a top speed of 1760 RPM is used to drive the lower pulley. The motor speed can be adjusted through a variable frequency speed control unit. Since the pulleys are dynamically balanced, virtually no vibration problems are encountered within the speed range of the motor. The motor top speed, chosen for safety considerations, results in a linear belt speed of 18.73 m/sec. For this maximum wall speed, proper simulation of the velocity triangles encountered in the actual turbine rotor for this blade required reduced inlet flow Reynolds number. Discussion on these operating conditions for the moving tipwall experiments is given in Chapter 7.

3.4 Data Acquisition System

The data acquisition system for measuring pressures consists of two pressure transducers, a Model 901 CPU and Intelligent Serial Interface (M 901) manufactured by Sciometric Instruments, a Model 81 Electronic Measurement System (M 81) produced by the same corporation and an IBM-PC compatible micro computer.

Rosemount variable capacitance differential pressure transducers convert the pressures to be measured into analog voltage signals. They possess an input range of ± 0.5 psid corresponding to a full scale output range of ± 2.5 volts DC and were calibrated for their complete range against inclined manometers. The resultant highly linear calibration curves are given in Appendix E. The calibrations were checked regularly during the experiments and were found to be extremely stable. One transducer is used for acquiring the wind tunnel contraction pressure difference while the other measures the test section pressure of interest. More information on these measured pressures will be given in the next section. The output signals from the transducers are transmitted to the M 81 unit described below.

The M 81 Module is activated by the M 901, which in turn is controlled by the micro computer through its serial port. The analog signal input range of the data acquisition system is controlled through software. This range is adjusted to the

appropriate value for each data point, providing the maximum possible resolution for the digitization of each analog signal. M 81 uses a 12 bit-plus-sign, dual slope, integrating analog to digital (A/D) converter. The integration renders a form of active filtering, eliminating high frequency noise. 12 bit-plus-sign provides a resolution of 0.024 percent of the selected range (eg., 1.22 mV for the ± 5 volt range and 2.4 μ V for the ± 10 mV range).

The M 901 unit, which provides the link between the computer and M 81, is connected to the computer through an RS232 serial interface port. A communication rate of 2400 bps was found to be most suitable for error free and time efficient transmission of the data. During a sampling period, individual transfer of every sample to the computer through the serial port is very slow. Therefore, the 24,000 bytes random access memory (RAM) available on M 901 is used for temporary storage of all the data acquired from each sampling period. This enables acquisition of two samples per second from each of the two transducers. Once the sampling period is completed, the data is transferred to the computer for averaging and further processing. More information on the sampling and processing of data is given in the following section.

3.4.1 Data Sampling

The wind tunnel contraction pressure difference is related to the reference inlet centerline dynamic pressure through calibration. This contraction pressure difference is sampled together with the test section pressure of interest. However, the data acquisition system described above can sample only one quantity at a time. Therefore, the test section and contraction pressures are sampled sequentially. Once all the data acquired throughout a complete sampling period are transmitted from the M 901 RAM to the computer, they are grouped to form measurement cycles consisting of "reference pressure - test section pressure - reference pressure." Subsequently, each test section pressure data point is non-dimensionalized by the average of the contraction pressure differences acquired in the same cycle. Finally, the individual non-dimensional samples are averaged.

As opposed to non-dimensionalizing averaged quantities, this method minimizes the effects of the small amounts of drift in the wind tunnel operating point and was found to give good repeatability for the data.

The source code of one of the data acquisition programs that provide communication with M 901 and process the resultant data in the above described manner is given in Appendix D.

A parameter of significance is the overall length of the sampling period. It should accommodate a large number of samples. Furthermore, it has to contain all possible frequencies in the measured pressures to provide accurate long term average values. The proper sampling time depends not only on the type of pressure measured, but also on the kind of flow in which the measurement is taken. Therefore, tipwall static pressures, blade static pressures, boundary layer and free stream pitot tube pressures, three-hole and seven-hole probe pressures were analyzed separately to determine the appropriate sampling period for each type of measurement. A sampling time of 40 seconds proved to be sufficient for the tipwall and blade static pressures. For the probe measurements, 60 seconds was found to be appropriate.

A 48 port Scanivalve is employed for all measurements other than pitot tube measurements and is stepped by the microcomputer. This control of the Scanivalve is performed by the data acquisition programs with signals sent through the parallel port data pins. Due to the internal volume in the Scanivalve, the volume of the actual transducer, and the tubing that connects these instruments, several seconds are necessary for the pressure reading to reach an asymptotic value. Therefore, a conservative waiting period of 10 seconds is used prior to each sampling period after stepping the Scanivalve.

Prior to the actual sampling period, samples are acquired for a period of 10 seconds without amplifying the analog signals. These samples are then used to determine the proper amplification level to be employed during the actual sampling period. This,

combined with the waiting period, actual sampling period, time for data transmission, and non-dimensionalizing and averaging of the data results in a total measurement time of about 100 to 110 seconds for each test section probe pressure data point, and 80 to 90 seconds for tipwall and blade pressure measurements.

3.5 Three-Hole Pressure Probe

A three-hole pressure probe was used in non-nulling mode for velocity magnitude, flow angle and total pressure measurements inside the tip clearance. The probe was also employed for periodicity checks and for inlet tipwall boundary layer traverses with the moving tipwall configuration. The probe geometry is shown in Figure 3.5.1. As seen from the figure, the probe head is only 0.7 mm thick and 2 mm wide. For a clearance size of 3.8 percent blade chord (9.5 mm), these probe dimensions enabled measurement of the gap flow with small probe interference. The s-shaped probe body is the only difference of the probe from the one used by Yaras (1987) for earlier measurements inside the tip gap.

Prior to the cascade experiments the three-hole probe was calibrated for yaw angles over the range of ± 70 degrees with an interval of one degree. This calibration was checked regularly during the experiments. A description of the calibration procedure and the resultant calibration curves are given in Appendix F. Least-square curves were fitted to the calibration data. This curve fitting is believed to be the dominant source of error in the probe measurements. Based on the error analysis given by Yaras (1987), the uncertainty in the measured yaw flow angle is estimated to be ± 1.5 degrees, and in the dynamic and total pressures 4 percent of the actual local dynamic pressure.

For pitch misalignments between the oncoming flow and the probe axis, the above quoted accuracies are no longer valid. Yaras (1987) showed that for pitch angles up to ± 30 degrees the measured yaw angles are accurate to within ± 4 degrees. The error in the

Figure 3.5.1 Three-hole pressure probe geometry.

measured dynamic pressure increases with increasing pitch misalignment. The magnitude of this error depends on the yaw angle as well. The parameter most affected by pitch misalignment is the total pressure. Significant underestimation of local total pressure was found by Yaras (1987) with increasing pitch misalignment. For instance, for ± 30 degrees pitch misalignment the local total pressure was underestimated by about 30 percent of the local dynamic pressure. This must be considered when evaluating the total pressure distribution measured inside the separation bubble on the blade tip.

Shear, wall proximity and turbulence are three other error sources. These effects do not appear to have been investigated for three-hole probes. However, if the correlations suggested for pitot probes are applied to the three-hole probe, the resultant errors for the tip-gap flow are considerably smaller than those discussed in previous paragraphs.

3.6 Seven-Hole Pressure Probe

A seven-hole pressure probe was used in the non-nulling mode for the measurement of the complex, three dimensional flow downstream of the blades. It was preferred over laser doppler velocimeter and hot wire probes mainly due to its ability to measure total pressure besides velocity. Cost, simplicity and ruggedness are other factors that contributed to this decision.

3.6.1 Probe Geometry

Multi-hole pressure probes are often manufactured with precise spherical or hemispherical shapes. Such probe tip shapes yield the simplification of using potential flow theories to relate the probe pressures to the local dynamic pressure, static pressure and flow direction. It is, however, practically impossible to manufacture tips without imperfections in surface geometry and hole location. Even if that were possible, pressure holes of finite size are sufficient to cause significant deviation from the potential theory.

To account for such imperfections, numerous empirical coefficients have to be substituted into the expressions derived from potential flow theories (eg., Kjelgaard, 1988). This in turn will require calibration of each probe, removing the advantage of having tip shapes for which the flow can be predicted. Recognizing this drawback, probes possessing spherical and hemispherical tip shapes have been replaced by more easy to manufacture tip geometries, and calibration methods that are insensitive to the exact probe tip shape have been devised.

The 7-hole pressure probe used in this study possesses such a tip geometry. It is made of seven pressure sensing stainless steel hypodermic tubes. The tubes are arranged with one at the center and the other six surrounding this center tube. The resultant bundle is contained in a brass tube. The tip of the probe with the space between the seven tubes filled with solder is tapered to a 30 degrees semi-apex angle. The overall probe tip diameter is 2.4 mm which is reasonably small compared to the scale of the flow structures that were measured in the cascade. The overall probe dimensions are given in Figure 3.6.1.1.

3.6.2 Comparison of Five-Hole and Seven-Hole Probes

The most commonly used pressure probe for the measurement of three dimensional flows has been the five-hole probe. The conventional calibration procedure for such probes is limited to about 25 degrees flow misalignment with respect to the probe axis (e.g., Judd, 1975). A modified calibration method permits the extension of the calibration range up to flow misalignment equal to probe tip cone semi apex angle (Samet and Einav, 1984). Thus, one could increase the probe tip apex angle to increase the range of flow angles that the probe can measure. This, however, would result in reduced probe sensitivity to flow angle. In the tip vortex downstream of the blades it is conceivable to have flow angle variations as large as 70 degrees. If such a flow is to be measured with a five-hole probe, it is likely that occasionally the probe will not be usable due to range limitations. To overcome this problem, Gerner and Maurer (1981) proposed the

Figure 3.6.1.1 Seven-hole pressure probe geometry.

application of 7-hole probes for which the calibration range is not limited by the probe tip cone semi-apex angle. The present probe was calibrated up to 50 degree flow misalignment with the probe axis using 5 degree steps. The calibration procedure and resultant calibration curves are given in Appendix G.

3.6.3 Error Sources in Seven-Hole Probe Measurements

3.6.3.1 Curve-Fitting to the Calibration Data

The main error source in seven-hole probe measurements is the curve-fitting to the calibration data. The five degree step size used during calibration limited the order of least-square polynomial that could be fitted to the data. Input of the calibration data into the program that uses the fitted calibration curves suggested accuracies of ± 1.5 degree for the measured flow angle, and ± 4 percent actual local dynamic pressure for the measured local dynamic and total pressures.

3.6.3.2 Spatial Resolution and Streamline Displacement

Many three dimensional flow fields, such as the tip vortex, contain high velocity and pressure gradient regions. When a 7-hole probe is used in such flows, spatial resolution problems arise since pressures from different ports are not measured at the same physical location. The error associated with this limitation is dependent on the magnitude of the gradients. Thus, most of the proposed correction schemes are based on the magnitude of the velocity and/or pressure gradients.

Another error source for probe measurements in flow regions with velocity gradients is the deflection of the streamlines toward the region of lower velocity. This deflection is a result of a "downwash velocity" from high streamwise velocity to low velocity regions.

Several schemes have been devised for correcting for these two error sources (eg., Sitaram et al., 1981; Ligrani et al., 1989). One drawback is that all of them are applicable to 5-hole probes only. In addition, the various empirical coefficients employed in these methods do not make them very attractive. Finally, none of the methods consider the two errors together. The effect of streamline displacement is simply a virtual displacement of the measurement point toward the higher velocity region. This suggests a close interaction between spatial resolution and streamline deflection problems. Therefore, a proper correction scheme should consider the two error sources in a combined fashion.

Recognizing the lack of accurate methods to correct for this problem, one has to try to reduce the magnitude of this error as much as possible. The only effective way to reduce the error is to manufacture the probe as small as possible, accepting the trade off in terms of larger pneumatic time constants. As was noted earlier, the 7-hole probe that was used in the cascade test section has an overall diameter of 2.4 mm, which is smaller than many 5- and 7-hole probes in use today.

An experimental facility to produce static pressure and velocity gradients of magnitudes comparable to those observed in the tip vortex was not available. Thus, there was no direct basis for estimating the magnitude of the errors that may have been introduced during the measurement of the tip-vortex flow. Nevertheless, internal checks on the data such as comparison of mass flow rates and circulations measured in different planes suggested that measurements were not seriously affected by these error sources.

3.6.3.3 Wall Proximity Effects

When a probe is near a solid surface, acceleration occurs in the region between the probe and the surface. Studies by Treaster and Yocum (1979), Sitaram et al. (1981), and Tamigniaux and Oates (1986) with 5-hole probes suggested that the error caused by this effect is negligible for probe to wall distances larger than three to four probe

diameters. With a 3.2 mm diameter five-hole probe in contact with a solid wall, Sitaram et al. noted errors of three and four percent in measured total pressure and velocity respectively. The errors for the seven-hole probe of similar size used in the cascade experiments are presumably of the same order of magnitude. Thus, the measurements acquired next to the tipwall will have slightly larger error bounds, and must be interpreted with this in mind.

3.6.3.4 Turbulence Effects

There does not appear to be a reliable scheme available for correcting for turbulent effects on the mean flow quantities extracted from multi-hole probes. Experimental studies by Sitaram et al. (1981) suggested errors in measured mean velocities about 0.33 percent of the actual mean velocity for turbulence intensities less than 10 percent. As will be discussed in Chapter 4, in the cascade experiments the turbulence level was considerably lower than this. Thus, errors due to turbulence can be neglected for the seven-hole probe measurements discussed herein.

3.6.3.5 Reynolds Number Effects

When the seven-hole probe calibration coefficients defined in Appendix G are used, little Reynolds number effects are observed. The effects were confined to the low misalignment range (zero to ten degrees pitch and yaw). As will be noted in Chapter 7, the relative tipwall motion experiments had to be undertaken with a cascade Reynolds number about a quarter of that for the stationary wall tests. Thus, even though Reynolds number effects on the probe were small, two probe calibrations corresponding to cascade exit mid-span, mid-passage velocity for stationary and moving tipwall configurations were performed.

3.7 Probe Traversing and Calibration Gears

For pitot, three-hole and seven-hole probe measurements in the cascade a traversing gear that provides linear motion in spanwise and pitchwise directions was employed. The gear is mounted to the back plate of the cascade and vertical slots filled with aluminum sliders provide access to the gear through the back plate. A picture of the gear is shown in Figure 3.7.1. Two 4 phase stepping motors provide the motion in the two mutually perpendicular directions. The motors are used in full stepping mode yielding one revolution with 200 steps. The rotating motion output from the stepper motors is translated into linear motion (0.00635 mm/motor step) through 1/4-20 threaded rods. The motors are controlled by DMA 64 controllers manufactured by Rapidsyn Precision Industries Inc.. The TTL compatible signals to drive the controllers are obtained from data pins on the parallel port of the microcomputer. The subroutines that provide these signals at the parallel port are built into the data acquisition programs (Appendix D). The speed of the motors (steps/sec) depends upon the rate at which signals are sent from the microcomputer. This in turn is a function of the computer clock speed. For a computer clock speed of 8 MHz, the motor speed was 157 steps per second. The overall accuracy of the positioning provided by this traversing mechanism was estimated to be ± 0.25 mm for both directions of motion.

For calibration of the three-hole and seven-hole probes the motorized calibration gear shown in Figure 3.7.2 was used. The stepping motors and controllers are identical to those used for the traversing gears. The calibration gear provides motion in both the pitch and yaw directions. High precision helical gears are used to translate the stepping motor rotation into probe rotation in the yaw and pitch directions at 0.01 degrees/motor step. The error in the angular location obtained with the traversing gear is less than 0.05 degrees. This high accuracy is due to virtually eliminated backlash and small "probe rotation to motor step" ratio, both provided by the high precision helical gears.

Figure 3.7.1 Probe traversing gear.

Figure 3.7.2 Probe calibration gear.

3.8 Tipwall and Blade Instrumentation

The plexiglass tipwall portion around the second blade from the test section bottom is instrumented with a closely-spaced grid of static pressure taps of 0.5 mm diameter. The spacing of the taps is smaller near the blade tip, providing a good resolution of the high pressure gradients experienced in that flow region. The grid extends over one blade pitch. The location of the taps with respect to the blade tip is shown in Figure 3.8.1. Repeatability tests for the tipwall static pressures suggested an accuracy of ± 0.035 for the static pressure coefficients.

The outer half of the aluminum center blade in the cascade is instrumented with 14 chordwise rows of static pressure taps of 0.51 mm diameter, each row containing 37 pressure side and 36 suction side taps. These provide a very detailed picture of the blade loading. The row locations relative to the blade tip and tap co-ordinates within each row are given in Appendix C. On each blade surface the taps at the same chordwise location are connected to the same channel inside the blade. Thus, during experiments only one row of taps can be used at a time and the other rows are sealed with a thin tape. Repeatability tests for the blade static pressures suggested an accuracy of ± 0.015 for the static pressure coefficients.

Figure 3.8.1 Tip-wall static pressure tap locations.

4. OPERATING CONDITIONS FOR STATIONARY TIPWALL EXPERIMENTS

In the experiments with plexiglass window tipwall (i.e, no relative blade motion simulated) measurements were performed for a Reynolds number of 4.3×10^5 based on the cascade inlet centerline velocity and the blade chord. During experiments total temperature of the flow entering the cascade varied substantially (up to 20 degrees Fahrenheit). As was noted in Section 3.4.1, the inlet centerline velocity is related to the contraction pressure difference through calibration. Thus, by monitoring the contraction pressure difference, adjustments could be made on the fan rotational speed to compensate for the temperature changes. In this way the Reynolds number was held constant to within $\pm 0.1 \times 10^5$. The inlet velocity corresponding to this operating point was about 30 m/sec, simulating essentially incompressible conditions. The inlet freestream turbulence intensity, $\frac{u'}{V_{cl}}$, measured with a hot wire, was about 1.5 percent.

In Section 3.2 it was noted that shims inserted between the tipwall and the side walls enabled the clearance size to be adjusted to any desired value. The slope of the ramp on the tipwall created by the insertion of shims varied with clearance size. Thus, larger clearances requiring a larger slope for the ramp resulted in more diffusion of the flow before entering the cascade. Consequently, the inlet tipwall boundary layer, measured 100 mm downstream of the ramp (170 mm upstream of the leading edge plane), varied with clearance. The integral parameters of the boundary layer on the tipwall at inlet were calculated from boundary layer traverses with a pitot tube of 0.8 mm tip outer diameter. The results are given in Table 4.1. The observed trends will be discussed in Section 6.3.

To identify the overall uniformity of the inlet flow, pitot tube traverses across the inlet span were performed 330 mm ($1.67c_x$) upstream of the leading edge plane, at seven vertical positions as shown in Figure 4.1. The resultant velocity distributions are given in Figure 4.2. In the free stream, deviations from the centerline dynamic pressure, q_{cl} ,

τ/c	0.0	0.015	0.020	0.028	0.055
τ (mm)	0.0	3.8	5.1	7.0	13.7
δ_x^* (mm)	3.6	6.5	4.8	6.8	2.8
θ_x (mm)	2.8	5.0	3.4	4.2	2.0
H_x	1.3	1.3	1.4	1.6	1.4

Table 4.1 Inlet tip-wall boundary layer parameters.

Figure 4.1 Inlet flow traverse locations.

Figure 4.2 Test section inlet velocity distribution at seven vertical locations.

are within 0.25 percent of q_{cl} . Toward the bottom of the inlet section the dynamic pressure is relatively high. This may be due to the use of a centrifugal fan in the tunnel.

5. DOWNSTREAM DEVELOPMENT OF THE TIP-LEAKAGE FLOW: VORTICITY FIELD

5.1 Introduction

In this section the strength of the tip vortex and its diffusion downstream of the trailing edge plane will be studied in terms of vorticity. Vorticity is the most suitable parameter for elucidating quantitatively the three dimensional structure of various vortices downstream of the blades. It is especially effective in distinguishing the boundary and/or interaction between adjacent vortical structures of opposite sense of rotation (eg., tip-leakage and passage/scraping vortices). Such structures can appear merged when viewed in terms of more conventional parameters such as total pressure. Inoue and Kuroumaru (1984) provided the first detailed study where vorticity is used in the evaluation of the flow field affected by tip leakage downstream of an axial flow compressor rotor.

As will be discussed in Chapter 6, significant tip-leakage losses occur due to mixing downstream of the blades. Therefore, studying the tip vortex diffusion in terms of vorticity should provide a valuable first step towards understanding this tip-leakage loss generation mechanism.

Section 6.8 will review the common tip-leakage loss prediction schemes. In that section it becomes evident that the tip-vortex circulation is one of the common parameters used in tip-leakage loss correlations. However, there is a definite lack of understanding of the parameters that influence the vortex strength. Discussions presented in this chapter will provide some physical insight into this aspect.

Finally, the cross-flow associated with the tip vortex and its decay with distance from the trailing edge plane is important in the sense that it will affect the incidence seen by a subsequent blade row. Based on the understanding of the vorticity field, a simple model for predicting this cross flow will be developed in Section 5.5.

5.2 Vorticity and Circulation Calculations

In cartesian co-ordinates the vorticity vector components are defined as follows:

$$\begin{aligned}\omega_x &= \frac{\partial w}{\partial y} - \frac{\partial v}{\partial z} \\ \omega_y &= \frac{\partial u}{\partial z} - \frac{\partial w}{\partial x} \\ \omega_z &= \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\end{aligned}\tag{5.2.1}$$

For the co-ordinate system and measurement planes shown in Fig. 5.2.1, the x-wise vorticity component can be calculated from the measurements in a single traverse plane. However, for the other two vorticity components it is necessary to take measurements in at least two closely spaced y-z planes. The measurement of the three dimensional flow field was done with the seven-hole pressure probe. In each plane, measurements were taken on an evenly spaced grid of one passage height and half span width. In the trailing edge plane (Plane B), where the tip vortex is confined to a relatively small space, a spacing between adjacent measurement points of 2.5 percent blade span gave sufficient resolution, leading to accurate results for integrated quantities. For measurement planes further downstream (Planes A, C₁, C₂ and D in Figure 5.2.1) a spacing of 5 percent blade span was found to be sufficient.

As mentioned, measurements in two closely spaced planes appear to be necessary to quantify the y and z vorticity components. However, this doubles the experiment time. Therefore, an alternative approach was evaluated. The conventional form of Euler's

Figure 5.2.1 Seven-hole probe traverse plane locations.

equation for steady incompressible flow written in the x direction is,

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial x}. \quad 5.2.2$$

By rearranging and adding extra terms to both sides of the equation,

$$v \frac{\partial v}{\partial x} - v \frac{\partial u}{\partial y} - w \frac{\partial u}{\partial z} + w \frac{\partial w}{\partial x} = \frac{1}{\rho} \frac{\partial p}{\partial x} + u \frac{\partial u}{\partial x} + v \frac{\partial v}{\partial x} + w \frac{\partial w}{\partial x} \quad 5.2.3$$

which can be rewritten,

$$v \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \right) - w \left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x} \right) = \frac{1}{\rho} \left(\frac{\partial p}{\partial x} + \frac{\rho}{2} \frac{\partial u^2}{\partial x} + \frac{\rho}{2} \frac{\partial v^2}{\partial x} + \frac{\rho}{2} \frac{\partial w^2}{\partial x} \right). \quad 5.2.4$$

The terms in parentheses on the left-hand-side are vorticities in the z and y directions, and the right-hand-side can be written in terms of total pressure. Thus, Euler's equation in the x direction can be written as,

$$v\omega_z - w\omega_y = \frac{1}{\rho} \frac{\partial P_o}{\partial x}. \quad 5.2.5$$

Similar expressions can be obtained for the y and z directions:

$$\begin{aligned} \frac{1}{\rho} \frac{\partial P_o}{\partial y} + u\omega_z - w\omega_x &= 0 \\ \frac{1}{\rho} \frac{\partial P_o}{\partial z} - u\omega_y + v\omega_x &= 0. \end{aligned} \quad 5.2.6$$

By rearranging Equations 5.2.6, the y and z vorticity components can be expressed as:

$$\begin{aligned}\omega_x &= \frac{1}{u}(w\omega_z - \frac{1}{\rho}\frac{\partial P_o}{\partial y}) \\ \omega_y &= \frac{1}{u}(v\omega_z + \frac{1}{\rho}\frac{\partial P_o}{\partial z}).\end{aligned}\tag{5.2.7}$$

From these simplified expressions the vorticities can be obtained from measurements in a single y-z plane only, reducing the experiment time substantially. However, Euler's equation is the momentum equation written under the assumption of inviscid flow. To check the validity of this assumption, vorticity components calculated from measurements in planes C_1 and C_2 were compared to the results obtained from Eqns. 5.2.7 using plane C_1 data. The comparisons are given in Figures 5.2.2 and 5.2.3 for clearance sizes of 2 and 5.5 percent of the blade chord. As seen from the figures, the agreement is very good for both vorticity components. The maximum deviation is below 10 percent of the vortex centerline streamwise vorticity values in the corresponding planes. Therefore, for determining ω_y and ω_z , Equations 5.2.7 were used throughout the experiments instead of the approach where measurements have to be taken in at least two closely spaced planes.

To non-dimensionalize the vorticity components, the velocity components were non-dimensionalized by the cascade inlet centerline velocity, the total pressure by the inlet centerline dynamic pressure and the distances by the blade chord length. Thus, the expressions for calculating the non-dimensional vorticities are as follows:

$$\begin{aligned}\omega'_x &= \frac{\partial(w/V_{cl})}{\partial(y/c)} - \frac{\partial(v/V_{cl})}{\partial(z/c)} \\ \omega'_y &= \frac{1}{u/V_{cl}}[\frac{v}{V_{cl}}\omega'_x + \frac{\partial C_{p_o}}{\partial(z/c)}] \\ \omega'_z &= \frac{1}{u/V_{cl}}[\frac{w}{V_{cl}}\omega'_x - \frac{\partial C_{p_o}}{\partial(y/c)}].\end{aligned}\tag{5.2.8}$$

Figure 5.2.2 Spanwise distribution of y-wise vorticity component through the tip vortex center.

Figure 5.2.3 Spanwise distribution of z-wise vorticity component through the tip vortex center.

This choice of non-dimensionalizing parameters will be justified later in this section.

The differential terms in Eqns. 5.2.8 were replaced by finite differences and were determined by central differencing, except at the boundaries of the measurement grid, where simple differencing was used. To evaluate the applicability of this finite differencing approach, for several clearances and measurement plane locations, rational surface splines were fit to u, v, w and C_p . Vorticities were determined using the velocity and total pressure gradients obtained from these rational splines. Various spline tension coefficients were tested ranging from simple cubic spline (zero tension) to straight lines between adjacent data points. The calculated vorticities showed virtually no variation with the selected tension coefficient. Therefore, linear variation of quantities between adjacent data points was assumed for calculating vorticities.

The circulation around the passage flow or a portion of it was calculated using the definition of circulation given by Stokes' theorem:

$$\Gamma = \oint \omega_n dA \quad 5.2.9$$

where ω_n is the vorticity component normal to area dA . Since the measurement planes were parallel to the y - z plane, the simplest approach was to integrate the x -wise vorticity over the area of interest. For the integration process a linear variation of vorticities was assumed within a panel formed by four grid points. When the vorticity is non-dimensionalized as described earlier and the area non-dimensionalized by the square of chord length, the parameter used for non-dimensionalizing the resultant circulation becomes $V_{CL} c$. To establish the appropriateness of this non-dimensionalizing parameter, a simple derivation for bound circulation is performed as follows.

Blade lift per unit span can be written as,

$$L = C_L \frac{1}{2} \rho V_m^2 c \quad 5.2.10$$

where lift can be expressed in terms of the bound circulation as,

$$L = \rho V_m \Gamma_B \quad 5.2.11$$

Substituting 5.2.11 into 5.2.10 and dividing both sides of Eqn. 5.2.10 by the centerline dynamic pressure at cascade inlet,

$$\Gamma'_B = \frac{\Gamma_B}{V_{cl} c} = \frac{C_L}{2} \cos \alpha_m \quad 5.2.12$$

where Γ'_B is the non-dimensional bound circulation and α_m is the mean flow angle as defined in the Nomenclature. Thus, $V_{cl} c$ does indeed appear to be the proper non-dimensionalizing parameter for the bound circulation.

5.3 General Observation of the Vorticity Field Downstream of the Blades

Before studying the tip vortex in detail, a general observation of the passage flow field downstream of the blades in terms of streamwise vorticity is in order.

The streamwise vorticity is determined from the vorticity components in the x and y directions using the expression:

$$\omega'_s = \omega'_x \cos \alpha - \omega'_y \sin \alpha \quad 5.3.1$$

where α is the local midspan flow direction in the measurement plane of interest. The contour plots of streamwise vorticity for a clearance size of 5.5 percent blade chord are given in Figures 5.3.1 and 5.3.2. The same contour interval is used on all three plots to provide a quick sense for the amount of diffusion with downstream distance. One drawback of these plots is that, particularly in planes C_1 and D, the blade wakes are not quite evident since little streamwise vorticity is associated with them. Therefore, the corresponding contours of total pressure loss coefficient are given in adjacent figures.

The vorticity contours show the presence of two vortices. The dominant one is the tip vortex covering about one third of the blade span, rotating clockwise when viewed from downstream. Comparable spanwise extents were quoted in compressor studies (eg., Storer, 1989; Schmidt et al., 1987a). The second vortex is the passage vortex located between the tip vortex and the tipwall. As expected, the sign of its vorticity indicates that it has a sense of rotation opposite to that of the tip vortex. Its size, however, is quite small. This is partly explained by the significant amount of inlet tipwall boundary layer entering the tip gap and contributing to the formation of the tip vortex, and will be elaborated upon in Chapter 6. However, even for zero clearance size (Figures 5.3.3 and 5.3.4) the passage vortex is quite small. This is presumably a result of the relatively low flow turning (about 45 degrees) on our cascade and the relative thinness of the tipwall boundary layer at inlet. Therefore, it may not be a general result.

Comparison of the streamwise vorticity values on the vortex centerline for the downstream measurement planes shows substantial diffusion over the first axial chord length downstream of the trailing edge plane (Figure 5.3.5). Hence, one should expect the majority of the mixing losses to occur over this distance. This will be confirmed in Chapter 6.

Figure 5.3.1

Streamwise vorticity and total pressure loss coefficient distributions for $\frac{\tau}{c} = 0.055$.

Figure 5.3.2 Streamwise vorticity and total pressure loss coefficient distributions for $\frac{\tau}{c}=0.055$.

Figure 5.3.3 Plane B total pressure loss coefficient distribution for $\frac{\tau_c}{c} = 0.0$.

Figure 5.3.4 Plane B non-dimensional streamwise vorticity distribution for $\frac{\tau_c}{c} = 0.0$.

Figure 5.3.5 Variation of streamwise vorticity with downstream distance.

The relative locations of the tip vortex with respect to the blade wakes in the three measurement planes suggest that the vortex centerline has a smaller pitch angle than the exit flow angle. In fact, one axial chord length downstream of the trailing edge plane the tip vortex starts to interact with the adjacent blade wake. The axial location where the interaction is initiated depends on the blade row solidity, thus the observation is not a general result. Further downstream, in plane D, the tip-leakage vortices from adjacent passages begin to interact with each other.

Another observation is that at the trailing edge plane the roll up of the tip vortex is already nearly complete. Moreover, if the angle between the vortex centerline and the measurement plane is taken into account, the vortex is nearly axisymmetric. These observations can be used to advantage in modelling the vortex and will be elaborated upon in Section 5.5. Lakshminarayana et al. (1987) noted that they did not observe a discrete tip vortex downstream of their compressor rotor. Thus, they suggested that cascades do not provide useful simulation of the actual flow encountered in turbomachinery rotors. On the other hand, Inoue et al. (1986) and Inoue and Kuroumaru's (1988) compressor rotor measurements did show a discrete tip vortex. Therefore, the degree of simulation of the actual turbomachinery rotor flow does not seem to be the explanation for the observed differences in the tip vortex structure. Moyle (1989) suggested that the radial component of pressure gradient near the tipwall is responsible for this variation. The author developed an analytical expression that relates the radial component of pressure gradient near the wall to the passage geometry, tip speed, skew, through-flow and wall friction coefficient. However, a drawback is that the expression is not applicable to cascades without relative motion.

The downstream flow field was measured for clearances varying from 1.5 to 5.5 percent of blade chord. The differences between smaller clearances (Figure 5.3.6) and the large clearance case studied in detail above are the size of the tip vortex and the vortex centerline pitch angle: as the clearance size is reduced the vortex size becomes

Figure 5.3.6 Streamwise vorticity and total pressure loss coefficient distributions for $\frac{\tau}{c}=0.015$.

Figure 5.3.7 Variation of tip vortex centerline orientation with clearance size.

smaller while the vortex centerline angle with respect to the inlet flow direction increases (Figure 5.3.7). The trend in the vortex centerline pitch angle, also observed by Inoue and Kuroumaru (1988) in a compressor rotor, is presumably explained by the reduced tip-gap mass flow rate resulting in less momentum normal to the blade chord as the tip-gap size is decreased. The variation in tip-vortex strength, while recognized for some time, has not been fully understood. Numerical results for the vortex strength, with a feasible explanation for the observed trend, are presented in the next section.

5.4 Effect of Clearance Size on Tip-Vortex Circulation

Lakshminarayana and Horlock (1962) were first to observe a reduction in the tip-vortex circulation as the gap size was decreased. The authors drew this conclusion from the lift at the blade tip obtained by extrapolating the spanwise blade pressure distribution to the blade tip for an isolated blade split at midspan. In the view of substantial spanwise pressure gradients on the blade surfaces and distortions in the pressure field caused by the tip vortex next to the blade suction surface, this does not seem to be a reliable approach.

A similar trend in the tip-vortex circulation was observed by Lewis and Yeung (1977) in a turbine cascade. These authors based their argument on static pressures measured on the tipwall along a contour following the blade shape. However, from the tipwall static pressures shown in Figure 5.4.1, the pressure field impressed on the tipwall is somewhat shifted in the pitchwise direction relative to the blade. Thus, Lewis and Yeung's approach is likely to give misleading values of the pressure differences experienced by the flow near tipwall.

Finally, Inoue et al. (1986) quoted similar trends for the tip vortex strength in a compressor rotor. The authors based their calculations on 5-hole pressure probe measurements downstream of the blade row. Therefore, in terms of the magnitude of the tip-vortex circulation this study is likely the most accurate one amongst the ones discussed so far. For comparison, the tip-vortex circulation was determined for the

Figure 5.4.1 Tip-wall static pressure distribution ($\frac{\tau}{c}=0.028$) .

present cascade by integrating the x -wise vorticity in Plane B. The same integration was performed in planes C_1 and C_2 as well. As seen from Figure 5.4.2, planes C_1 and C_2 give somewhat lower tip vortex circulation compared to the trailing edge plane. This is presumably a result of the interaction of the tip vortex with the passage vortex and the transverse tipwall boundary layer perpendicular to the undisturbed exit flow direction. Thus, it appears more appropriate to consider the tip-vortex circulation in Plane B where any interaction of the tip vortex with other vorticity fields downstream of the blades is excluded.

The calculated circulations and published data are given as fractions of bound circulation in Figure 5.4.3. A consistent trend is observed for all configurations. As was indicated by the plots in the previous section, little vorticity is associated with the blade wakes. Therefore, one would expect essentially all the bound vorticity to be contained in the tip vortex. However, the trend in Figure 5.4.3 suggests that this is only true for very large clearances where a finite wing configuration is approached. Lakshminarayana and Horlock (1962) proposed that part of the bound vorticity jumps across the gap and ends on the tipwall instead of shedding downstream as part of the tip vortex. Dean (1954) noted this possibility for small clearances only where the flow in the gap is viscous. However, according to the results to be discussed in Section 6.4, for practical clearances a significant portion of the gap flow between the blade tip and the tipwall is inviscid. Consequently, in the absence of a mechanism by which vorticity can be transmitted, it is not possible for any of the bound vortex lines to jump across the gap. An alternative, physically more feasible explanation follows.

Figure 5.4.4 shows the blade midspan pressure difference and the portion of the blade pressure difference reflected to the tipwall. The latter is obtained by determining the peak to peak pressure difference normal to the blade chord direction from the tipwall static pressure measurements. The pressure differences experienced by fluid near the tipwall are seen to be comparable to the blade midspan loading. They are somewhat reduced for the largest examined clearance, which is an expected trend since pressure

Figure 5.4.2 Variation of tip-vortex circulation with downstream distance.

Figure 5.4.3 Variation of bound circulation fraction retained at the blade tip with tip clearance size.

Figure 5.4.4 Blade midspan and tip-wall driving pressure differences.

differences sensed by fluid near the tipwall will eventually become zero at very large clearances. In agreement with these recent results, for practical clearance sizes significant pressure gradients on the tipwall were also observed by Inoue and Kuroumaru (1988) in a compressor rotor. In the absence of a tipwall, an approximate velocity distribution resulting from the pressure gradients in the proximity of the blade tip would be as in Figure 5.4.5. If a tipwall is placed at any point between A and B, it will experience gradients normal to the blade chord line and thus, vorticity will be generated on the wall in the chordwise direction. This vorticity will have a sense of rotation opposite to that of the bound vorticity shedding from the blade tip (Figure 5.4.6). This is supported by Bindon's (1987b) flow visualization studies where the leakage flow nearest the tipwall was found to mingle with the passage vortex instead of joining the tip vortex. Interaction of the vorticity field on the tipwall with the bound vorticity shedding from the blade tip may provide a feasible explanation for the variation of the tip-vortex circulation as a fraction of the bound circulation. The magnitude of the gradient sensed by the wall and thus the amount of vorticity generated, is a function of the distance between the blade tip and the wall, i.e the tip clearance size. Therefore, tip clearance size is likely a dominant parameter influencing the tip-vortex circulation.

The remaining question is the nature and location of the interaction between the bound vorticity on the blade tip and the vorticity generated on the tipwall. The fact that the core of the gap flow is inviscid eliminates the possibility of interaction within the gap. The remaining feasible interaction region is past the tip-gap exit where the leakage flow merges with the passage stream. Even though absence of data within that region prevents a definite conclusion, the fact that the tip vortex is located next to the tipwall transverse boundary layer and adjacent to the passage vortex that is strengthened by the transverse boundary layer on the tipwall makes this a feasible explanation.

Figure 5.4.5 Velocity distribution normal to the blade chord in the vicinity of the blade tip without a tip wall.

Figure 5.4.6 Bound vorticity at the blade tip and vorticity associated with the transverse boundary layer on the tip wall.

In the above discussions it was pointed out that tip clearance size is an important parameter that affects the tip-vortex circulation. Thus, it seems appropriate to plot the tip-vortex circulation against the clearance size as in Figure 5.4.3. For large blade thicknesses and relatively small gap heights it is possible that the vorticity layers on the tipwall and the blade tip may grow enough to cause interaction inside the gap. Thus, blade thickness to chord ratio may be another parameter influential on the tip-vortex circulation especially for small $\frac{\tau}{t_{MAX}}$ ratios. This may be a candidate topic for future research.

Due to reasons mentioned earlier, the agreement of the recent data with Lakshminarayana and Horlock's (1962) and Lewis and Yeung's (1977) data in Figure 5.4.3 is probably fortuitous. On the other hand, Inoue et al.'s (1986) compressor rotor data, which is likely to be more reliable, is considerably below those for the present stationary wall configuration. This difference will be elaborated upon in Section 8.4.1 where the effects of relative wall motion are examined.

5.5 Modelling of the Tip Vortex: Prediction of Vorticity and Cross-Flow Velocity Distributions

In Section 5.3 it was observed that the tip vortex nearly completed its roll-up by the trailing edge plane and it had an axially symmetric shape. Thus, it should be possible to model the vorticity and cross flow velocity field associated with the vortex with a simple scheme. This would enable one to estimate the spanwise variation in incidence seen by the subsequent blade row, permitting it to be designed accordingly.

The simplest possible model that one can suggest is Lamb's (1932) solution for the laminar diffusion with time of a simple line vortex. In the solution the tangential velocity distribution is given by,

$$v_{\theta} = \frac{\Gamma}{2\pi r} \left(1 - e^{-\frac{r^2}{4\nu(\frac{z}{V_s})}} \right) \quad 5.5.1$$

where, Γ is the tip-vortex circulation, s is the distance from the effective vortex origin as shown in Figure 5.5.1, V_2 is the undisturbed velocity at the cascade exit and ν is the molecular viscosity.

The term $(\frac{s}{V_2})$ in Eqn. 5.5.1 represents the approximate elapsed time for the convection of the vortex over length s . Figure 5.5.2a shows a substantial variation in the axial convection velocity through the vortex. This is a consequence of the static pressure rise along the vortex centerline (Figure 5.5.2b) which leads to relative flow up the center of the vortex. In addition, the results discussed in Section 5.3 showed that the vortex centerline is not exactly aligned with the undisturbed mean flow angle at the cascade exit. Thus, the implication in Eqn. 5.5.1 that the vortex is convected downstream from its effective origin with the downstream undisturbed mean velocity is not precise.

Lamb's solution is for laminar vortex diffusion. However, in the tip vortex the flow field is definitely turbulent. In fact, from measurements downstream of a compressor rotor Inoue and Kuroumaru (1984) found that the turbulence quantities are very high in the leakage vortex, presumably because the high mean velocity gradients are an effective source of turbulence energy generation. Therefore, it is necessary to replace the molecular viscosity in Eqn. 5.5.1 by a parameter that will take into account the turbulent momentum transport. In a study of turbulent wing tip vortices, Squire (1965) suggested the replacement of ν by $\nu + \nu_e$ where ν_e is the eddy viscosity. The author proposed the correlation of $\frac{\nu_e}{\nu}$ through the following equation:

$$\frac{\nu_e}{\nu} = a_s \left(\frac{\Gamma}{\nu} \right) \quad 5.5.2$$

where a_s is a constant. Squire noted that the circulation round the vortex is the principal permanent characteristic of the vortex and it should be reasonable to assume that it dominantly determines the character of the motion. Furthermore, Inoue and Kuroumaru (1984) found that, due to conversion of mean vortex energy into turbulence energy, high

Figure 5.5.1 Tip-leakage vortex as a diffusing line vortex.

Figure 5.5.2a Spanwise distribution of axial velocity through the center of the tip vortex.

Figure 5.5.2b Spanwise distribution of static pressure
through the center of the tip vortex.

turbulence stresses remain effective in the tip vortex despite long effective flowing distances. Thus, it seems reasonable to identify a single eddy viscosity value effective in the vortex through a relation such as Eqn. 5.5.2. However, as seen from Figure 5.5.3, a_t is dependent on the clearance size. El-Ramly (1972) noted that Owen (1970), who compared the results of various laboratory experiments, found that even though a_t was roughly constant for a given core, it differed by as much as a factor of ten from one core to another. Thus, Owen (1970) suggested an alternative parameter,

Λ , to be used in the following expression:

$$\frac{v_t}{v} = \Lambda^2 \sqrt{\frac{\Gamma}{v}} \quad 5.5.3$$

The comparison of various experimental results by Owen suggested relatively little variation in the value of this parameter compared to a_t . The recent cascade studies for different clearance sizes, resulting in tip vortices of different strengths, support this observation (Figure 5.5.3). Thus, Eqn. 5.5.3 employing Λ seems to be a more suitable correlation for the calculation of the eddy viscosity. It should be noted, however, that the recent Λ value around 1.3 is slightly higher than the values compiled by Owen (0.7 to 1.2) for wing tip vortices.

For the present cascade experiments the constants a_t and Λ together with the effective vortex origin were determined from the streamwise vorticity. From Lamb's solution,

$$\omega_s = \frac{\Gamma}{4\pi v(\frac{s}{V_2})} e^{\frac{-r^2}{4\pi(\frac{s}{V_2})}} \quad 5.5.4$$

which for $r=0$ (the vortex centerline) becomes,

$$\omega_s = \frac{\Gamma}{4\pi v(\frac{s}{V_2})} \quad 5.5.5$$

Figure 5.5.3 Variation of vortex model parameters with tip clearance size.

at the vortex centerline. Considering the appropriate non-dimensionalizing parameters for ω_z and Γ (Section 5.2), replacing the molecular viscosity by $\nu + \nu_e$, and writing s and V_2 in terms of their axial (x) components, Eqn. 5.5.5 can be expressed in non-dimensional form as follows:

$$\omega'_z = \frac{\Gamma}{4\pi} \frac{Re_c}{(1 + \frac{\nu_e}{\nu}) \cos\gamma (\frac{x}{c_x} - \frac{x_o}{c_x})} \quad 5.5.6$$

Thus for a given clearance size the two unknowns, the effective vortex origin and eddy viscosity, can be determined from the streamwise vorticity values at the vortex center at two different axial locations, and the tip-vortex circulation. As was discussed in Section 5.4, the tip vortex circulation was obtained for each clearance by integrating the x-wise vorticity in Plane B. This circulation and vortex centerline ω'_z values in planes B and C₂ resulted in eddy viscosity values about 500 to 650 times the molecular viscosity whereas the effective vortex origin location was found to depend on clearance size (Figure 5.5.3).

Having quantified all the required parameters, the radial variations of streamwise vorticity and tangential velocity at a desired axial location can be determined using Equations 5.5.4 and 5.5.1 respectively. The comparison of these predictions with experimental data is presented in the next section.

5.5.1 Comparison of Experimental and Predicted Results

For the largest clearance of 5.5 percent blade chord, downstream measurements were taken at plane D in addition to planes B and C. Thus, it was possible to check the accuracy of the model in predicting the axial variation of streamwise vorticity at the vortex centerline. The predictions using Eqn. 5.5.6 are compared to experimental results in Figure 5.5.1.1. The dashed curves in the figure separate the four vorticity component

Figure 5.5.1.1 Variation of vorticity components with downstream distance.

data from each other for clarity. As is seen from the figure, the measured streamwise vorticity at the vortex centerline in Plane D is quite accurately predicted (solid curve). The other three vorticity components are given in the figure for completeness.

The tangential distribution of streamwise vorticity is predicted accurately as well (Figure 5.5.1.2). It should be noted, however, that with the exception of Plane D for 5.5 percent clearance, the predictions in this figure are implicitly forced to pass through the centerline experimental ω'_z value since this parameter was used to quantify the unknown variables in the model. Therefore, the predictions for planes B and C₁ are only a partial test of the model.

Figure 5.5.1.3 shows the experimental and predicted tangential velocity distributions for two clearance sizes. The experimental tangential velocities were determined as the velocity components in the plane normal to the vortex centerline. As for the streamwise vorticities in Figure 5.5.1.2, the results for all axial locations are presented with the vortex centers coincident. The cross-flow in plane B is substantial resulting in a velocity vector swing of about 75 degrees. Despite the simplifying assumptions discussed earlier, the model seems to provide very reasonable predictions of the cross flow distributions at all three axial locations. There appears slight disagreement outside the vortex core. This is partly due to the induced effect of the tip vortices in adjacent passages, not considered in the prediction scheme. Furthermore, it was found that the magnitude of the tangential velocities in the far field are sensitive to the direction selected for the vortex centerline. Thus, slight inaccuracy in the vortex centerline orientation may be part of the explanation for the observed disagreement between experimental and predicted results in the figure.

Before concluding this section it is important to recognize that the model presented herein is for stationary blades. Having determined that the proposed model is successful for this configuration, the next step would be to identify its applicability to the more realistic case with relative blade motion. This issue will be dealt with in Section 8.5.

Figure 5.5.1.2 Spanwise distribution of streamwise vorticity through the center of the tip vortex.

Figure 5.5.1.3 Spanwise distribution of tangential velocity through the center of the tip vortex.

6. EVALUATION OF TIP-LEAKAGE LOSSES

6.1 Introduction

Tip-leakage losses account for a substantial portion of the total aerodynamic losses in turbomachinery blade rows. In turbines the tip-leakage penalty and its rate of change with tip clearance are even greater due to higher blade loadings. In small turbines, where relative tip clearance size tends to be larger, tip-leakage losses may account for up to one third of the total aerodynamic losses in the stage.

The tip-leakage flow and its interaction with the rotor passage flow field has only recently received significant attention due to the relative complexity of the flow. Consequently, for a long period tip-leakage loss prediction schemes employed for design purposes have been simple, empirical correlations. Recognizing the lack of understanding of the loss mechanisms associated with this flow, several detailed experimental studies have been initiated within the last decade (eg., Lakshminarayana et al., 1986; Lakshminarayana et al., 1987; Schmidt et al, 1987a; Dishart and Moore, 1989; Bindon, 1989). In these studies, the following possible loss mechanisms associated with the tip-leakage flow were explored:

- (i) the loss mechanisms inside the tip gap;
- (ii) the sudden expansion and mixing losses as the tip gap flow merges with the passage flow;
- (iii) and the losses associated with mixing-out of the tip vortex downstream of the blades.

Thus tip-leakage flow measurements providing information inside the tip gap, within the blade passage and downstream of the blade row are necessary to identify the presence, nature and relative contribution of these possible tip-leakage loss mechanisms. However,

each of the published studies dealing with this issue concentrated on only one or at most two of the noted flow regions.

The present study, together with the results of Yaras (1987), does provide such a comprehensive evaluation of tip-leakage loss mechanisms. The experimental results and/or the conclusions drawn will be compared to those provided by other studies wherever possible.

6.2 Mass-Averaged Loss Coefficient Definitions and Experimental Uncertainty

To obtain an overall, representative value for a quantity of interest and to compare it to results obtained in other measurement locations, mass-averaging was performed. In this process, two requirements have to be met for the study to provide valuable results:

- a) the measurement plane should be selected large enough to contain the complete flow field of interest;
- b) all measurement planes whose results are to be compared to each other should be mass-averaged over the same stream tube.

In this case tip-leakage loss is the quantity of interest. The first requirement noted above was met by measuring the complete flow field inside the tip gap (blade tip to tipwall; leading edge to trailing edge) and by using the measurement planes downstream of the blade row which are one pitch high and a half span wide. To satisfy the second condition, a reference mass flow rate corresponding to free stream fluid passing through an area of one half span and one pitch at the cascade inlet was selected. All losses are referred to this reference mass flow rate. The downstream measurements showed 5 to 10 percent less mass flow rate than at the inlet. This was presumably due to the displacement effect of the tip vortex in the outer half of the passage. The fluid displaced is expected to be largely free stream fluid. Thus, for the mass-averaging calculations in

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the downstream planes, the mass flow rate was topped up with loss free fluid until the reference mass flow rate was reached.

The following sections will study the mass-averaged total pressure loss and mass-averaged kinetic energy coefficients. In the light of the information given in the previous paragraph, these coefficients can be defined as follows:

$$C_{P_t}'' = \frac{\int_{\Delta x} \int_{\Delta y} C_{P_t}(y,z) \rho V_n dy dz}{\dot{m}_{REF}} \quad 6.2.1$$

$$C_q'' = \frac{\int_{\Delta x} \int_{\Delta y} C_q(y,z) \rho V_n dy dz}{\dot{m}_{REF}}$$

where the local total pressure loss coefficient, C_{P_t} , and the local kinetic energy coefficient, C_q , are defined in the nomenclature. For mass-averaged losses, chordwise distributions inside the gap and pitchwise distributions at a particular spanwise location in the downstream planes may also be of interest. For this purpose a local mass-averaged loss coefficient is defined. For the tip-gap flow it is given as:

$$C_{P_t}' = \frac{\int_0^r C_{P_t} \rho V_n dz}{\int_0^r \rho V_n dz} \quad 6.2.2$$

Error bounds for probe measurements were discussed in Sections 3.5 and 3.6. The uncertainty in mass-averaged quantities is more difficult to quantify. Based on repeatability tests and individual probe measurement accuracies, the mass-averaged quantities at the inlet plane are estimated to be accurate within ± 0.01 . In certain portions of the flow inside the gap and downstream of the blade row high shear effects should result in larger uncertainty. In these flow regions the error bound for mass-averaged quantities is estimated to be ± 0.03 .

6.3 Cascade Inlet, Profile and Secondary Losses

The only loss source at the outer half of the cascade inlet flow is the tipwall boundary layer. As was discussed in Chapter 4, an increase in tip clearance size required a larger ramp slope on the cascade inlet tipwall. Thus, one should expect a thicker wall boundary layer as the tip clearance size is increased. This in turn should increase the mass-averaged loss coefficient at the inlet plane. However, the results quoted in Table 6.3.1 do not suggest such a trend. This is presumably caused by the smoothness at the ramp ends that could not be matched for all clearances. The mass-averaged loss coefficient is observed to be slightly on the positive side for zero clearance. This is because losses are measured relative to the total pressure at the inlet centerline. As shown in Chapter 4, there is some non-uniformity in the inlet free stream flow with the center-line being in a relatively low energy region, which explains the anomalous loss coefficient.

Profile losses were determined by calculating the local pitchwise mass-averaged loss coefficient from the passage flow traverse at midspan. The results quoted in Table 6.3.1 are based on Plane C_1 measurements. Diffusion losses associated with profile losses were found to be small (Goobie, 1989). Thus, Plane C_1 profile losses were taken to apply to all of the downstream traverse planes.

For zero tip clearance size, secondary losses can be determined by subtracting the mass-averaged profile losses and inlet plane losses from overall mass-averaged losses measured in downstream planes. According to Mobarak et al. (1988), analysis of cascade loss data by Sharma and Butler (1987) has shown that inlet end-wall boundary layer losses convect through the blade passage without causing additional loss. Thus, the inlet end-wall boundary layer losses are additive and can easily be distinguished from the secondary losses. Since the passage vortex is next to the blade suction surface and subsequently adjacent to the blade wake (Figure 6.3.1), interaction between secondary and

		Clearance Size τ/c				
		0.0	0.015	0.020	0.028	0.055
Location						
Inlet						
	C''_{p_s}	0.0067	-0.0787	-0.0355	-0.0455	-0.0196
Downstream						
	q_2/q_1	2.13	1.95	2.04	----	1.82
	$C''_{p_s} (Profile)$	-0.063	-0.050	-0.055	----	-0.042

Table 6.3.1 Quantities extracted from flow field measurements.

Figure 6.3.1 Plane B total pressure loss coefficient distribution for $\frac{\tau}{c} = 0.0$.

profile loss mechanisms is inevitable. However, the traditional approach has been to ignore this interaction.

The above described additive approach has also been used for the more complex case of nonzero tip clearance where there are the additional tip-leakage loss mechanisms (eg., Bindon, 1989; Dishart and Moore, 1989). In this case, corresponding profile losses can still be determined from measurements at midspan. However, as will be discussed in Section 6.6, a strong interaction is present between secondary and tip-leakage loss mechanisms. Ignoring this interaction would not be consistent with the flow physics. Nevertheless, to tabulate the results of the conventional approach for comparison, it was necessary to quantify the secondary loss that would have been observed for each non-zero tip clearance configuration in the absence of the tip-leakage flow. According to Dunham's (1970) correlation of secondary loss against inlet boundary layer displacement thickness, the secondary loss would vary by at most ± 15 percent for the boundary layer thickness range observed herein. Thus, the secondary loss measured for zero clearance was used for all clearances to compute the tip-leakage loss based on the traditional non-interactive (additive) approach.

6.4 Losses Inside the Tip Clearance

The flow field inside the tip gap was measured for clearances of 2, 2.8 and 5.5 percent blade chord using a three-hole probe. A detailed discussion of the measurement technique and results for the velocity field were presented by Yaras (1987). In this section only the loss generation will be focused upon.

Figure 6.4.1 shows the total pressure loss coefficient distribution along the blade meanline for a clearance size of 2.8 percent blade chord. Substantial losses exceeding loss coefficient values of -4.0 are observed inside the separation bubble on the blade tip. However, as determined by Yaras (1987), a very small fraction of the tip-gap mass flow rate occurs in this region. In fact, the bubble was found to contain a vortical flow with

Figure 6.4.1 Tip gap total pressure loss coefficient distribution along the blade mean line ($\frac{\tau}{c} = 0.028$) .

quite strong chordwise velocity components, as also observed by Bindon (1987c). Thus, in mass-averaged terms the losses generated inside the gap should turn out to be quite modest. This is confirmed by the chordwise distribution of mass-averaged losses inside the gap (Figure 6.4.2). The figure shows a substantial increase in the mass-averaged losses at quarter chord length from the leading edge. In this region the blade loading reaches its peak, resulting in large tip-leakage driving pressure differences. The resulting high tip-leakage velocities are likely to introduce larger shear stresses in the bubble, and therefore higher entropy generation.

Another loss source in the tip gap is the boundary layer on the tipwall. Due to mechanical limitations, the measurement station closest to the tipwall was about 5 percent gap height away from the tipwall for 2.8 percent clearance size. This prevented the quantification of the tipwall boundary layer. On the other hand, the losses associated with this boundary layer within the gap are likely to be considerably smaller than those associated with the separation bubble on the blade tip. Thus, the error in the quoted results is expected to be small.

While the above discussed measurements can quantify the tip-gap losses up to the blade meanline (camberline), the information of interest is all the loss generated inside the gap until the flow reaches the tip-gap exit. Tip-gap three-hole probe measurements taken across the blade thickness at selective chordwise positions helped to clarify this issue. Velocity vectors extracted from these measurements showed that the separation bubble reattaches some distance before the gap exit. Thus one would expect little loss generation beyond the blade meanline in the tip gap. This is confirmed by Figure 6.4.2, which shows that the losses obtained at the gap outlet agreed well with those measured on the blade mean line.

Another point of interest is the effect of tip clearance size on the losses generated in the tip gap. The results of Yaras (1987) showed an increase in the bubble size relative to the gap height as the gap was opened. In support to this, Bindon's (1987b) flow

Figure 6.4.2 Local mass-averaged losses inside the tip gap for $\frac{\tau}{c} = 0.028$.

visualization study indicated that the separation bubble becomes too small to observe visually for very small clearances. As the clearance size is increased, the bubble (initially) appears to grow faster than the clearance and thus, the mass-averaged losses inside the tip gap rise with increasing clearance size. However, as the tip gap is increased, eventually the bubble will reach its natural size, in which case its relative size will decrease with further increases in gap height. This in turn should bring the mass-averaged losses down. This argument is supported by the mass-averaged loss coefficients in the tip gap shown in Table 6.4.1. The data indicates that the peak loss occurs at somewhat less than 2 percent clearance height, which presumably corresponds to the condition at which the relative separation bubble size reaches its maximum value.

Extensive calibrations by Yaras (1987) showed that the three-hole probe underestimates the total pressure when the flow is not in the plane of the probe. This flow condition is likely to occur in the tip gap when measurements are taken inside the separation bubble. Thus, the mass-averaged total pressure values for the tip gap in Table 6.4.1 are presumably somewhat overestimated. This observation must be kept in mind when the losses in the gap are compared to losses generated in other portions of the tip-leakage flow.

Following the argument in Section 6.2, to enable the comparison of the losses generated inside the gap to losses measured at other regions, the tip gap flow was topped up with the portion of the reference flow rate that enters the passage instead of contributing to the tip-gap flow. The mass-averaged losses corresponding to this reference mass flow are quoted in Table 6.4.1.

It is important to realise that the mass-averaged losses for the gap flow quoted in the third row of the tip-gap section of Table 6.4.1 would represent the losses generated inside the gap only if the flow entering the tip gap were loss free. This, however, is not true since a considerable portion of the inlet tipwall boundary layer is known to enter the tip clearance. This is observed from the leading edge portion of Figure 6.4.1 and was

		Clearance Size τ/c				
		0.0	0.015	0.020	0.028	0.055
Location						
Inlet	C_p^H	0.0067	-0.0787	-0.0355	-0.0455	-0.0196
Tip Gap	m_g/m_{REF}	----	----	0.073	0.083	0.161
	$C_p^H (for m_g)$	----	----	-0.805	-0.461	-0.230
	$C_p^H (for m_{REF})$	----	----	-0.0584	-0.0382	-0.0370
	$C_p^H (gap=0.25in)$	----	----	-0.0496	-0.0268	-0.0321
	$C_{e_s}^H$	----	----	0.164	0.176	0.331
Downstream	q_2/q_1	2.13	1.95	2.04	----	1.82
	$C_p^H (Profile)$	-0.063	-0.050	-0.055	----	-0.042

Table 6.4.1 Quantities extracted from flow field measurements.

supported by smoke injected into the tipwall boundary layer at the cascade inlet. To quantify the net losses generated inside the gap, losses associated with the portion of the inlet tipwall boundary layer entering the gap need to be subtracted from the losses measured inside the gap. Sjolander and Amrud's (1987) flow visualization studies suggested that about a quarter of the inlet boundary layer enters the gap. While this value was quoted as very approximate, in the absence of other data it was used for determining the net tip-gap losses given in Table 6.4.1.

Having quantified the losses generated inside the gap, the next step is to identify the losses generated as the tip-leakage flow expands at the tip-gap exit and merges with the passage flow.

6.5 Tip-Leakage Losses in the Blade Passage

Total pressure contour plots in Plane B for two clearances and zero clearance are given in Figure 6.5.1. Consistent with the vorticity field studied in Section 5.3, the tip vortex is the dominant structure in the outer half of the passage while the passage vortex squeezed between the tip vortex and tipwall is quite small. Compared to the zero clearance configuration, the losses associated with the passage vortex are reduced significantly as the tip gap is opened. This is true even for the small clearance size and is the consequence of a significant portion of the inlet tipwall boundary layer entering the gap instead of contributing to the formation of the passage vortex. In addition, the small passage vortex is located next to the tip vortex, which is likely to have an effect on the mixing losses that would have been produced by the two vortices in the absence of any interaction. In relation to this, Dishart and Moore (1989) observed redirection of the passage vortex secondary velocities upon interaction of the passage vortex with the tip vortex. Bindon's (1987b) flow visualization study in a turbine cascade showed that the leakage flow nearest tipwall does not join the tip vortex but bypasses it to mingle with the passage vortex. Finally, Yamamoto's (1988) tests in a linear turbine cascade indicated significant interaction between the leakage and passage vortices especially

Figure 6.5.1 Plane B total pressure loss coefficient distribution

at large clearances. These observations suggest that the traditional approach, which assumes no interaction between secondary and tip-leakage flows, is not consistent with the physics of the flow.

A more appropriate approach is to consider the "end-loss", which is the loss measured in the trailing edge or any other downstream plane less the inlet and profile losses. This argument will be further justified in the next section where the downstream flow field is studied. In the mean time, the loss generation inside the passage will be evaluated in terms of end-losses.

Mass-averaged losses inside the gap, the end-losses at the trailing edge, and corresponding secondary kinetic energies are given in Figures 6.5.2 and 6.5.3 for two clearances. The kinetic energy at the tip-gap exit is based upon the velocity component normal to the blade chord. There are several approaches that have been used to identify the secondary flow velocities downstream of the blades. Inoue et al. (1986) and Lakshminarayana et al. (1986) employed the relative design flow angle as the reference direction; Dishart and Moore (1989) defined the reference direction at the cascade exit as the metal angle at the blade trailing edge; Dring (1981) used the measured passage-averaged direction and Gregory-Smith (1982) utilized the local exit flow angle at midspan. Despite the differences in definition, resultant secondary velocities should exhibit little variation with the type of approach selected. Herein Gregory-Smith's approach is used.

From Figure 6.5.2 it is observed that for the smaller clearance the end-loss generated in the passage is only about 20 percent of the losses introduced inside the tip gap. On the other hand, for the larger clearance the passage end-loss is as much as four times the tip-gap losses. This significant difference between the two clearances is due to both reduction of tip-gap losses and increase of passage end-losses with increasing clearance size. The former was discussed in the previous section. The growth of passage end-losses with increasing clearance can be explained by the amount of kinetic energy

Figure 6.5.2 Mass-averaged losses and secondary kinetic energies ($\frac{\tau}{c}=0.020$) .

Figure 6.5.3 Mass-averaged losses and secondary kinetic energies ($\frac{\tau}{c}=0.055$) .

available at the gap exit. As illustrated on the bar chart, mass-averaged kinetic energy normal to the blade chord increases rapidly with increasing clearance size. This results in increased mixing losses as the leakage flow ejects into the passage. It also will raise the losses associated with the tipwall boundary layer inside the passage. Finally, it may increase the losses in the boundary layer on the blade suction surface beneath.

Intuitively, the portion of the secondary kinetic energy at the tip-gap exit that is not used for end-loss generation in the passage might be expected to appear as secondary kinetic energy at the trailing edge plane. However, for either clearance size on the bar chart, when the trailing edge plane secondary kinetic energy is combined with the end-losses generated in the passage, the total kinetic energy at the tip-gap exit normal to blade chord is not accounted for. This is probably due to the orderly shear interaction between the gap and the passage flow, instead of a sudden expansion process at the gap exit, which allows a significant portion of the gap kinetic energy to be recovered, presumably as static pressure. However, as will be evident from the discussions in the next section, this recovery is temporary.

6.6 Losses Downstream of the Blades and Fully Mixed-out Losses

To evaluate the end-loss production downstream, the total pressure data in two closely spaced measurement planes about one axial chord length downstream of the trailing edge plane were used (Planes C_1 and C_2). Once the mass-averaged overall loss coefficient was obtained in each plane, the mass-averaged end-loss was calculated in the same manner as for the trailing edge plane. The results for the two clearances are given in Figures 6.6.1 and 6.6.2 together with the tip-gap and passage end-losses. As before, the mass-averaged secondary kinetic energy in each measurement plane is shown by an additional bar section on top of that for the end-loss.

For both clearances the end-losses appear to double over the one axial chord length between planes B and C_1 . It is also observed that about half of the secondary

Figure 6.6.1 Mass-averaged losses and secondary kinetic energies ($\frac{\tau}{c}=0.020$) .

Figure 6.6.2 Mass-averaged losses and secondary kinetic energies ($\frac{\tau}{c}=0.055$) .

kinetic energy available at the trailing edge plane is lost in the process. To determine the additional diffusion losses beyond plane C_2 , mixing loss calculations were performed by assuming that the flow mixes out at constant area to uniform static pressure and streamwise velocity and with no remaining secondary velocity components. Thus, the mixing losses connected with all components of velocity are included. The mixing calculations involve application of conservation of mass and momentum relations. The choice of control volume that covers the complete passage pitch provided periodic side boundaries. Thus the mass and momentum flux through these side boundaries could be ignored. The resultant expressions for mixed-out velocity, static pressure coefficient and total pressure loss coefficient are as follows:

$$\begin{aligned} \frac{V_{mixed}}{V_{CL}} &= \frac{1}{A_{mixed}} \int_{A_{mixed}} \left(\frac{V_2}{V_{CL}} dA \right) \\ C_{P_{mixed}} &= \frac{1}{A_{mixed}} \left\{ \int_{A_{mixed}} C_P dA - 2 \left(\frac{V_{mixed}}{V_{CL}} \right)^2 A_{mixed} + 2 \int_{A_{mixed}} \left(\frac{V_{mixed}}{V_{CL}} \right)^2 dA \right\} \\ C_{P_{total}} &= C_{P_{mixed}} + \left(\frac{V_{mixed}}{V_{CL}} \right)^2 \end{aligned} \quad 6.6.1$$

where,

$$dA = (dy \cos(\alpha_2)) dz \quad 6.6.2$$

and A_{mixed} is the exit plane of the control volume measured perpendicular to the axial (x) direction. If the flow were to enter a downstream blade row in a turbine, less loss would be experienced from mixing since the mixing process would take place in an accelerating flow. Thus, the mixed-out loss calculated as described above is only chosen for convenience and is not necessarily the loss that would occur in a real turbomachine.

Resultant fully mixed-out losses are shown in Figures 6.6.3 and 6.6.4. Comparison of the end-loss in planes B, C_2 and the fully mixed-out plane shows that majority of loss generation is completed by the time the flow reaches about an axial chord

Figure 6.6.3 Mass-averaged losses and secondary kinetic energies ($\frac{\tau}{c}=0.020$) .

Figure 6.6.4 Mass-averaged losses and secondary kinetic energies ($\frac{\tau}{c}=0.055$) .

length downstream of the trailing edge plane. In relation to this observation, Lakshminarayana et al. (1987) found that the leakage flow measured in a compressor rotor mixes completely within about one to two chord lengths downstream the rotor blades.

From Figures 6.6.3 and 6.6.4 the sum of end-loss and secondary kinetic energy in Plane C_2 is seen to be larger than that in Plane B. In this context Dishart and Moore (1989) found that the secondary kinetic energy measured 40 percent axial chord length downstream of turbine blades could not account for the mixing losses downstream of this measurement plane. The authors explained this observation by the possible dissipation of primary kinetic energy in addition to secondary kinetic energy. However, Figures 6.6.3 and 6.6.4 suggest that the energy loss unaccounted for between planes B and C_2 is too large to be caused by the mixing-out of axial non-uniformity in the flow. To confirm this, the fully mixed-out losses were calculated based on Plane B measurements and compared to those calculated from plane C_2 as shown in Figures 6.6.5 and 6.6.6. For both clearances the mixed-out losses are larger when calculated from the plane C_2 data. It is important to recognize that this difference is very close to the discrepancy between the sum of end-loss and secondary kinetic energy in planes B and C_2 . This observation and the fact that mixing calculations take into account losses connected with all components of velocity suggest the presence of an additional loss mechanism between measurement planes B and C_2 , other than conventional diffusion of either axial or secondary velocities. One possible loss source is shear on the tipwall, which was not considered in the momentum relations for the mixing calculations. In support to this argument, Mobarak et al.'s (1988) experimental study of secondary flows in a turbine cascade suggested that velocity gradients on the tipwall amplified by secondary flows appear to be a possible loss generation mechanism downstream of the blades. However, the extra losses are still too large to be accounted for by this mechanism. Inaccuracies in the measurement of plane B due to small blade wakes and tip-vortex size can be part of the explanation. On the other hand, the possibility of another loss mechanism in effect downstream of the blade row cannot be ruled out.

Figure 6.6.5 Mass-averaged losses and secondary kinetic energies ($\frac{\tau}{c}=0.020$) .

Figure 6.6.6 Mass-averaged losses and secondary kinetic energies ($\frac{\tau}{c}=0.055$) .

From Figures 6.6.5 and 6.6.6 it is seen that the kinetic energy normal to the blade chord at the tip-gap exit is quite close to the non-gap end-losses in Plane C_2 (end-loss evaluated in C_2 less the losses generated inside the tip gap). Since in the present study the passage vortex, which is one of the mechanisms contributing to the non-gap end-losses, was quite small compared to the tip vortex, the above noted observation suggests that the losses associated with the tip vortex seem to account for the kinetic energy normal to the chord line at the tip-gap exit. A similar result was observed by Dishart and Moore (1989) from measurements 40 percent axial chord downstream of turbine blades. This agreement could to some degree be fortuitous since, as was noted in the previous section, some of the gap kinetic energy is in fact recovered and part of the final loss is apparently due to wall shear stress. On the other hand, the greater the recovery of the gap energy, the stronger is the tip vortex and the greater the loss production on the end-wall. The recovery could therefore be regarded as temporary, leading to loss mechanisms which assure that the energy is ultimately lost. This observation provides a basis for predicting the end-losses, as will be described in a later section.

In the previous section the end-loss was suggested to be the appropriate quantity to study as opposed to tip-leakage loss. This is supported by the downstream measurements. Figure 6.6.7 shows the variation of net mass-averaged losses (overall losses minus the inlet losses) with downstream distance for several clearances, including zero clearance. It is noticed that for small clearances the losses are greater than for zero clearance by roughly the same amount over the first axial chord length. Thus, if losses are subdivided in the conventional way (i.e., subtracting profile and zero clearance secondary loss from the net total loss to obtain tip-leakage loss) the secondary losses alone seem responsible for the growth of losses downstream of the blade row. Considering the size of the tip vortex with respect to the passage vortex, this is difficult to accept physically. Thus, it seems more appropriate to study end-losses instead of tip-leakage and secondary losses individually, confirming the conclusion drawn in the previous section.

Figure 6.6.7 Variation of net total pressure loss with downstream distance.

The mass-averaged losses evaluated in different sections of the tip-leakage flow are summarized for reference in Table 6.6.1.

6.7 Relative Contribution of Loss Mechanisms

Having evaluated all of the end-loss mechanisms, it is appropriate to do a final overall comparison of the relative magnitudes of the different mechanisms. This is particularly important for the designer since it will provide valuable guidance for design modifications leading to effective loss reduction. The comparison is given in Figure 6.7.1. Since the results discussed in the previous section showed that non-gap tip-leakage losses seem to scale on the mass-averaged kinetic energy normal to the chordline at the gap exit, it was found appropriate to use the gap mass flow rate non-dimensionalized by the reference flow rate as the independent variable in the figure.

The trends of the individual curves have already been discussed in previous sections. As far as relative magnitudes of the two losses is concerned, it is evident that the losses due to mixing inside the passage and downstream of the blade row are considerably larger than the tip-gap losses. This is in agreement with Dishart and Moore's (1989) work but contradicts Bindon's (1989) work that suggested comparable magnitudes for the two loss mechanisms. Bindon was confronted with accuracy problems regarding the measurements inside the tip gap. In addition, he acquired downstream flow data in the trailing edge plane only and did not do a mixing calculation. Thus it seems that a significant reduction in the overall end-losses can be achieved by reducing the mass flow rate through the gap, even if the design modification to achieve this goal should result in somewhat increased tip-gap losses. More discussion and experimental studies dealing with design modifications to reduce end-losses will be presented in Chapter 9.

		Clearance Size τ/c				
		0.0	0.015	0.020	0.028	0.055
Location						
Inlet						
	C_{p_e}''	0.0067	-0.0787	-0.0355	-0.0455	-0.0196
Tip Gap						
	m_f/m_{REF}	----	----	0.073	0.083	0.161
	C_{p_e}'' (for m_f)	----	----	-0.805	-0.461	-0.230
	C_{p_e}'' (for m_{REF})	----	----	-0.0584	-0.0382	-0.0370
	C_{p_e}'' (gap-0.25in)	----	----	-0.0496	-0.0268	-0.0321
	C_{p_e}''	----	----	0.164	0.176	0.331
Downstream						
	q_2/q_1	2.13	1.95	2.04	----	1.82
	C_{p_e}'' (Profile)	-0.063	-0.050	-0.055	----	-0.042
	C_{p_e}'' (Plane B)	-0.0365	-0.174	-0.158	----	-0.202
	C_{p_e}'' (Plane C ₁)	-0.137	-0.251	-0.261	----	-0.392
	C_{p_e}'' (Plane C ₂)	-0.137	-0.256	-0.262	----	-0.410
	C_{p_e}'' (Plane B)	0.0145	0.0301	0.0389	----	0.0889
	C_{p_e}'' (Plane C ₁)	0.0053	0.0116	0.0127	----	0.0485
	C_{p_e}'' (Plane C ₂)	0.0055	0.0110	0.0118	----	0.0471
	C_{p_e}'' (mixed B)	-0.073	-0.241	-0.241	----	-0.364
	C_{p_e}'' (mixed C ₁)	-0.152	-0.290	-0.307	----	-0.490
	C_{p_e}'' (mixed C ₂)	-0.150	-0.294	-0.307	----	-0.507

Table 6.6.1 Quantities extracted from flow field measurements.

Figure 6.7.1 Variation of tip gap losses and non-gap end-losses with tip leakage flow rate.

6.8 Prediction of Tip-Leakage Losses

Recent developments in computer technology have initiated significant amounts of research toward the numerical solution of the three-dimensional Navier-Stokes equations for turbomachinery flows (eg., Hah, 1986; Moore and Moore, 1985; Dawes, 1988). This approach could indeed provide detailed information for the loss distribution in the flow field of interest. However, the highly three dimensional and complex nature of the flows encountered in turbomachines require very fine spatial resolution to achieve acceptable accuracy. This in turn demands very long computation times. For instance, Dawes's computational scheme, which solves the three-dimensional Reynolds-averaged Navier-Stokes equations by a time-marching finite-volume algorithm, was used to predict the flow in the present linear turbine cascade. Depending on the chosen spatial resolution, the prediction of the passage flow between midspan and the tipwall required 45 to 90 CPU hours on an Apollo Domain 4000 computer equipped with a 68881 floating point processor. Therefore, such methods are still too time consuming to be used for routine design. Instead, quick and simple procedures are still the most desirable tools for the designer.

The development of semi-empirical schemes applicable to a wide range of designs require a very good understanding of the parameters affecting the flow of interest. There is an extensive collection of semi-empirical prediction schemes in the literature. A significant portion of these schemes are simple correlations based on geometrical parameters only. There are some models based on assumptions regarding the tip-leakage flow physics. However, up to now lack of a full understanding of the tip-leakage flow has prevented the evaluation of the accuracy of these assumptions.

With the present work, a much improved understanding of the tip-leakage flow has been developed. Thus, it seems appropriate to review the various tip-leakage models in an attempt to evaluate their accuracy both in terms of numerical results and simulation of the flow physics.

The simplest scheme suggested for tip-leakage loss prediction takes the form,

$$\Delta\eta = -C\left(\frac{\tau}{c}\right) \quad 6.8.1$$

where C is a constant representing the sensitivity of stage efficiency to changes in clearance size. However, the compressor data summarized by Senoo and Ishida (1986) in Figure 6.8.1 clearly suggest that the sensitivity of stage efficiency to changes in clearance size varies substantially from one turbomachine to another. In the light of the recent experimental data, suggesting complex loss mechanisms associated with tip-leakage flow, it is not surprising to see that such a simple correlation provides poor accuracy.

Moyle (1988) stated that if the efficiency variation with clearance size is analyzed at constant shaft power instead of constant flow coefficient, the above noted sensitivity is reduced. Based on the data available in the literature, Moyle suggested the following expression.

$$\Delta\eta = 0.214\left(\frac{\tau}{h}\right)^{0.227} - 0.005 \quad 6.8.2$$

However, the approach remains an empirical correlation. Therefore, it is optimistic to expect the scheme to exhibit good performance in predicting the tip-leakage losses accurately for today's turbomachine stages, whose geometric and aerodynamic parameters vary over a wide range.

A related approach has been to correlate the efficiency drop due to tip leakage to the zero clearance stage efficiency through several geometric parameters. This was first suggested by Amann et al. (1963). Their expression, applicable to turbine blade rows, is as follows:

Figure 6.8.1 Efficiency variation with tip clearance for a range of compressors (reproduced from Senoo and Ishida, 1986)

$$\Delta\eta = \frac{-r_t}{r_m \cos\alpha_2} \eta_o \frac{\tau}{h} \quad 6.8.3$$

where r_m and r_t are mean and tip annulus radii respectively. The scheme is restricted to cases where whirl at the turbine blade row exit is small and the tip clearance section of the flow is essentially undeflected. Rogo (1968) stated that the prediction accuracy for this correlation is reduced with increasing clearance size. Moreover, Kacker and Okapuu (1982) noted that the results from this correlation had to be reduced by seven percent to obtain better agreement with more recent experimental data. Clearly the scheme is far from being universal mainly because it does not attempt to model the physics of the flow in any respect.

Having reviewed the prediction schemes based on simple correlations, two approaches that are somewhat more physical are examined.

6.8.1 Models Based on Momentum Considerations

These models predict the drag on the blade by the tip-leakage flow by analogy with the induced drag for a finite wing. To convert the resultant momentum deficiency into total pressure loss it is necessary to assume that the loss is distributed uniformly over the passage mass flow. In other words, the flow is assumed to have mixed-out completely. Furthermore, it is assumed that the mixing is completed before the flow enters the subsequent blade row. These assumptions are also used in the other models described in later sections. Experimental results in Section 6.6 suggested that, at least for the present cascade configuration, the approximate mixed-out flow condition is reached about one axial chord length downstream the trailing edge. Thus, in a strict sense the models discussed herein are applicable to turbomachines with spacing between adjacent blade rows of this order of magnitude.

The work of Betz (1926) seems to have initiated this approach. In this study the drag induced by tip leakage was evaluated by considering the shed vortices at the blade trailing edge produced by the gradient of the bound circulation distribution across the blade span. However, experiments have shown that for the case of finite tip clearance almost all the vorticity observed downstream of the blade sheds from the blade tip instead of the trailing edge, resulting in an immediate roll-up of the vortex sheet.

A commonly used tip-leakage loss model for axial turbines is that of Ainley and Mathieson (1951). It is an adaption of the theory developed by Carter (1948) for secondary flows. Carter determined the incidence induced by secondary flow by adapting the conventional theory for calculating the induced incidence on finite wings with trailing vortices. The resultant expression given by Ainley and Mathieson is,

$$Y_{tip} = \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} \frac{1}{S/c} 0.5 \left(\frac{\tau}{S} \right) \frac{C_L^2}{(h/c)} \quad 6.8.1.1$$

In Figure 6.8.1.1 the predictions are compared to the present end-loss data. Since the model predicts the tip-leakage loss only, the underestimation of end-loss data which includes the secondary losses is reasonable. When the secondary loss predicted with a scheme proposed by the same authors is added to their tip-leakage loss, the agreement with the data is good. It should be noted, however, that this agreement is probably fortuitous since the simple decomposition of the losses, which is assumed by Ainley and Mathieson's approach, is not supported by the experimental results. Another problem associated with the model is that it assumes the tip-vortex circulation to be equal to bound circulation. The results presented in Section 5.4 showed that the circulation in the tip vortex was substantially less than the bound circulation.

Figure 6.8.1.1 Comparison of tip leakage loss models to cascade data (D & C: Dunham and Came, 1970; A & M: Ainley and Mathieson, 1951; L & H: Lakshminarayana and Horlock, 1965).

Dunham and Came (1970) modified Ainley and Mathieson's model in the light of some later cascade data, which suggested non-linear variation of tip-leakage losses with clearance size. The resultant expression is as follows:

$$Y_{tip} = \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} \frac{1}{(S/c)^2} 0.47 \left(\frac{\tau}{c} \right)^{0.78} \frac{C_L^2}{(h/c)} \quad 6.8.1.2$$

However, as seen from Figure 6.8.1.1, the model substantially overestimates the losses. This was also observed by Kacker and Okapuu (1982).

In their scheme for predicting tip-leakage losses in compressors, Lakshminarayana and Horlock (1965) followed a procedure somewhat similar to that of Ainley and Mathieson. They simulated the tip vortex with a Rankine vortex and determined the drag induced on the blade by this vortex. It is a more realistic model compared to Ainley and Mathieson's in the sense that the location and alignment of the vortex relative to the blade, its initiation point and its strength are modelled specifically. The expression suggested by the authors is:

$$Y_{tip} = \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} \frac{1}{(S/c)} \frac{C_L^2 (1-K)}{8\pi (h/c)} (A-B) \quad 6.8.1.3$$

where

$$A = \ln \left(\frac{e^{\frac{2\pi h}{S}} - 1}{(1 + \coth \frac{2\pi \tau}{S})(e^{\frac{2\pi h}{S}} - 1) + 2} \right) \quad 6.8.1.4$$

$$B = \ln \left(\frac{e^{\frac{\pi d}{S}} - 1}{(1 + \coth \frac{2\pi \tau}{S})(e^{\frac{\pi d}{S}} - 1) + 2} \right) \quad 6.8.1.5$$

and $(1-K)$ is the fraction of the bound circulation shed in the tip vortex. The model was found to be sensitive to the vortex diameter. Figure 6.8.1.1 shows that the trend in the losses predicted by this model is not in good agreement with the experimental results. Quantitative comparison requires that a secondary loss component be added to the prediction. If, for example, the Ainley and Mathieson secondary loss model were used for this purpose, the predicted end-loss would clearly be substantially higher than the experimental results.

In a modified version of this model, Lakshminarayana (1970) took into account the losses associated with the spanwise flow in the blade surface boundary layer near the tip. However, this will increase the predicted tip-leakage loss which would make the disagreement with experimental results in Figure 6.8.1.1 even worse.

It should be noted that the models which predict the tip-leakage loss through induced drag calculations are inviscid schemes. Results discussed in Section 6.6 suggested that there is additional viscous loss production on the tipwall due to the high shear stresses that occur beneath the tip vortex. This implies that an extra loss should be added to the purely inviscid loss component derived from the induced drag analysis. However, Figure 6.8.1.1 shows that the losses are over-predicted without this additional viscous loss being included. Thus, addition of a viscous loss component, as suggested by the physics, would only make the agreement poorer.

The induced drag models discussed so far were introduced as tip-leakage loss rather than end-loss prediction schemes. Meldahl (1941) appears to be the first to model the secondary and tip-leakage losses for turbines in a combined fashion. Betz's model formed the basis of Meldahl's scheme. The author identified two terms in the resultant end loss expression: one is dependent on clearance size, hence the author referred to it as the tip-leakage loss. The second term, independent of clearance size, was identified as a "fixed end-loss." The expression is as follows:

$$Y_{end} = 0.25 \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} \left(\frac{c}{S} \right) \left(\frac{\tau}{S \sin \alpha_2} + \frac{\ln 2}{\pi} \right) \frac{C_L^2}{(h/c)} \quad 6.8.1.6$$

Vavra (1960) gave a physical interpretation to this fixed end-loss and referred to it as the loss introduced by the secondary flow. The author found it necessary to replace Meldahl's fixed end-loss term by Carter's (1948) secondary loss expression since the former implied a secondary loss larger than that for zero clearance:

$$Y_{end} = 0.25 \frac{\cos \alpha_2}{\cos^3 \alpha_m} \left(\frac{\tau}{h} \right) \left(\frac{c}{S} \right)^2 C_L^2 + 0.04 C_L^2 \left(\frac{S}{h} \right) \left(\frac{c}{S} \right)^2 \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} \quad 6.8.1.7$$

While this approach regards secondary and tip-leakage losses together, the secondary loss is not reduced by the presence of tip clearance. Hence, the formulation does not reflect the interaction between the two loss mechanisms. The agreement of the predictions based on Eqn. 6.8.1.7 with the recent experimental data was found to be poor.

6.8.2 Models Based on Energy Considerations

Another approach, quite different from the ones discussed so far, was proposed by Rains (1954). The author suggested that the tip-leakage flow rolls up into a vortex whose energy cannot be recovered. In other words, the leakage kinetic energy in the gap normal to the blade chord is lost eventually and it accounts for all of the tip-leakage loss. As was discussed in Section 6.6, this is supported by experimental results. Rains' analysis was somewhat involved. The approach was simplified by Vavra (1960) who proposed the following expression for the tip-leakage loss coefficient:

$$Y_{tip} = \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} \frac{1}{S/c} \frac{4\sqrt{2}}{5} k \Omega^3 \frac{\tau}{h} C_L^{1.5} \quad 6.8.2.1$$

where, Ω is a factor that takes account of the flow resistance in the gap and k is the contraction factor for the flow through the gap. Hesselgreaves (1969) and Lewis and Yeung (1977) followed Vavra's approach except they set Ω and k equal to 1. Interestingly, Eqn. 6.8.2.1 suggests a weaker dependence of tip-leakage losses on blade loading than the previously discussed momentum approach.

As shown in Figure 6.8.2.1, Vavra's model underestimates the experimental data substantially and the difference is too large to be accounted for by secondary losses, especially for large clearances. The modifications by Hesselgreaves, and Lewis and Yeung made the models applicable to rotating turbomachinery only since the final expressions were in terms of efficiency rather than loss coefficient, Y . Thus, their models' accuracy could not be assessed based on the recent cascade data. According to Schmidt et al. (1987b), the suggested modifications do not improve the accuracy. On the other hand, the experimental results justify the assumption about the flow physics upon which Vavra's model is based. Hence, recognizing the potential of the underlying idea, further examination of Vavra's model is undertaken in the next section.

6.8.3 An Improved End-Loss Model

The kinetic energy associated with the tip-leakage velocity component normal to the blade chord can be written as:

$$\Delta E = \int_{m_t} \frac{V_n^2}{2} dm_t \quad 6.8.3.1$$

As was summarized in Chapter 1, for determining the velocity component normal to the blade chord, Yaras et al. (1989) showed that the application of Bernoulli's equation based on the undisturbed (2-D) blade pressures gives satisfactory results:

$$V_n = \sqrt{\frac{2(P_{PS} - P_{SS})}{\rho}} \quad 6.8.3.2$$

Figure 6.8.2.1 Comparison of tip leakage loss models to cascade data (D & C: Dunham and Came, 1970; A & M: Ainley and Mathieson, 1951; L & H: Lakshminarayana and Horlock, 1965).

This would not be applicable to very small clearance sizes where the flow in the gap is no longer predominantly inviscid. A criterion for determining the size of clearances where viscous forces are important is given by Rains (1954). For very large clearances a strong interaction between streamwise momentum and leak momentum takes place (Booth et al., 1982). This results in fluid particles being swept downstream before they can migrate from the pressure to the suction surface. Tipwall static pressure data, presented in Section 5.4, show that the pressure gradient driving the tip leakage close to the tipwall is still comparable to the undisturbed loading for a clearance height of 5 percent of blade chord. Thus, only for clearances considerably larger than 5 percent of blade chord would Eqn. 6.8.3.2 not be valid. The range of clearances within which Eqn. 6.8.3.2 is applicable should be kept in mind in the application of any loss model based upon this expression. In Vavra's (1960) loss model the velocity obtained from Eqn. 6.8.3.2 was multiplied by a flow resistance factor, Ω . This implies a loss of total pressure inside the gap which is not supported by experimental results. This is likely one factor responsible for the poor accuracy of Vavra's model.

The mass flow rate term in Eqn. 6.8.3.1 can be expressed as,

$$dm_g = C_D \rho V_n \tau dx' \quad 6.8.3.3$$

where C_D is a discharge coefficient. Vavra used a constant contraction factor value of 0.5. However, the measurements of Yaras et al. (1989) demonstrated that the magnitude of the discharge coefficient is quite different and furthermore varies with the aspect ratio of the tip gap. The variation of the discharge coefficient with the gap aspect ratio is given in Fig. 6.8.3.1. The values in the figure are somewhat larger than the ones quoted by Yaras et al. (1989). In the calculation of the discharge coefficient, Yaras et al. (1989) considered the leakage mass flow rate normal to the blade mean line. However, since the leakage kinetic energy discussed herein is based on the tip-gap velocity component normal to the blade chord, it is more appropriate to use the same convention for the tip-leakage mass flow rate calculations. While the discharge coefficient shows some

Figure 6.8.3.1 Gap discharge coefficient variation with tip clearance size.

chordwise variation, this may be different depending on the blade loading distribution. Thus, it is probably more reasonable to use an average value of around 0.75. Dishart and Moore (1989) found a value of 0.7 based on their measurements while Booth et al. (1982) suggested a value about 0.8.

By substituting equations 6.8.3.2 and 6.8.3.3 into Eqn. 6.8.3.1, the gap exit kinetic energy becomes,

$$\Delta E = \sqrt{\frac{2}{\rho}} C_D \tau \int_0^c (P_{PS} - P_{SS})^{1.5} dx' \quad 6.8.3.4$$

To perform the integration, a chordwise distribution for the blade pressures has to be assumed. Vavra assumed a triangular distribution for a front-loaded blade. As seen from Figure 6.8.3.2, this is an acceptable approximation for the present cascade blades as well. Hence, the analysis will be continued with this loading distribution. The effect of loading distribution will be discussed later. Thus, for the triangular loading distribution in Figure 6.8.3.2,

$$P_{PS} - P_{SS} = \Delta P_{\max} \left(1 - \frac{x'}{c}\right) \quad 6.8.3.5$$

where $\Delta P_{\max}$ is adjusted to give the same total loading as the actual pressure distribution. Substituting this into the expression for kinetic energy,

$$\Delta E = \sqrt{\frac{2}{\rho}} C_D \tau (\Delta P_{\max})^{1.5} \int_0^c \left(1 - \frac{x'}{c}\right)^{1.5} dx' \quad 6.8.3.6$$

If the lift force is assumed to be roughly normal to the blade chord, from the definition of lift coefficient, the triangular distribution gives

Figure 6.8.3.2 Measured blade midspan pressure distribution and assumed triangular loading.

$$C_L = \frac{\Delta P_{max} \frac{c}{2}}{\frac{1}{2} \rho V_m^2 c} \quad 6.8.3.7$$

Hence, substituting for ΔP_{max} in Eqn. 6.8.3.6 and performing the integration will result in the following expression:

$$\Delta E = \frac{2\sqrt{2}}{5} \rho c C_D \tau V_m^3 C_L^{1.5} \quad 6.8.3.8$$

The loss coefficient Y , for a cascade flow where absolute and relative frame of references are the same, is defined as follows,

$$Y = \frac{\Delta E}{E} = \frac{\Delta E}{\dot{m} \frac{V_2^2}{2}} \quad 6.8.3.9$$

where E is the passage kinetic energy based on outlet conditions. E can also be written in terms of the mean velocity through the passage:

$$E = \frac{\rho h s}{2} \frac{V_m^3 \cos^3 \alpha_m}{\cos^2 \alpha_2} \quad 6.8.3.10$$

Thus, the tip clearance loss coefficient becomes,

$$Y_{tip} = \frac{4\sqrt{2}}{5} \frac{c\tau}{hs} C_D \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} C_L^{1.5} \quad 6.8.3.11$$

In the development of this expression a front-loaded blade with a triangular loading distribution was assumed. For the case of a rear-loaded blade with triangular load distribution the same expression applies. If a uniform chordwise loading is used, the

constant $\frac{4\sqrt{2}}{5}$ in the above equation becomes 1. Thus, in generalized form Eqn. 6.8.3.11 can be written as,

$$Y_{tip} = K_E \frac{c\tau}{hs} C_D \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} C_L^{1.5} \quad 6.8.3.12$$

where K_E is a blade loading parameter and is equal to 1.132 for a front- or rear-loaded blade, and 1.0 for a uniform-loaded blade.

The difference between the two cases is only about 10 percent, indicating relatively small sensitivity of the tip-leakage loss to a particular chordwise blade loading distribution.

The loss coefficient given in Eqn 6.8.3.12 accounts for the losses associated with the merging of the tip leakage with the passage flow and the subsequent diffusion of the tip vortex downstream of the blade row. The remaining end-loss components to be accounted for are the losses introduced inside the gap and the secondary losses. Thus, the total end-loss for the outer half of the blade passage can be written as,

$$Y_{end} = Y_{gap} + k_s Y_{s,0} + Y_{tip} \quad 6.8.3.13$$

where $Y_{s,0}$ is the secondary loss for zero clearance and k_s is the fraction of secondary loss that remains identifiably associated with the passage vortex when the tip gap is opened.

In Section 6.7 it was concluded that the losses generated inside the gap are smaller than the non-gap end-losses. However, for completeness it is desirable to have an

expression for Y_{gap} as well. From Figure 6.7.1, it was observed that for practical clearances the tip-gap mass-averaged loss coefficient based on the reference mass flow rate is fairly constant. On the other hand, the tip-gap loss coefficient based on the gap flow rate, Y'_{gap} , appeared to vary roughly inversely with the tip-gap flow rate (see Table 6.6.1). Thus, one can write,

$$Y_{gap} = Y'_{gap} \frac{\dot{m}_g}{\dot{m}_p} \quad 6.8.3.14$$

Calculating the tip-gap mass flow rate from Eqn 6.8.3.3 and writing the reference passage flow rate as $\rho V_m \cos \alpha_m h S$, the above relation can be expressed as,

$$Y_{gap} = Y'_{gap} K_G \frac{\frac{c}{S} C_D \sqrt{C_L}}{\cos \alpha_m} \left(\frac{\tau}{h} \right) \quad 6.8.3.15$$

where $K_G = 0.943$ for front- and rear-loaded blades and
 $= 1.0$ for uniform-loaded blades.

For clearance sizes from 2 to 5 percent of blade chord, the data suggest that the variation of Y'_{gap} can be approximated by,

$$Y'_{gap} = -8.89 \left(\frac{\tau}{c} \right) + 0.55 \quad 6.8.3.16$$

which is obtained by a first order least-square fit to the data. Thus, equation 6.8.3.15 becomes,

$$Y_{gap} = \left(-8.89 \left(\frac{\tau}{c} \right) + 0.55 \right) K_G \frac{C_D \sqrt{C_L}}{\cos \alpha_m} \frac{\tau c}{h S} \quad 6.8.3.17$$

This expression indicates that, as for Y_{tip} , Y_{gap} increases with blade loading, though with considerably less sensitivity. It should be pointed out that there is relatively large uncertainty in the experimental values of Y'_{gap} (Table 6.6.1) since a rough assumption had to be made regarding the amount of inlet tipwall boundary layer fluid entering the tip gap.

The final end-loss component that remains to be modeled is the secondary losses generated by the inlet tipwall boundary layer that enters the passage and forms the passage vortex. As was noted earlier, in Equation 6.8.3.13, $Y_{s,0}$ is the secondary loss for zero clearance and k_s is the fraction of secondary loss that remains identifiably associated with the passage vortex when the tip gap is opened. $Y_{s,0}$ can be determined by a conventional model, such as the one proposed by Ainley and Mathieson (1951). The relatively difficult part is to quantify the factor k_s . One parameter that may likely affect this factor is the inlet boundary layer thickness relative to the gap height. However, the small range of inlet endwall boundary layers studied in the recent experiments did not provide a large enough data base to draw reliable conclusions. Yamamoto's (1989) experiments suggested that the interaction between the leakage and passage vortices is dependent on the incidence. Clearly more research is necessary in this area. Fortunately, due to the relatively small flow turning of about 45 degrees in the present cascade, the zero clearance secondary loss was small relative to the overall end-loss (Table 6.6.1). Thus, for the present configuration the accuracy of the end-loss model given in equation 6.8.3.13 can still be assessed accurately without considering the secondary loss term, $k_s Y_{s,0}$. The comparison of the predicted end-loss to the experimental data is given in Figure 6.8.3.3. It is observed that the improved model provides considerably more accurate predictions than other schemes. While Ainley and Mathieson's model appears to render comparable accuracy, it is inconsistent with the flow physics in several ways, as was previously discussed.

From Figure 6.8.3.3 it is seen the model was not applied to clearances smaller than about one percent blade chord. For small clearances the viscous forces in the tip gap are

Figure 6.8.3.3

Comparison of tip leakage loss models to cascade data (D & C: Dunham and Came, 1970; A & M: Ainley and Matheson, 1951; L & H: Lakshminarayana and Horlock, 1965).

bound to become increasingly important. In that circumstance the assumptions upon which the model is based will no longer be valid. As an approximation, one could interpolate between the final value of the present model for Y_{end} at, say, $\frac{\tau}{c}=0.01$ and at $\frac{\tau}{c}=0$ where Y_{end} is just the secondary loss. It should be noted, however, that it is possible to have significant nonlinearity within this region. This may be especially true for compressors: due to the diffusive nature of the flow in compressors, for zero tip clearance the low energy fluid build-up in the blade suction side - tipwall corner results in significant losses. For small clearances the opposition of the tip-leakage flow to the secondary flow may prevent this build-up, thus reducing the end-losses. The possible existence of an optimum clearance size in compressors has been explored in several studies (eg., Spencer, 1956; Storer, 1989).

A final noteworthy comment on the present model and on all the other schemes discussed in previous sections is with regard to the temperature drop experienced in actual turbines. As Denton (1987) pointed out, for large stage temperature drops there will be additional entropy generation due to the temperature difference between the hot leakage flow and the relatively cool main flow. None of the prediction schemes have taken this into account. Hence, they may underestimate the actual tip-leakage losses for turbine stages where the temperature drop across the stage is substantial.

7. OPERATING CONDITIONS FOR EXPERIMENTS WITH MOVING TIPWALL

As was noted in Section 3.3, the maximum attainable linear speed for the moving belt is 18.7m/sec. Thus, to simulate the engine equivalent ratio of wall speed to inlet axial speed (2.164), it was necessary to keep the inlet velocity at 8.6 m/sec. This velocity corresponds to a Reynolds number of 1.4×10^5 as opposed to the 4.3×10^5 used for the stationary tipwall experiments. For this low Reynolds number a separation bubble formed on the blade suction surface close to the leading edge. A trip wire of 0.37 mm diameter placed between suction side pressure taps #2 and #3 (see Appendix B for tap locations) was adequate to remove the bubble. The reduced Reynolds number is expected to increase the separation bubble size on the blade tip at the chordwise position where the blade loading is a maximum (Yaras, 1987). This will be evident from the discharge coefficient values presented in Section 8.3. Finally, for zero belt speed the downstream flow field may not be the same as that observed with plexiglass tipwall, due to a tipwall boundary layer somewhat altered by the small opening at the belt leading edge (Figure 7.1) and relatively low Reynolds number. If the discrepancy is substantial, it will be difficult to relate the relative wall motion experiments to the previously discussed stationary tipwall results. To clarify this issue, the streamwise vorticity distribution in Plane C_1 , for 2.4 percent clearance size, 4.3×10^5 Reynolds number and plexiglass window tipwall, is compared in Figure 7.2 to the distribution for the same clearance size, a Reynolds number of 1.4×10^5 and zero belt speed. The tip vortex structures appear to be very similar, eliminating any concerns regarding significant differences in the downstream flow field caused by the above mentioned factors.

For an actual turbine rotor with a blade tip speed of U , representative velocity triangles are shown in Figure 7.3 for the absolute and relative frame of references. The sketch shows an idealized case of a coplanar velocity profile in the absolute frame of reference. For the coplanar case, the tip of the relative velocity vectors lie on a straight

Figure 7.1 Moving belt leading edge configuration.

$Re = 1.4 \times 10^5$
belt tip-wall
0% EES

$Re = 4.3 \times 10^5$
plexiglass tip-wall

Figure 7.2 Plane C_1 non-dimensional streamwise vorticity distribution for $\frac{\tau}{c} = 0.024$.

Figure 7.3 Inlet tip-wall boundary layer in relative and absolute frame of references.

Figure 7.4 Superimposed diagram of velocity vectors for a coplanar inlet tip-wall boundary layer.

line when the velocity triangles are superimposed as in Figure 7.4. For simplicity, in designing the belt Danias (1987) aimed toward a coplanar velocity profile at the leading edge plane.

Experiments were performed for 1.5, 2.4 and 3.8 percent clearance sizes and three wall speeds corresponding to 0, 60 and 100 percent engine equivalent wall speed to inlet axial velocity ratios (EES). For each clearance the inlet tipwall boundary layer was traversed with a three-hole probe at mid-passage in the leading edge plane. The integral parameters for the inlet axial and transverse boundary layers are given in Table 7.1. The superimposed velocity profiles for 100 percent EES are shown in Figure 7.5. It is seen that even though an exact coplanar inlet tipwall boundary layer could not be simulated, particularly for clearances of 2.4 and 3.8 percent this condition is closely approximated.

τ/c	0.0	0.015		0.024		0.038	
V_{wall}/V_{cl}	0	0	2.164	0	2.164	0	2.164
τ (mm)	0.0	3.7	3.7	6.0	6.0	9.5	9.5
δ_x^* (mm)	4.7	2.7	1.3	3.0	1.2	4.5	1.4
θ_x (mm)	2.6	1.1	0.3	1.7	0.3	2.7	0.3
H_x	1.8	2.4	4.3	1.8	4.0	1.6	4.7
δ_y^* (mm)	----	----	1.5	----	1.4	----	1.4
θ_y (mm)	----	----	0.4	----	0.3	----	0.4
H_y	----	----	3.8	----	4.7	----	3.5
θ_{xy} (mm)	----	----	0.7	----	0.6	----	0.8
θ_{yx} (mm)	----	----	0.2	----	0.1	----	0.2

Table 7.1 Inlet tip-wall boundary layer parameters.

Figure 7.5 Superimposed velocity diagrams for the inlet tip wall boundary layer. (100% EES)

8. TIP-LEAKAGE FLOW STUDIES WITH RELATIVE TIPWALL MOTION

8.1 Introduction

While stationary cascade data provide valuable guidance to the design of turbomachinery blade rows, it is important to evaluate their relation to the more realistic configuration with relative blade motion. Experimental tip-leakage studies available in the literature have used either stationary or rotating configurations. However, there are only a few studies where the comparison of the tip-leakage flow for the two configurations was done to a limited extent (Dean, 1954; Rechbach, 1960; Gearhart, 1966; Graham, 1986 and Morphis and Bindon, 1988). Having completed a study of the tip-leakage flow for the stationary blade configuration, it is appropriate to examine in detail the changes introduced by relative wall motion on the present cascade.

In this chapter these changes will be studied based on measurements of the flow field inside the tip gap and downstream of the blades as well as the blade pressures. Since the relative blade motion is simulated with a moving tipwall, the centrifugal forces in effect in actual turbomachinery rotors will not be present. Nevertheless, the results should still provide valuable information particularly on the viscous drag effects of relative wall motion.

8.2 Measurement of the Tip-Gap Flow

Yaras (1987) measured the tip-gap flow for a stationary tipwall. Based on the results, a number of observations were made regarding the flow physics. It is conceivable that introduction of relative tipwall motion may alter the tip-gap flow field substantially. This in turn may affect the relative significance of tip-leakage loss mechanisms previously discussed for the stationary wall configuration. It also may have an impact on the choice of design modification for reducing the tip-leakage losses.

In relation to the effects of tipwall motion on the tip-gap flow, the compressor flow simulations of Dean (1954) and Gearhart (1966) with a moving-belt endwall concentrated on wall motion effects on the tip-gap mass flow rate. Graham's (1986) study on a water flow turbine cascade went somewhat further and provided some insight to the type of flow present in the gap through dye injection. Finally, Morphis and Bindon (1988) examined the effect of tipwall motion on the pressure distribution over the blade tip.

In turbines relative wall motion will tend to reduce the tip-gap mass flow rate. Two possible mechanisms may contribute to this effect, as shown schematically in Figure 8.2.1. The shear force at the tipwall may "drag" mass back over the tip, or the pressure difference driving the flow through the gap may be reduced in some way by the rotation. Commonly the shear effect has been assumed to dominate the flow and has been invoked to explain the reduction in mass flow rate (e.g, Rechbach, 1960, Kingcombe et al., 1990).

The gap flow was measured in the present cascade for a 3.8 percent clearance size using a three-hole probe along the blade mean line. The relatively large clearance size was selected to reduce the probe size with respect to the gap height to lessen the interference of the probe with the tip-gap flow. In addition, the minimum distance between the probe and the moving tipwall was kept at three times the probe tip thickness to prevent accidental contact between the probe and the belt. Selection of a large clearance size reduced the relative size of the part of the gap flow that would not be measured. Measurements were performed for wall speeds of 0, 60 and 100 percent EES.

Velocity vectors extracted from the measurements for all three wall speeds are shown in Figure 8.2.2. The presence of the separation bubble on the blade tip is evident for all three cases. The vortical structure of the flow inside the bubble is also evident as previously observed by Yaras et al. (1989) and Bindon (1987c). Over the first quarter chord length the separation bubble does not seem to extend up to the blade mean line.

Figure 8.2.1 Possible physical mechanisms by which rotation reduces tip leakage in turbines.

Figure 8.2.2 Tip gap velocity field ($\frac{\tau}{c}=0.038$).

At quarter chord length, where the blade loading is at its peak, the bubble grows suddenly and remains relatively large up to the trailing edge. This result is consistent for all three wall speeds. Comparison of the data for the three wall speeds suggests a reduction in the bubble size as the wall speed is increased. This can be seen from the results in the measurement plane at $\frac{z}{\tau} = 0.147$. As speed is increased the velocity vectors at this traverse plane reverse direction relative to the blade mean line. This suggests that the centerline of the vortex has shifted towards the blade tip, apparently passing through the 14.7 percent plane at about 60 percent EES. The physics behind this trend in the separation-vortex will be discussed following the evaluation of the tip-gap main stream flow.

Outside the separation bubble the velocity vector magnitude is reduced substantially as the wall speed is increased. As a consequence, a significant reduction in the tip-gap flow rate is observed (Figure 8.2.3). The reduction in the velocity magnitude is distributed uniformly over the gap height. Also, the rotation of the velocity vector induced by the wall motion as the tipwall is approached is small even for a wall speed of 100 percent EES. The limiting velocity vector on the tipwall has to be at 90 degrees relative to the cascade inlet flow direction, due to the no-slip boundary condition. Thus, a significant rotation of the tip-gap velocity vector has to occur eventually. However, according to the measurements this swing of the velocity vector is confined to a very thin transverse boundary layer on the tipwall. If viscous drag in the tip gap were the dominant mechanism by which tipwall motion affects the gap flow, substantial rotation of the velocity vector would have occurred all across the gap. Moreover, in that case the effect of relative tipwall motion would not have been distributed uniformly across the tip gap. These observations suggest that the relative wall motion alters the gap flow through a dominant mechanism other than shear.

A possible mechanism is a global reduction of the tip-leakage driving pressure difference. This would not only agree with the observed behaviour of the tip-gap free stream flow, but also would explain the variation in the blade tip separation-vortex size

Figure 8.2.3 Variation of tip gap mass flow rate with relative wall speed
for $\frac{\tau}{c} = 0.038$.

with wall speed: as the main stream velocity in the gap is reduced with increasing wall speed, the vorticity jump through the vortex sheet on the blade tip decreases. This in turn results in weakening and shrinking of the tip separation vortex. Without additional measurements it is difficult to speculate on how the reduction in the leakage driving pressure difference is induced by the relative wall motion. Therefore, seven-hole probe traverses were performed downstream of the blades. Evaluation of these measurements will be presented in Section 8.4.

It is important to realise that the observations are for a relatively large clearance size of 3.8 percent of blade chord. For small clearances it is feasible that viscous forces will dominate the tipwall motion effect on the tip-gap flow. In fact, Graham's (1986) particle tracing experiments in a linear turbine cascade with a clearance size of 0.8 percent blade chord suggested that the clearance flow was dominated by wall shear effects, for 100 percent EES. However, this was true for a large blade thickness only ($\frac{t_{MAX}}{c}=0.175$). For a smaller blade thickness of $\frac{t_{MAX}}{c}=0.042$ and same clearance size the tip-gap velocity magnitude was reduced without a change in direction, consistent with the present observations. This suggests that the wall motion - tip-gap flow interaction may be dominated by viscous forces for small clearances with relatively large blade thicknesses.

A shortcoming of the present experiment was that the total pressure losses inside the gap could not be studied. This was the result of an unknown amount of work being done on the flow by the moving belt. Obviously, the same drawback applied to the downstream measurements as well. Fortunately, evaluation of the vorticity field downstream of the blades provided implicit information on the possible effect of relative tipwall motion on end-losses. This aspect will be addressed in Section 8.4.

8.3 Prediction of the Tip-Gap Flow

In this section a model for predicting the tip-gap flow field for the relative wall motion configuration is developed. The model is found to be useful in providing insight into the flow physics. It may also furnish the basis for a scheme to be used for evaluating blade tip and tipwall treatments whose effectiveness may depend on the relative wall speed.

A simple model was developed by Yaras et al. (1989) for the prediction of the inviscid flow inside the tip gap for the case of a stationary tipwall. From measurements, the authors found little variation in chordwise momentum as the flow passed through the gap. Thus, they concluded that the tip-leakage driving pressure difference was used almost entirely to accelerate the flow normal to the blade chord. Moreover, the majority of the fluid passing through the gap remained loss free up to the gap exit. Therefore, Bernoulli's equation was applied to the acceleration process. The authors also found that the undisturbed blade loading appeared to approximate the tip-leakage driving pressure difference quite well. This made the model more attractive since the undisturbed blade pressure distribution is relatively easy to predict. Once the tip-gap velocity component normal to the blade chord was predicted in this manner, it was used to determine the ideal tip-gap mass flow rate. The ratio of the measured and ideal mass flow rates was then defined as a discharge coefficient that accounts for viscous effects in the gap (i.e., separation bubble on the blade tip and the tipwall boundary layer). This coefficient varied with clearance size, mainly due to changes in the relative separation bubble size (Figure 6.8.3.1).

In Section 8.2 it was concluded that the main effect of relative tipwall motion on the tip-gap flow is a global reduction of the tip-leakage driving pressure difference. This in turn results in a reduced separation bubble size on the blade tip. Thus, when modifying the tip-gap flow model to incorporate tipwall motion effects, two issues have

to be dealt with. One is the variation of the discharge coefficient with wall speed. The other is the dependence of the tip-leakage driving pressure difference on wall speed.

In the stationary wall model, the velocity component normal to the blade chord is determined from Bernoulli's equation as follows:

$$\frac{V_n}{V_{CL}} = \sqrt{C_{P_n} - C_{P_s}} \quad 8.3.1$$

where $C_{P_n} - C_{P_s}$ is the undisturbed blade pressure difference nondimensionalized by the inlet centerline dynamic pressure. As seen in Figure 8.3.1, relative to the blade midspan pressure difference, the tip-gap driving pressure difference is shifted rearward by about 20 percent of chord length. This displacement is to be applied to the blade pressure differences before being used in Eqn. 8.3.1. Even though the shift observed in Figure 8.3.1 may not be a general result, the predictions are still quite good when it is not taken into account. The reduction in the driving pressure difference due to relative wall motion can be incorporated through a "pressure reduction coefficient", $C_{\Delta P}$. Then, for moving tipwall Eqn. 8.3.1 becomes,

$$\frac{V_n}{V_{CL}} = \sqrt{C_{\Delta P}(C_{P_n} - C_{P_s})} \quad 8.3.2$$

It might be necessary to allow $C_{\Delta P}$ to vary along the blade chord. This possibility will be considered after comparison of the experimental results with the predictions based on Eqn. 8.3.2 with a constant $C_{\Delta P}$ value. To determine the value of this constant, the following procedure is employed.

Figure 8.3.1 Blade midspan and tip wall driving pressure differences.

The ideal tip-gap mass flow rate for zero wall speed can be written as,

$$\left\{ \frac{\dot{m}_g}{\dot{m}_p} \right\}_{ideal} = \frac{\int_{A_t} \frac{V_n}{V_{CL}} dA}{A_p} \quad 8.3.3$$

Using Eqn. 8.3.1, Eqn. 8.3.3 can be rewritten as,

$$\left\{ \frac{\dot{m}_g}{\dot{m}_p} \right\}_{ideal} = \frac{\int_{A_t} \sqrt{C_{P_n} - C_{P_w}} dA}{A_p} \quad 8.3.4$$

For non-zero wall speed, equation 8.3.4 becomes,

$$\left\{ \frac{\dot{m}_g}{\dot{m}_p} \right\}_{ideal, V_w \neq 0} = \frac{\int_{A_t} \sqrt{C_{\Delta P} (C_{P_n} - C_{P_w})} dA}{A_p} \quad 8.3.5$$

Taking the ratio of Equations 8.3.5 and 8.3.4 results in an expression for the pressure reduction coefficient, $C_{\Delta P}$:

$$C_{\Delta P} = \left\{ \frac{\dot{m}_{g(ideal, V_w \neq 0)}}{\dot{m}_{g(ideal, V_w = 0)}} \right\}^2 \quad 8.3.6$$

The ideal mass flow rates for zero and non-zero wall speeds in the above expression have to be extracted from experimental data that is not influenced by viscous effects. Any of the tip-gap measurement planes that is outside the separation bubble and the tipwall boundary layer can be used for this purpose. Figure 8.2.2 suggests that the

measurement plane 57.3 percent gap height away from the blade tip satisfies this condition for all three wall speeds. Thus, using the measurements in this plane, variation of C_{AP} with wall speed is calculated. The results are given in Figure 8.3.2. As seen from the figure, for 100 percent EES the driving pressure difference is reduced by as much as 50 percent.

Due to the separation bubble on the blade tip and the tipwall boundary layer, the actual tip-gap mass flow rate is lower than the ideal one calculated in Eqn. 8.3.5. This difference is dependent on wall speed due to reduction of the separation bubble size and augmentation of tipwall viscous drag as the wall speed is increased. Thus, the discharge coefficient, C_D , representing these viscous effects is expected to depend on the wall speed. The relation for calculating the discharge coefficient is,

$$C_D = \frac{\dot{m}_{t_{\text{meas}}}}{\dot{m}_{t_{\text{ideal}}}} \quad 8.3.7$$

Applying this relation to all three wall speeds, the variation of C_D with relative wall speed is established (Figure 8.3.3). From the figure it is seen that despite the reduction in the separation bubble size with increasing wall speed, C_D decreases as the wall speed is increased. This is a consequence of the fluid dragged back over the tip in the tipwall boundary layer against the natural tip-leakage flow direction. Interestingly, the dependence of the coefficient onto wall speed is nearly linear.

The work of Yaras et al. (1989) with stationary tipwall suggested C_D values around 0.7 to 0.8. However, Fig. 8.3.3 gives a slightly lower C_D value of 0.65 for zero wall speed. The Reynolds number was about three times lower than that simulated by Yaras et al. (1989). Yaras (1987) observed an increase in the separation bubble size as the Reynolds number is reduced. Thus, this may be an explanation for the observed difference in the values of C_D . The slightly different inlet boundary layer thickness on

Figure 8.3.2 Variation of the pressure reduction coefficient with relative wall speed for $\frac{\tau}{c}=0.038$.

Figure 8.3.3 Variation of gap discharge coefficient with relative wall speed
for $\frac{\tau}{c}=0.038$.

the stationary belt, compared to the plexiglass tipwall configuration, may have been a factor as well.

A slight dependence of C_D on tip-clearance aspect ratio was suggested in Figure 6.8.3.1. The possible dependence of $C_{\Delta p}$ on the gap size cannot be established at this time since tip-gap flow measurements were performed for one clearance size only. Nevertheless, indirect evidence from downstream flow measurements discussed in the next section suggest that for practical clearance sizes the mechanism by which the wall motion reduces the tip-leakage driving pressure difference should have little dependence on the gap size.

Having established the variation of $C_{\Delta p}$ and C_D , the tip-gap velocity component normal to the blade chord can be predicted using Eqn. 8.3.2. This and the undisturbed velocity on the blade pressure side can be used to determine the velocity vector in the tip gap. Finally, the tip-gap mass flow rate can be predicted using the equation:

$$\dot{m}_t = C_D \rho \tau \int_0^c V_n dx' \quad 8.3.8$$

For all three wall speeds the predicted mass flow rate distribution is compared to experimental data in Figure 8.3.4. The prediction is very satisfactory regardless of the wall speed. To ensure that this accuracy is a result of proper simulation of the tip-gap flow field, the predicted velocity magnitude and flow angle in the gap is compared to experimental values in Figures 8.3.5 and 8.3.6 respectively. Again, the results are satisfactory.

An alternative might be that the proportion by which tipwall motion reduces the tip-leakage driving pressure difference does not remain constant along the blade chord. This was tested by allowing $C_{\Delta p}$ to vary along the chord. The prediction accuracy showed only marginal improvement over that for constant $C_{\Delta p}$ along the blade.

Figure 8.3.4 Comparison of predicted and experimental tip gap flow rate distribution for $\frac{\tau}{c} = 0.038$.

Figure 8.3.5 Comparison of predicted and experimental tip gap velocity magnitude distribution for $\frac{\tau}{c} = 0.038$.

Figure 8.3.6 Comparison of predicted and experimental tip gap flow angle distribution for $\frac{\tau}{c} = 0.038$.

8.4 Downstream Flow Field in the Vicinity of the Blade Tip

To fully understand the effect of relative tipwall motion on the tip-leakage flow, measurements inside the tip gap were supplemented by seven-hole probe traverses downstream of the blades. These traverses were done for two clearances of 2.4 and 3.8 percent blade chord, and for three wall speeds of 0, 60 and 100 percent EES. As was noted in Section 8.2, the total pressure field cannot be interpreted due to the unknown amount of energy given to the flow by the moving tipwall. Therefore, the vorticity field will be examined instead.

The streamwise vorticity distribution 47.5 percent axial chord length downstream of the trailing edge plane (Plane A) is shown in Figure 8.4.1. The same contour interval is used on all three plots for easy comparison. As the tipwall speed is increased from 0 to 60 percent EES, the tip-vortex strength is reduced significantly. Simultaneously, the passage vortex adjacent to the tip vortex is somewhat enhanced by the scraping effect of the blades on the tipwall boundary layer. It is noteworthy to point out that for this scraping effect to occur the blade tips do not have to intrude into the tipwall boundary layer physically; the pressure field created by the blades is sufficient to introduce this effect. This was noted earlier by Allen and Kofskey (1955) who performed smoke visualization studies in a low speed turbine rotor. Because of viscous drag, wall motion seems to push the passage vortex, and thus the tip vortex, toward the suction side of the passage. In the process, part of the enhanced passage vortex squeezes between the tipwall and the tip vortex. As wall speed is further increased to 100 percent EES, the tip-leakage vortex strength continues to decrease while the passage vortex is further strengthened. The two vortices continue to move towards the suction side of the passage. However, the displacement is not as large as that observed between 0 percent and 60 percent EES. These features were also observed for 2.4 percent clearance size. Of course for very large clearances the blade or its pressure field cannot produce a scraping effect on the tipwall. In this case the observed flow with relative motion is likely to approach that of stationary configuration.

Figure 8.4.1 Plane A non-dimensional streamwise vorticity distribution for $\frac{\tau}{c} = 0.038$.

These observations for turbine blades can be contrasted with the observed effects of rotation in compressors, where the direction of rotation is opposite to that for turbines. Dean's (1954) study in a linear compressor cascade with moving tipwall and Lakshminarayana et al.'s (1987) work in a compressor rotor showed a pitchwise shift of the tip vortex toward the pressure side of the passage as wall speed was increased. In view of opposite directions of the relative wall motions in turbines and compressors, these observations seem to agree with the trend encountered in the present turbine cascade. Furthermore, unlike in turbines, in compressors the scraping effect of the blades is expected to produce a "scraping vortex" on the leading (pressure) side of the blade with a sense of rotation the same as that of the tip vortex. This was observed by Phillips and Head (1980) who performed flow visualization in an axial compressor, and confirmed through measurements by Inoue et al. (1986) and Inoue and Kuroumaru (1988).

In Section 8.2 it was noted that the tip-gap flow measurements suggest a global reduction in the tip-leakage driving pressure difference with increasing wall speed. The mechanism by which this reduction is introduced is evident from the downstream flow field described above. As wall speed is increased, the enhanced passage vortex, and thus the tip vortex, are pushed toward the blade suction side. This should introduce a blocking or throttling effect on the tip-gap flow, reducing the tip-leakage driving pressure difference. A rise in the tip-gap exit pressure level, presumably caused by this throttling effect, was observed by Morphis and Bindon (1988) in a simulated turbine rotor flow from measurements on the blade tip in the fully attached region at the gap exit. From the same argument, in compressors the scraping vortex forming on the pressure side of the blade tip should raise the pressure level at the gap inlet, increasing the tip-gap flow rate. This increase in the tip-gap inlet pressure was observed by Inoue and Kuroumaru (1988) in their compressor rotor.

8.4.1 Effect of Relative Wall Motion on Tip-Vortex Circulation

Figure 8.4.1 showed significant reduction in tip-vortex strength with increasing wall speed. To quantify this effect, the vortex circulation was determined by integrating the x-wise vorticity component within the tip vortex flow region in Plane A. The results are shown for both clearances in Figure 8.4.1.1. A substantial reduction in the vortex strength is evident. The non-linear nature of this trend is also apparent from the figure. The zero belt speed results lie slightly below those for the plexiglass tipwall configuration: For the belt-wall configuration, a narrow flow region (1.2 percent of passage span) next to the tipwall was not measured. Thus, a small portion of the vorticity field associated with the tip vortex was not accounted for in the calculation of the tip-vortex circulation. This was only true for the zero wall speed since for non-zero wall speeds the tip vortex is pushed away from the tipwall by the scraping vortex (Figure 8.4.1).

An argument for the variation of the tip-vortex strength with wall motion can be made in terms of the amount of vorticity shedding from the blade tip. In the discussion of the stationary wall results in Section 5.4, the tip-vortex circulation being less than the bound circulation was explained by the interaction of the vorticity generated on the tipwall with the vorticity of opposite sign shedding from the blade tip. With the introduction of wall motion, the vorticity associated with the transverse boundary layer on the tipwall becomes considerably larger. Therefore, with wall motion the vorticity layer on the wall is likely to have a larger effect on the resultant vorticity that remains in the tip-vortex structure. In addition to this mechanism, the wall motion influences the amount of vorticity shed from the blade tip by altering the pressure field in its vicinity: as discussed earlier, wall motion enhances and relocates the passage vortex and reduces the pressure difference driving the tip-leakage flow. This in turn reduces the velocity in the gap main stream and thus the vorticity generated on the blade tip. By comparison, in compressors the direct effect of the wall motion will be to increase the inviscid core velocity in the tip gap, leading to increased vorticity shedding from the blade tip.

Figure 8.4.1.1 Effect of relative tip-wall speed on tip-vortex circulation.

This is consistent with the values of tip-vortex circulation observed by Inoue et al. (1986) in their compressor rotor: they were higher than the values obtained at the same clearances for stationary tipwalls by Yaras and Sjolander (1989), Lakshminarayana and Horlock (1962), and Lewis and Yeung (1977).

The main information of interest is the effect of relative tipwall motion on end-losses. Since loss correlations based on stationary cascade data have been widely used in turbomachinery design, it is essential to quantify this effect. Even though it is not possible to extract losses from the recent data, the downstream vorticity field provides useful insight to this aspect. In Chapter 6 it was established that mixing of the passage and tip-leakage vortices, and the separation bubble inside the gap, are the primary loss mechanisms associated with end-losses. It was observed that with the introduction of relative tipwall motion the size of the separation bubble on the blade tip is reduced and the tip vortex is weakened due to the reduced gap flow rate. On the other hand, the passage vortex is enhanced. Thus, there are expected to be increased losses associated with the strengthened passage vortex but a reduction in the loss caused by the tip vortex. Similar argument may hold for the reduced size of the separation vortex on the blade tip against enhanced tipwall boundary layer losses inside the gap. Therefore, in overall terms tipwall motion seems to cause opposing effects on end-losses. However, on the basis of the present data it is not possible to say whether there be a net increase or decrease in losses. Rechbach (1960) appears to be the only one who provided quantitative results on this effect. The author performed experiments in an annular turbine rig. Even though the flow field was not explored in detail, drag measurements on the rotor provided sufficient information to conclude that relative tipwall motion had little effect on end-losses.

8.5 Modelling of the Tip Vortex with Relative Tipwall Motion

The results discussed in the previous section indicate a significant reduction in the tip vortex strength as the relative wall motion is introduced. Fortunately, the axisymmetric nature of the vortex is not altered substantially by the tipwall motion. This

is especially true for the smaller of the two examined clearances (2.4 percent of blade chord). Thus, it may be possible to modify the vortex model introduced in Section 5.5 to apply to this more realistic configuration.

As in Section 5.5, the eddy viscosity and the effective vortex origin are determined from the measured tip-vortex centerline vorticity in planes A and C₁, and the vortex circulation:

$$\omega'_s = \frac{\Gamma'}{4\pi} \frac{Re_c}{(1 + \frac{v_o}{v}) \cos\gamma (\frac{x}{c_x} - \frac{x_o}{c_x})} \quad 8.5.1$$

The eddy viscosity was found to depend on wall speed to some extent, increasing from about 200 times the molecular viscosity to 550 times the molecular viscosity as wall speed is raised from 0 to 100 percent EES.

For the stationary tipwall the eddy viscosity constant, Λ , appeared to be around 1.3 regardless of the clearance size. As seen from Figure 8.5.1, for zero belt speed this value is approximated closely. The slight discrepancy is presumably due to the underestimated tip vortex circulation for zero belt speed, discussed earlier. With increasing wall speed the value of Λ increases to about 2.0 for 100 percent EES. However, as seen from the figure, this variation in Λ is still substantially less than that of the alternative constant, a_s , introduced in Section 5.5. Figure 8.5.1 shows that the effective vortex origin shifts towards the leading edge as the clearance size is increased from 2.4 to 3.8 percent of blade chord, confirming the previously observed trend for solid tipwall. The figure also shows that increasing wall speed has the same effect as decreasing the clearance size. This is plausible since both have a similar influence on the vortex strength.

Figure 8.5.1 Variation of vortex model parameters with wall speed.

By using the eddy viscosity, vortex origin, and tip-vortex circulation values in the following expression, the cross flow associated with the vortex was calculated for both clearances and all three wall speeds:

$$v_{\theta} = \frac{\Gamma}{2\pi r} \left(1 - e^{-\frac{r^2}{4(v+\nu)X\frac{r}{V_t}}} \right) \quad 8.5.2$$

The results are compared to experimental data in Figures 8.5.2 and 8.5.3. The results are presented with the vortex center in both planes coincident. As seen, the peak to peak variation in the cross flows and the diameter of the vortices are well predicted for all speeds. There is some discrepancy at the outer edges of the induced velocity field. This is partly explained by the induced effects of tip vortices from adjacent blades, which are not included in the model. Another, presumably dominant, reason is the uncertainty in the experimental tangential velocity values. These velocities are commonly defined as the velocity components that lie in a plane normal to the vortex centerline direction. As seen from Figure 8.5.4, the vortex centerline angle increases with wall speed. However, taking into account this variation, the application of the above noted tangential velocity definition to non-zero wall speed cases resulted in tangential velocity distributions across the vortex that do not change sign at the vortex center. This is thought to be a result of the tip vortex structure being modified by the wall dragging effect as shown in Figure 8.5.5. Therefore, for determining the tangential velocity components the midspan exit flow direction was used as reference instead of the vortex centerline. For compatibility, this was done for the zero wall speed case as well. It was found that the choice of reference direction alters the cross-flow distribution in a global fashion only; it has very little effect on the peak to peak variation of the inferred cross-flow, which is the quantity of particular interest.

Figure 8.5.2 Spanwise distribution of tangential velocity through the center of the tip vortex ($\frac{\tau}{c} = 0.024$).

Figure 8.5.3 Spanwise distribution of tangential velocity through the center of the tip vortex ($\frac{\tau}{c} = -0.038$).

Figure 8.5.4 Variation of tip-leakage vortex centerline orientation with relative wall speed.

Figure 8.5.5 Effect of relative wall motion on tip vortex.

8.6 Effects of Relative Tipwall Motion on Blade Loading

The blade pressures were measured for 0, 60 and 100 percent EES. The comparison of the results at midspan for 1.5 and 3.8 percent clearances (Figures 8.6.1 and 8.6.2) shows no variation of midspan pressures with wall speed. Near the blade tip (rows 1 and 2), the pressure side is somewhat influenced by wall motion. However, the dominant effect is seen on the suction side. As the wall speed is increased, considerable unloading is evident. Somewhat sparse blade pressure measurements of Graham (1986) also appeared to show such unloading. The amount of unloading seems to increase with increasing clearance size. For both clearances the dominant effects of tipwall motion are confined to within a few gap heights from the tip. This is in agreement with the compressor cascade data of Dean (1954). Downstream measurements discussed in Section 8.4 show reduced tip vortex strength with increasing wall speed. Since the suction peak in the proximity of the blade tip seen in Figures 8.6.1 and 8.6.2 is a consequence of the tip vortex, it is expected that its magnitude will drop with reduced tip vortex strength. On the other hand, this unloading should not be viewed as a detrimental effect. Sjolander and Amrud (1987) found that the increased loading due to the vortex-induced suction peak is accompanied by a rotation of the load vector toward the axial direction. Thus, the additional loading appears as an axial thrust on the rotor instead of a tangential force augmenting the rotor torque. As was noted in Section 8.4, in compressors the relative tipwall motion increases tip-vortex strength and raises the pressure level on the pressure side. Therefore, unlike the present results for the turbine configuration, the relative tipwall motion in compressors would be expected to increase the loading at the blade tip. This trend was observed by Dean (1954).

Row 14
(Midspan)

Row 1

Row 2

Figure 8.6.1 Blade static pressure distribution for $\frac{\tau}{c} = 0.015$.

Figure 8.6.2 Blade static pressure distribution for $\frac{\tau}{c} = 0.038$.

9. DESIGN MODIFICATIONS TO REDUCE TIP-LEAKAGE LOSSES: WINGLETS

9.1 Introduction

Results presented in previous chapters show that tip leakage is responsible for significant aerodynamic losses. Moreover, it causes significant variations in the flow across the annulus which in turn results in off-design flow at a downstream blade row. On the other hand, there is at least one case where tip-leakage flow is desirable: Farokhi (1987) showed that an axial exhaust diffuser downstream of a turbine rotor enjoys an enhancement in its pressure recovery since the tip-leakage flow emerges as a highly energetic wall jet. This, however, is applicable to last stages only. Therefore, almost always the overall effects of tip leakage are detrimental and designers have examined various design modifications to reduce these effects.

There have been several studies that examined design modifications on the rotor casing for compressors (eg., Moore and Osborn, 1977; Takata and Tsukuda, 1977). The driving force for this has been the desire to improve the stall margin for the compressor rotor rather than reducing tip-leakage losses. Skewed slots on the tipwall were found to be most effective by both references (Figure 9.1.1). If relative dimensions are chosen properly, it is possible that such a configuration may provide some reduction in tip-leakage losses for both turbines and compressors. This will be a consequence of an effective labyrinth type seal being formed by the blade tip and the treated wall. It may be possible that the effectiveness of such a configuration can be increased by aligning the slots with the blade chord line since the results of Yaras et al. (1989) showed that the tip-gap velocity vector is normal to the chord line for practical clearance sizes. Recessed casing was found to be another wall treatment that contributed to tip-leakage loss reduction both in compressors and turbines (Kofskey, 1961; Haas and Kofskey, 1977; Wisler and Beacher, 1989).

Figure 9.1.1 Casing treatment, skewed slot (reproduced from Moore and Osborn, 1977)

An effective approach for reducing tip-leakage losses is the incorporation of a complete shroud at the rotor tip. However, there are penalties associated with this design approach in terms of increased rotor blade centrifugal stresses and increased manufacturing costs. Furthermore, rotor tip shrouding introduces a different type leakage in the form of axial flow through the labyrinth seals. This leakage, however, is not as detrimental as the conventional one since it does not roll into a vortex to produce significant mixing losses and disturb the flow for the subsequent blade row. An additional advantage of shrouds is that they add rigidity to the blades. An alternative to shrouds that has received some attention in the last decade is the use of mini-shrouds (winglets) which may provide a good trade-off between weight and/or complexity of the treatment, and better aerodynamic performance. To the author's knowledge the only studies in this area have been done for turbines. Patel (1980) quoted improvements in stage efficiency of 1.2 percent at the design point with the application of winglets. Booth et al. (1982) studied various blade tip treatments in a water cascade. They found winglets to be superior to grooved blade tips. Subsequently, the authors tested the winglets on a rotor and an efficiency gain of 0.6 percent was recorded. Recognizing the potential of this treatment, several tests were performed in the present turbine cascade in attempt to identify the mechanisms by which the winglets provide loss reduction. Such information could prove useful for optimizing the winglet shape and/or combining the winglet with other treatments.

9.2 Winglet Geometry

No general rules seem to exist for choosing the relative dimensions of the winglets. Thus, as a starting point, a suction-side winglet, a pressure-side winglet and a third one that is a combination of the two were examined on the present linear cascade. The winglet geometry for each type was obtained by extending half the local blade thickness in the appropriate direction (suction, pressure or both). The leading and trailing edge portions of winglets were somewhat smoothed after being generated in this fashion. The resultant winglet shapes are shown to scale in Figure 9.2.1.

Figure 9.2.1 Examined blade tip treatments.

9.3 Studies with Stationary Tipwall

Winglet effects with stationary tipwall were studied for two clearance sizes of 2.4 and 3.7 percent of blade chord. The inlet tipwall boundary layer integral parameters, as obtained from pitot tube traverses 170 mm upstream of the leading edge plane, are given for reference in Table 9.3.1. Measurements in the test section consisted of seven-hole probe traverses in two downstream planes (Planes B and C₁), blade static pressures and tipwall static pressures.

The total pressure loss coefficient contours in plane B are shown in Figure 9.3.1 for all blade tip configurations. For easy comparison, the same contour interval is used for all configurations. Surprisingly, there appears to be very little change in the size and location of the tip vortex for any of the winglets tested. This result is supported by the streamwise vorticity contours in Figure 9.3.2, where the passage and tip-leakage vortices can be distinguished more clearly. Similar results were observed for 3.7 percent clearance.

To evaluate the effect of each winglet on end-losses more precisely, the mass-averaged end-losses were determined for the measurement planes and the hypothetical mixed-out plane. Deduction of the end-losses from the overall losses was performed as described in Section 6.5. The mass-averaged losses used in these calculations are tabulated for the two clearances in Tables 9.3.2 and 9.3.3. The mixed-out end-losses are given in Figures 9.3.3 and 9.3.4 for both clearances and all blade tip configurations. As seen, each tip treatment provides only marginal reduction in the end-losses. The pressure-side winglet is less effective in relation to the other two treatments. In this context, comparison of the two figures suggests that each winglet provides about the same amount of reduction in end-losses (in terms of $\Delta C_p''$) regardless of the clearance size. This implies that winglets do not affect the sensitivity of end-losses to the tip clearance size, which supports Patel's (1980) observation.

τ/c	0.024	0.037
τ (mm)	6.0	9.2
δ_x^* (mm)	3.4	4.4
θ_x (mm)	2.4	3.1
H_x	1.4	1.4

Table 9.3.1 Inlet tip-wall boundary layer parameters.

Figure 9.3.1 Plane B C_p distribution for $\frac{\tau}{c}=0.024$.

- a) regular tip
- b) suction-side winglet
- c) pressure-side winglet
- d) double winglet

Figure 9.3 ~ Plane B ω' distribution for $\frac{\tau}{c} = 0.024$.

a) regular tip

b) suction-side winglet

c) pressure side winglet

d) double winglet

		Blade tip configuration			
		Regular tip	Suction winglet	Pressure winglet	Double winglet
Location					
Inlet					
	C_{P_s}''	-0.034	-0.034	-0.034	-0.034
Downstream					
	q_2/q_1	2.02	2.06	2.03	2.06
	C_{P_s}'' (Plane B)	-0.208	-0.195	-0.195	-0.183
	C_{P_s}'' (Plane C ₁)	-0.289	-0.257	-0.275	-0.250
	C_{q_s}'' (Plane B)	0.049	0.060	0.061	0.063
	C_{q_s}'' (Plane C ₁)	0.017	0.029	0.022	0.030
	C_{P_s}'' (mixed B)	-0.311	-0.292	-0.299	-0.288
	C_{P_s}'' (mixed B, C ₁)	-0.339	-0.309	-0.331	-0.303

Table 9.3.2 Quantities extracted from flow field measurements ($\frac{\tau}{c} = 0.024$) .

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS
STANDARD REFERENCE MATERIAL 1010a
(ANSI and ISO TEST CHART No. 2)

		Blade tip configuration			
		Regular tip	Suction winglet	Pressure winglet	Double winglet
Location					
Inlet					
		-0.038	-0.038	-0.038	-0.038
Downstream					
	q_2/q_1	1.91	1.99	1.95	1.95
	C_{P_s}'' (Plane B)	-0.246	-0.207	-0.231	-0.197
	C_{q_s}'' (Plane B)	0.066	0.101	0.080	0.100
	C_{P_s}'' (mixed B)	-0.377	-0.348	-0.360	-0.347

Table 9.3.3 Quantities extracted from flow field measurements ($\frac{\tau}{c} = 0.037$) .

Figure 9.3.3 Comparison of mixed-out end-losses based on Plane B ($\frac{\tau}{c}=0.024$).

Figure 9.3.4 Comparison of mixed-out end-losses based on Plane B ($\frac{\tau}{c}=0.037$).

In view of burn-out problems encountered at the edges of the turbine blade winglets, increased centrifugal stresses in the blades and extra manufacturing costs, the minor gain in aerodynamic performance provided by winglets seems too small to justify their application. However, it is possible that winglets may be more effective when used in combination with other blade and/or tipwall treatments. To guide the development of such treatment combinations, an understanding of the flow physics by which winglets provide the observed loss reduction is essential.

The downstream development of the end-losses for all the examined blade tip configurations are given in Figure 9.3.5. From the figure it is seen that the suction-side and double winglets provide some reduction in the downstream mixing losses while the pressure-side winglet has little effect on this loss component. These trends are rather surprising since the secondary kinetic energy associated with the tip vortex is considerably larger for the treated blade tips than for the regular tip (Tables 9.3.2 and 9.3.3). This observation regarding the trend in secondary kinetic energy is supported by the relative magnitudes of the tip-vortex circulations shown in Figures 9.3.6 and 9.3.7. In this case one would expect increased mixing losses with the application of winglets, contrary to the trends in Figure 9.3.5. A possible explanation for this apparent contradiction is that with the employment of winglets the increase in the secondary kinetic energy is accompanied by an axial non-uniformity less severe than the one for the regular tip. In this case the increased mixing losses due to larger secondary kinetic energy may be more than offset by the reduced losses associated with the diffusion of axial non-uniformity.

The results for Plane B in Figure 9.3.5 suggest that a significant portion of the loss reduction due to winglets occurs before the flow emerges from the blade passage. To explore the physics behind this, tipwall static pressures were measured for all tip configurations. The results are given in Figures 9.3.8 to 9.3.10 as peak-to-peak tipwall pressure differences normal to the blade chord. Corresponding blade midspan pressure differences are also plotted on the same figures for reference. The results suggest a

Figure 9.3.5 Variation of end-losses with downstream distance ($\frac{\tau}{c} = 0.024$).

Figure 9.3.6 Tip-leakage vortex circulation measured in Plane B
 $(\frac{\tau}{c}=0.024)$.

Figure 9.3.7 Tip-leakage vortex circulation measured in Plane B
 $(\frac{\tau}{c}=0.037)$.

Figure 9.3.8 Blade midspan and tip wall driving pressure differences for $\frac{\tau}{c} = 0.024$.

Figure 9.3.9 Blade midspan and tip wall driving pressure differences for $\frac{\tau}{c} = 0.024$.

Figure 9.3.10 Blade midspan and tip wall driving pressure differences for $\frac{\tau}{c} = 0.024$.

reduction in the tip-leakage driving pressure difference for the winglets that extend on the blade suction side (i.e, suction-side and double winglets). As seen from the contour plot in Figure 9.3.11, the pressure gradient near the blade tip is large. Thus, when the suction-side winglet is attached to the blade tip, the suction pressure experienced by the tip-gap flow is reduced. In Section 6.6 it was found that the gap mass-averaged kinetic energy normal to the blade chord line, which scales on the tip-leakage driving pressure difference, closely approximates the additional mixing losses after the gap flow discharges into the passage. The reduction in driving pressure difference seen for the suction-side and double winglets would thus be expected to lead to a reduction in losses, as observed.

According to Figure 9.3.8 the pressure-side winglet does not reduce the tip-leakage driving pressure difference. The remaining mechanisms through which the pressure-side winglet can contribute to loss reduction are:

- (i) decreasing the losses inside the tip gap or,
- (ii) a mechanism other than reduced tip-gap driving pressure that will result in smaller mixing losses inside the blade passage.

Reduction of losses generated inside the tip gap can be achieved mainly by altering the separation bubble size on the blade tip. However, since the pressure-side winglet forces the flow to follow a more difficult path than for the pressure-side corner of the regular blade tip, intuitively one would expect the bubble size to increase with the application of the pressure-side winglet. Thus, one should anticipate an increase in the tip-gap losses, if anything. On the other hand, the increased separation bubble size would augment the blockage in the gap, reducing the tip-leakage mass flow rate. For a given tip-leakage driving pressure difference, this would reduce the mass-averaged kinetic energy available at the tip-gap exit, thus lessening the mixing losses as the tip-leakage flow merges with the passage stream. Since, the results in Section 6.7 suggested that the mixing losses are the dominant component of the end-losses, it is conceivable that the reduction in mixing losses more than offsets the increased losses inside the tip gap. In

Figure 9.3.11 Tip wall static pressure distribution ($\frac{\tau}{c}=0.028$) .

conclusion, an enlarged separation bubble on the blade tip seems to provide a feasible explanation for the beneficial effect of pressure-side winglets. This mechanism should be in effect for the double-winglet as well.

9.4 Studies with Relative Tipwall Motion

The moving tipwall studies discussed in Chapter 8 showed substantial changes in the tip-leakage flow as well as in the strength and location of the passage and tip leakage vortices as relative blade motion is introduced. Thus, it is feasible that the effectiveness of the winglets may be altered by relative tipwall motion. To study this possibility, all three winglets and the regular blade tip were tested for 2.4 percent clearance height and a wall speed of 100 percent EES.

With relative wall motion, the losses could not be examined due to the unknown amount of work done by the belt on the flow. Therefore, the vorticity distribution was studied instead. As will be evident from the following discussions, observation of this quantity was sufficient to draw useful conclusions on the influence of relative wall motion on the effectiveness of winglets. The streamwise vorticity contours for plane B are shown in Figure 9.4.1. Compared to the previously discussed stationary wall configuration, the tip-vortex strength is reduced, the passage vortex is enhanced, and both vortices are shifted toward the suction side of the passage. However, for all blade tip geometries the tip vortex is of similar size and strength, as in the stationary wall tests. The tip-leakage vortex circulations shown in Figure 9.4.2 further support this observation.

One difference between the trends in Figure 9.4.1 and the stationary tipwall results is with regard to the tip vortex location. For the moving wall case the suction side and double winglets are more effective in moving the tip vortex away from the blade suction side. Thus, the losses associated with the boundary layer on the blade suction surface should be somewhat smaller.

Figure 9.4.1

Plane B ω' distribution for $\frac{\tau}{c} = 0.024$ and 100% EES.

a) regular tip

b) suction-side winglet

c) pressure-side winglet

d) double winglet

Figure 9.4.2 Tip-leakage vortex circulation measured in Plane B
 (100%*EES*, $\frac{\tau}{c} = 0.024$) .

9.4.1 Blade Loading

The effects of winglets on the blade pressure distribution was studied for all tip configurations, 2.4 percent clearance size and 100 percent EES wall speed. The results are shown in Figures 9.4.1.1 to 9.4.1.3.

As anticipated, at midspan the blade pressure distribution is essentially unaltered by blade tip treatment. Near the blade tip the blade pressure-side pressure distribution is not affected by the application of the winglets. The same is true for the suction side when the pressure-side winglet is employed. The downstream measurements showed that the suction side and double winglets move the tip vortex away from the blade suction side. This is evident in the blade pressures as well: with the introduction of these winglets the vortex-induced suction peak is removed. While this seems to be a detrimental effect in terms of blade tip loading, one has to keep in mind that the additional loading caused by the presence of the tip vortex is accompanied by a rotation of the load vector toward the axial direction (Sjolander and Amrud, 1987). A gain provided by the relocation of the vortex away from the blade surface is the reduction of losses associated with the blade suction surface boundary layer near the blade tip.

In conclusion, the stationary tipwall tests indicated only marginal reduction in end-losses with the application of the pressure-side, suction-side and double winglets. In addition, the winglet studies with relative wall motion suggested little impact of wall motion on the effectiveness of the winglets. Thus, in view of the detrimental effects of winglets (e.g, increased manufacturing costs, larger centrifugal stresses in the blades), the gain in the aerodynamic performance provided by the winglets does not seem to be enough to justify their utilization. It is feasible, however, that winglets may become more effective if they are used in combination with other treatments such as grooves on the blade tip or on the tipwall.

Figure 9.4.1.1 Blade static pressure distribution for $\frac{\tau}{c} = 0.024$

Figure 9.4.1.2 Blade static pressure distribution for $\frac{\tau}{c} = 0.024$.

Figure 9.4.1.3 Blade static pressure distribution for $\frac{\tau}{c} = 0.024$.

10. CONCLUSIONS AND RECOMMENDATIONS

10.1 Conclusions

The main conclusions drawn from the present work are as follows.

For the stationary tip wall, vorticity measurements downstream of the blades indicated that the circulation in the tip vortex is substantially smaller than the bound circulation for the blade. It was found that, as a result of the pressure field impressed onto the tip wall, significant amount of chordwise vorticity is generated on the wall. Interaction of this vorticity layer with the bound vorticity of opposite sign shedding from the blade tip was proposed as a feasible explanation for the observed reduced tip-vortex strength. Measurements suggested that the majority of this interaction occurs inside the blade passage after the leakage flow has emerged from the tip gap. The size of the tip clearance was found to have considerable influence on this interaction: decreasing clearance size resulted in increased interaction, and thus reduced tip-vortex strength.

The tip vortex was found to have completed its roll-up by the time it has reached the trailing-edge plane and even in that plane the structure was nearly axisymmetric. In addition, the vortex demonstrated rapid diffusion with downstream distance. A very simple model based on the diffusion of a line vortex was found to provide reasonable estimates of the lateral diffusion of the tip vortex, the cross-flow velocities through it, and the growth of its diameter with downstream distance.

Regarding the aerodynamic losses, the tip-leakage and secondary aspects of the flow were found to be essentially inseparable for all clearances examined. Therefore, it was concluded that the overall loss generated in the outer half of the passage, less the profile losses, should be treated as end-losses, with the secondary losses being the limiting value as the clearance size tends to zero. In the presence of clearance, the secondary loss is taken to be the loss which is associated with identifiable secondary flow, namely the passage vortex and the tipwall boundary layer on the pressure side of the passage. It was

concluded that this secondary loss decreased substantially with increasing clearance but it was not possible to quantify this effect precisely.

It was found that the losses generated inside the gap were small in comparison to the final end-loss. This final end-loss was found to have been largely reached at one axial chord length downstream of the trailing edge plane. Furthermore, the component of the final end-loss caused by mixing of the tip-leakage flow with the main stream was found to scale closely upon the leakage kinetic energy normal to blade chord at gap outlet. Due to the orderly roll-up of the leakage fluid into a tip vortex, much of this kinetic energy was recovered by the time the leakage flow has reached the trailing edge plane. Evidently, however, this recovery was temporary since it led to a strong tip-vortex structure which in turn caused significant mixing losses downstream of the blades as well as higher shear stresses on the tip wall.

The commonly used tip-leakage loss models were reviewed in the light of the experimental observations. All models were found to have a common shortcoming in that they do not account for any interaction between secondary and tip-leakage losses. The momentum based methods, i.e. models which assume that the tip-leakage loss is due to an induced drag, have the additional drawback in that they do not account for the increased viscous loss production on the tipwall caused by the presence of the tip vortex. The models based on energy considerations assume that the leakage kinetic energy carried by the normal component of the gap velocity is ultimately lost. This was confirmed in the present experimental work. However, in these schemes the leakage kinetic energy was not modelled properly and the interaction between secondary and tip-leakage losses was not taken into account. Thus, an improved end-loss model based on this underlying idea was developed in the present study.

The reduction in the tip-gap mass flow rate caused by the relative wall motion in turbines has commonly been assumed to be the result of shear effects dominating the tip-gap flow. However, the present measurements indicated that the majority of the gap flow

consists of an inviscid core. The shear layer on the tipwall remained quite thin for all wall speeds tested. With increasing wall speed a global reduction of the inviscid core velocity was observed. This was apparently the result of a reduction in the pressure difference driving the flow through the gap. Downstream measurements suggested that this "throttling" of the gap flow was caused by the tip vortex which was dragged towards the tip gap exit by wall motion, thus partially blocking the gap exit. The dominance of the pressure field in the gap allowed the gap-flow model developed by Yaras (1987) for a stationary tip wall to be extended to relative tipwall motion in a very simple way. The chordwise distribution of flow rate, core velocity and core flow angle were predicted quite accurately for all relative wall speeds tested.

The tip vortex was found to weaken considerably as relative wall speed is increased. At the same time, the scraping effect of the blades on the tipwall boundary layer resulted in significant enhancement of the passage vortex. A separate scraping vortex, commonly encountered in compressor rotors, was not observed. Both the passage and tip vortices were dragged towards the suction side of the passage by the relative motion of the wall. Reduction in the inviscid core velocity inside the gap, caused by the relative wall motion, resulted in less vorticity being generated on the blade tip. This explained the observed reduction in the tip-vortex circulation with increasing wall speed.

The tip-vortex model developed for stationary tip walls was extended to the rotating configuration. Once the effects of rotation on the various model parameters were accounted for, the cross-flow associated with the tip vortex was predicted accurately.

Several winglets were evaluated as blade tip treatments to reduce the tip-leakage losses. The suction-side winglet was found to decrease the pressure difference driving the tip-leakage flow. This resulted in less kinetic energy at the gap outlet, and thus reduced mixing losses as the leakage flow merges with the main stream. Evidently, the pressure-side winglet causes the same effect by increasing the separation bubble size on the blade tip, thereby reducing the leakage flow rate. The double-winglet seemed to

benefit from both effects. However, the gain in aerodynamic performance by the application of any of these winglets was only marginal.

10.2 Suggestions for Future Research

As was noted in the introduction, within the last decade extensive research has been undertaken to explore the tip-leakage flow and its effects on stage performance. The present work provides considerable progress towards this goal through experimental observation, prediction, and design modifications to reduce the detrimental effects of this flow. However, there are still many aspects of the tip-leakage flow that are to be investigated. Several such aspects have become apparent during the course of the present work.

All the experiments in the present work were done for a fairly lightly loaded blade (for turbine standards). Furthermore, the blade was strongly forward loaded. Modelling indicated that various tip-leakage loss mechanisms are dependent on the magnitude of the blade loading and its chordwise distribution. It would be desirable to verify this effect by undertaking tests with different values of overall blade loading and chordwise loading distributions.

In terms of prediction schemes, one area that needs further attention is the completion of the end-loss model proposed in the present work. As was mentioned in Section 6.8.3, the losses associated with the passage vortex for non-zero clearance remain to be modelled. This in turn demands further experimentation to understand the effects of tip clearance on secondary flow, and the parameters governing this interaction. The parameters that are likely to have a significant effect on this interaction are the tipwall boundary layer thickness at inlet, which affects the secondary flow structure, and the flow turning through the blade passage, which influences both the passage-vortex and the tip-vortex strengths.

As far as design modifications to reduce the detrimental effects of tip leakage are concerned, the two possibilities are treatment of the blade tip and treatment of the tipwall.

In Chapter 9 it was found that winglets do not provide sufficient gain in aerodynamic performance to justify the trade off in terms of mechanical, heat transfer and economical considerations. An alternative would be grooved blade tips. This would increase the losses generated inside the tip gap but would reduce the kinetic energy available at the gap exit. As was concluded in the present work, the dominant tip-leakage loss mechanism is mixing of the tip leakage with the main stream, which was found to scale on the kinetic energy at the gap exit. Therefore, the overall effect of grooved blade tips would likely be a reduction in the total tip-leakage loss. Wadia and Booth (1982) measured the overall effect of several grooved blade tips. However, the optimization of the groove geometry based on detailed experimentation remains to be done. One other alternative for blade tip treatment is injection of coolant air from the blade tip. In this configuration, the tip-leakage flow is sealed off (or reduced) by the cooling air discharge. There has been several studies that have already examined this alternative to some extent. (Wadia and Booth, 1982 and Ewen et al., 1973). Having developed a good understanding of the tip-leakage flow physics for the plain tip gap, it seems the appropriate time to further evaluate and optimize this blade tip treatment through detailed experimentation.

A tipwall treatment that is likely to be effective in reducing tip-leakage losses is grooved tipwalls. This treatment has been commonly examined in terms of its effects on stall margin in compressors (Appendix A). As far as reduction in tip-leakage losses is concerned, the mechanism in effect for this configuration would be the same as that for the grooved blade tips. It may be beneficial to employ winglets at the same time, since this would increase the number of grooves present in the gap flow field at a given instant, thus maximizing the reduction in the kinetic energy at the gap exit.

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APPENDICES

Appendix A

Review of Studies on Tip-Leakage Effects in Axial Turbomachinery

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NOMENCLATURE

English Symbols

A	Aspect ratio of the blade
c	Nominal blade chord
C_D	Discharge coefficient
C_{D,t}	Equivalent drag coefficient for the tip clearance losses
C_f	Skin friction coefficient
C_L	Lift coefficient
C_p	Static pressure coefficient
e	Ratio of tip clearance height to maximum blade thickness at the blade tip
ΔE	Power loss due to the tip clearance flow
h	Blade height
k	Contraction factor for the flow through the tip gap
K	Retained lift coefficient
m	Mass flow rate
P	Static pressure
q	Dynamic pressure
r	Annulus radius
r_c	Tip leakage vortex sheet radius
s	Blade spacing
S_t	Solidity based on tip radius
t	Tip clearance height
t[*]	Dimensionless time (defined in eqn. A.2.10.12)
U	Blade speed
V	Velocity
W	Relative velocity
z_v	Co-ordinate along the vortex sheet starting from the blade leading edge

Greek Symbols

α_2	Exit absolute/relative flow angle for vane/blade
β^*	Stagger angle
β	Tip gap flow angle relative to the blade chord
α_m	Mean flow angle: $\tan^{-1} ((\tan\alpha_1 + \tan\alpha_2)/2)$
Γ^*	Dimensionless circulation ($2\Gamma/V_\infty C_L c$)
τ	Blade thickness
ϵ	Maximum thickness to chord ratio at the blade tip
ϕ	Flow coefficient
ψ	Work coefficient
ν	Kinematic viscosity
δ	Boundary layer thickness
δ^*	Displacement thickness
θ	Momentum thickness
ρ	Density
η	Efficiency

Subscripts

c	Exit
t	Tip
T	Tangent to the Blade Pressure Surface
h	Hub
m	Mean
N	Normal to blade meanline
R	Retained
∞	Undisturbed value of a particular quantity
o	Value of a quantity at zero clearance
2D	Two dimensional
1	Blade row inlet
2	Blade row exit

A.1. INTRODUCTION

In the rotating blade row of turbomachine stages a clearance must be kept between the casing and the blade tip so as to avoid rubbing of the blades against the annulus wall. In some applications the effects of tip clearance have been reduced by using shrouded blades. However, in many cases this configuration is not suitable due to technical and/or economical reasons. The presence of this clearance causes a significant drop in the performance of the stage. The comparison of the order of magnitudes of the loss elements in a turbomachine stage, shown in Fig. A.1.1, clearly indicates the relative importance of the effects of tip leakage.

Today, there is a tendency to increase the pressure ratio of the compressors and turbines so as to increase the engine performance. This results in very small annulus heights at the last stages of compressors and at the first stages of turbines, which implies substantially larger tip clearances relative to the blade height, i.e., more significant tip clearance losses. Hence, if serious attempts to reduce the tip clearance effects are not made in the near future, the distribution in Fig. A.1.1 is likely to change rapidly in favour of the losses due to tip leakage. Therefore, it seems to be the appropriate time to concentrate on the tip clearance flow field and the losses associated with it. While there has been some work done on tip leakage, due to the complexity of the tip clearance flow field and the dependence of tip leakage losses on other loss elements, the understanding of the phenomenon appears to be incomplete in many aspects. The detailed review of the work that has been done on tip clearance given herein, can be used as a guide for determining the areas that need further investigation.

For aircraft applications with high power requirements axial turbomachines are preferred over radial ones due to size and efficiency considerations. On the other hand, radial turbomachines are able to run at relatively high efficiencies for low flow rate and high pressure ratio applications. From this point of view, radial

Figure A.1.1 Losses in a turbomachine stage (reproduced from Booth (1985))

turbomachinery seems to be the right choice for small engines. However, considerable difficulties have been encountered in the application of internal cooling to radial rotor blades. Therefore, in small engines with elevated turbine inlet temperatures axial turbines are preferred at least for the high pressure compressor turbine. Hence, it is seen that for most applications where the aerodynamic losses related to the tip clearance are significant, the axial turbomachinery is still the preferable choice. Therefore, in this review most emphasis is given to tip leakage effects in axial turbomachines.

A.2. REVIEW OF TIP CLEARANCE STUDIES

In general, the work that has been done on tip clearance effects can be grouped as follows:

- (i) Experimental studies leading to a detailed picture of this complicated flow field;
- (ii) Methods for predicting the flow field and the losses associated with the tip leakage;
- (iii) Studies on preventing or at least reducing the losses and other negative effects of the tip leakage flow.

The review in this chapter is arranged along these three groups.

A.2.1 Configurations Used for Tip Clearance Flow Studies

The tip clearance flow is very complicated and there is a large number of parameters affecting the flow. Therefore, different configurations have been used for tip clearance studies so as to investigate various effects separately. The configurations that have been used are introduced in the following paragraphs in order of increasing accuracy in simulating the real flow.

The simplest configuration is an isolated airfoil mounted between plane walls, with a tip clearance at one end of the airfoil. Sugiyama (1971) and Lewis and Yeung (1977) used this type of a configuration. In some cases, instead of simulating the tip clearance at one end of the airfoil, the airfoil is split at midspan so as to simulate the tip clearance flow without the effect of endwall boundary layers. Note that due to the symmetry, using the method of images, the flow in a desired tip clearance height can be investigated with a split height twice that of the clearance height. The work of Lakshminarayana and Horlock (1962), done with this type of configuration, was an

important step in the understanding of the tip clearance flow. Finally, it is also possible to simulate the relative endwall motion while still using an isolated aerofoil. Gearhart (1966), who investigated the cavitation problem associated with the tip leakage flow, performed experiments using a single blade where the relative blade motion was simulated with a moving belt.

A rectilinear cascade is probably the most widely used configuration for tip clearance studies. In some of these studies the endwall motion was not introduced and the tip clearance was simulated with a gap between the blades and one of the endwalls (e.g., Dishart and Moore (1989), Sjolander and Amrud (1986)). In some other cases the tip clearance is simulated with blades split at midspan so as to dispense the endwall boundary layer effect (see discussion in previous paragraph). For example, Lakshminarayana and Horlock (1965), Khabbaz and Senoo (1961), Yokoyama (1961) used this configuration. The next step was the introduction of relative endwall motion to rectilinear cascades using moving belts (Dean (1954), Graham (1986)). Another configuration that has been used to introduce the relative endwall motion while still having the blade row stationary is an annular cascade with rotating casing (Rehbach (1960), Morphis and Bindon (1988)).

The configuration that is closest to the real case is undoubtedly rotating blade rows. However, until the beginning of 1980's, because of the difficulties encountered in taking measurements within rotating blade rows, information on tip clearance effects from this type of test section was limited to measurements at inlet and at exit of the blade row. This information has been obtained either directly in the form of engine performance (e.g., Kofskey (1961), Kofskey and Nusbaum (1967), Szanca et al. (1974)) or from the detailed distribution of various quantities at the exit of the blade row (e.g., Hutton (1955), Haas and Kofskey (1977), Inoue et al. (1986), Schmidt et al. (1987)). Murthy and Lakshminarayana (1986), Lakshminarayana et al. (1987), Inoue and Kuroumaru (1988), and Kingcombe et al. (1990) have recently obtained detailed information on the leakage flow within compressor and turbine rotors using periodic

sampling with Laser Doppler Velocimeters, hot wires and high frequency response miniature pressure transducers.

A.2.2 Flow in Rotor Blade Passages

In this section the passage flow picture, in the absence of the tip clearance, is briefly presented so that the detailed discussions on the tip clearance flow given in later sections can be followed more easily.

From the secondary flow studies it is known that the endwall boundary layer rolls up into a vortex inside the passage (passage vortex) which, due to the pressure gradient in the passage, is located at the blade suction surface-endwall corner. Furthermore, due to the interaction of the endwall boundary layer flow with the blade leading edge, a horseshoe vortex is formed. The pressure side leg of this horseshoe vortex rotates in the same direction as the passage vortex, and Langston et al. (1977) proposed that this leg merges with the passage vortex. This was also observed in the flow visualization studies of Sieverding and Bosche (1983). The rotation of the suction side leg of the horseshoe vortex is the opposite of the passage vortex and Klein (1966) stated that it is wrapped around the passage vortex while Sjolander (1975) and Langston et al. (1977) suggested that it is located at the corner of the endwall and blade suction surface, i.e., underneath the passage vortex. The flow visualization studies by Sieverding and Bosche (1983) support Klein's (1966) model. In the light of these observations, the passage flow in the case of zero tip clearance can be presented schematically as shown in Fig. A.2.2.1.

In stationary blade rows, the above flow picture is seen both at the hub and at the tip. However, in unshrouded rotating blade rows, while the flow in the inner part of the passage is still as discussed in the previous paragraph, in the outer part, the flow picture is substantially different due to the tip clearance. The remainder of this review includes a detailed assessment of the studies which have improved the

Figure A.2.2.1 Endwall flow models by Klein (1966) and Langston et al. (1977) (reproduced from Sieverding (1985))

understanding of the flow in this region.

A.2.3 Tip Clearance Flow in Compressors and Turbines

Reduced blade height, blade unloading towards the tip, and mixing of the tip gap flow with the mainstream resulting in considerable total pressure losses are the significant negative effects of tip clearance which cause substantial reduction in the performance of the machine. In compressors, as a result of the decelerating flow through the blade passages, the tip clearance flow may have an even more significant effect on the aerodynamic losses and it can also affect the stability of the machine. This is probably the reason that most of the extensive tip leakage studies have been performed in compressors.

There are a number of factors to be considered when applying the observations obtained from tip clearance studies in decelerating flows to accelerating flows and vice versa. One of them is the fact that the endwall boundary layer will behave differently under accelerating and decelerating free stream conditions. Furthermore, on compressor blades the maximum load is generally concentrated near the leading edge while on turbine blades, loading tends to be distributed more or less uniformly over the blade length. One more difference, which, as far as the tip leakage flow is concerned, is probably the most significant one, is the relative endwall motion. In compressors and turbines this motion is in opposite directions and therefore, their effect on the leakage flow will be different (Fig. A.2.3.1). Nevertheless, tip clearance effects observed from accelerating flow stages can still provide insights into the clearance effects in decelerating flow stages and vice versa. In order to confirm this statement, Yokoyama (1961) performed tip clearance experiments with no endwall motion in two different cascades, one of compressor blades and one of turbine blades. The blade angles were set such that:

$$\alpha_1(\text{comp.}) = \alpha_2(\text{turb.}) ; \alpha_2(\text{comp.}) = \alpha_1(\text{turb.}) \quad \text{A.2.3.1}$$

(a)

(b)

Figure A.2.3.1 Endwall motion effect on the tip clearance flow
a) Turbine b) Compressor

Figure A.2.3.2 Velocity triangles in a) turbine Stage b) compressor Stage
(reproduced from Moustapha (1986))

The reason for this angle setting can clearly be seen in Fig. A.2.3.2. In each cascade a number of blades close to the centre were split at midspan so as to investigate the tip clearance effect without endwall boundary layer effects. The blade surface pressure distributions and spanwise lift coefficient distributions showed very similar trends for both cascades, confirming the similarity between tip clearance effects on passage flows in compressor and turbines in the case of no endwall motion.

A.2.4 Tip Clearance Studies Using Isolated Blades and Rectilinear Cascades without Endwall Motion

A.2.4.1 The Effect of the Tip Leakage Vortex on the Blade Suction Surface Pressure Distribution

The first extensive study with a rectilinear cascade appears to have been performed by a number of graduate students in the M.I.T. Gas Turbine Laboratory in 1953-1954. Using this accumulated data, Dean (1954) presented a detailed picture of the flow at a compressor blade tip. In the experiments, clearance sizes up to 10 percent of chord length were investigated and with increasing clearance size a continuous increase in the magnitude of the suction peak near the blade tip (caused by the tip leakage vortex) was observed. Furthermore, the experimental results indicated a continuous rearward shift of this peak with increasing clearance size. The same trend in the peak location has subsequently been observed by Khabbaz and Senoo (1961) in a compressor cascade where the endwall boundary layer effect was avoided by using a midspan gap, by Lakshminarayana and Horlock (1962) with a single airfoil split at midspan and by Sjolander and Amrud (1986) in a turbine cascade with tip clearance at one end of the blades.

In agreement with Dean's (1954) result, using a planar cascade of turbine blades, Sjolander and Amrud (1986) and Yaras (1987) observed an increase in the suction peak near the blade tip with increasing clearance size. This was also seen by

Lakshminarayana and Horlock (1962) using a single airfoil split at midspan. The enhancement of the suction peak is clearly an indication of the strength of the tip leakage vortex. For very large clearances one would expect this peak to reduce, since the finite wing configuration is approached. This is evident in the results of Lakshminarayana and Horlock (1962).

In some of the cascade tests multiple vortices at the blade tip can be seen from the presence of multiple peaks in the blade surface static pressure distribution near the blade tip (e.g. the work of Khabbaz and Senoo (1961) where a compressor cascade accommodating blades split at midspan was used and the turbine cascade study of Sjolander and Amrud (1986)). In the latter work, the presence of these multiple vortices was commented on by the authors. Detailed flow field measurements at various axial locations inside the blade passage will be necessary for verifying the possible presence of multiple vortices observed from blade pressures.

A.2.4.2 Spanwise Distribution of the Blade Lift Coefficient

Sugiyama (1975) used an isolated wing in a rectangular test section to determine the endwall boundary layer effect on spanwise lift and drag distribution on a wing with tip clearance. The suction side pressure distribution close to the wing tip indicated considerable variation with changes in the inlet endwall boundary layer displacement thickness while the effect on the pressure surface was negligible. Even though as the endwall boundary layer thickness is increased, the chordwise distribution of the suction surface pressure near the blade tip does not vary in a certain trend, the global effect was a considerable reduction in the local lift. This phenomenon has already been observed from secondary flow studies with no tip clearance. As was noted in Section A.2.2, the endwall boundary layer consisting of fluid with relatively low momentum rolls into a passage vortex after it enters the blade passage. Due to the pressure gradient in the blade passage, this vortex is located close to the suction side of the blade. Hence, as a result of the presence of the low energy fluid close to

the blade surface a local reduction in the blade lift will occur. As the inlet endwall boundary layer is thickened, the passage vortex becomes larger which simply means more substantial local reduction of the blade lift. From Sugiyama's results it is also seen that the presence of the tip clearance increases the spanwise blade portion over which the lift distribution is affected by the variations in the endwall boundary layer parameters. In the case of nonzero clearance, very close to the blade tip the lift rises rapidly before finally dropping as the tip is approached. More detailed discussion on this observation is given in the later part of this section. Sugiyama makes use of the concept of retained lift at the wing tip, first introduced by Lakshminarayana and Horlock, (1962) (see Section A.2.10). This is a hypothetical lift obtained graphically by neglecting the additional increase in the lift due to the clearance flow very close to the tip and extending the spanwise lift coefficient distribution curve to the wing tip (Fig. A.2.4.2.1). The author states that the parameter formed as $(c_{L\infty} - c_L)/(c_{L\infty} - c_{L(\text{tip})})$, where $c_{L\infty}$ is the lift under uniform, two dimensional flow conditions and $c_{L(\text{tip})}$ is the retained lift at the blade tip, does not vary with the angle of attack and endwall boundary layer displacement thickness and appears to be a function of the tip clearance height only.

The work of Khabbaz (1959) where the tip clearance effect is investigated using the mirror image technique, indicated that the loading slightly increases towards the blade tip. This has been observed by all investigators who obtained spanwise lift distributions from their experiments with nonzero tip clearance (eg. Lakshminarayana and Horlock (1962), Sugiyama (1975) and Sjolander and Amrud (1986)), (see Fig. A.2.4.2.1). Khabbaz (1959) explains this increase in the loading at the blade tip using a momentum flux theory (Fig. A.2.4.2.2). Since in compressors the pressure at the blade outlet (p_2) is higher than at the inlet (p_1), considering the control volume in Fig. A.2.4.2.2, the momentum flux into the volume is larger than the momentum flux leaving the volume. Khabbaz noted that this implies that some of the air must leave the cascade by passing through the passage so as to satisfy continuity. He states that presumably this additional fluid is passing through the blade row near the blade tip,

Figure A.2.4.2.1 Spanwise lift distribution at various gap-to-chord ratios
(reproduced from Lakshminarayana and Horlock (1962))

Figure A.2.4.2.2 Momentum flux theory suggested by Khabbaz (1959)

increasing the loading in that region. Using the same argument, he also notes that in the case of a turbine the loading at the tip should decrease with increasing clearance. However, from the turbine cascade experiments of Yokoyama (1961) and Sjolander and Amrud (1986), it is clear that the loading near the blade tip increases as the clearance size is increased. This clearly undermines Khabbaz's argument.

Furthermore, Yokoyama (1961), from his compressor cascade experiments (using blades split at midspan) observed that the flow angle and axial velocity at the blade row exit is reduced towards the blade tip, and Sjolander and Amrud (1986) from their turbine cascade experiments found that the increase in the loading towards the blade tip is in the axial component rather than in the tangential component. This again shows that the real flow picture at the blade tip cannot be explained with Khabbaz's (1959) argument. The increase in the loading very close to the blade tip is simply explained by the presence of the tip leakage vortex located close to the suction side, which ties in both Yokoyama's (1961) and Sjolander and Amrud's (1986) observations.

As the clearance is increased, the blade will approach the finite wing situation. In this case the unloading towards the blade tip is expected to increase. Lakshminarayana and Horlock's (1962) and Sugiyama's (1970) results clearly indicate this trend. However, as seen from Fig. A.2.4.2.1, for the clearance sizes seen in practice (i.e, up to about 4 percent of the blade height) this unloading is quite small. From this figure it is also seen that even at very large clearances the increased loading at the blade tip due to the tip leakage vortex is still present; its magnitude is, however, substantially reduced, indicating that the classical wing tip vortex situation is approached.

A.2.4.3 Compressibility Effects on Tip Clearance Flow

Apparently, the work of Moore et al. (1988) is the only one where the high Mach number tip gap flow is investigated. This is quite surprising since in practice

turbine rotors commonly operate at transonic regimes. The authors used a water table where a turbine tip gap is simulated with an arrangement similar to the single isolated blade split at midspan, discussed earlier. In order to determine the shock patterns in the gap, Johnson's (1964) hydraulic analogy between free surface liquid flow and 2D compressible flow was used. As a result of the over expansion at the pressure side corner of the blade tip, oblique shocks were observed inside the gap for blade suction surface Mach numbers larger than 0.78. As the shock wave hit the shroud wall, depending on the value of the blade suction side Mach number, either regular reflection or Mach reflection occurred.

A.2.5 Simulation of Relative Blade Motion Using Moving Belts

In the above discussed configurations the pressure difference across the gap is the dominant factor determining the tip leakage flow. However, when endwall motion is taken into account, the viscous drag of the wall also appears to be an important factor affecting the flow through the clearance (see Fig. A.2.3.1). Dean (1954) appears to have been the first to investigate endwall motion using a rectilinear compressor cascade with a moving endwall. His belt covered a region from 0.4 chord length upstream of the leading edge to the cascade exit. From his experiments he concluded that for the case of a compressor the endwall motion shifts the leakage vortex core towards the pressure side of the passage since the endwall motion increases the mass flow rate through the gap. However, he noted that this shift does not continue above a certain wall speed and the vortex remains at the same place. Peacock (1982) noted that this observation appears to contradict the results of Gearhart (1966) who performed experiments with a moving endwall and a single blade. Gearhart's results show that the mass flow rate through the gap increases with a rising rate as the endwall speed is increased. However, it should be noted that Dean's experiment includes the effect of the pressure gradient inside the passage which causes the passage vortex (secondary flow) to form. This secondary flow was absent in

Gearhart's experimental set up and as discussed in the next paragraph, probably explains the contradiction.

As was noted in Section A.2.2, in the absence of the tip clearance, the passage vortex is located at the blade suction surface-endwall corner. When the clearance is opened, due to the formation of the tip clearance vortex, this passage vortex is shifted towards the pressure side of the passage. Allen and Kofskey's (1955) visualization studies at the tip region of a turbine rotor clearly shows this interaction. Undoubtedly, the passage vortex will be a factor that prevents the tip leakage vortex from moving too far away from the suction surface. In fact, Lakshminarayana and Horlock's (1965) vorticity measurements at the exit of a compressor cascade indicate that the secondary flow causes the leakage vortex to be located nearer to the suction surface than observed when there was no endwall in the tip gap. This probably explains the behaviour observed by Dean.

Gearhart's (1966) study was mainly aimed towards reducing the cavitation problem in hydraulic turbomachine rotors. He distinguished between two types of cavitation. One of them occurs in the separation bubble at the pressure side corner of the blade tip while the other one is related to the tip leakage vortex (note that cavitation bubbles are a good way of visualizing the tip leakage vortex which was in fact used by Rains (1954)). In order to avoid the formation of a separation bubble on the blade tip he suggested rounding the pressure surface corner. For reducing the strength of the tip leakage vortex, which in turn will reduce the amount of cavitation in that vortex, he suggested a tip clearance gap which is slightly divergent. It should be noted that, while reducing the gap cavitation effect, the former geometric modification (i.e, rounding the corner) will cause a considerable increase in the tip clearance mass flow rate. However, the fact that whether this is undesirable or not is not entirely clear: it might be preferable to have somewhat increased mass flow rate through the tip gap if this fluid can be made to have a much lower total pressure loss. Hence, in order to arrive with the appropriate tip gap configuration, one has to

determine whether the majority of the losses occur inside the gap or as the leakage flow merges with the main stream. One more point worth mentioning is that, according to Gearhart (1966), the experiments performed by Shalnev (1958) indicated that the size of the separation bubble on the blade tip does not change with Reynolds number. However, this is not the case with the results of Yaras (1987); a reduction in Reynolds number caused an increase the bubble size at the location where it reaches its peak size (25 percent chord length from the leading edge).

Using a turbine cascade, Klein (1966) examined the effect of tip leakage and endwall motion on the low energy fluid region at the blade suction surface-endwall corner of the passage. He observed that when the clearance size is increased, the low energy fluid region (passage vortex) shifts towards the pressure side of the passage and the losses associated with this secondary flow are substantially reduced. The losses near the wall are further reduced with endwall motion which is due to the reenergizing effect of the endwall motion (viscous drag effect).

The most recent work with this type of configuration is that by Graham (1986), who performed experiments using high pressure axial turbine blades in a water analogy rig. This study is mainly oriented to the effects of tip clearance size and relative blade motion on blade unloading. He notes that, as a result of a weaker tip leakage vortex, the unloading at the blade tip is increased with endwall motion. At the engine equivalent endwall speed this increase in unloading is not very large; however, it appears to be very large for twice the engine equivalent endwall speed. As expected, the experiments indicated a reduction in the clearance flow with decreasing clearance size. On the other hand, it is noted that, regardless of the belt speed, the reduction of the leakage is nonlinear. Based on the results reported by Yaras et al. (1988), this can be attributed to the fact that the separation bubble size relative to the clearance height varies for different clearance sizes. As the belt speed was increased, a decrease in the leakage flow was observed. When the endwall

motion was introduced, the local increase in the loading at the blade tip appeared to be reduced which is clearly an indication of decreased vortex strength.

Although it was done with no endwall motion, the work of Bindon (1986) using a rectilinear turbine cascade is worth mentioning in this section. As was noted above, Gearhart suggested rounding the pressure surface corner of the blade tip in order to avoid the formation of the separation bubble and, therefore, the cavitation associated with it. In turbomachines with air as the working fluid the existence of the separation bubble has been recognized for some time, but surprisingly has received very little attention. Bindon (1986), is probably the first to investigate this phenomenon in detail. From his measurements he observed very high velocities at the pressure side corner of the blade tip. The author notes that this explains the burnout problem at the blade tip pressure surface corner seen in some existing turbines. Even though rounding the pressure surface corner may solve this problem, the author suggests the alternative of blanketing the corner with a cooling air film discharged radially near the tip of the pressure surface. The latter may be a better solution since it will not increase the tip clearance mass flow rate, as would be the case with the rounded corner, and may even decrease it by partly blocking the gap flow. In fact, the work of Ewen et al. (1973) on a small axial turbine rotor indicated that the blade tip cooling air injection provides an increase in efficiency because the tip leakage flow is effectively sealed off by the tip cooling air discharge.

As was noted in the discussion on Gearhart's (1966) work, the appropriate tip gap configuration will depend on where the majority of the losses associated with tip leakage occur. Bindon (1988), from experiments in a rectilinear turbine cascade with no endwall motion, determined the losses that occur inside the gap and the losses associated with the merging of the leakage flow with the main stream. While the determination of each of these loss components was subject to considerable error margins, it appeared that the mixing losses until the flow reaches the trailing edge plane are larger than the losses that occur inside the gap. Since, there will be

substantial mixing losses downstream of the trailing edge plane as well, it appears that the losses associated with merging of the leakage with main stream are considerably larger than the ones that occur inside the tip gap. This was confirmed by Dishart and Moore (1989) based on measurements in a linear turbine cascade. In that case, in order to reduce the losses associated with the tip leakage, it seems to be appropriate to focus on tip gap configurations that will reduce the leakage mass flow rate. This will also reduce the spanwise blade portion that is disturbed by the leakage vortex. Discussion on the investigated tip gap configurations to serve this purpose will be presented in Section A.2.9.

Finally, as an extension of Bindon's earlier studies with no endwall motion discussed above, Morphis and Bindon (1988) performed experiments using an annular turbine cascade with a rotating casing to simulate the relative endwall motion. The results indicated that the previous observations about the pressure distribution at the blade tip corner are also valid for the case with endwall motion. During these tests the authors also found out that rounding off the pressure side corner of the blade tip (radius = $2.5 \times$ clearance height) prevents the formation of the separation bubble. However, while this geometric modification will substantially reduce the aerodynamic losses inside the gap, considerable increase in the leakage mass flow rate will occur. According to the argument given in the previous paragraph, this should result in increased overall tip leakage losses.

A.2.6 Rotating Turbomachinery

When the relative motion of the blades is considered, in addition to the leakage vortex and passage vortex, a so called scraping vortex is formed (presumably this name is used because the vortex is formed by the scraping effect of the blade on the endwall boundary layer). Allen and Kofskey (1955), from their smoke flow visualization studies in a low speed turbine, observed the presence of this vortex. They also noted that even when the blade tip is out of the endwall boundary layer

region, the endwall boundary layer is rolled into a scraping vortex. Of course, in this case, the scraping vortex is a result of the motion of the pressure field rather than the physical presence of the blade itself. Phillips and Head (1980) also noted the presence of the scraping vortex from their visualization studies in an axial-flow compressor. It is important to note that in the case of a compressor the scraping vortex is on the pressure side of the blade and rotates in the same direction as the leakage vortex while in turbines it forms on the same side as the leakage vortex with a rotational direction opposite to the leakage vortex.

Commonly, the passage vortex formed by the endwall boundary layer under the effect of the passage pressure gradient rotates in the opposite direction to the leakage vortex. Hence, the two vortices do not merge with each other. However, Inoue and Kuroumaru (1988) note that their flow field measurements in a compressor rotor did not indicate the presence of a passage vortex. They explain this by the fact that the direction of the passage vortex rotation may be reversed due to the streamwise component of the vorticity at the inlet of the rotor. As a result of this, the two vortices will be rotating in the same direction, and hence, quite likely will merge with each other. According to the authors, the possibility of the secondary flow to rotate in the opposite of the expected direction was first pointed out by Horlock (1963).

Most of the information obtained within the blade row has been limited to flow visualization experiments since taking quantitative measurements within a rotating blade row is quite difficult. Recently, with the application of the latest technology to the measurement of the flow field (e.g. periodic sampling using Laser Doppler Velocimeter, hot wires anemometers and high response pressure transducers interfaced with complex data processors), quantitative measurement of the flow field inside the rotor passages has become possible (e.g. Murthy and Lakshminarayana (1986), Lakshminarayana et al. (1987), Inoue and Kuroumaru (1988), Kingcombe et al. (1990)). On the other hand, quite valuable quantitative information on tip leakage effects has also been obtained from the overall performance variation and from radial

surveys at the rotor exit. A review of all these studies is given in the following sections.

A.2.6.1 Effect of Tip Clearance on Performance

Data obtained from various radial and axial impellers indicate a roughly linear decrease in efficiency with increasing clearance size, as seen in Figures A.2.6.1.1 and A.2.6.1.2. These impeller data were summarized by Senoo and Ishida (1986) who derived a loss correlation for centrifugal and axial impellers. A similar linear variation is also seen in Fig. A.2.6.1.3 which includes mixed flow and axial flow turbomachines, compiled by Hesselgreaves (1969), for the purposes of checking the accuracy of his loss correlation (for the notation used on the horizontal axis of the graph, refer to Eqn. A.2.10.2.2). The data of Ewen et al. (1973) obtained from a small axial turbine indicated deviation from linearity in the efficiency variation with clearance size at very small clearances. The authors explain this deviation with the fact that at very small clearances the boundary layers on the blade tip and endwall comprise a very significant portion of the clearance, hence substantially altering the effective clearance size. An important feature observed from Figures A.2.6.1.1 to A.2.6.1.3, is that the data is not simply scattered but there are systematic differences between the results obtained from different machines at the same relative clearance. This implies that the tip clearance size is not the only factor affecting the tip clearance losses. One of these factors appears to be the degree of reaction at the blade tip, as discussed in detail in the following section.

Figure A.2.6.1.1

Efficiency variation with tip clearance, centrifugal impellers (reproduced from Senoo and Ishida (1986))

Figure A.2.6.1.2

Efficiency variation with tip clearance, axial impellers (reproduced from Senoo and Ishida (1986))

Figure A.2.6.1.3 Tip clearance loss correlation suggested by Hesselgreaves (1969)

A.2.6.1.1 Effect of Blade Tip Reaction on the Performance Variation With Tip Clearance Size

Kofskey and Nusbaum (1967) investigated the tip clearance effect on a two-stage turbine designed with 0.862 tip reaction. The results indicated a 3.7 points efficiency decrease for every 1 percent increase of the tip clearance to annulus height ratio. Furthermore, Szanca et al. (1974), who investigated the tip clearance effect on a research turbine with 0.805 tip reaction, quoted an efficiency reduction of 1.9 points for every 1 percent increase in the tip clearance. If these two results are compared to the 1.75 points efficiency drop for every 1 percent tip clearance increase obtained by Kofskey (1961) from an impulse design turbine stage, it is seen that increasing reaction at the blade tip increases the rate of efficiency reduction with increasing tip clearance. This is caused by the effect of increased relative velocity through the rotor on the mass flow rate through the tip clearance: high reaction at the blade tip results in a large pressure difference across the tip gap, thus causes higher kinetic energy associated with the tip leakage flow normal to the blade chord. The observations of Ewen et al. (1973) support this argument by showing an increased tip leakage as the reaction is increased.

A.2.6.2 Effect of Tip Clearance on Passage Flow Field Inside and Downstream of the Blade Row

Hutton (1955), from his measurements downstream of a single stage axial compressor impeller, noted that when the tip clearance size is increased, the pressure rise all along the blade is reduced, with the maximum drop being close to the tip region. He also investigated the variation of the axial velocity with tip clearance and concluded that increasing the clearance size reduces the axial velocity close to the tip, which was also seen by Yokoyama (1961) in his cascade tests. Furthermore, Hutton (1955) observed a significant increase in the axial velocity close to hub. This clearly shows that the region in which the flow conditions are affected by the tip clearance

extend over a substantial portion of the annulus height. Patel (1980) performed pressure and temperature measurements downstream of an axial gas generator turbine so as to determine the tip clearance effect on the local efficiency at various spanwise locations. As is seen in Fig. A.2.6.2.1, the radial variation of efficiency changes very significantly as the clearance size is increased, although the efficiency penalty is seen mainly between midspan and the casing. The results of Ewen et al. (1973) confirm these results. Recently, Schmidt et al. (1987) and Lakshminarayana et al. (1987) presented the spanwise distribution of mass averaged loss coefficient obtained downstream of isolated compressor rotors. The results were similar to the ones given by Ewen et al. (1973) and Patel (1980) for turbine rotors. From Schmidt et al.'s (1987) results it is observed that the peak loss region moves away from the casing as the clearance size is increased. This is due to the increase of the vortex core diameter as the clearance size is increased. Lakshminarayana et al. (1987) presented results for planes inside the blade passage as well. The results showed that the vortex starts to form at about quarter chord length from the leading edge and continues to grow as it approaches the blade row exit. The spanwise passage flow portion effected by the presence of the clearance seems to remain the same once the vortex flow exits the blade row.

Furthermore, from measurements at the exit of a two stage turbine, very close to the outer wall, Kofskey and Nusbaum (1967) observed a slight increase in total pressure with increasing clearance size. This is explained by the substantial underturning seen in this region: for large clearances the fluid close to the endwall passes through the stage with relatively small turning. Therefore, this fluid contributes little to the stage work output and retains much of its energy. The underturning was also observed by Davino and Lakshminarayana (1982), Szanca et al. (1974) and Haas and Kofskey (1977).

The studies of Inoue and Kuroumaru (1984) and Inoue et al. (1986) are noteworthy since very detailed vorticity measurements were performed at the exit of

Figure A.2.6.2.1 Effect of tip clearance on the radial distribution of efficiency (reproduced from Patel (1980))

the rotor. This provides detailed information about the relative sizes of the various vortex structures.

The measurements of Inoue and Kuroumaru (1984) were taken at the exit of an axial fan. The radial velocity distribution near the hub clearly shows the trailing vortex, while near the tip the leakage vortex is observed. The radial velocity distribution near the tip also indicates another relatively small vortex structure with the same rotational direction as the leakage vortex; this is the so called scraping vortex. Note that even though they appear to be located very close to each other, since the considered turbomachine is a fan, the scraping vortex is formed on the pressure side of the blade while the tip leakage vortex is on the suction side of the blade. The authors also note that the downstream measurements indicated higher turbulence quantities in the leakage vortex which they explain by the fact that turbulence energy is generated by the mean vortex motion.

The measurements by Inoue et al. (1986) at the exit of an axial compressor gave very interesting results regarding the location and the size of the tip leakage vortex. For the clearance sizes examined (up to 5 percent of the blade tip chord), with increasing clearance the vortex moves towards the pressure side of the adjacent blade. This apparently contradicts the result of Lakshminarayana and Horlock (1962), who performed experiments on an isolated airfoil split at midspan. Lakshminarayana and Horlock's downstream vorticity measurements indicated that for very small clearances the vortex is far from the blade suction surface, and it moves towards the blade as the clearance size is increased. This result was verified by the measurements of Lakshminarayana et al. (1987) within and downstream of an isolated compressor rotor. On the other hand, the observation of Inoue et al. (1986) is supported by the flow field measurements of Inoue and Kuroumaru (1988) inside the tip clearance and downstream of the blade row in an axial compressor rotor. Hence, it appears that this issue needs more investigation.

Inoue and Kuroumaru (1988) repeated their hot wire measurements inside the tip clearance of an axial compressor rotor for a thickened inlet boundary layer. The results indicated essentially no change in the structure of the tip leakage flow with changes in the endwall boundary layer displacement thickness. This is explained by the smoke visualization study of Sjolander and Amrud (1986) which showed that the majority of the endwall boundary layer fluid passes through the gap close to the leading edge and the rest of the gap flow is essentially free stream fluid.

Inoue and Kuroumaru (1988) also investigated the effect of solidity on the tip leakage flow. They found that as the solidity is increased the leakage vortex reaches the pressure side of the adjacent blade and interacts with its wake which is intuitively an expected result.

Finally, the casing static pressures measured by Inoue and Kuroumaru indicate that as the clearance size is increased the peak wall pressure location moves downstream. This observation is in agreement with the results of Yaras et al. (1988) who measured the endwall static pressure in a rectilinear cascade of turbine blades with no endwall motion. The blade pressures measured close to the tip by a number of investigators follow this trend as well (see Section A.2.4.1).

As far as the effect of relative endwall motion on the vortex location is concerned, from measurements inside and downstream of a compressor rotor, Lakshminarayana et al. (1987) note that due to relative endwall motion the leakage vortex is convected towards the pressure side of the blade passage. This is consistent with the results obtained from linear compressor cascades with relative endwall motion (see Section A.2.5).

A.2.7 Optimum Clearance Size

At a certain clearance size the tip vortex strength will just be enough to remove the low energy fluid from the endwall - suction surface corner and the passage and tip vortex strengths will be equal. It has been suggested that this clearance size represents an optimum clearance and that it depends on parameters such as tip leakage vortex strength and endwall boundary layer thickness (which will determine the strength of the passage vortex and the scraping vortex). Whether it is an accelerating or decelerating flow stage is also important, since it effects the direction of rotation of the scraping vortex and its location relative to the blade. The latter is significant as far as the interaction between the tip leakage vortex and the scraping vortex is concerned. Furthermore, for small clearances, besides the passage vortex, the vorticity associated with the horseshoe vortex also has to be considered in the determination of the optimum clearance size : the existence of the horseshoe vortex for clearances smaller than 1 percent was observed by Sjolander and Amrud (1986) and Bindon (1987). The presence of an optimum clearance is important because it shows that the smallest possible clearance is not always desirable. The first experimental data where the optimum clearance is observed is the data of Dean (1954) (Fig. A.2.7.1). Hubert's (1960) data also confirms the existence of an optimum clearance (Fig. A.2.7.1). From their compressor cascade experiments with strong secondary flow, Lakshminarayana and Horlock (1965) concluded that the optimum clearance size is 4 percent of chord length. Finally, based on compressor cascade tests, Storer (1989) suggested an optimum clearance size between 0.5 and 1 percent of the blade chord. Lakshminarayana (1970), based on his modelling of the passage flow in the blade tip region, gives an equation for determining the optimum clearance size based on the sum of the circulations associated with the tip leakage, passage and scraping vortices being equal to zero.

There is also some evidence of an optimum clearance in rotating machinery experiments. Hutton (1955), from his experiments with an axial-flow impeller, quotes

Figure A.2.7.1 Variation of loss coefficient with tip clearance (reproduced from Lakshminarayana and Horlock (1962))

an optimum clearance of 2 percent of the blade height for which the efficiency reaches its peak value.

Even though the above discussion indicates the possible existence of an optimum clearance, Senoo and Ishida (1986) presented the data obtained from seven different axial impellers (Fig. A.2.6.1.2) which shows that for practical clearance sizes the efficiency continuously improve as the clearance size is decreased. This implies that even though for some configurations an optimum clearance may exist, in order to draw more general conclusions, more research has to be done on the parameters that affect the optimum clearance size.

A.2.8 Tip Clearance Effect on Compressor Stall

As was indicated at the beginning of this chapter, the tip clearance flow in compressor stages is particularly important due to its influence on surge and stall. Therefore, a short review of the studies on the tip clearance effect on compressor stall is included.

McNair (1958) attempted to draw some general conclusions on the tip clearance effects on stalling pressure rise, using the available data in the literature. However, he found a large variation in the sensitivity of various compressors to tip clearance (Fig. A.2.8.1). He attempted to correlate the pressure rise variation with a number of parameters, and eventually concluded that if the slopes of the curves in Fig. A.2.8.1 (which appear to be nearly linear within the clearance range examined) are plotted against $S_t/(V_z/U_t)$ where S_t is the solidity at the tip, V_z the average axial velocity and U_t the blade tip speed, a fairly good correlation is obtained (Fig. A.2.8.2). However, this correlation should be used with caution since it is not supported by physical reasoning.

Figure A.2.8.1 Peak pressure rise vs. clearance area/annulus Area (reproduced from McNair (1960))

Figure A.2.8.2 Correlation suggested by McNair (reproduced from McNair (1960))

Khabbaz and Senoo (1961), from their experiments in a rectilinear cascade with a midspan gap, found that the presence of tip clearance is not harmful but in fact shifts the stalling at the blade tip to slightly higher angles of attack. This is presumably due to the fact that the tip leakage vortex energizes the suction surface boundary layer fluid close to the blade tip, thus delaying the separation. However, in real situations where the effects of endwall boundary layer and relative endwall motion are present, this will not be the case. As observed from Dean's (1954) results, obtained in a rectilinear cascade with endwall motion, and from the rotor exit measurements in an axial-flow compressor by Inoue et al. (1986), due to the viscous drag effect the endwall motion moves the tip leakage vortex away from the suction surface. As a result, the stall angle of attack at the blade tip will no longer be improved by the reenergizing effect of the tip vortex. Furthermore, in rotating machinery the oncoming endwall boundary layer will frequently be skewed, which will increase the angle of attack of the blade near the tip. This, of course, will cause the blade to stall first in the tip region.

One of the early studies in this field was performed by Hutton (1955) who investigated the tip clearance effect on an axial-flow impeller. The author states that the stall starts first at the tip, which is explained by the argument given in the previous paragraph. His results indicate that at small clearances the stall is sudden while at larger clearances it occurs more gradually and, interestingly, at relatively larger flow coefficients (Fig. A.2.8.3). However, as Jefferson and Turner (1958) point out, the occurrence of stall at higher flow coefficients as the clearance size is increased, appears to be the case for single stage compressors; for multistage compressors a reverse trend is seen. Hutton also noted that the hysteresis observed at small clearances disappears when the clearance size is increased (Hysteresis : the case where the blade would not unstall until a higher flow rate than that at which it had originally stalled). Huppert and Benser (1953) noted that hysteresis is seen in the case of rotating stall (i.e, a substantial portion of the blade span stalls simultaneously and is accompanied by a sharp drop in pressure ratio). Thus, it can be concluded that as the

Figure A.2.8.3 Effect of tip clearance on stalling and wall pressure rise (reproduced from Hutton (1955))

clearance size is increased the blade tip section stalls prior to the other sections of the blade, resulting in a progressive stall with gradual drop in the pressure ratio instead of rotating stall. This is explained by the fact that when the clearance is increased, the work done on the endwall boundary layer by the blades decreases in which case the adverse pressure gradient effect becomes dominant. The fact that at larger clearances the tip section stalls before the other sections implies a reduction in stall margin. This has been observed by a number of investigators. Phillips and Head (1980) performed flow visualization experiments in the tip region of a rotating blade row so as to understand the endwall boundary layer behaviour through the blade row. Their experiments showed that increasing the tip clearance causes stalling at a lighter blade loading. The same result was also obtained by Bettner and Elrod (1983) who, from their experiments in a highly loaded single stage compressor, concluded that a 12.5 percent decrease in the stall margin occurred when the tip clearance-to-span ratio was doubled. Furthermore, Moore and Osborn (1977) observed a similar trend from their experiments with a transonic fan stage.

A.2.9 Casing and Blade Tip Treatment

A.2.9.1 Casing Treatment in Axial Compressors

One of the most effective ways of improving the stall margin of a compressor is casing treatment, that is using a casing with a series of holes or slots above the blade tip.

Bettner and Elrod (1983), observed that at small clearances (about 1.5 percent of blade span) wall roughness causes a 6.5 to 7.5 percent reduction in the stall margin while for large clearances (about 3 percent of blade span), interestingly, a 2 to 4 percent stall margin improvement was observed with a rough endwall. This shows that the sensitivity of the endwall boundary layer to the casing treatment is dependent on the tip clearance size. From the transonic fan stage experiments, Moore and

Osborn (1977) observed that a casing accommodating skewed-slots over the blade row (Fig. A.2.9.1.1) increased the stall margin and it was increased further by opening the slots. The skewed-slot configuration was also one of the configurations examined by Takata and Tsukuda (1977). From their experiments in a number of low speed axial-flow compressors, the authors found that this treatment gave the best improvement in the stall margin. In effectiveness the axial-slot configuration followed the skewed-slot while a circumferentially grooved casing had only little improvement in stall margin. The treatments found useful by Takata and Tsukuda are given in Fig. A.2.9.1.2. However, unlike Moore and Osborn (1977), they concluded that with the application of the skewed-slot or axial-slot treatment the stall margin became uncoupled from the clearance size (i.e, changes in the clearance size did not affect the stall margin). From these results it is clear that, as a basis for choosing the most effective configuration, the mechanism through which the casing treatment leads to improvements in stall margin must be understood.

Realizing this, Takata and Tsukuda (1977) performed detailed hot wire measurements, including within the slots of the treated casing. From their measurements behind the rotor they observed that the application of effective casing treatments removes the low energy fluid from the blade tip region. This prevents the blade tip from stalling prior to the other sections of the blade. Thus, with the application of casing treatment, the stall is seen at a lower flow rate and when it occurs, the whole blade span stalls simultaneously resulting in a rotating stall. From their measurements within the slots they were able to give the mechanism that delays the stall at the blade tip as follows:

"A part of the fluid in the main stream near the rotor tip enters the treatment slots and the backchamber, being pushed by the pressure surface near the trailing edge of the blades at every blade passing. High static pressure around the outlet of the blade row also helps the flow go into the slots. The fluid moves within the slots or the chamber towards the upstream and then is discharged at every blade passing as a

Figure A.2.9.1.1 Casing treatment, skewed-slot (reproduced from Moore and Osborn (1977))

Figure A.2.9.1.2 Casing treatments (reproduced from Takata and Tsukuda (1977))

thick jet from the part of the slots facing the upstream part of the blade chord into the middle of the blade spacing."

From the above given flow picture, it is evident that the jet out of the slot helps removing the low energy fluid close to the blade tip region as a result of momentum interchange. In fact, Takata and Tsukuda note that the effect of this jet is seen over a substantial radial portion of the annulus. They also state that this jet out of the slot does in fact work on to the main stream and increases the total pressure in some regions which explains the reduced total pressure loss when casing treatment is used.

Now, in the light of the above given flow picture, the reason for the skewed-slot configuration to be effective while the circumferential grooves to be not, can easily be understood.

Various compressor casing configurations have been examined in attempts to increase the stall margin. Recently, however, recessed casings in compressors were examined in terms of their effects on stage efficiency (Wisler and Beacker (1989)). The authors proposed a configuration that provided a 0.5 point improvement in efficiency without adversely affecting the stall margin.

A.2.9.2 Casing Treatment in Axial Turbines

Because of the stalling problem in decelerating-flow stages, casing treatment has been used widely in fans and compressors. Even though stall is not a problem in accelerating-flow stages, casing treatment can be used to reduce the tip clearance losses.

One of the early studies in this area was done by Kofskey (1961), who tested different casing treatments in a single stage turbine. The configurations tested are

(a) Reduced blade height.

(b) Recessed casing

(c) Shrouded rotor

Figure A.2.9.2.1 Configurations investigated by Kofskey (1966)

Figure A.2.9.2.2 Effect of tip clearance on performance at design equivalent speed and pressure ratio (reproduced from Kofskey (1961))

given in Fig. A.2.9.2.1 and the resulting efficiency variations with increasing clearance height are given in Fig. A.2.9.2.2. The trend in efficiency with tip clearance is essentially the same as discussed in Section A.2.7.1. Apparently due to the reduced blade tip unloading and reduced mass flow rate through the clearance, the labyrinth type configuration shows the best performance. The author explains the performance improvement of the recessed-casing configuration, compared with the smooth wall, by the fact that the blade height is larger in the former case. It is worth noting that when the recessed-casing is used there should be a reduction in the tip mass flow and blade tip unloading, even though it will not be as large as for the labyrinth type configuration. One more point worth mentioning is that according to the experiments by Haas and Kofskey (1977) on the recessed-casing configuration, the case where the rotor tip diameter is the same as stator tip diameter gives better performance than when the blade is extended in the groove. This may be due to the fact that when the blade height is extended into the groove, the work done by the blade will not increase in proportion to its length, since the blade portion within the groove sees a flow with relatively less energy. Haas and Kofskey also state that whether the recessed-casing extends after the blade trailing edge or ends right after the trailing edge does not have any effect on the efficiency.

2.9.3 Blade Tip Treatment

This is one of the areas where, according to the available literature, not much work has been done. As is well known, one way of reducing the tip leakage effects to very low levels is to use shrouded rotors. However, this has its own disadvantages, such as additional weight and the resulting increased radial stresses in the blades. Realizing these disadvantages, investigators have examined the use of a partial shroud (winglet). This is simply a small plate of some shape attached to the blade tip (for example, see Fig. A.2.9.3.1). Beside of winglet's apparent advantages on performance, discussed in the following paragraph, for the first turbine stages the problem of burnout at the pressure side edge of the winglet can reduce the life of the

Figure A.2.9.3.1 Partial shrouded blade (reproduced from Patel (1980))

component. There might thus be a trade off between aerodynamic performance and component life.

Patel (1980) apparently conducted one of the first studies on the effects of winglets on aerodynamic performance. The experiments were performed with an axial gas generator turbine with the winglet shape shown in Fig. A.2.9.3.1. As was discussed in Section A.2.7.2, considerable underturning at the blade tip region was observed by a number of investigators (eg. Yokoyama (1961), Davino and Lakshminarayana (1982) and Takata and Tsukuda (1977)), indicating that in that region a relatively smaller portion of the fluid energy is converted into shaft work output. Patel's results (Fig. A.2.9.3.2) show that, with the exception of a very small region close to the casing, the underturning in the outer 50 percent of the annulus is substantially reduced. This indicates that with the use of winglets, the effect of the tip leakage flow on the pressure and suction side of the blade tip is considerably reduced. However, according to the results of Patel's study, unlike the casing treatments, the winglet does not reduce the rate of change of the efficiency with increasing clearance height.

It is also important to determine whether the presence of winglets reduces the mass flow rate through the tip gap and hence the losses associated with the tip leakage flow. This aspect was investigated to some extent by Booth et al. (1982), who performed a number of experiments, some of which examined winglets, discussed in the following paragraph.

The experiments were performed in a single-side discharge rig, and a water table cascade with turbine blades (Fig. A.2.12.1). The blade tip treatments examined during the discharge rig experiments are shown in Fig. A.2.9.3.3. The authors noted that the grooved blade tip configuration appeared to be the most effective one since it had the lowest discharge coefficient (Fig. A.2.9.3.4). However, since the discharge coefficient is based on the actual pressure drop across the tip, using the discharge

Figure A.2.9.3.2 Effect of partial shrouds on the radial distribution of the stage exit swirl angle (reproduced from Patel (1980))

Figure A.2.9.3.3 Blade tip treatments investigated by Booth et al. (discharge rig experiments) (reproduced from Booth (1985))

Figure A.2.9.3.4 Results of the discharge rig experiments (reproduced from Booth (1985))

coefficient as a basis for comparing the configurations may give misleading results because a reduction in the leakage flow due to reduced pressure drop across the gap will not reflect in the value of the discharge coefficient. This fact was considered in the water-table cascade experiments, where the optimal tip treatment was further investigated: this time, in the analysis of the results the authors used the dimensional value of the leakage flow as a basis for comparison. The results indicated that the winglet is somewhat superior to the grooved configuration in reducing the leakage flow, since in addition to the reduced discharge coefficient, the driving pressure difference across the gap is also reduced. The results for the configurations shown in Fig. A.2.9.3.5 are given in Fig. A.2.9.3.6.

Wadia and Booth (1982) further continued the study on blade tip treatments both experimentally and analytically. The calculation scheme to determine the discharge coefficient for various tip treatment configurations analytically was simply the solution of the two dimensional incompressible laminar flow through the gap using time dependent finite difference equations. Except for the case of no tip treatment, the experimental discharge coefficients for various blade tip treatments were not predicted very accurately. As the authors noted, this might be a result of a number of shortcomings such as an accurate modelling of the singularities at the corners within the gap. Nevertheless, as the authors pointed out, this analysis can be taken as a starting point and be improved by eliminating the present limitations.

Figure A.2.9.3.5 Tip configurations investigated by Booth et al. (water-table cascade experiments) (adapted from Booth (1985))

Figure A.2.9.3.6 Results of the water-table cascade experiments (reproduced from Booth et al. (1982))

2.10 Tip Clearance Loss Analyses

In Section A.2.6.1 the tip clearance effect on performance was discussed. As was seen from the results obtained from various turbomachines, the variation of the efficiency drop due to tip clearance with clearance size is close to linear. Therefore, it appears to be possible to give a simple correlation based on geometry only, i.e.:

$$\Delta\eta = -C_1 \left(\frac{t}{c} \right) \quad \text{A.2.10.1}$$

where C_1 is a constant. However, as was discussed in A.2.6.1, this constant which represents the sensitivity of stage efficiency to changes in the clearance size varies from one turbomachine to another, indicating the presence of factors other than the clearance height affecting the efficiency loss due to tip clearance. This clearly shows the necessity of more advanced models that account for all of the parameters effecting the losses due to tip clearance. The studies that have been done in this area are summarized in this section.

As far as the analysis of the tip clearance losses is concerned, Betz (1926) appears to have been one of the first to perform detailed studies in this area. In his analysis he evaluated the induced drag from the shed vortices at the blade trailing edge produced as a result of the gradient of the circulation distribution. Prandtl commented on this analysis and noted that the formation of the tip leakage vortex, i.e., the curling of the vortex sheet towards the blade tip, should be included. In reply to this comment, Betz stated that assuming a flat vortex sheet or a sheet that is curled towards the blade tip should not make a difference as far as the lost energy is concerned. However, Betz's model implies that the flow direction in the tip clearance is parallel to the chordline which can only be valid where the blade aspect ratio and the tip clearance to chord ratio are large. Therefore, this analysis may give reasonable results only for fans with large blade aspect ratio and clearance size.

Rains' (1954) work is probably the first study where a reasonably accurate modelling of the flow and a good correlation of the losses were achieved. In fact, many subsequent studies were essentially improvements of his analysis. Therefore, a detailed summary of his work is given in the following paragraphs.

Rains (1954) stated that the application of the lifting line theory for predicting the tip clearance losses is not adequate. This is because the flow on a wing of high aspect ratio is not an adequate approximation to the flow on a blade of small aspect ratio with a tip clearance which is only a few percent of the chord length. He stated that for a blade of small aspect ratio with tip clearance, the vorticity is mainly shed from the tip along the chord rather than from the trailing edge. Therefore, he suggested applying the Kutta condition to the edge of the clearance rather than to the trailing edge.

Since the pressure gradient along the blade is considerably small relative to the one across the clearance, the acceleration of the tip leakage flow can be assumed to be in the direction normal to the chord only. Based on this assumption Rains gave an expression for the calculation of the velocity component normal to the blade chord, which is simply Bernoulli's equation applied between the conditions far upstream and downstream of the tip gap :

$$V_N = \left(\frac{2(P_1 - P_2)}{\rho} \right)^{1/2} \quad \text{A.2.10.2}$$

(for nomenclature refer to Fig. A.2.12.3). Note that the above expression is valid only for the case where viscous forces are negligible. Rains, from his experiments with a single hydrofoil where endwall pressure measurements were taken along a contour following the blade shape, noted that this perfect fluid model where viscous forces are not taken into account is valid for gap to blade thickness ratios larger than 0.07. For clearances smaller than this, the pressure difference across the gap is almost

completely balanced by the viscous forces. Certainly the Reynolds number must also have some effect on the maximum clearance size where viscous effects start to become dominant. Considering this point, Rains derived a more general parameter to be used for determining the ranges where the viscous and/or inertia forces are dominant: in order to have the same dimensionless blade tip pressure distribution from one Reynolds number to another, the nondimensional clearance velocity must be constant. This leads to the fact that the viscous pressure drop across the gap has to remain fixed also. Thus, calculating the viscous pressure drop from laminar flow analysis, Rains arrived at the following parameter :

$$L = e^2 \epsilon Re \quad \text{A.2.10.3}$$

where e is the gap height to maximum blade thickness ratio, and ϵ is the maximum blade thickness to chord ratio at the blade tip. Based on his measurements, Rains concluded that for cases where this parameter is larger than 125, inertia forces are dominant, while for values smaller than 11 a pure viscous solution is required. As a second approach he derived an equilibrium laminar flow model which further confirms the ranges where friction and inertia forces are dominant. The above discussion implies that for the cases where this parameter is between 11 and 125, both inertia and viscous forces are effective. For this range Rains introduced another equation to be used for gap velocity calculation. This expression is derived from the Bernoulli equation with added loss terms that take into account the losses at the gap entrance (i.e., the separation bubble) and the turbulent or laminar friction loss within the gap. The resulting expression is as follows :

$$\frac{V_N}{V_\infty} = \left[\frac{\Delta C_p - \left(k + 2 \frac{\tau_i}{l} C_f \right) \left(\frac{V_N}{V_\infty} \right)^2}{1 + k + \left(2 \frac{\tau_i}{l} C_f \right)} \right]^{1/5} \quad \text{A.2.10.4}$$

where k is a constant that depends on the sharpness of the entrance corner of the clearance ($0 \leq k \leq 0.25$). Depending on whether the flow is laminar or turbulent, the skin friction coefficient, C_f , is calculated from one of the following equations:

for laminar flow approximation,

$$C_f \equiv \frac{0.078}{\left[\frac{t}{V} (V_T^2 + V_\infty^2 \Delta C_p)^{0.5} \right]^{0.25}} \quad \text{A.2.10.5}$$

for turbulent flow approximation,

$$C_f \equiv \frac{12\nu}{t[V_T^2 + V_\infty^2 \Delta C_p]^{0.5}} \quad \text{A.2.10.6}$$

Rains (1954) also analyzed the effect of the relative endwall motion and from his experiments he concluded that the viscous drag effect of the endwall motion cannot be analyzed separate from the pressure difference, since the pressure field is considerably modified by the endwall motion. Therefore, he derived an expression applicable to clearance flow velocity calculation for the cases with endwall motion. The starting equation for this derivation was again Bernoulli's equation with added loss terms mentioned in the previous paragraph. The resultant expression is as follows:

$$\frac{V_N}{V_\infty} = \left[\frac{\Delta C_p + \left(\frac{V'}{V_\infty} \right)^2 - (k + 2C_{f_t} \frac{\tau_t}{t}) \left(\frac{V''}{V_\infty} \right)^2}{1 + k + 2C_{f_t} \frac{\tau_t}{t}} \right]^{0.5} \quad \text{A.2.10.7}$$

where V' is the component of the free stream velocity normal to the blade and V'' is the component along the blade. C_t is still defined as in Eqns. A.2.10.5 and A.2.10.6 with the exception that in those expressions V_T is replaced by V'' . It is worth mentioning that while Eqn. A.2.10.7 is quite similar to Eqn. A.2.10.4, it is not restricted to a certain clearance range. This implies that with relative endwall motion the viscous forces become more significant. However, Rains noted that the errors caused by neglecting the viscosity terms from Eqn. A.2.10.7 do not exceed 20 percent, which indicates that for practical clearance sizes inertia forces are still dominant even when endwall motion is taken into account.

Based on these tip clearance flow computations, Rains (1954) introduced an expression for the tip clearance loss in which he assumed that the kinetic energy associated with the velocity component normal to the blade chord cannot be recovered. Even though he recognized that frictional effects within the gap also have some effect on the losses, he assumed them to be negligible for the clearance sizes seen in practice.

$$\Delta E = \frac{2 \left(\frac{t}{r_t} \right)}{\left(1 - \left(\frac{r_h}{r_t} \right)^2 \right) \cos(\beta_t^*)} \left[\frac{S_t}{c_t} \frac{\phi \psi}{\cos(\beta_t^*)} \right]^{1/2} \times \left[1 + \frac{c_t \cos^3(\beta_t^*)}{S_t \phi \psi} \left(\frac{t}{2\delta^*} \right) \right]^{1/2} \quad \text{A.2.10.8}$$

Comparison of the theoretical results with the experimental ones from his pump experiments and from another reference fan showed very good agreement. The data covered the ranges where inertia and viscous forces are both important ($11 < L < 125$) and the range where inertia forces are dominant ($L > 125$)).

Rains (1954) also developed a model for the flow outside the tip clearance where he considered the rate of rolling up of the vortex. The model, which is based on assumptions such as constant lift along the blade chord and inviscid flow, enables the calculation of the angle between the vortex axis and the chord line:

$$\beta_v = \tan^{-1} \left(\frac{4C_{L_t}^{1/2}}{4 - C_{L_t}} \right) \quad \text{A.2.10.9}$$

the circulation associated with the vortex core:

$$\Gamma = e\epsilon \left(\frac{4 + C_{L_t}}{C_{L_t}} \right)^{1/2} t^{*0.72} \quad \text{A.2.10.10}$$

and the diameter of the core at a desired location:

$$\frac{r_c}{t} = 0.14 t^{*0.85} \quad \text{A.2.10.11}$$

where t^* , the dimensionless time used in the above equations, is defined as follows:

$$t^* = \frac{\left(\frac{z_v}{c_t} \right) (C_{L_t})^{1/2}}{e\epsilon} \quad \text{A.2.10.12}$$

Undoubtedly, one can argue on the validity of some of the assumptions used in this model. For example, constant lift distribution along the chord is probably not a very realistic assumption. Furthermore, as Rains himself pointed out, the size of the vortex is only a rough estimation because it is based on a stiff tip clearance jet assumption: since the jet velocity and free stream velocity magnitudes are comparable, "stiff" jet is not a good approximation for the tip clearance flow. Even though, as the author stated, this model could be improved, as will be discussed later in this section,

in some of the recent prediction methods the size of the vortex is still obtained from this analysis.

Finally, it should be noted that all of Rains' study discussed so far is restricted to compressor and pumps, thus its validity for turbines is still to be investigated.

2.10.1 Tip Clearance Loss Models Proposed As An Equivalent Drag Coefficient

Vavra (1960) introduced a simplified loss prediction method for axial compressors which uses the tip clearance velocity expression for stationary blade case, given by Rains (1954). Again, following Rains' argument, it is assumed that the tip clearance loss is the result of the kinetic energy normal to the blade chord not being recoverable. In the analysis a triangular blade loading is assumed, which is a reasonable approximation, particularly for compressor blades. However, even though the resulting expression is simpler than Rains' loss expression, it includes a contraction factor of the flow through the clearance (k) and another parameter that takes into account the flow resistance (Ω) in the clearance. These have to be determined experimentally. The power loss is expressed as follows:

$$\Delta E = \frac{2\sqrt{2}}{5} k \Omega^3 \rho \pi c V_{\infty}^3 C_L^{3/2} \quad \text{A.2.10.1.1}$$

The author continued his analysis so as to express this loss in terms of a drag coefficient; the result is as follows:

$$C_{D,t} = \frac{4\sqrt{2}}{5} k \Omega^3 \frac{t}{h} C_L^{3/2} \quad \text{A.2.10.1.2}$$

For the constants k and Ω Vavra suggests the values 0.5 and 0.8 respectively. (Thus, the resultant constant in Eqn. A.2.10.1.2 is 0.29). The same expression was also used

by Fujie (1962): based on his experimental data, Fujie suggested expressions for the profile, secondary flow and annulus drag coefficients. On the other hand, he pointed out that the tip clearance losses are affected by many variables, thus his data was not enough to derive an expression for these losses. For these losses he suggested Vavra's formula to be used. However, as Lakshminarayana and Horlock (1962) showed (Fig. A.2.10.1.2), this expression does not provide very good results.

Lakshminarayana and Horlock (1962) gave an expression where the tip clearance losses are predicted again in terms of an equivalent drag coefficient. The authors note that their expression is applicable to small clearance to chord ratios: they suggest that for small values of the clearance to chord ratio, unlike the finite wing case, some of the lift is retained at the blade tip as a result of viscous effects. This means that only a portion of the vorticity is shed at the blade tip. Therefore, they state that using the common lifting line theory, which assumes zero retained lift at the blade tip, is not adequate, particularly for small clearance to chord ratios.

Lakshminarayana and Horlock (1962) thus had to determine the amount of retained lift at the blade tip. They obtained this information by extrapolating the spanwise lift distribution curves to the blade tip. The retained lift obtained in this way was then nondimensionalized by the two-dimensional lift coefficient for the blade shape and was called the "retained lift coefficient". This retained lift coefficient together with the mean value of the blade bound vortex circulation enabled the determination of the amount of vorticity shed from the blade tip, which was then used in the calculation of the induced drag.

In their induced drag model Lakshminarayana and Horlock (1962) assumed constant lift along the blade span, which implies that there is no vorticity shed from the trailing edge. It should be noted that this is a reasonable approximation for small aspect ratio blades only. Fig. A.2.10.1.1 schematically shows the model used for this analysis. The authors compared the predicted induced drag values with the

Figure A.2.10.1.1 Model used by Lakshminarayana and Horlock (1962) for induced drag prediction

Figure A.2.10.1.2 Comparison of the experimental and theoretical values of $C_{D,t}$ at low gap-to-chord ratios (reproduced from Lakshminarayana and Horlock (1962))

experimental ones which they obtained from a single aerofoil split at midspan. As can be seen from Fig. A.2.10.1.2, the agreement is quite good particularly when compared to the accuracy obtained with Vavra's (1960) analysis.

In 1965, based on their cascade experiments, the same authors introduced another expression for the induced drag coefficient. This analysis differs from the previous one in that the induced velocity is obtained from a different vortex system and, for better accuracy, the diameter of the tip vortex (determined experimentally) is taken into account when determining the limits for the spanwise integration of the induced drag. Furthermore, the authors observed that for moderate clearance-to-chord ratios the variation of the induced drag with the passage aspect ratio, which appears as a parameter in the expression, is quite small. Thus, as a result of simplifications, the induced drag coefficient was expressed as a function of the blade lift coefficient, blade aspect ratio and the retained lift coefficient only (Fig. A.2.10.1.3). The authors noted that for clearance sizes seen in turbomachinery practice, the values of the retained lift coefficient can be approximated to be 0.5. Thus, their expression reduces to:

$$C_{D,i} = 0.7 \left(\frac{t}{s} \right) \frac{C_L^2}{A} \quad \text{A.2.10.1.3}$$

The authors noted that the agreement between the results of this equation and their experimental data is reasonably good.

The expressions discussed so far, were suggested for compressors. Ainley and Mathieson (1951) gave an expression to be used for evaluating the tip leakage losses in turbines. The expression is as follows:

$$C_{D,i} = 0.5 \left(\frac{t}{s} \right) \frac{C_L^2}{A} \quad \text{A.2.10.1.4}$$

As is seen, this expression is very similar to that of Lakshminarayana and Horlock (1965) with the exception of the value of the constant (Ainley and Mathieson noted

Figure A.2.10.1.3 Variation of $C_{D,t}$ with tip clearance
(reproduced from Lakshminarayana and Horlock (1965))

that this constant is suggested for plain tip clearance and should be replaced by 0.25 for shrouded blades).

The expression proposed by Meldahl (1941) differs from the ones discussed so far by the fact that the tip clearance drag coefficient is also related to the flow angle at the exit of the blade row :

$$C_{D,t} = 0.25 \frac{1}{\sin(\alpha_2)} \left(\frac{t}{s} \right) \frac{C_L^2}{\bar{A}} \quad \text{A.2.10.1.5}$$

All the analyses discussed so far assume a linear dependence of tip clearance loss on clearance size. Dunham and Came (1970) noted that the cascade data of Hubert (1960) suggested a slight deviation from this linearity. Thus, they modified the expression given by Ainley and Mathieson (Eqn. A.2.10.1.4) and proposed the following expression:

$$C_{D,t} = 0.47 \frac{c}{s} \left(\frac{t}{c} \right)^{0.78} \frac{C_L^2}{\bar{A}} \quad \text{A.2.10.1.6}$$

For shrouded blades the constant should be replaced by 0.37. The authors noted that this expression showed good agreement with turbine results as well.

Comparison of some of the loss models discussed in this section is given in Fig. A.2.10.1.4.

2.10.2 Tip Clearance Loss Models Introduced in Terms of Percentage Reduction in Efficiency

In some of the tip clearance loss prediction studies the tip clearance effect has been expressed in terms of a percentage decrease in the efficiency of the stage.

Figure A.2.10.1.4 Variation of $C_{D,t}$ with tip clearance for a number of proposed models (reproduced from Lakshminarayana (1985))

Based on Vavra's analysis, Hesselgreaves (1969) derived this type of an expression: as is known, the reduction in the work coefficient as a consequence of the tip clearance flow is caused by three factors:

- i) Increase in drag due to the clearance flow;
- ii) Reduction in blade loading;
- iii) Reduction of blade size.

Hesselgreaves noted that if the efficiency at zero clearance is sufficiently high and the work coefficient reduction due to (i) is comparable to the sum of the reductions due to (ii) and (iii), the decrease in efficiency can be written in terms of (i) only:

$$\Delta\eta \equiv \eta_o \frac{\Delta\psi_o}{\psi_o} \quad \text{A.2.10.2.1}$$

Then using the power loss expression given by Vavra (1960) (with the exception of flow resistance and gap contraction factor constants) he derived the following relation:

$$\Delta\eta \equiv \left(\frac{t}{h}\right) f \quad \text{A.2.10.2.2}$$

where,

$$f = \eta_o \frac{2\sqrt{2}}{5} \frac{1}{\psi_o \phi} \left(\frac{V_\infty}{U_t}\right)^3 \frac{2S_t C_L^{3/2}}{1 + \frac{r_h}{r_t}}$$

is referred to as a sensitivity factor.

As was shown in Fig. A.2.6.1.3, the agreement between the predicted efficiency drops and experimental results is reasonably good. The author also commented on the triangular blade loading assumption used by Vavra: he stated that assuming a uniform loading instead of a triangular one causes only a reduction of 11 percent in the power

loss due to tip leakage. Thus he concluded that the type of assumed loading is apparently not very critical.

Another loss correlation was suggested by Lewis and Yeung. The authors used the power loss expression given by Vavra (1960) (with the exception of flow resistance and gap contraction factor constants). In their correlation the tip leakage loss is nondimensionalized as follows:

$$TL = \frac{\Delta E}{\rho C_x \pi (r_t^2 - r_h^2) \Delta P_o} \quad \text{A.2.10.2.3}$$

where ΔP_o is the total pressure rise through the stage. Schmidt et al. (1987) noted that even though Hesselgreaves' and Lewis and Yeung's models are both developments of Vavra's model, one of them overpredicted their experimental results while the other underestimated them. This clearly shows that the procedure used to nondimensionalize the power loss due to tip leakage has to be selected carefully.

Another study where the tip clearance losses are expressed in terms of a decrease in efficiency is the one of Amann et al. (1963). Based on the following assumptions, Amann et al. proposed the expression given in Eqn. A.2.10.2.4 to be used for turbines.

- The mean angular momentum changes are representative of the overall stage performance.
- The whirl at the turbine exit is very small.
- The stator has zero clearance.
- In the tip clearance section the flow is essentially undeflected.

$$\Delta \eta = \frac{-2\Gamma r_f}{A_{throat}} \eta_o \quad \text{A.2.10.2.4}$$

With a reasonable approximation this expression can be simplified as follows:

$$\Delta\eta = \frac{-r_t}{r_m \cos(\alpha_2)} \eta_o \frac{t}{h} \quad \text{A.2.10.2.5}$$

Based on experimental investigations in a small turbine, Rogo (1968) noted that the above given expression predicts the experimental results (at design speed and work conditions) quite accurately, provided the clearance size is not large.

Kacker and Okapuu (1982) also suggested Amann's expression with the exception that they reduce the result obtained from Eqn. A.2.10.2.4 by 7 percent. They noted that the results correlated with most of the available experimental data within 15 percent.

As was discussed in Section A.2.6.1.1, degree of reaction at the blade tip appears to be an important parameter affecting the magnitude of the tip clearance losses. Roelke (published by Glassman (1973)) compared some of the experimental results available in the literature to the predicted ones using the correlation given by Hong and Groh (1966) (Fig. A.2.10.2.1) where the effect of reaction on the tip clearance losses is taken into account. (In the figure, W and V are the relative and absolute velocities respectively). Roelke noted that the predictions were reasonably accurate for clearance sizes seen in practice.

Another correlation for the tip clearance losses was proposed by Craig and Cox (1970-71). The authors considered a special application where a sudden enlargement is present between stationary and moving blade rows (see Fig. A.2.10.2.2). This application is frequently seen in steam turbines. While there is a loss due to the sudden enlargement, the static pressure reduction caused by the enlargement will result in a considerable decrease in reaction, thus reducing the losses due to the tip gap flow. The correlation is as follows:

Figure A.2.10.2.1 Effect of reaction on the efficiency variation with tip clearance (reproduced from Glassman (1973))

Figure A.2.10.2.2 Variation of the efficiency debit factor with the rotor tip leakage coefficient (reproduced from Craig and Cox (1970-71))

$$\Delta\eta = \frac{-1.5F_k r_t}{r_m \cos(\alpha_2)} \eta_o \frac{t}{h} \quad \text{A.2.10.2.6}$$

where F_k is a factor that accounts for the effect of reaction and recess on the tip clearance loss. The correlation for F_k is given in Figure A.2.10.2.2. As is seen, at zero reaction the efficiency drop due to tip leakage appears to be nonzero. The authors explain this with the fact that some windage loss from the shroud band still remains at zero clearance.

As a continuation of the work by Lakshminarayana and Horlock (1965), Lakshminarayana (1970) introduced a semi-empirical expression for calculating the efficiency drop due to tip clearance effects. He noted that the losses presented in the form of an induced drag coefficient do not account for all of the losses since there is substantial spanwise flow in the blade surface boundary layer near the tip. Therefore, he also included this effect in his semi-empirical expression for efficiency decrease due to tip leakage. The resultant expression is as follows:

$$\Delta\eta = \frac{0.7\psi}{\cos(\alpha_m)} \left[1 + 10 \left(\frac{1}{\cos(\alpha_m)} \frac{\phi}{\psi} \frac{h}{c} \frac{t}{h} \right)^{1/2} \right] \frac{t}{h} \quad \text{A.2.10.2.7}$$

The author compared a number of experimental results for pumps, compressors and turbines with predicted values (Fig. A.2.10.2.3), which showed quite good agreement. However, as can be seen from Fig. A.2.10.2.4, the results of one recent study (Inoue et al. (1986)) are not predicted very accurately. Lakshminarayana (1970) noted that even though his comparisons gave good agreement, there are a number of parameters that are not included in the correlation, such as blade thickness at the blade tip, Reynolds number and endwall boundary layer thickness, which are likely to influence the losses. The disagreement seen in Fig. A.2.10.2.4 might be a result of this. Schmidt et al. (1987) compared their experimental results obtained from an isolated compressor rotor to the prediction of Lakshminarayana's model. The model

a)

b)

c)

Figure A.2.10.2.3 Stage efficiency variation with clearance size
 a) axial-flow compressors b) axial-flow fans
 c) axial pump and turbine
 (reproduced from Lakshminarayana and Horlock (1970))

Figure A.2.10.2.4 Efficiency variation with tip clearance (adapted from Inoue et al. (1986))

overpredicted the efficiency. Considering the fact that the model was able to predict some cases available in literature with similar flow coefficients to the ones examined by Schmidt et al. (Fig. A.2.10.2.3), Schmidt et al. concluded that Lakshminarayana's model must be sensitive to stage loading. Furthermore, the authors noted that while Lakshminarayana's retained lift coefficient is a function of gap height only, their data clearly shows that the lift at the blade tip is significantly affected by the flow coefficient. This may be another reason for the model not to be able to predict every test case accurately. Finally, as part of this study, Schmidt et al. (1987) suggested some modifications to the loss prediction models of Hesselgreaves (1969), Lakshminarayana (1970) and Lewis and Yeung (1977). However, the modifications are in the form of constants substituted into the models without sufficient justification.

Another correlation for efficiency reduction with tip clearance in axial compressors was proposed by Senoo and Ishida (1986). It seems to be the most complex correlation as far as the number of parameters included into the expression is concerned. Unlike the models discussed so far, it is necessary to know the conditions at zero clearance and the loss coefficients due to causes other than the tip leakage in order to use the correlation.

In a recent study, Moyle (1988) pointed out that the available data obtained from various compressors clearly indicate that the sensitivity of the efficiency to tip clearance size varies substantially from one case to another (e.g., see Fig. A.2.6.1.2). Realizing the complexity of the proposed models so as to account for this sensitivity, he proposed another approach described in the following paragraph in an attempt to arrive with a simplified correlation.

Most of the efficiency versus clearance size data available in literature was derived at constant flow coefficient. And since the flow coefficient is a significant factor effecting the sensitivity, this resulted in near-linear curves with a different slope for each compressor tested. The author stated that this can be avoided if the

efficiency drop with clearance size is analyzed at constant shaft power input rather than constant flow coefficient. Since for most of the test cases available in literature, the given information was not enough to reobtain the efficiency drop at constant shaft input power, the author used the changes in peak efficiency with clearance size as a substitute to that approach. The results are given in Fig. A.2.10.2.5. It is observed that the data obtained from various compressors seems to fall on a single curve. The author also suggested a correlation to this curve as follows:

$$\Delta\eta = 0.214(t/h)^{0.227} - 0.005 \quad \text{A.2.10.2.8}$$

As is seen, the gap height is the only variable in the correlation. Hence, it appears that after all, in affecting the sensitivity of efficiency to tip clearance variations, the geometry and flow parameters may not be as important as it was thought they were. Another important observation from Fig. A.2.10.2.5 is that the sensitivity of efficiency to clearance size is not constant; higher sensitivity is present for smaller clearances. In some of the data presented at constant flow rate this could be observed as well. But it seems that, in general, the clearance effect on efficiency determined at constant flow rate does not reflect this nonlinear character very well.

Figure A.2.10.2.5 Peak efficiency variation with tip clearance (reproduced from Moyle (1988))

2.11 Prediction of Blade Surface Pressures Near the Blade Tip

In the case of small aspect ratio and finite clearance size, the suction surface pressure distribution is substantially affected by the presence of the tip leakage vortex. Therefore, it is not possible to use the usual lifting surface theory for predicting the pressure distribution on a turbomachinery blade. Yokoyama (1961) introduced a quite simple method for determining the suction surface pressure distribution. In his calculation he assumed a linear variation in bound vorticity along the chord line, which corresponds to a triangular chordwise lift distribution. The tip leakage vortex is assumed to be a continuation of the bound vortex which implies that none of the vortex lines cross the gap between the blade tip and the endwall (i.e., no retained lift at the blade tip is assumed). Two different models were used for the vortex shedding at the blade tip. In one of them the vortices are taken to be parallel to each other and are aligned with the main stream direction. In the other one, they join a vortex which starts from the leading edge and is in the main stream direction (see Fig. A.2.11.1). Then, the suction surface velocity is determined by superimposing the velocities induced by the bound vortex and the tip leakage vortices. Comparison of the results with experimental data showed reasonable agreement for both spanwise and chordwise pressure distributions particularly when the tip vortices were assumed to join the vortex which starts from the leading edge.

Probably the agreement of Yokoyama's model with experimental data could be improved if the vortex starting point were taken as approximately quarter of the chord length from the leading edge since, despite the slight shifts with various factors, that is the approximate location where the tip leakage vortex starts to form (Lakshminarayana et al. (1987)). In fact, in the potential vortex model introduced by Lakshminarayana and Horlock (1965), which was used for predicting the blade pressure distributions, the vortex was started at 25 percent chord length from the leading edge. In this model the circulation distribution at the blade tip was varied linearly with x/c and the circulation in the first 25 percent chord length region was assumed to be retained at the blade tip

Figure A.2.11.1 Yokoyama's (1961) models for blade suction surface pressure distribution prediction

Figure A.2.11.2 Lakshminarayana and Horlock's model for blade surface pressure distribution prediction (reproduced from Lakshminarayana and Horlock (1965))

(see Fig. A.2.11.2). Another improvement in Lakshminarayana and Horlock's model (compared to the model used by Yokoyama) is that the potential vortex is not only at an angle to the suction surface but is also inclined to the blade tip. Considering the conical shape of the vortex, this is certainly more realistic. The two angles were determined experimentally. One more respect in which Lakshminarayana and Horlock's model is more powerful than Yokoyama's model is that the former is able to predict the pressure distribution on both sides of the blade while Yokoyama notes that the boundary condition that he used in his model is not applicable for predicting the pressures on the pressure surface of the blade.

Since Yokoyama was not able to predict the pressure distribution on the blade pressure surface, in order to overcome this shortcoming he introduced a lifting line theory which is modified from Prandtl's original lifting line theory so as to account for the tip leakage vortex effect (Fig. A.2.11.3). Particularly close to the blade tip, the agreement between the calculated spanwise lift distribution and experimental data is not very good; nevertheless the expected trend is predicted.

Lewis and Yeung (1977) reviewed a number of methods that have been used for predicting the lift distribution (Fig. A.2.11.4): the vortex lattice method has been successfully used in the lift distribution prediction for large aspect ratio lifting surfaces. However, it is not suitable for small aspect ratio blades since it does not account for the presence of the cumulatively developed rolled up tip vortices. Furthermore, the lift distribution, normalized against the overall lift coefficient, predicted by the vortex lattice method, is not affected by the angle of attack which does not agree with the experimental results for turbomachine blades.

The tip vortex method shown in Fig. A.2.11.4 appears to give better results as far as the aspects noted in the previous paragraph are concerned. In fact, this model, with the exception of the trailing vortices, was the basis of one of the two models used by Yokoyama (1961) to predict the blade suction surface pressure distribution.

Figure A.2.11.3 a) Model used in Prandtl's lifting line theory
 b) Yokoyama's model (reproduced from Yokoyama (1961))

(A) VORTEX LATTICE METHOD

(B) TIP VORTEX METHOD

(C) CONCENTRATED TIP VORTEX METHOD

Figure A.2.11.4 Diagrammatic view of different models (reproduced from Lewis and Yeung (1977))

However, the tip vortex method fails to predict the localized increase in the loading close to the blade tip, caused by the tip vortex located against the suction surface of the blade. As a solution to this problem Lewis and Yeung (1977) suggest a concentrated tip vortex method (Fig. A.2.11.4) which will be discussed later in this section.

During their work on models to be used for predicting the loading distribution, Lewis and Yeung (1977) performed some experiments with an isolated aerofoil. The static pressure on the endwall was measured along a contour following the blade surface edge (see Fig. A.2.11.5). From these measurements they determined the retained lift at the endwall and formed the retained lift coefficient as defined by Lakshminarayana and Horlock (1962). Their data appear to be close to that determined at the blade tip by Lakshminarayana and Horlock (1962). Since Lewis and Yeung's (1977) data covered a larger gap to chord ratio range, following Lakshminarayana and Horlock's (1962) correlation, they presented another correlation for the retained lift coefficient. Both correlations together with experimental data are given in Fig. A.2.11.5. Even though the correlations appear to give similar results, Lewis and Yeung's correlation can be criticized since data for high gap to chord ratios were also taken into account whereas Lakshminarayana and Horlock indicate that this type of retained lift calculation is only valid for low gap to chord ratios.

Using this retained lift coefficient, Lewis and Yeung (1977) modified the tip vortex method given in Fig. A.2.11.4 such that only part of the bound vorticity at the blade tip is shed. They also used a different expression for calculating the angle between the shed vortex and the blade chord. This improved tip vortex model is shown in Fig. A.2.11.6. Comparison of the predictions with the experimental results of Lakshminarayana and Horlock (1962) shows that the modified tip vortex method gives much better results than the vortex lattice and the original tip vortex methods, particularly at small clearances where the amount of retained lift at the blade tip is significant.

Figure A.2.11.5 Retained lift coefficient variation with clearance size (reproduced from Lakshminarayana (1985))

Figure A.2.11.6 Lewis and Yeung's modified tip vortex method (reproduced from Lewis and Yeung (1977))

As noted by Lewis and Yeung, the concentrated tip vortex method (Fig. A.2.11.4) is probably more realistic than the ones discussed so far. With the exception of the trailing vortices, and the strength of the tip vortex (note that in the concentrated vortex model the tip vortex starts to shed at the leading edge which means all the circulation at the blade tip is shed), this is in fact the model used by Lakshminarayana and Horlock for blade surface pressure distribution prediction (Fig. A.2.11.2), discussed earlier.

In the work of Lakshminarayana (1970), the author noted that the potential vortex model used for predicting the blade surface pressure distribution (Fig. A.2.11.2) is not realistic enough to enable prediction of the blade-to-blade outlet angle and loss distribution at the tip region of the blade. Therefore, the author presented another model (see Fig. A.2.11.7). In this calculation scheme the diameter of the tip leakage vortex is obtained from Rains' (1954) analysis. From his comparisons with various experimental results he concluded that the model predicts the over and under turning quite accurately while the loss distribution is predicted only qualitatively. A qualitative agreement between the measured and predicted flow angle was also observed for Inoue et al.'s (1986) measurements at the exit of an axial compressor. It should also be noted that in this analysis the determination of the circulation associated with the tip leakage vortex is based on an inviscid tip leakage flow with a stationary blade. Gearhart (1966) and Graham's (1986) experimental results showed that due to viscous drag effect, the endwall motion has a significant effect on the leakage flow. Therefore, it is likely that the circulation associated with the leakage vortex (thus the retained lift coefficient) is altered by the endwall motion, and this effect should be included into the analysis. In fact, from the data obtained in a compressor rotor by Inoue et al. (1986), it appeared that the measured circulation associated with the tip leakage vortex is much larger than the one predicted using Lakshminarayana and Horlock's (1962) correlation for the retained lift coefficient. This is an expected result since in compressors the endwall motion increases the tip leakage mass flow rate which will likely result in a stronger leakage vortex. Thus, the

Figure A.2.11.7 Modelling of the passage flow in the blade tip region (reproduced from Lakshminarayana (1970))

effect of endwall motion on the retained lift coefficient is one aspect that should be investigated in the near future.

Recently, Chen et al. (1990) proposed a similarity analysis for predicting the tip-leakage vortex structure in compressors. In the inviscid scheme, the steady, three-dimensional tip-leakage flow is viewed as a two-dimensional unsteady flow. The model was found to provide reasonable predictions of the velocity and flow angle distribution within the tip-vortex structure as well as the vortex core trajectory. Thus, it seems that the model can be used as part of a scheme for predicting the effects of the tip vortex on the blade pressures.

2.12 Modelling of the Flow Inside the Tip Gap

In Section A.2.10 a review of the loss models was given. The analyses that were based on Rains' (1954) assumption that the kinetic energy associated with the normal component of velocity of the leakage flow is lost, require a model for the velocity field within the gap. As discussed, one such model was introduced by Rains (1954) and is simply based on a tip leakage velocity calculation using Bernoulli's equation, with the assumption that the tangential velocity is conserved across the gap.

Even though a few leakage models exist in the literature, the one indicated above being one of them, they were introduced as part of other calculation schemes, such as loss correlations and/or modelling of the flow emerging from the clearance region. Although some of these calculation schemes appeared to be successful, this does not necessarily mean that the leakage flow inside the tip gap is modeled accurately and in none of these studies was this analyzed in detail. Even though the prediction of the losses and the flow disturbance near the blade tip is important, since the main objective is to reduce the negative effect of the tip clearance flow on the performance of the machine, there is an obvious need for a model which enables the

prediction of the flow field within the tip clearance. This could lead to configurations which reduce the leakage flow and hence the losses associated with it.

Booth et al.'s (1982) appears to have been the first study where the leakage flow was modeled for the above indicated purpose. The authors used Rains' (1954) inviscid analysis (see Section A.2.10) as the basis for their model, since their theoretical and experimental analyses seemed to support the assumptions built into that analysis (i.e., the dominance of inertia forces for practical clearance sizes, the conservation of tangential momentum across the clearance, etc.). In their experiments they used three test sections which are shown in Fig. A.2.12.1. One of the interesting results of the experiments with the single-side discharge rig is that the discharge coefficient (defined for the tip leakage mass flow rate) is independent of the clearance size. This appears to contradict Graham's (1986) results which, as was noted in Section A.2.5, showed a nonlinear variation of the leakage mass flow rate with clearance size. The recent results of Yaras et al. (1988) support Graham's observation: the experimental results obtained in a turbine cascade indicated that as the clearance size varied, the separation bubble size on the blade tip changes, causing a nonlinear variation in the leakage mass flow rate (Fig. A.12.2). From their double-sided discharge rig Booth et al. concluded that the effect of the suction side velocity on the leakage flow was of second-order. This helps to justify the assumption of conservation of the tangential momentum through the gap. Finally, they performed some experiments in a water table cascade with turbine blades where they measured the leakage flow velocity and direction using dye clusters and a high speed camera. However, as noted by the authors, there were some errors due to this measurement technique. One of the striking results was the straight streamlines through the gap, which they then used in their leakage model. Based on these observations the authors presented a model for determining the velocity magnitude and flow angle distribution inside the tip gap.

Figure A.2.12.1 Water rig configurations (reproduced from Booth et al. (1982))

Figure A.2.12.2 Discharge coefficient variation with clearance size
(reproduced from Yaras (1987))

The recent work of Yaras et al. (1988) provided considerably more detailed and accurate data associated with the flow field inside the tip gap. Using this data, the assumptions built into the model of Booth et al. (1982) were criticised. Realizing that the relatively complex calculations in Booth's model are not really justified, a much simpler model was suggested. The basis of the model was to use Bernoulli's equation to determine the gap velocity component normal to the blade meanline, which was also suggested by Rains. From measurements, the driving pressure difference to be used in this calculation was found to be equal to the local undisturbed blade pressure difference, a quantity that can be determined easily. With the experimentally justified assumption that the tangential velocity is conserved through the gap, the chordwise distribution of gap velocity and flow angle was then determined. Comparison with experimental results indicated that the model was able to predict the flow field quite accurately. In relation to this work, Storer and Cumpsty (1990) measured the tip gap flow in a linear compressor cascade. They confirm the validity of the tip-gap model evaluated by Yaras et al. (1988). However, they suggest that the tip-leakage flow should depend on the blade pressure distribution near the tip, which contradicts Yaras et al.'s experimental observations.

Recently, Moore and Tilton (1987) presented two theories for predicting the pressure distribution through the gap as well as experimental data which they obtained from measurements in a rectilinear turbine cascade with 2.1 percent clearance-to-span ratio. In addition to the work discussed in the previous section, Rains (1954) also introduced an expression based on potential theory which, once the tip gap velocity is determined using Bernoulli's equation (see Rains' analysis discussed in Section A.2.10), can be used to determine the pressure distribution within the gap. Rains did not go into the details of this theory since his main objective was to obtain a loss correlation and the gap velocity obtained from Bernoulli's equation was good enough for that purpose. Moore and Tilton analyzed this potential theory and concluded that it contradicts the one given by Rayleigh in 1876. Their further analysis showed that the discrepancy was in Rains' analysis and they presented the correct expression.

Figure A.2.12.3 Rains' idealized tip gap flow model (reproduced from Moore and Tilton (1987))

Figure A.2.12.4 Actual tip gap flow (reproduced from Moore and Tilton (1987))

Undoubtedly, as seen in Fig. A.2.12.4, the real flow is quite different from the ideal one given in Fig. A.2.12.3. Therefore, Moore and Tilton used the potential theory only for the flow prediction up to the clearance entrance. The experimental pressures, both from the endwall and blade pressure surface, appeared to be predicted quite accurately. In order to predict the whole gap flow, the authors presented a second method which was a two-dimensional potential flow theory combined with a mixing analysis. With this model the flow up to the point where the stream tube inside the gap reaches its minimum height was predicted quite accurately. Downstream of this section, the final pressure value after the pressure rise due to mixing was also predicted accurately. However, in the intervening region of rising pressure the agreement was not very good.

Moore and Tilton calculated the discharge coefficient which appeared to be close to the value of about 0.8 quoted by Booth et al. (1982). They were also able to predict this discharge coefficient from an expression obtained from the latter theory mentioned above. The predicted discharge coefficient appears to be reasonably accurate. The theoretical discharge coefficient is a function of the contraction ratio only. The theory gives a constant contraction factor of 0.611 so that the discharge coefficient has a constant value of 0.8435. However, as was noted earlier, the results of Yaras et al. (1988) clearly indicated a variation of discharge coefficient with gap size. In Moore and Tilton's case the experimental value of the contraction factor appeared to be close to the theoretical one, explaining the good accuracy of the predicted pressure distributions. However, in cases where the contraction factor deviates significantly from the theoretical one, the results should not be as good.

Finally, in this section it is worth mentioning that a number of models have been developed to predict the entire flow field in rotor passages using three-dimensional numerical schemes. One of them was developed for compressors by Hah (1986). In this model the Reynolds averaged Navier Stokes equation is solved in elliptic form. From comparison with experimental data, the author stated that the

velocity vectors inside the tip gap were predicted quite accurately. Furthermore, the overall tip clearance effects on casing wall boundary layer development, relative outflow angle and overall compressor performance were predicted reasonably well as well. One shortcoming of the model is that it is limited to compressors only. A more recent model proposed by Dawes (1988) does not have this shortcoming and is applicable to any type of turbomachinery. However, Dawes' model is limited to compressible flow. Furthermore, in the model the tip clearance geometry is not simulated very accurately. The algorithm cannot handle sharp corners at the blade tip, hence, the blade entered to the model will have to have rounded corners. Therefore, it seems that as it stands Dawes' model can only predict the effects of tip leakage on the passage flow field in a qualitative manner.

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Appendix B

a) Geometric specifications of the Pratt and Whitney turbine blade:

- Leading edge wedge angle	= 54.10 deg.
- Trailing edge wedge angle	= 1.92 deg.
- Maximum thickness	= 0.1163"
- Blade stagger	= 35.96 deg.
- Metal turning	= 60.63 deg.

b) Geometric specifications of the linear turbine cascade blade:

- Blade scale (model/P&W prototype)	= 12.29:1
- Nominal blade chord	= 250 mm
- Blade height	= 202.3 mm
- Blade spacing	= 150 mm
- Blade maximum thickness / chord	= 0.0984
- Metal turning	= 59.2 deg.
- Stagger angle	= 37.6 deg.
- Incidence	= -11.1 deg.

Appendix C**a) Blade surface pressure tap row locations relative to the blade tip**

Row #	(Distance from blade tip)/h
1	0.0212
2	0.0460
3	0.0707
4	0.0954
5	0.1201
6	0.1448
7	0.1695
8	0.1943
9	0.2190
10	0.2684
11	0.3178
12	0.3673
13	0.4167
14	0.4661

b) Tap locations relative to the co-ordinate system in Figure C.1

Tap #	X (inches)	Y (inches)
<hr/>		
(Suction side)		
1	-3.2052	2.8267
2	-2.9803	3.0725
3	-2.7554	3.1290
4	-2.5317	3.1364
5	-2.2872	3.0971
6	-2.0844	3.0283
7	-1.8472	2.9140
8	-1.6112	2.7689 (*)
9	-1.4220	2.6276
10	-1.2155	2.4543
11	-1.0400	2.2884
12	-0.8677	2.1053
13	-0.6932	1.9049
14	-0.5334	1.7083
15	-0.3748	1.4982
16	-0.2175	1.2794
17	-0.0578	1.0483
18	0.1020	0.8136
19	0.2618	0.5715
20	0.4203	0.3232
21	0.5801	0.0701
22	0.7399	-0.1905
23	0.8984	-0.4550
24	1.0582	-0.7276
25	1.2179	-1.0053
26	1.3765	-1.2892
27	1.5363	-1.5768
28	1.6948	-1.8718
29	1.8533	-2.1704
30	2.0131	-2.4752
31	2.1716	-2.7825
32	2.3290	-3.0946
33	2.4875	-3.4105
34	2.6448	-3.7288
35	2.8033	-4.0496
36	2.9275	-4.3089
37	3.0541	-4.5682

(Pressure side)

38	-3.0664	2.6104
39	-2.8980	2.5145
40	-2.7394	2.4187
41	-2.5809	2.3167
42	-2.3830	2.1827
43	-2.1999	2.0512
44	-2.0070	1.9025
45	-1.8312	1.7599
46	-1.6555	1.6100
47	-1.4809	1.4502
48	-1.3052	1.2818
49	-1.1454	1.1208
50	-0.9857	0.9537
51	-0.8271	0.7780
52	-0.6673	0.5961
53	-0.5063	0.4068
54	-0.3466	0.2114
55	-0.1880	0.0098
56	-0.0283	-0.1966
57	0.1315	-0.4093
58	0.2913	-0.6268
59	0.4510	-0.8480
60	0.6108	-1.0741
61	0.7706	-1.3015
62	0.9304	-1.5338
63	1.0889	-1.7673
64	1.2474	-2.0033
65	1.4072	-2.2429
66	1.5670	-2.4850
67	1.7648	-2.7947
68	2.0426	-3.2360
69	2.2331	-3.5506
70	2.4064	-3.8455
71	2.5784	-4.1442
72	2.7272	-4.4109
73	2.8734	-4.6825

(*) : Not used during measurements

Figure C.1 Co-ordinate system for blade static pressure tap locations.

Appendix D

Source codes for data acquisition programs described in Chapter 3.

The source code for the seven-hole probe traverse and data acquisition program is listed in the following pages. The code is written in Turbo-Pascal Version 3.0 programming language. All other data acquisition programs used for purposes such as probe calibration, three-hole probe traverse, tip-wall static pressure, blade static pressure and boundary layer measurements have very similar program structures. Thus, they are not listed herein to conserve space. The "included" library file given subsequent to the main program contains the subroutines that are common for all data acquisition programs used during experiments. Communication routines account for a major part of this library file. Turbo Pascal Ver. 5.0 simplifies communication through the serial port substantially. Thus, it may be possible to simplify the code listed herein by adapting it to this more recent version.

Program Seven_HoleProbe_Traverse;

{ Data acquisition program for seven hole probe traverses

- Because of memory limitations, results are written to the output file after every measurement station.
- Maximum number of samples per channel during sampling period is 1000. To change this, modify 'type array1'.
- In addition to data acquisition, the movement of the probe in two mutually perpendicular directions is controlled by this program as well.
- The 7 probe ports should be connected to ports 2 to 8 on the Scanivalve.
- First and ninth Scanivalve ports are to be connected to the reference pressure to check on the transducer zero reading. }

```
Type      STR      = String[25];
          array1    = Array [1..1000] of real;
          String19   = String[19];
          String6    = String[6];
```

(\$I Library.sub) { -> should be on the same disk as this program }

```
Var
Port, SBits, DBits                               : Byte;
Par, Baud_Rate, CPU_Option, Waiting_Time, Sampling_Time : Integer;
TipClr, Contraction, Patm, T0, CalcofP, CalcofQ       : Real;
TestDate, Answer, outfill1, outfill2                : String[25];
Channel1, Channel2                                  : Char;
```

```
{Port      : Serial port number on the micro computer (usually 1)
SBits      : Number of stop bits used during communication
DBits      : Number of data bits used during communication
Par        : Parity
CPU Option : Parameter that specifies the CPU Module in use (901 or 902)
TipClr     : Tip clearance height/chord length
Patm       : Atmospheric pressure in mm mercury
T0         : Total temperature of the air entering the test section
CalcofP    : Calibration coefficient for the transducer that measures the
             test section pressure of interest
CalcofQ    : Calibration coefficient for the transducer that measures the
             contraction pressure difference
Channel1    : Data acquisition channel on the M 81 module which receives the
             contraction pressure difference signal
Channel2    : Data acquisition channel on the M 81 module which receives the
             test section pressure signal }
```

{-----Beginning of procedure SHProbe_Traverse-----}

```
Procedure SHProbe_Traverse (outfill1 : STR; CPU_Option, Waiting_Time,
                             Sampling_Time : Integer; TipClr, Contraction : Real;
                             Channel1, Channel2 : Char; TestDate : STR;
                             Patm, T0 : Real; CalcofP, CalcofQ : Real);
```

Var

```
Outfile1      : Text;
Psc2_Psc1, P_Pref : Array1;
Q1, Q2, Q3, Q4, Q5, Q6, Q7, P1, P2, P3, P4, P5, P6, P7,
Cp1, Cp2, Cp3, Cp4, Cp5, Cp6, Cp7, InitP, InitQ,
AverageP, AverageQ, AverageCp, TotalDistance, Sum_P_Pref,
Sum_Q, Sum_Cp, Probe_Angle, Zinit, Yinit, Span_Step_Size,
Pitch_Step_Size : Real;
Choice, S11, P11, I2, PN1, SN1, N2, Counter, Counter2,
Inp1, Inp2, Delta_inp, P1I2, P2I2, P3I2, P4I2, P5I2,
P6I2, Enter_Init_Slot, Controller_Option, Clock_Speed : Integer;
S_Direction, P_Direction, A1, A2 : String[6];
DataError     : Boolean;
```

{

Psc2_Psc1 :
Contraction pressure difference samples acquired within a sampling period.

F_Pref :
Test section data acquired within a sampling period.

DataError :
Flag to indicate any error that occurs during data acquisition or data transmission. In this case data acquisition for that data is repeated.

P1I2, P2I2, P3I2, P4I2, P5I2, P6I2 :
Used for printing the average data onto the screen.

Enter_Init :
Interval for entering transducer zero readings.

Zinit, Yinit :
Co-ordinates of the probe location at the beginning of the traverse.

S_Direction :
Rotation direction for the motor that provides the spanwise linear motion; Clockwise (HCW) or Counter-clockwise (HCCW).

P_Direction :
Rotation direction for the motor that provides the pitchwise linear motion; Clockwise (VCW) or Counter-clockwise (VCCW).

Slot :
Slot number on the cascade back-wall used during traversing.

Controller_Option :
Used to identify the motor controller employed during the experiment.

Clock_Speed :
Computer clock speed.

Inp1, Inp2 :
Scanivalve transducer zero reading acquired before and after 7 port readings.

Counter :
Counter for the seven-hole probe ports.

Counter2 :
Counter for identifying the time for transducer zero reading checks.

InitQ :
Transducer zero reading for the transducer that measures the contraction pressure.

InitP :
Transducer zero reading for the transducer that measures the test section data.

SN1 :
Number of measurement stations in spanwise direction.

PN1 :
Number of measurement stations in pitchwise direction.

SI1, PI1 :
Counters for spanwise and pitchwise measurement stations.

Choice :
Spanwise motion, pitchwise motion, or both.

N2 :
Number of samples obtained from each channel during a sampling period.

Pitch_Step_Size :
Step size for the pitchwise motion.

Span_Step_Size :
Step size for the spanwise motion.

P1, P2, P3, P4, P5, P6, P7 :
7-hole probe pressures.

Q1, Q2, Q3, Q4, Q5, Q6, Q7 :
Inlet dynamic pressure corresponding to P1 to P7.

Cp1, Cp2, Cp3, Cp4, Cp5, Cp6, Cp7 :
7-hole probe non-dimensional pressures. }

```

Begin;
  Clrscr;
  Enter_Init := 1;
  Writeln (' Enter seven hole probe angle relative to the horizontal cascade');
  Write   (' axis in degrees (negative when probe tip is facing down) : ');
  Readln (Probe_Angle);
  Writeln (' ');
  Writeln (' Enter vertical distance of probe stem head from point A2 on the');
  Write   (' back plate -in mm- (above A2 is positive) : ');
  Readln (Yinit);
  Writeln (' ');
  Writeln (' Enter horizontal distance of probe stem head from tip-wall');
  Write   (' (in mm) : ');

```

```

    Readln (Zinit);
    Writeln (' ');
    Writeln (' Enter seven hole probe traverse slot on the cascade; far ');
    Write (' downstream slot is slot # 5 : ');
    Readln (Slot);
    Writeln (' ');
    Write (' Enter zero reading of transducer for Psc1-Psc2 (volts) : ');
    Readln (InitQ);
    Writeln (' ');
    Write (' Do you want horizontal motion, vertical motion',
           ' or both (0,1,2) ? ');
    Readln (Choice);

    Case Choice of
    0 : begin
        Write (' Enter step size for spanwise motion (mm) : ');
        Readln (Span_Step_Size);
        Write (' Enter number of measurement stations : ');
        Readln (SN1);
        Writeln (' Clockwise turn will be away from the backplate ');
        Write (' Enter the motor turning direction (HCW or HCCW) : ');
        Readln (S_Direction);
        PN1 := 1; { used as a dummy value }
        end;
    1 : begin
        Write (' Enter step size for pitchwise motion (mm) : ');
        Readln (Pitch_Step_Size);
        Write (' Enter number of measurement stations : ');
        Readln (PN1);
        Writeln (' Clockwise turn will be downwards motion ');
        Write (' Enter the motor turning direction (VCW or VCCW) : ');
        Readln (P_Direction);
        SN1 := 1; { used as a dummy value }
        end;
    2 : begin
        Write (' Enter step size for spanwise motion (mm) : ');
        Readln (Span_Step_Size);
        Write (' Enter number of measurement stations in this direction : ');
        Readln (SN1);
        Writeln (' Clockwise turn will be away from the backplate ');
        Write (' Enter the motor turning direction (HCW or HCCW) : ');
        Readln (S_Direction);
        Write (' Enter step size for pitchwise motion (mm) : ');
        Readln (Pitch_Step_Size);
        Write (' Enter number of measurement stations in this direction : ');
        Readln (PN1);
        Writeln (' Clockwise turn will be downwards motion ');
        Write (' Enter the motor turning direction (VCW or VCCW) : ');
        Readln (P_Direction);
        end;
    end;
    Clrscr;
    Writeln (' Controller 1 : Contains two DMA 64');
    Writeln (' Controller 2 : Contains one DMA 64 and one CMD 40');
    Write (' Enter option (1/2) : '); Readln (Controller_Option);
    Write (' Enter computer clock speed (MHz) (8 or 4) : ');
    Readln (Clock_Speed);

    S11 := 1; P11 := 1; Counter2 := 0;
    Assign (outfile1, outfile1);
    Rewrite (outfile1);
    Writeln (outfile1, ' Test Date = ', TestDate);
    Writeln (outfile1, ' Clearance size (t/c) : ', TipClr:10:5);
    Writeln (outfile1, ' Tunnel contraction factor : ', Contraction:10:5);
    Writeln (outfile1, ' Transducer calibration coef. for Psc2-Psc1 : ', CalcofQ:5:2);
    Writeln (outfile1, ' Transducer calibration coef. for P-Pref : ', CalcofP:5:2);
    Writeln (outfile1, ' Waiting period (sec) : ', Waiting_Time:3);
    Writeln (outfile1, ' Sampling period (sec) : ', Sampling_Time:3);
    Writeln (outfile1, ' Traverse slot # : ', Slot:3);

```

```

Writeln (outfile1,' Initial location of the probe stem head : ');
Writeln (outfile1,' Distance from endwall (mm) : ',Zinit:10:4);
Writeln (outfile1,' Pitchwise distance from A2 (mm) : ', Yinit:10:4);
Writeln (outfile1,' ');
Writeln (outfile1,' Probe angle (deg) : ',Probe_Angle:5:2);
Writeln (outfile1,' Patm in mmHg, T0 in Kelvin and the other dimensional ');
Writeln (outfile1,' parameters in volts ');
If Choice = 0 then begin
  Writeln (outfile1,' Spanwise step size (mm) : ',Span_Step_Size:7:4);
  Writeln (outfile1,' Turning direction : ',S_Direction);
  Writeln (outfile1,' Number of measurement stations : ',SN1:3);
end;
If Choice = 1 then begin
  Writeln (outfile1,' Pitchwise step size (mm) : ',Pitch_Step_Size:7:4);
  Writeln (outfile1,' Turning direction : ',P_Direction);
  Writeln (outfile1,' Number of measurement stations : ',PN1:3);
end;
If Choice = 2 then begin
  Writeln (outfile1,' Spanwise step size (mm) : ',Span_Step_Size:7:4);
  Writeln (outfile1,' Turning direction : ',S_Direction);
  Writeln (outfile1,' Number of spanwise measurement stations : ',SN1:3);
  Writeln (outfile1,' Pitchwise step size (mm) : ',Pitch_Step_Size:7:4);
  Writeln (outfile1,' Turning direction : ',P_Direction);
  Writeln (outfile1,' Number of pitchwise measurement stations : ',PN1:3);
end;
Writeln (outfile1,' ');

Writeln (outfile1,' 1st (111) V# H# InQcntr Inp Q1cntr P1 Q2cntr P2');
Writeln (outfile1,' 2nd (222) Q3cntr P3 Q4cntr P4 Q5cntr P5 Q6cntr P6 Q7cntr P7');
Writeln (outfile1,' 3rd (333) V# H# Patm T0 Cp1 Cp2 Cp3 Cp4 Cp5 Cp6 Cp7');
Writeln (outfile1,'-----');
Writeln (outfile1,' ');

Repeat
  Clrscr;
  GotoXY(30,15); Writeln (' Initializing the Scanivalve ');
  Step_Home_Scanivalve (0,1);

  Counter := 0;
  Counter2 := Counter2+1;

  Delay (1000);
  GotoXY(40,15); Writeln ('

Repeat
  { Calling the Communication procedure to start sampling : }
Repeat
  Communication (CPU Option, Waiting Time, Sampling_Time, Channell, Channel2,
    Psc2_Psc1, P_Pref,N2,DataError);
Until (DataError = False);
{ Returning from this procedure are Psc2_Psc1[1..1000], P_Pref[1..1000] and N2}

{ Preparing the computer screen for data presentation : }
Clrscr;
If { Counter > 0 } and { Counter < 8 } then begin
  GotoXY(1,1); Writeln (' Waiting Period : ',Waiting_Time:3);
  GotoXY(1,2); Writeln (' Sampling Period : ',Sampling_Time:3);
  GotoXY(6,5); Writeln ('Q (Dyn P.)= volts');
  GotoXY(6,7); Writeln ('P1 - Pref = volts');
  GotoXY(6,9); Writeln ('P2 - Pref = volts');
  GotoXY(6,11); Writeln ('P3 - Pref = volts');
  GotoXY(6,13); Writeln ('P4 - Pref = volts');
  GotoXY(6,15); Writeln ('P5 - Pref = volts');
  GotoXY(6,17); Writeln ('P6 - Pref = volts');
  GotoXY(6,19); Writeln ('P7 - Pref = volts');
end;

```

```

( Initializing some variables : )
Sum_P_Pref := 0.0; Sum_Q := 0.0; Sum_Cp := 0.0;

( Processing the acquired data : )
For I2 := 1 to N2 do begin
  Sum_P_Pref := Sum_P_Pref + P_Pref[I2];
  Sum_Q := Sum_Q + Psc2_Psc1[I2];
  If ((Counter > 0) and (Counter < 8)) then
    Sum_Cp := Sum_Cp + (P_Pref[I2] - Inp1)*CalCofP/
      ((Psc2_Psc1[I2] - InitQ)*CalCofQ*Contraction);
end;
AverageP := Sum_P_Pref/N2;
AverageQ := Sum_Q*Contraction/N2;
If ((Counter > 0) and (Counter < 8)) then AverageCp := Sum_Cp/N2;

  If (Counter = 0) then Inp1 := AverageP;
  If (Counter = 8) then Inp2 := AverageP;
  If (Counter <> 0) and (Counter <> 8) then begin
    If (Counter = 1) then begin
      Q1 := AverageQ;
      P1 := AverageP;
      Cp1 := AverageCp;
      GotoXY(2,7); Writeln(I2:3);
      GotoXY(18,7); Writeln (AverageP:12:8);
      P1I2 := I2;

      end;
    If (Counter = 2) then begin
      Q2 := AverageQ;
      P2 := AverageP;
      Cp2 := AverageCp;
      GotoXY(2,9); Writeln(I2:3);
      GotoXY(18,9); Writeln (AverageP:12:8);
      P2I2 := I2;

      GotoXY(2,7); Writeln(P1I2:3);
      GotoXY(18,7); Writeln (P1:12:8);

      end;
    If (Counter = 3) then begin
      Q3 := AverageQ;
      P3 := AverageP;
      Cp3 := AverageCp;
      GotoXY(2,11); Writeln(I2:3);
      GotoXY(18,11); Writeln (AverageP:12:8);
      P3I2 := I2;

      GotoXY(2,9); Writeln(P2I2:3);
      GotoXY(18,9); Writeln (P2:12:8);
      GotoXY(2,7); Writeln(P1I2:3);
      GotoXY(18,7); Writeln (P1:12:8);

      end;
    If (Counter = 4) then begin
      Q4 := AverageQ;
      P4 := AverageP;
      Cp4 := AverageCp;
      GotoXY(2,13); Writeln(I2:3);
      GotoXY(18,13); Writeln (AverageP:12:8);
      P4I2 := I2;

      GotoXY(2,11); Writeln(P3I2:3);
      GotoXY(18,11); Writeln (P3:12:8);
      GotoXY(2,9); Writeln(P2I2:3);
      GotoXY(18,9); Writeln (P2:12:8);
      GotoXY(2,7); Writeln(P1I2:3);
      GotoXY(18,7); Writeln (P1:12:8);

      end;
    If (Counter = 5) then begin
      Q5 := AverageQ;
      P5 := AverageP;
      Cp5 := AverageCp;
      GotoXY(2,15); Writeln(I2:3);
      GotoXY(18,15); Writeln (AverageP:12:8);
      P5I2 := I2;

      GotoXY(2,13); Writeln(P4I2:3);

```

```

GotoXY(18,13); Writeln (P4:12:8);
GotoXY(2,11); Writeln(P3I2:3);
GotoXY(18,11); Writeln (P3:12:8);
GotoXY(2,9); Writeln(P2I2:3);
GotoXY(18,9); Writeln (P2:12:8);
GotoXY(2,7); Writeln(P1I2:3);
GotoXY(18,7); Writeln (P1:12:8);

                                end;
If (Counter = 6 ) then begin
    Q6 := AverageQ;
    P6 := AverageP;
    Cp6 := AverageCp;
    GotoXY(2,17); Writeln(I2:3);
    GotoXY(18,17); Writeln (AverageP:12:8);
    P6I2 := I2;

    GotoXY(2,15); Writeln(P5I2:3);
    GotoXY(18,15); Writeln (P5:12:8);
    GotoXY(2,13); Writeln(P4I2:3);
    GotoXY(18,13); Writeln (P4:12:8);
    GotoXY(2,11); Writeln(P3I2:3);
    GotoXY(18,11); Writeln (P3:12:8);
    GotoXY(2,9); Writeln(P2I2:3);
    GotoXY(18,9); Writeln (P2:12:8);
    GotoXY(2,7); Writeln(P1I2:3);
    GotoXY(18,7); Writeln (P1:12:8);

                                end;
If (Counter = 7 ) then begin
    Q7 := AverageQ;
    P7 := AverageP;
    Cp7 := AverageCp;
    GotoXY(2,19); Writeln(I2:3);
    GotoXY(18,19); Writeln (AverageP:12:8);

    GotoXY(2,17); Writeln(P6I2:3);
    GotoXY(18,17); Writeln (P6:12:8);
    GotoXY(2,15); Writeln(P5I2:3);
    GotoXY(18,15); Writeln (P5:12:8);
    GotoXY(2,13); Writeln(P4I2:3);
    GotoXY(18,13); Writeln (P4:12:8);
    GotoXY(2,11); Writeln(P3I2:3);
    GotoXY(18,11); Writeln (P3:12:8);
    GotoXY(2,9); Writeln(P2I2:3);
    GotoXY(18,9); Writeln (P2:12:8);
    GotoXY(2,7); Writeln(P1I2:3);
    GotoXY(18,7); Writeln (P1:12:8);

                                end;

                                GotoXY(2,5); Writeln(I2:3);
                                GotoXY(18,5); Writeln (AverageQ:12:8);

                                end;

{ Stepping the Scanivalve : }
Step_Home_Scanivalve(0,128);

Counter := Counter + 1;
Delay (1000); { wait for 1 second }
If ( Counter = 9 ) then begin
    Delta_Inp := Abs(Inp1-Inp2);
    If (Delta_Inp > 0.003) then begin
        Writeln(' This set has to be repeated; do not change angle setting');
        Writeln(' ');
        Writeln(' Zero reading at the beginning (volts) : ', Inp1:10:7);
        Writeln(' ');
        Writeln(' Zero reading at the end (volts)           : ', Inp2:10:7);
        Writeln(' ');
        Writeln(' Press any key to continue ... ');
        Repeat Until KeyPressed;
        Clrscr;
        GotoXY(30,15); Writeln ( ' Initializing the Scanivalve ');
        Step_Home_Scanivalve(0,1);
    end;
end;

```

```

Counter := 0;
Delay (1000); ( wait for 1 second )
GotoXY(40,15); WriteLn (' ');
end else begin
  InitP := InPl;
  ( Write results to output files : )

  WriteLn (outfile1,'111 ',
    P11:3,' ',S11:3,' ',InitQ:9:6,' ',InitP:9:6,' ',
    Q1/Contraction:9:6,' ',P1:9:6,' ',Q2/Contraction:9:6,
    ',P2:9:6);
  WriteLn (outfile1,'222 ',
    Q3/Contraction:9:6,' ',P3:9:6,' ',Q4/Contraction:9:6,
    ',P4:9:6,' ',Q5/Contraction:9:6,' ',P5:9:6,' ',Q6/Contraction:9:6,
    ',P6:9:6,' ',Q7/Contraction:9:6,' ',P7:9:6);
  WriteLn (outfile1,'333 ',
    P11:3,' ',S11:3,' ',Patm:5:2,' ',T0:5:2,' ',
    Cp1:10:7,' ',Cp2:10:7,' ',Cp3:10:7,' ',Cp4:10:7,' ',
    Cp5:10:7,' ',Cp6:10:7,' ',Cp7:10:7);

Case Choice of
0 : begin
  If (S11 = SN1) then begin
    { move the span motor to initial position }
    A1 := 'HCW'; A2 := 'HCCW';
    TotalDistance := Span_Step_Size*(SN1-1);
    If (S_Direction = A1) then
      Motor_Step(Controller_Option,A2,TotalDistance,Clock_Speed) else
      Motor_Step(Controller_Option,A1,TotalDistance,Clock_Speed);
    end else begin
      { move the span motor to next station }
      Motor_Step(Controller_Option,S_Direction,Span_Step_Size,Clock_Speed);
    end;
    S11 := S11+1;
  end;
1 : begin
  If (P11 = PN1) then begin
    { move the pitch motor to initial position }
    A1 := 'VCW'; A2 := 'VCCW';
    TotalDistance := Pitch_Step_Size*(PN1-1);
    If (P_Direction = A1) then
      Motor_Step(Controller_Option,A2,TotalDistance,Clock_Speed) else
      Motor_Step(Controller_Option,A1,TotalDistance,Clock_Speed);
    end else begin
      { move the pitch motor to next station }
      Motor_Step(Controller_Option,P_Direction,Pitch_Step_Size,Clock_Speed);
    end;
    P11 := P11+1;
  end;
2 : begin
  If (S11 < SN1) or (P11 < PN1) then begin
    If (S11 = SN1) then begin
      { move the span motor to initial position }
      { move the pitch motor to the next station }
      P11 := P11 + 1;
      S11 := 1;
      A1 := 'HCW'; A2 := 'HCCW';
      TotalDistance := Span_Step_Size*(SN1-1);
      If (S_Direction = A1) then
        Motor_Step(Controller_Option,A2,TotalDistance,Clock_Speed) else
        Motor_Step(Controller_Option,A1,TotalDistance,Clock_Speed);
      Motor_Step(Controller_Option,P_Direction,Pitch_Step_Size,Clock_Speed);
    end else begin
      { move the span motor to next station }
      S11 := S11+1;
      Motor_Step(Controller_Option,S_Direction,Span_Step_Size,Clock_Speed);
    end;
  end else begin

```

```

S11 := S11 + 1; { to be used to exit the Repeat-Until loop }

{ Moving the probe to where it was at the beginning of the experiment }
A1 := 'HCW'; A2 := 'HCCW';
TotalDistance := Span_Step_Size*(SN1-1);
If (S_Direction = A1) then
  Motor_Step(Controller_Option,A2,TotalDistance,Clock_Speed) else
  Motor_Step(Controller_Option,A1,TotalDistance,Clock_Speed);
A1 := 'VCW'; A2 := 'VCCW';
TotalDistance := Pitch_Step_Size*(PN1-1);
If (P_Direction = A1) then
  Motor_Step(Controller_Option,A2,TotalDistance,Clock_Speed) else
  Motor_Step(Controller_Option,A1,TotalDistance,Clock_Speed);
end;
end;
end;
end;

If ( Round(Counter2/Enter_Init)*Enter_Init = Counter2 ) then begin
{ after every Enter_Init measurement stations enter the following values }
  Writeln (' The init'al reading for Psc2-Psc1 was : ',InitQ:7:5);
  Writeln (' ');
  Write (' Enter initial reading for Psc2-Psc1 (volts) : ');
  Readln (InitQ);
  Write (' Enter temperature in K : ');
  Readln (T0);
  Write (' Enter pressure in mmHg : ');
  Readln (Patm);
  Writeln (' Zero reading entering period is every ',Enter_Init);
  Write (' stations; enter new (integer) value : ');
  Readln (Enter_Init);
  Counter2 := 0;
end;
end;
Until (Counter = 9) and (Delta_Inp < 0.003);
Until (P11+S11 > PN1+SN1);
Close (outfile);
end;

{-----End of procedure SHProbe_TrV-----}

Begin { Beginning of the main program }
{ Initializing the serial port : }
Clrscr;
Writeln(' ');
Writeln('*****');
Writeln('**      Port Number      :    1 , 2          **');
Writeln('**      Baud Rate bit/s   :    300,600,2400,4800,9600    **');
Writeln('**      Parity           :    (0) None , (1) Even , (2) Odd**');
Writeln('**      Stop Bits        :    1 , 2          **');
Writeln('**      Data Bits         :    7 , 8 (if parity is enabled, **');
Writeln('**                               it should be set to 8)**');
Writeln('*****');
Writeln (' ');
Writeln (' The following data entrances should match the M 901 settings');
Writeln (' ');
Write(' Enter the Port Number : '); Readln (Port);
Write(' Enter the Baud Rate   : '); Readln (Baud_Rate);
Write(' Enter the Parity       : '); Readln (Par);
Write(' Enter the Stop Bits    : '); Readln (SBits);
Write(' Enter the Data Bits    : '); Readln (DBits);
Clrscr;
Writeln (' First data acquisition channel (0) is not available; ');
Writeln (' it is used for Auto Zero if enabled');
Write (' Enter first data acquisition channel number (1-7) : ');
Readln (Channel1);
Write (' Enter second data acquisition channel number (1-7) : ');
Readln (Channel2);
Writeln (' Enter the waiting period (sec)');
Write (' (maximum 60 seconds) : '); Readln(Waiting_time);

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Writeln (' Enter the sampling period (sec)');
Write   (' (maximum 240 seconds) : '); Readln(Sampling_time);
Clrscr;

If (Baud_Rate = 300) and (Par = 0) then
  RS_Initialize(Port,B300,None,SBits,DBits);
If (Baud_Rate = 2400) and (Par = 0) then
  RS_Initialize(Port,B2400,None,SBits,DBits);
If (Baud_Rate = 4800) and (Par = 0) then
  RS_Initialize(Port,B4800,None,SBits,DBits);
If (Baud_Rate = 9600) and (Par = 0) then
  RS_Initialize(Port,B9600,None,SBits,DBits);

If (Baud_Rate = 300) and (Par = 1) then
  RS_Initialize(Port,B300,Even,SBits,DBits);
If (Baud_Rate = 2400) and (Par = 1) then
  RS_Initialize(Port,B2400,Even,SBits,DBits);
If (Baud_Rate = 4800) and (Par = 1) then
  RS_Initialize(Port,B4800,Even,SBits,DBits);
If (Baud_Rate = 9600) and (Par = 1) then
  RS_Initialize(Port,B9600,Even,SBits,DBits);

If (Baud_Rate = 300) and (Par = 2) then
  RS_Initialize(Port,B300,Odd,SBits,DBits);
If (Baud_Rate = 2400) and (Par = 2) then
  RS_Initialize(Port,B2400,Odd,SBits,DBits);
If (Baud_Rate = 4800) and (Par = 2) then
  RS_Initialize(Port,B4800,Odd,SBits,DBits);
If (Baud_Rate = 9600) and (Par = 2) then
  RS_Initialize(Port,B9600,Odd,SBits,DBits);

Writeln (' ');
Writeln ('      Serial Port is initialized to :');
Writeln ('      ,Port:1,',',',Baud_Rate,',',',Par,',',',SBits:1,',',',DBits:1);
Writeln (' ');
Writeln (' Press any key when CPU Module and measurement system are set...');
Writeln (' ');
Repeat Until KeyPressed;

{ Checking the Software Version of the CPU Model :
M-901 -> Version 2.81 software, M-902 -> Version 3.00 software }
Delay (250); { Wait 0.25 seconds }
Writeln (AUX,'VER');
Answer := Get Answer;
Writeln (' The version of the software in the CPU Module is :',Answer);
Answer := Copy(Answer,1,1);
If (Answer = '3') then CPU_Option := 902 else CPU_Option := 901;

{ Erase any commands in M 901/902 memory : }
Delay (250);
Writeln (AUX,'@CLRLIST');
Answer := Get Answer;
Writeln (' M ,CPU_Option:3,
      ' memory is cleared of commands; response : ',Answer);

{ Clear any data that might be present in M 901/902 memory initially : }
Delay (250);
Writeln (AUX,'@CLRDATA');
Answer := Get Answer;
Writeln (' M ,CPU_Option:3,' memory is cleared of data; response :
      ',Answer);

{ Enable automatic temperature compensation for better accuracy;
particularly useful at low level voltage measurements }
Delay (250);
Writeln (AUX,'SETAZ 0,1'); {assumes channel # 0 is used for auto-zero}
Answer := Get Answer;
Writeln (' Auto-zero is enabled; response : ',Answer);

```

```

( Set date in M 901/902 (do not change this date setting) )
Delay (250);
Writeln (AUX,'SETDATE 0,1,1,0');
Answer := Get_Answer;
Writeln (' Date is set to SUN-1-JAN-190; response : ',Answer);

    Writeln (' ');
    Writeln (' Press any Key to continue ... ');
    Repeat Until KeyPressed;
Clrscr;

Writeln (' ');
Write (' Enter output file name for data          : ');
Readln (outfill);
Write (' Enter tip clearance size (t/c)              : ');
Readln (TipClr);
Write (' Enter tunnel contraction factor              : ');
Readln (Contraction);
Write (' Enter initial atmospheric pressure (mmHg)      : ');
Readln (Patm);
Write (' Enter initial total temperature (K)            : ');
Readln (T0);
Write (' Enter test date                                : ');
Readln (TestDate);
Write (' Enter transducer calibration coef. for P-Pref    : ');
Readln (CalCofP);
Write (' Enter transducer calibration coef. for Psc1-Psc2 : ');
Readln (CalCofQ);
Clrscr;

( Go to procedure SHProbe Trv for data acquisition : )
SHProbe_Trv(outfill, CPU_Option, Waiting_Time, Sampling_Time,
            TipClr, Contraction, Channell, Channel2, TestDate,
            Patm, T0, CalcofP, CalcofQ);

Write (' Do you want to perform warm or cold boot to M '
        ,CPU_Option:3,' (W/C) : ');
Readln (Answer);
If (Answer = 'C') or (Answer = 'c') then
Writeln (AUX,'CPURESET 1') else Writeln (AUX,'CPURESET 0');
If (CPU_Option = 901) then
Answer := Get_Answer;
Write (' In 30 seconds the M ',CPU_Option:3,' will reboot');
If (CPU_Option = 901) then Writeln ('; response : ',Answer);

Writeln (' ');
Writeln ('Serial Port Status: ',Binary(SerialStatus(1)));

( Cleaning up all the changes made to the interrupt vectors )
RS_Cleanup;
end.

```

SUBROUTINE LIBRARY

```

Type Registers = Record
    Case Integer of
        1: (AX,BX,CX,DX,BP,SI,DI,DS,ES,Flags:Integer);
        2: (AL,AH,BL,BH,CL,CH,DL,DH:Byte);
    End;
    Long_String = String[255];
    Pointer = ^Byte;

Var Regs: Registers;
{-----}
Function Get_Time:Long_String;
Var
    Hour,Min,Sec,M:String[2];
    I:Byte;
Begin
    Regs.AH:=$2C;
    MSDos(Regs);
    Str(Regs.CL:2,Min);
    Str(Regs.DH:2,Sec);
    For I := 1 to 2 do
        Begin
            If Min[I]=' ' then Min[I]:='0';
            If Sec[I]=' ' then Sec[I]:='0';
        End;
    Case Regs.CH of
        0: I:=12;
        13..23: I:=Regs.CH-12;
        else I:=Regs.CH;
    End; {of case}
    Str(I:2,Hour);
    If Regs.CH < 12 then
        M:='AM'
    else
        M:='PM';
    Get_Time:=Hour+':'+Min+':'+Sec+' '+M;
End; {of get_time}
{-----}
Type Interrupt_Block=Record
    Interrupt_OldIP:Integer;
    Interrupt_OldCS:Integer;
    Interrupt_Number:Byte;
End;

Const Interrupt_Ds:Integer = 0;

Procedure Install_Interrupt(Interrupt:Byte; Interrupt_Handler:Integer;
    Var Block:Interrupt_Block);
Begin
    Interrupt_DS:=DSeg;
    Block.Interrupt_Number:=Interrupt;
    Regs.AH:=$35;
    Regs.AL:=Interrupt;
    MSDos(Regs);
    Block.Interrupt_OldCS:=Regs.ES;
    Block.Interrupt_OldIP:=Regs.BX;
    Regs.AH:=$25;
    Regs.AL:=Interrupt;
    Regs.DS:=CSeg;
    Regs.DX:=Interrupt_Handler;
    MSDos(Regs);
End; {of Install_Interrupt}

Procedure Uninstall_Interrupt (Var Block:Interrupt_Block);
Begin
    Regs.AH:=$25;
    Regs.AL:=Block.Interrupt_Number;
    Regs.DS:=Block.Interrupt_OldCS;
    Regs.DX:=Block.Interrupt_OldIP;

```

```

    MSDos(Regs);
    End; {of Uninstall_Interrupt}

type INS8250 = Record
    THR : Integer;    {Transmit holding register}
    RBR : Integer;    {Receive holding register}
    IER : Integer;    {Interrupt enable register}
    LCR : Integer;    {Line control register}
    MCR : Integer;    {Modem control register}
    LSR : Integer;    {Line status register}
    MSR : Integer;    {Modem status register}
    End;

    RS_Baud = (B110,B150,B300,B600,B1200,B2400,B4800,B9600);
    RS_Parity = (None,Odd,Nevermind,Even);

Const RS232:Array[1..2] of INS8250 =
    ((THR:$3F8;RBR:$3F8;IER:$3F9;LCR:$3FB;MCR:$3FC;LSR:$3FD;MSR:$3FE),
     (THR:$2F8;RBR:$2F8;IER:$2F9;LCR:$2FB;MCR:$2FC;LSR:$2FD;MSR:$2FE));
    RS_Buffer_Size = 4095;
Var RS_Error:Byte;
    RS_Buffer:Array[0..RS_Buffer_Size] of Byte;
    RS_Buf_Head,RS_Buf_Tail:Integer;
    RS_Com:Byte;
    RS232_Block:Interrupt_Block;
    AuxIn,AuxOut:Integer;
    {-----}
Procedure RS232_Interrupt;
Begin
    Inline ($50/           {PUSH AX}
           $53/           {PUSH BX}
           $51/           {PUSH CX}
           $52/           {PUSH DX}
           $57/           {PUSH DI}
           $56/           {PUSH SI}
           $06/           {PUSH ES}
           $1E/           {PUSH DS}
           $FB/           {STI}
           $2E/$A1/Interrupt_DS/ {MOV AX,CS:Interrupt_DS}
           $8E/$D8);      {MOV DS,AX}
    RS_Error:=Port[RS232[RS_Com].LSR] and $1E;
    RS_Buffer[RS_Buf_Tail]:=Port[RS232[RS_Com].RBR];
    RS_Buf_Tail:=(RS_Buf_Tail+1) mod (RS_Buffer_Size+1);
    Inline ($FA);        {CLI}
    Port[$20]:=$20;      {CLEAR INTERRUPT FLAG}
    Inline ($1F/         {POP DS}
           $07/         {POP ES}
           $5E/         {POP SI}
           $5F/         {POP DI}
           $5A/         {POP DX}
           $59/         {POP CX}
           $5B/         {POP BX}
           $58/         {POP AX}
           $8B/$E5/      {MOV SP,BP}
           $5D/         {POP BP}
           $CF);         {IRET}
End; {of RS232_Interrupt}
    {-----}
Procedure RS_Break;
Begin
    Port[RS232[RS_Com].LCR]:=Port[RS232[RS_Com].LCR] or $40;
    Delay(200);          {1/5 SECONDS}
    Port[RS232[RS_Com].LCR]:=Port[RS232[RS_Com].LCR] and $BF;
End; {of RS_Break}
    {-----}
Function RS232_Avail:Boolean;
Begin
    RS232_Avail:=RS_Buf_Head <> RS_Buf_Tail;
End; {of RS232_Avail}

```

```

Function RS232_In:Char;
Begin
  While RS_Buf_Head = RS_Buf_Tail do Delay(10);
  RS232_In:=Char(RS_Buffer[RS_Buf_Head]);
  RS_Buf_Head:=(RS_Buf_Head+1) mod (RS_Buffer_Size+1);
End; {of RS232_In}
{-----}
Procedure RS232_Out(Param:Char);
Var I:Byte;
Begin
  While ((Port[RS232[RS_Com].LSR] and $20) <> $20) {Transmit Reg is empty}
    do Delay(1);
  Port[RS232[RS_Com].MCR]:= $0B; { Set Request to send}
  While ((Port[RS232[RS_Com].MSR] and $30) <> $30) {DSR and CTS}
    do Delay(1); {Wait a while}
  Port[RS232[RS_Com].THR]:=Byte(Param);
  RS_Error:=0;
End;
{-----}
Procedure RS_Initialize(Com:Byte;Speed:RS_Baud;P:RS_Parity;Stop,Length:Byte);
Begin
  RS_Com:=Com;
  Regs.DX:=RS_Com-1;
  Regs.AX:=Ord(Speed)*32 + Ord(P)*8 + (Stop-1)*4 + Length-5;
  Intr($14,Regs);
  Install_Interrupt($0D-Com,Ofs(RS232_Interrupt),RS232_Block);
  If RS_Com = 1 then Port[$21]:=Port[$21] and $EF
  else Port[$21]:=Port[$21] and $F7;
  Port[RS232[RS_Com].LCR]:=Port[RS232[RS_Com].LCR] and $7F;
  Port[RS232[RS_Com].IER]:=1;
  Port[RS232[RS_Com].MCR]:= $0B;
  AuxIn:=AuxInPtr;
  AuxOut:=AuxOutPtr;
  AuxInPtr:=Ofs(RS232_In);
  AuxOutPtr:=Ofs(RS232_Out);
  RS_Buf_Head:=0;
  RS_Buf_Tail:=0;
End; { of RS_Initialize}
{-----}
Procedure RS_Cleanup;
Begin
  Port[$21]:=Port[$21] or $18;
  Port[RS232[RS_Com].LCR]:=Port[RS232[RS_Com].LCR] and $7F;
  Port[RS232[RS_Com].IER]:=0;
  Port[RS232[RS_Com].MCR]:=0;
  Uninstall_Interrupt(RS232_Block);
  AuxInPtr:=AuxIn;
  AuxOutPtr:=AuxOut;
End; {of RS_Cleanup}
{-----}
Procedure _Int14(PortNumber,Command,Parameter: Integer);
{ Do a BIOS COM driver interrupt }
Begin
  With Regs Do
  Begin
    DX:=PortNumber-1;
    AH:=Command;
    AL:=Parameter;
    Flags:=0;
    Intr($14,Regs);
  End;
End;
{-----}
Function SerialStatus(PortNumber: Integer): Integer;
{ Return the status of a COM port }
Begin
  _Int14(PortNumber,3,0);
  SerialStatus:=Regs.AX;
End;

```

```

Function Binary(V: Integer): Long String;
{ Converts an integer number to binary format; it is used to convert the
  output of function "serial status" into binary form. }
Var
  I: Integer;
  B: Array [0..3] Of String[4];
Begin
  For I:=0 To 15 Do
    If (V And (1 Shl (15-I)))<>0 Then B[I Div 4][(I Mod 4)+1]:='1'
    Else B[I Div 4][(I Mod 4)+1]:='0';
  For I:=0 To 3 Do B[I][0]:=Chr(4);
  Binary:=B[0]+' '+B[1]+' '+B[2]+' '+B[3];
End;
-----}
Procedure SetTime (Hours, Minutes, Seconds, Hundreths:Integer; var Status: byte);
{ Sets the system clock to the time specified.

Parameters:
  Hours      (in) - hour of the day (0-23)
  Minutes    (in) - (0-59)
  Seconds    (in) - (0-59)
  Hundreths  (in) - (0-99)
  Status     (out)- sucess status of the procedure 0 = successful
                                           1 = invalid time
  (invalid times will have no effect on the system clock)}

begin
  with REGS do
    begin
      { set the registers to set the time in the system clock }
      AX := $2D00; { Dos function Code }
      CX := (HOURS shl 8) + Minutes;
      DX := (Seconds shl 8) + Hundreths;

      { invoke the DOS interrupt that sets the current time}
      MSDos (REGS);

      { return the sucess status based upon the current time }
      If (Lo(AX) = $FF) then
        Status := 1
      else
        Status := 0;
    end;
  end;
-----}
Procedure GetTime (var Hours, Minutes, Seconds, Hundreths: Integer);
{ Accesses the time of day clock to obtain the current time of day.
  Parameters: similar to the ones in procedure SetTime }

begin
  with REGS do
    begin
      {set AH to request the current time contained in the time of day clock}
      AX := $2C00; { DOS function code }

      { invoke the DOS function that returns the current time }
      MSDos (REGS);

      { return time as hours, minutes, seconds and hundreths of seconds}
      Hours := Hi(CX);
      Minutes := Lo(CX);
      Seconds := Hi(DX);
      Hundreths := Lo(DX);
    end;
  end;
end;

```

```

Procedure Step_Home_Scanivalve (K1, K2:byte);
( Sends the byte loaded into register byte DL to the parallel port using the
  interrupt function $21.)

Begin
  With Regs Do
    Begin
      DX:= 0;
      AH:= $05;
      DL:= K1;
      Flags:=0;
      MSDOS(Regs); ( same as Intr($21,Regs) )

      DX:= 0;
      AH:= $05;
      DL:= K2;
      Flags:=0;
      MSDOS(Regs);

      DX:= 0;
      AH:= $05;
      DL:= K1;
      Flags:=0;
      MSDOS(Regs);
    End;
  End;
}-----}
Function Get_Answer:Long_String;
( Is used to receive responses from M 901/902 )

Var
  Character : Char;
  AW       : String[25];
Begin
  Aw := '';
  Character := ' ';
  Repeat
    If RS232_Avail then
      Begin
        Read(AUX,Character);
        If (Character <> #00) then
          Aw := Aw + Character;
        End;
      Until (Character = #50A);
    Get_Answer := Aw;
  End;
}-----}
( Procedure for communication with interface CPU module M 901 or M 902

Differences between the two modules :

- M 902 samples 2.5 times faster than M 901.
- M 902 has a larger RAM (random access memory) than M 901 to accomodate larger
  amount of data.
- Cold and warm reset effects are slightly different depending on whether 901 or
  902 is used.
- Turning the module power off has different effects on 901 and 902; 902 is
  protected against a power failure; it does not loose any data or commands, and
  continues sampling when power comes back on.
- There are some differences between the commands accepted by M 901 and M 902.
- M 902 has a more advanced software built into its ROM (Ver 3.00). )

Procedure Communication(CPU_Option, Waiting_Time,Sampling_Time: Integer;
  Channell,Channel2:Char;Var Data1,Data2: Array1;
  Var L :Integer; Var Data_Error : Boolean);

( This procedure performs sampling for a specified period, scanning two
  channels sequentially

```

```

Input :      CPU_Option : has a value of 901 or 902 depending on which
               interface CPU module is used
               Waiting period (seconds)
               Sampling period (seconds)
               Channel numbers to be read (1-7) (Channel 0 is used for
               Auto-Zero, if enabled)

Output :      Data1 : Array containing samples from the first channel
               Data2 : Array containing samples from the second channel
               L      : Number of samples obtained from each channel )

Var
Keep_Sc, Keep_Mn, Counter2,
J, JJ, K, Code, Mn, Sc, G1, G2,
Hours, Minutes, Seconds, Hundreths
Comma_Location, Status
Data
Message, Info
Answer, Answer2
Command1, Command2
Gain1_init, Gain2_init, Gain1, Gain2, Ch
Data1_Max, Data2_Max, Data1_Min, Data2_Min
Increase_Gain, First_Pass
: Integer;
: Byte;
: Array [1..1000] of String[20];
: String[40];
: String[10];
: String[25];
: Char;
: Real;
: Boolean;

Const
Temp_Sc = 10; Temp_Mn = 0;

Begin
Data_Error := False;

( - For this procedure to execute properly, the CPU Module (M 901 or 902)
  should be set to stand-alone mode.
  - The procedure is written so that readings are taken sequentially from two
  channels specified by the user. If this is to be changed, the number of
  "DCV para1,para2" commands loaded into the M901/902 memory and the output
  format have to be adjusted. For example if readings are to be taken from 3
  channels sequentially, three DCV para1,para2 commands will have to be loaded
  into the M 901/902 memory prior to sampling. The output from M901/902 will
  consist of one information line followed by data lines, same as two channel
  sampling. The difference will be that each data line will contain 3 samples
  separated by commas instead of two samples. Thus the portion of the procedure
  where the received data strings are divided into individual samples must be
  modified.)

If (Sampling_time < 60) then begin
  Mn := 0;
  Sc := Sampling_time;
end;
If (Sampling_time >= 60) then begin
  Mn := 1;
  Sc := Sampling_time - 60;
end;
If (Sampling_time >= 120) then begin
  Mn := 2;
  Sc := Sampling_time - 120;
end;
If (Sampling_time >= 180) then begin
  Mn := 3;
  Sc := Sampling_Time - 180;
end;

  { initially assuming Gain=0 for channel1 and channel2 }
  Gain1 := '0'; Gain2 := '0';
  First_Pass := True;
  Counter2 := 1;

```



```

Repeat
  { Control returns here after the initial 10 seconds data is collected
    and analyzed to determine the appropriate gain for the sampling period }

  If (First_Pass = True) then begin
    { Sample for 10 seconds to determine the appropriate gain for each channel}
    Keep_Sc := Sc;
    Keep_Mn := Mn;
    Sc := Temp_Sc;
    Mn := Temp_Mn;
  End;

  { Clear any commands loaded to M 901/902 memory : }
  Clrscr;
  Writeln ( ' ');
  Delay (250);
  Writeln (AUX,'@CLRLIST');
  Answer := Get_Answer;
  Writeln ( ' M ',CPU Option:3,
    ' memory is cleared of commands; response : ',Answer);

  { Load the necessary commands to the M 901/902 memory }
  Command1 := '+A DCV '+Channel1+', '+Gain1;
  Command2 := '+A DCV '+Channel2+', '+Gain2;
  Delay (250);
  Writeln (AUX,Command1);
  Answer := Get_Answer;
  Writeln ( ' First command is loaded to M '
    ,CPU Option:3,' memory; response : ',Answer);
  Delay (250);
  Writeln (AUX,Command2);
  Answer := Get_Answer;
  Writeln ( ' Second command is loaded to M '
    ,CPU Option:3,' memory; response : ',Answer);

  If ( First_Pass = True ) then begin

    { Wait for Waiting_time seconds : }
    GetTime (Hours, Minutes, Seconds, Hundreths);
    Writeln ( ' Time : ',Hours:2,',':',Minutes:2,',':',Seconds:2,',':',Hundreths:2);
    Repeat
      SetTime (Hours, Minutes, 0, 0, Status); { Set seconds to 00 }
    Until (Status = 0);
    Writeln ( ' Time initialized (seconds set to 00)');
    Writeln ( ' Waiting period is started');
    Repeat
      GetTime (Hours, Minutes, Seconds, Hundreths);
    Until (Seconds >= Waiting_time);
    Writeln ( ' Waiting period is completed ');
  end;

  { Start scanning : }
  Delay (250);
  Writeln (AUX,'@SCAN -1,0.01,0.01');
  Answer := Get_Answer;
  Writeln ( ' ');
  Writeln ( ' Scanning is started; response : ',Answer);

  { Set minutes and seconds to 00 }
  GetTime (Hours, Minutes, Seconds, Hundreths);
  Writeln ( ' Time : ',Hours:2,',':',Minutes:2,',':',Seconds:2,',':',Hundreths:2);
  Repeat
    SetTime (Hours, 0, 0, 0, Status);
  Until (Status = 0);
  Writeln ( ' Time initialized (minutes and seconds set to 00)');

  Repeat
    GetTime (Hours, Minutes, Seconds, Hundreths);

```

```

Until (Seconds >= Sc) and (Minutes = Mn);

{ sampling period is completed; stop scanning }
Writeln (AUX,'@AUTOSCAN 0');
Answer := Get_Answer;
GetTime (Hours, Minutes, Seconds, Hundreths);
Writeln (' Time : ',Hours:2,':',Minutes:2,':',Seconds:2,':',Hundreths:2);
Writeln (' Scanning is stopped; response : ',Answer);
Writeln (' ');
Clrscr;
Writeln (' Complete output transfered from M ',CPU_Option:3);

    { Initialize parameters for loop to transfer data to the computer }
    J := 1;
    JJ := 1;
    Message := '';

    { Send results to the computer }
    If (CPU_Option = 901) then
        Writeln (AUX,'@DUMP 0')
    else
        Writeln (AUX,'@DUMP 0,0,10000');
    Delay (250);
Repeat
    While (RS232_Avail and (J < JJ+2)) do begin
        Repeat
            Read(AUX,Ch);
            Write(Ch);
            If (Ch <> #00) and (Ch <> #50A) then { 50A = CReturn byte }
                Message := Message+Ch;
        Until (Ch = #50A);
        If (Message <> '') then begin
            If J = 1 then begin
                Info := Message;
                If (Length(Info) < 10) then J := J-1 else begin
                    {Length(Message) returns string length+1}
                    Message := Copy(Message,21,Length(Message)-21);
                    Val (Message,JJ,Code);
                    { JJ is the number of data acquired from each channel }
                    If (Code <> 0) then Writeln (' ERROR!!!');
                end;
            end
            else
                Data[J-1] := Message;
            Message := '';
            J := J+1;
        end;
    end;
Until (J = JJ+2);

    L := J-2;

{ The following section of the procedure extracts the channell and
channel2 readings as real numbers from the data strings. It makes use of
the fact that the data strings are always in one of the following formats:
    Gain = 0 :
        #.####, #.####
        -#.####, #.####
        #.####, -#.####
        -#.####, -#.####

    Gain = 1 :
        #.#####, #.#####
        -#.#####, #.#####
        #.#####, -#.#####
        -#.#####, -#.#####

    Gain = 2 :
        #.#####, #.#####

```

```

-#.#####, #.#####
#.#####,-#.#####
-#.#####,-#.#####

Gain = 3 :
#.#####, #.#####
-#.#####, #.#####
#.#####,-#.#####
-#.#####,-#.##### }

{ Converting the gain values from character to integer : }
Val(Gain1,G1,Code); Val(Gain2,G2,Code);

For K := 1 to L do begin
  Comma_Location := Pos(',',Data[K]);
  If Comma_Location = 6 + G1 then
    begin
      { Data from first channel is positive }

      Val(Copy(Data[K],1,5+G1),Data1[K],Code);
      If (Code <> 0) then begin
        Writeln (' Error at ',Code:2,' when converting data1 # ',
          K:3,' into a real number ');
        Data_Error := True;
      end;

      If (Copy(Data[K],7+G1,1) = '-') then begin
        { Data from second channel is negative }
        Val(Copy(Data[K],7+G1,6+G2),Data2[K],Code);
        If (Code <> 0) then begin
          Writeln (' Error at ',Code:2,' when converting data2 # ',
            K:3,' into a real number ');
          Data_Error := True;
        end;
      end
    end
  else begin
    { Data from second channel is positive }
    Val(Copy(Data[K],7+G1,5+G2),Data2[K],Code);
    If (Code <> 0) then begin
      Writeln (' Error at ',Code:2,' when converting data2 # ',
        K:3,' into a real number ');
      Data_Error := True;
    end;
  end
end
else
begin
  { Data from first channel is negative }

  Val(Copy(Data[K],1,6+G1),Data1[K],Code);
  If (Code <> 0) then begin
    Writeln (' Error at ',Code:2,' when converting data1 # ',
      K:3,' into a real number ');
    Data_Error := True;
  end;

  If (Copy(Data[K],8+G1,1) = '-') then begin
    { Data from second channel is negative }
    Val(Copy(Data[K],8+G1,6+G2),Data2[K],Code);
    If (Code <> 0) then begin
      Writeln (' Error at ',Code:2,' when converting data2 # ',
        K:3,' into a real number ');
      Data_Error := True;
    end;
  end
end
else begin
  { Data from second channel is positive }
  Val(Copy(Data[K],8+G1,5+G2),Data2[K],Code);
  If (Code <> 0) then begin

```

```

        Writeln ( ' Error at ',Code:2,' when converting data2 # ',
                  K:3,' into a real number ');
        Data_Error := True;
      end;
    end;
  end;
end;

If ( First_Pass = True ) then begin
{ Determining the extreme readings from each channel : }

  Data1_Max := Data1[1];
  Data1_Min := Data1[1];
  Data2_Max := Data2[1];
  Data2_Min := Data2[1];
  For K := 1 to L do begin
    If Data1[K] > Data1_Max then Data1_Max := Data1[K];
    If Data2[K] > Data2_Max then Data2_Max := Data2[K];
    If Data1[K] < Data1_Min then Data1_Min := Data1[K];
    If Data2[K] < Data2_Min then Data2_Min := Data2[K];
  end;

  { Check if it is possible to read any of the channels with a
  higher gain : }

  Gain1_init := Gain1; Gain2_init := Gain2;

  If (Data1_Max < 0.450) and (Data1_Min > -0.450) then Gain1 := '1';
  If (Data1_Max < 0.040) and (Data1_Min > -0.040) then Gain1 := '2';
  If (Data1_Max < 0.007) and (Data1_Min > -0.007) then Gain1 := '3';

  If (Data2_Max < 0.450) and (Data2_Min > -0.450) then Gain2 := '1';
  If (Data2_Max < 0.040) and (Data2_Min > -0.040) then Gain2 := '2';
  If (Data2_Max < 0.007) and (Data2_Min > -0.007) then Gain2 := '3';

  If ( Gain1 <> Gain1_init ) or ( Gain2 <> Gain2_init ) then
    Increase_Gain := True else Increase_Gain := False;

  First_Pass := False;
  Sc := Keep_Sc;
  Mn := Keep_Mn;
End;

If (Increase_Gain = True) and (Counter2 < 2) then begin
  Writeln ( ' ');
  Writeln ( 'Channel # ',Channel1,' is increased to Gain : ',Gain1);
  Writeln ( 'Channel # ',Channel2,' is increased to Gain : ',Gain2);
  Writeln ( ' ');
end;

Counter2 := Counter2 + 1;

Until ( Counter2 > 2 );

Writeln ( ' ');
Delay (250);
Writeln (AUX,'@CLRLIST');
Answer := Get_Answer;
Writeln ( ' Commands in M ',CPU_Option:3,
          ' memory are erased; response : ',Answer);
Writeln ( ' ');
Clrscr;
End;

```

```

Procedure Motor_Step(Controller_Option:Integer;
                    Which_Motor_And_Direction:long_string; Distance:real;
                    Clock_Speed:Integer);

Procedure Send_To_Port (K:byte);
{ This procedure Will send the byte loaded into register byte DL to the
  parallel port using the Interrupt function $21 }
Begin
  With Regs Do
    Begin
      DX:= 0;
      AH:= $05;
      DL:= K;
      Flags:=0;
      MSDOS(Regs); { same as Intr($21,Regs) }
    End;
  End;

Var
  I, N, CS                : Integer;
  K1, K2                  : Byte;

Const
  HOR_MOTOR_CALB_COEF  = 0.00635; {MM/PULSE}
  VER_MOTOR_CALB_COEF  = 0.00635; {MM/PULSE}
  PITCH_MOTOR_CALB_COEF = 0.01; {DEG/PULSE}
  YAW_MOTOR_CALB_COEF  = 0.01; {DEG/PULSE}

BEGIN
  If (Clock_Speed = 4) then CS := 0 else CS := 1;

  If (Which_Motor_And_Direction = 'HCW') or
    (Which_Motor_And_Direction = 'HCCW')
  then begin
    N := Round(Distance/Hor_Motor_Calb_Coef);
    Writeln (' ');
    Writein (' sending ',N:3,' pulses to the span linear motion motor ');
    Writeln (' ');
  end;

  If (Which_Motor_And_Direction = 'VCW') or
    (Which_Motor_And_Direction = 'VCCW')
  then begin
    N := Round(Distance/Ver_Motor_Calb_Coef);
    Writeln (' ');
    Writein (' sending ',N:3,' pulses to the pitch linear motion motor ');
    Writeln (' ');
  end;

  If (Which_Motor_And_Direction = 'PCW') or
    (Which_Motor_And_Direction = 'PCCW')
  then begin
    N := Round(Distance/Pitch_Motor_Calb_Coef);
    Writeln (' ');
    Writein (' sending ',N:3,' pulses to the pitch angular motion motor ');
    Writeln (' ');
  end;

  If (Which_Motor_And_Direction = 'YCW') or
    (Which_Motor_And_Direction = 'YCCW')
  then begin
    N := Round(Distance/Yaw_Motor_Calb_Coef);
    Writeln (' ');
    Writein (' sending ',N:3,' pulses to the yaw angular motion motor ');
    Writeln (' ');
  end;

  If (Controller_Option = 1) then begin

```

```

If (Which_Motor_And_Direction = 'VCW') or
  (Which_Motor_And_Direction = 'PCW') Then begin
  K1 := 32; K2 := 96;
  For I := 1 TO N Do Begin
    Delay(CS);
    Send_to_port (K1);
    Delay(CS);
    Send_to_port (K2);
  End;
End;
If (Which_Motor_And_Direction = 'VCCW') or
  (Which_Motor_And_Direction = 'PCCW') then begin
  K1 := 32; K2 := 48;
  For I := 1 TO N Do Begin
    Delay(CS);
    Send_To_Port (K1);
    Delay(CS);
    Send_To_Port (K2);
  End;
End;
If (Which_Motor_And_Direction = 'HCW') or
  (Which_Motor_And_Direction = 'YCW') then begin
  K1 := 4; K2 := 12;
  For I := 1 TO N do begin
    Delay(CS);
    Send_To_Port (K1);
    Delay(CS);
    Send_To_Port (K2);
  end;
end;
If (Which_Motor_And_Direction = 'HCCW') or
  (Which_Motor_And_Direction = 'YCCW') then begin
  K1 := 4; K2 := 6;
  For I := 1 TO N do begin
    Delay(CS);
    Send_To_Port (K1);
    Delay(CS);
    Send_To_Port (K2);
  end;
end;
end;

If (Controller_Option = 2) then begin

  If (Which_Motor_And_Direction = 'VCW') or
    (Which_Motor_And_Direction = 'PCW') Then begin
    K1 := 32; K2 := 96;
    For I := 1 TO N Do Begin
      Delay(CS);
      Send_to_port (K1);
      Delay(CS);
      Send_to_port (K2);
    End;
  End;
  If (Which_Motor_And_Direction = 'VCCW') or
    (Which_Motor_And_Direction = 'PCCW') then begin
    K1 := 32; K2 := 48;
    For I := 1 TO N Do Begin
      Delay(CS);
      Send_To_Port (K1);
      Delay(CS);
      Send_To_Port (K2);
    End;
  End;
  If (Which_Motor_And_Direction = 'HCW') or
    (Which_Motor_And_Direction = 'YCW') then begin
    K1 := 8; K2 := 10;
    For I := 1 TO N do begin

```

```

        Delay(CS);
        Send_To_Port (K1);
        Delay(CS);
        Send_To_Port (K2);
    end;
end;
If (Which_Motor_And_Direction = 'HCCW') or
   (Which_Motor_And_Direction = 'YCCW') then begin
K1 := 12; K2 := 14;
    For I := 1 TO N do begin
        Delay(CS);
        Send_To_Port (K1);
        Delay(CS);
        Send_To_Port (K2);
    end;
end;

end;
K1 := 0;
Send_To_Port (K1);    (Turn the motors off)
end;

```

Appendix E

Transducer calibration curves.

Figure E.1 Rosemount (serial no. 591) pressure transducer calibration curve.

Figure E.2 Rosemount (serial no. 633) pressure transducer calibration curve.

Appendix F

Three-hole pressure probe calibration procedure and calibration curves.

A method outlined by Lewis (1966) is followed for calibrating the three-hole probe. In that procedure the following coefficients are employed:

$$\begin{aligned}
 K_1 &= \frac{(P_1 - P)}{q} \\
 K_{12} &= \frac{(P_1 - P_2)}{q} \\
 K_{13} &= \frac{(P_1 - P_3)}{q} \\
 K_{23} &= \frac{(P_2 - P_3)}{q} \\
 K_{\phi 2} &= \frac{(P_1 - P_2)}{(P_1 - P_3)} \\
 K_{\phi 3} &= \frac{(P_1 - P_3)}{(P_1 - P_2)}
 \end{aligned}
 \tag{F.1}$$

where the pressure indices refer to the port numbering shown in Figure F.1. The three-hole probe can only be used for measuring the yaw angle in the plane of the probe. During the reduction of probe data the sign of the flow angle relative to the probe axis is determined by comparing the probe pressures P_2 and P_3 . The magnitude of the angle is obtained from the coefficients $K_{\phi 2}$ and $K_{\phi 3}$. The employment of two coefficients instead of one provides better accuracy in extracting the flow angle. As seen from Figure F.2, $K_{\phi 2}$ is not suitable for flow angle ranges other than -50° to 0° and 50° to 70° . Therefore, for the remaining part of the flow angle range considered, the coefficient $K_{\phi 3}$ is used instead (Figure F.3).

Figure F.1 Three-hole pressure probe

Figure F.2 Calibration data for variation of K_{ϕ_2} with flow angle relative to the probe axis.

Figure F.3 Calibration data for variation of K_{ϕ_3} with flow angle relative to the probe axis.

Once the flow angle is determined, the local dynamic pressure is extracted from the coefficients K_{12} , K_{13} or K_{23} . As seen from Figure F.4, K_{12} is Reynolds number dependent in the positive flow angle range. The same is true for negative flow angles for K_{13} (Figure F.5). Therefore, for positive and negative flow angles K_{12} and K_{13} are used respectively. There is an exception to this whereby these two coefficients approach zero. Since in the calculation of the dynamic pressure the coefficient appears as the denominator, if it is close to zero, a small change in its value will result in large changes in the dynamic pressure. This in turn will increase the uncertainty in the calculated dynamic pressure. Therefore, for regions where the coefficients K_{12} and K_{13} are close to zero, which correspond to -30° to -10° and 10° to 30° flow angle ranges, K_{23} is used instead (Figure F.6).

Next to dynamic pressure, the static pressure is extracted from the coefficient K_1 . As seen from Figure F.7, the coefficient does not appear to be dependent on Reynolds number throughout the calibration range.

Piecewise least-square polynomials were fitted to the above described calibration curves. The order of the polynomials were determined by trial and error and were typically within 6 to 8.

Figure F.4 Calibration data for variation of K_{12} with flow angle relative to the probe axis.

Figure F.5 Calibration data for variation of K_{13} with flow angle relative to the probe axis.

Figure F.6 Calibration data for variation of K_{23} with flow angle relative to the probe axis.

Figure F.7 Calibration data for variation of K_1 with flow angle relative to the probe axis.

Appendix G

Seven-hole pressure probe calibration procedure and calibration curves.

The calibration procedure described herein was first proposed by Gerner and Maurer (1981). For small angles relative to the probe axis for which the flow remains attached at all seven pressure ports the following non-dimensional parameters are defined:

$$\begin{aligned} C_{pitch_1} &= \frac{P_4 - P_1}{P_7 - P_M} \\ C_{pitch_2} &= \frac{P_3 - P_6}{P_7 - P_M} \\ C_{pitch_3} &= \frac{P_2 - P_5}{P_7 - P_M} \end{aligned} \tag{G.1}$$

where P_M is the average of the six surrounding pressures. The corresponding pressure port numbering is given in Figure G.1. These coefficients are respectively sensitive to flow angle variations in three different planes passing through the probe axis. The denominator in the above expressions can be referred to as "pseudo-dynamic pressure" since P_7 approximates the local total pressure and P_M approximates the local static pressure. Only two of the three coefficients are necessary to define the flow direction uniquely. Instead of a random elimination of one coefficient, it is more appropriate to resolve the three coefficients into two coefficients defined within the tangential co-ordinate system (Figure G.2) as follows:

$$\begin{aligned} C_{pitch} &= \frac{1}{3}(2C_{pitch_1} + C_{pitch_2} - C_{pitch_3}) \\ C_{yaw} &= \frac{1}{\sqrt{3}}(C_{pitch_2} + C_{pitch_3}) \end{aligned} \tag{G.2}$$

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MICROCOPY RESOLUTION TEST CHART
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Figure G.1 Seven-hole probe port numbering.

Figure G.2 Tangential co-ordinate system for small flow misalignment.

These two variables can then be related to the local yaw angle (β) , pitch angle (α) , total pressure coefficient (C_{P_t}) and dynamic pressure coefficient (C_q) through the following relations:

$$\begin{aligned}\alpha &= f_1(C_\alpha, C_\beta) \\ \beta &= f_2(C_\alpha, C_\beta) \\ C_{P_t} &= f_3(C_\alpha, C_\beta) \\ C_q &= f_4(C_\alpha, C_\beta)\end{aligned}\tag{G.3}$$

which are determined by calibrating the probe in a known flow. The total and dynamic pressure coefficients in these expressions are defined as:

$$\begin{aligned}C_q &= \frac{P_\gamma - P_M}{P_o - P} \\ C_{P_t} &= \frac{P_o - P_\gamma}{P_\gamma - P_M}\end{aligned}\tag{G.4}$$

This calibration procedure is used as long as the center port reads the highest pressure among the seven ports, which ensures that the flow is attached over all side ports. For the seven-hole probe used in the cascade experiments, the low angle calibration data for a local velocity of 14 m/sec are given in terms of the above described coefficients in Figures G.3 to G.6. This calibration data was checked regularly throughout the study.

For large misalignments the flow will tend to detach over the downstream portion of the probe (Figure G.7). In this case the pressure coefficients must be defined so that they contain only the port pressures in the attached flow region. For this purpose the six side ports of the probe are grouped into six 60-degree pie-shaped sectors with each sector containing the central port and three of the side ports. A

Figure G.3 Calibration data for C_α in sector 7.

Figure G.4 Calibration data for C_β in sector 7.

Figure G.5

Calibration data for C_q in sector 7.

Figure G.6 Calibration data for C_p in sector 7.

Figure G.7 **Flow over the 7-hole probe tip for large flow misalignment.**

particular sector is employed for extracting the flow parameters if the side port on which the sector is centered has the highest pressure reading. For this large misalignment configuration it is more appropriate to use polar co-ordinates (Figure G.8). In this case the non-dimensional calibration coefficients can be defined as follows:

$$\begin{aligned} C_{\theta_n} &= \frac{P_n - P_7}{P_n - 0.5(P_{n^+} + P_{n^-})} \\ C_{\phi_n} &= \frac{P_{n^+} - P_{n^-}}{P_n - 0.5(P_{n^+} + P_{n^-})} \end{aligned} \quad \text{G.5}$$

where subscript n indicates the side port number (Figure G.1) onto which a particular side sector is centered, and n^+, n^- are the ports adjacent to port n . The parameters related to these coefficients are the pitch angle, roll angle, total pressure coefficient and dynamic pressure coefficient:

$$\begin{aligned} \theta_n &= f_1(C_{\theta_n}, C_{\phi_n}) \\ \phi_n &= f_2(C_{\theta_n}, C_{\phi_n}) \\ C_{P_{t_n}} &= f_3(C_{\theta_n}, C_{\phi_n}) \\ C_{q_n} &= f_4(C_{\theta_n}, C_{\phi_n}) \end{aligned} \quad \text{G.6}$$

where the total and dynamic pressure coefficients are defined as:

$$\begin{aligned} C_{P_{t_n}} &= \frac{P_n - P_o}{P_n - 0.5(P_{n^+} + P_{n^-})} \\ C_{q_n} &= \frac{P_n - 0.5(P_{n^+} + P_{n^-})}{P_o - P} \end{aligned} \quad \text{G.7}$$

Similar to the G.3 relations, the functions in G.6 are again obtained by calibrating the probe in a known flow. Sector 1 calibration data for a local velocity of 14 m/sec are given in Figures G.9 to G.12. The other five sectors had similar calibration curves with compatible smoothness.

Figure G.8

Polar co-ordinate system for large flow misalignment.

Figure G.9 Calibration data for C_{θ} in sector 1.

Figure G.10 Calibration data for C_ϕ in sector 1.

Figure G.11

Calibration data for C_q in sector 1.

Figure G.12

Calibration data for C_p in sector 1.

END

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FIN