The Pennsylvania State University
The Graduate School
Department of Aerospace Engineering

STUDY OF MITIGATION OF TURBINE BLADE TIP LEAKAGE FLOWS
USING TIP LEAKAGE INTERRUPTER

A Thesis in
Aerospace Engineering
by
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Submitted in Partial Fulfillment
of the Requirements
for the Degree of

Master of Science

August 2017
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ABSTRACT

The gap between the turbine rotor blade tip and turbine casing surface is essential to prevent rubbing of these two components. Due to presence of this gap, the differential pressure across the blade tip drives fluid flow from pressure side to suction side of the blade tip. This fluid flow is called tip leakage flow and is an important sources of multidisciplinary problems. The aerodynamic losses due to tip leakage flows in a turbine blade accounts for about one third of the overall aerodynamic loss of a present day turbine stage. The tip leakage fluid in general do not go through a significant expansion and work generation process. The solid surfaces in the tip area encounter the fluid areas with the highest absolute total temperature existing in the rotor frame of reference. The tip leakage flows also increase the thermal stresses, induce further thermal oxidation on the blade tip surfaces and on the casing surface. In addition, they may cause significant structural problems and also induce unsteadiness in the turbine flow field. Hence understanding of these tip leakage flows and reducing them are of outmost importance for improving energy efficiency, lifetime and reliability of any future turbine design.

The current study uses Tip Leakage Interrupters (TLI) to mitigate some of the adverse effects of the tip leakage flows and improve the efficiency of the turbine stage. The TLI introduced in this study has the ability to alter the tip region flows on the airfoil surfaces. They operate by inducing controlled vortical structures originating from strategically shaped/oriented multiple and sub-miniature vortex generators. The TLIs in this investigation were attached on the suction side of the rotating blade tip sections in the Axial Flow Turbine Research Facility (AFTRF). The AFTRF is a high-pressure, single-
stage cold-flow turbine with a 29 bladed rotor at the Department of Aerospace Engineering in the Penn State University. Three different experimental studies were completed on the TLI in which three different parameters such as the mounting location of TLI on the airfoil tip region, the number of TLIs mounted on the blade and the specific orientation of TLI were varied. The influence of the TLI on the leakage flow system was experimentally observed and interpreted using turbine exit total pressure maps obtained with very high spatial resolution. The time accurate and phase-locked total pressure data from the downstream of the modified turbine blade was collected by a high-time-response total pressure probe using a (100 KHz) dynamic pressure sensor. High resolution total pressure maps in a just downstream location of the AFTRF rotor provided detailed scans of the turbine exit field for further interpretations of flow field physics resulting from tip region flows and TLI induced flow systems. The current span-wise measurement resolution is about one percent of the blade span of 123 mm. The system generates 6000 equally spaced measurement points along one revolution of the rotor at a selected span-wise position. A phase-locked measurement system fills the 6000 bins during each revolution of the turbine rotor using a mil-spec optical shaft encoder with high encoding resolution. Each measurement bin occupies a 0.06 degrees in the circumferential direction. The system can collect about 206 data point along one blade pitch in one of the 29 rotor passages.

The results obtained from these experiments was used to draw conclusion about the effectiveness of TLIs in reducing the tip leakage flows in the turbine tip region flow field. The ultimate goal of the TLI installations is to increase the total-to-total efficiency of axial flow turbines via the suppression or mitigation of individual tip vortices from each rotor blade.
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ACKNOWLEDGEMENTS

I want to thank my advisor, Professor Cengiz Camci, for all his support and guidance in helping me achieving my academic goals. I am very grateful for the encouragement, ideas and knowledge he has provided me over the course of this research work.

Thanks to Benjamin Enders, Kirk Heller, James Miller and Rick Auhl for their prompt and outstanding technical in overcoming the technical difficulties. I also want to thank Harry Houtz and Nick Doroschenko for the maintenance and modification of the research facility.

Many thanks to Dr. Jason Town, Gohar Khokhar and Daniel Doleiden for their help in completing my research work. A very special thanks to SOLAR TURBINES and SIEMENS for their financial and technical assistance.

Last but not least I want to thank my parents for all the encouragement and support that they have given me throughout the years. I could not have accomplished this without them.
Chapter 1

Introduction

Turbomachinery industry has undergone a significant change over the past few decades. As the industry evolved from the simple impulse steam turbines to multi stage gas turbine engines there has been substantial improvement in the performance, reliability and efficiency of these machinery, with the recent gas turbines breaking the 60% efficiency barrier. Turbomachinery has a wide range of applications from small pump used in refrigerators to large gas turbine engines used in aircrafts and ground based electricity generators. They are also used to power ships, submarines, trucks, high speed trains and atmospheric flight vehicles. They are instrumental in generating electricity from different types of sources such as stream, coal, gas and wind. Even though there has been a huge improvement in understanding of these machineries there is still a long way to go to fully understand the complex flow interactions inside these systems. With new stringent environmental laws and intense industry competition there is an increasing demand for more compact, reliable and efficient turbomachinery systems. All these factors put pressure on the industry to improve the understanding of these systems. One of the major components in turbomachinery are turbines. They play a significant role in power generation and are used in various applications. As the turbomachinery industry is a global industry any small improvements in performance or efficiency of turbines would have a significant worldwide economic impact.
The main function of turbines is to convert fluid aero-thermal energy to mechanical energy through the rotation of blades. They are the main component of almost all power generation systems (wind turbines, steam turbine, gas power plant, coal power plant) and an integral part of aircraft engines. There are two frequently used types of turbines: axial turbines and radial turbines. Most modern gas turbines employed in aerospace and electricity generation use axial turbines. Great strides have been made in the design of turbines. The modern turbines are optimized for the efficient conversion of energy from fluid enthalpy to rotational energy as shown in reference [1]. In case of gas turbine engines, the turbines are cascaded in series to enhance the expansion of the fluid and thus resulting in a greater efficiency. From reference [2] it can be seen that the higher the inlet temperature of the gas, the higher the efficiency of the turbine, but is often limited by the material properties of the turbine. This is due to the properties of a typical gas turbine thermodynamic cycle which is also called the Joule Brayton cycle as shown in reference [3]. The recent development in the materials and turbine design have protected the turbines from extreme thermal and mechanical stresses during their operation while permitting the use of high temperature gas. Even though modern turbine designers can generate a turbine configuration more advanced compared to the designs of 80s and 90s, the turbine design process still lacks the complete understanding of the turbine aero-thermal flow physics as explained in reference [1]. One such flow field which needs understanding improvements is the fluid flow in between rotating turbine blades. As the turbines are collection of highly three-dimensional turbine airfoils arranged around a hub, the complete understanding of the viscous, highly rotational, unsteady and usually turbulent and compressible flow in this region is essential for the optimization of future turbine designs.
As shown in reference [4] the flow field in between the turbine blades or the flow field in the turbine passage is complex due to presence of different types of flow such as highly three-dimensional, unsteady, laminar, transitional, turbulent and vortical flows. These flow features usually interact with each other. A clear understanding of these flow fields is essential for the optimization of turbine design in terms of performance and efficiency. But due to the complex nature of the flows and the difficulty in measuring these flow fields, it is one of the least understood interaction in the field of turbomachinery. The blades in a turbine continuously expand and change the direction of the flow and in doing so, extracts the energy from the fluid. Reference [5] shows that the efficiency of this process is negatively affected by certain flow irreversibility. One such irreversibility is the tip leakage flows, which accounts for about one third of the overall aerodynamic loss in a turbine stage. Despite its significant role in loss generation, the tip leakage phenomenon still remains not fully understood.

The tip leakage flows in turbine blades are one of the most challenging flow fields to study in turbomachinery. Boyle et al [6] shows that even though the tip leakage vortex originating from each turbine blade takes up small physical space (less than 1.5% of the blade span) and has negligible mass flow rate compared to the core flow, it accounts for over 30% of overall aerodynamic losses in the turbine stage. It also poses heat transfer and structural problem for the casing and blade tip as it increases the local temperature of the flow. In addition, it also introduces unsteadiness in the flow which propagate to next turbine stage. The tip leakage flows decreases the efficiency, reliability, performance and life time of the turbine engine. Hence any design of a turbine with superior performance would require clear understanding of the tip leakage flows. Unfortunately the effect of
various geometry and flow parameters in the leakage flows are not yet fully known due to complexity of the flow field coupled with very small physical space which limits the use of large scale measuring systems in the field.

1.1  Statement of Problem

Maintaining tight gap between the rotor blade tip and outer casing is essential for maximization of performance of any turbomachinery design. This tip clearance gap cannot be reduced below certain height as it would result in mechanical complications such as rubbing of the rotor blades and casing surface. Due to the pressure difference across the pressure side and suction side of the blade tip, the flow leaks through the tip clearance and swirls to form vortex near the suction side of the blade. With the trend of increasing gas temperature and blade loading in turbines, the aerothermal knowledge of tip leakage flow is crucial. The interaction between tip vortex, blade boundary layer and the annulus wall boundary layer results in a complex flow. These complexities of the tip leakage flow field, and the difficulty to measure it, makes tip leakage one of the least understood phenomena.

Designing turbine blade tip is a rigorous process as it has to meet multidisciplinary requirements such as weight limitation, Aerodynamic and structural requirement, static and dynamic stress considerations, operating and off design conditions, and maintenance requirement. In doing so it also has to provide better performance and durability with reduced cost. But this is a difficult process as the tip leakage flows pose several problems in the turbine stage. Of those three important problems are thermal and structural problems, pressure loss across the stage and additional flow unsteadiness.
1.1.1 Thermal and Structural Problems

The inlet to the typical industrial and aviation gas turbines usually contains a high velocity hot stream of gas from the combustor. Due to high temperature and high rotational force, the turbine blades are placed in an extreme environment where they experience both thermal and mechanical stress. The flow leaking through the turbine blade tips increases the local temperature and thermal loading of the blades. The trailing edge of the blade tips are the most affected region due to its thin structure. The blade tips often melt if sufficient cooling is not provided to them, this in turn increases the tip gap height allowing more leakage mass flow and more thermal load. This phenomenon is called tip-burnout and was first studied by Bindon et al. [10]. In off design and transient conditions, the blade tip may reach even higher temperature leading to oxidation and erosion of some material. All these thermal problems can be solved by simply over cooling the blade tips or reducing the operating flow temperature, both of which negatively impacts the efficiency and performance of the turbine system. Over-cooling in general steals excessive amounts of air from the main-gas path of the high-pressure compressor stage. The measurable amounts of air extracted from the gas turbine cycle for cooling purposes may adversely affect the overall efficiency of any gas turbine system. The tip leakage flows also causes severe problems on the casing surface near the blade tip. The heat flux on the casing due to tip leakage flows are much higher than those due to passage flows as can been seen in reference [11]. It was also found that the temperature of the tip leakage flows were higher than the total temperature of the inlet flow. This was due to an extra mechanism called “rotor compressive heating” where the rotor does the work on the leakage flow by turning
and accelerating it to a whirl velocity greater than the inlet velocity. This also means that the tip leakage flows imposes severe penalty on the efficiency and performance of the system as more amount of the fluid is removed from the compressor to cool the casing surface. The tip leakage flows, when coupled with the temperature transverse migration caused by the non-uniform temperature distribution of the inlet flow from the combustor, could enhance the rubbing of blade tips against the casing during the transient operations due to the radial growth of the blades related to thermal and mechanical stress. Mentioned effect compounds the thermal and structural loading on the blades and the casing. As a result, the performance of the engine is negatively affected. Hence the flow physics behind the tip leakage flows has to be clearly understood for the management of thermal and structural loads and to improve the design of the engines.

1.1.2 Aerodynamic Problems

Since the beginning of the study of tip leakage flows, there have been several efforts done to understand and reduce its aerodynamic loss. Despite all these efforts it is yet to be completely understood. From the study of leakage flows in reference [6], it can be seen that the aerodynamic losses due to tip leakage flows accounts for 30% of the overall losses in the turbine stage. Even though these results have been generated in 1984, they are still valid for many of the current turbomachinery components, especially with and depending on the turbomachinery component, their overall efficiency can be dropped by 2% - 4% as shown in reference [1]. For example, according to Stakolich, E.G and Sromberg, W.J [12], the tip leakage flows in the turbomachinery components of the JT9D engine increases the
specific fuel consumption by 1.1%. As explained by Harvey in [1] for every 1.5% increase in tip clearance, the turbine stage efficiency is reduced by 1%. The tip leakage flows are of greater importance with high pressure turbomachinery components, like HP turbines, where even slight change of tip clearance results in significant reduction in efficiency and performance of the components.

The tip leakage flows often changes the flow properties, thus forcing the turbine blades to operate in a non-optimum condition. In reference [7] it was found that due to tip leakage vortex, the flow angle can vary up to ±40 degrees from the mean flow. The losses due to tip leakage usually propagate to its succeeding turbine stage. Hence the inlet flow to the stator row blades will have variation in both pressure and incidence in the blade-span wise direction due to the tip leakage flows from the preceding rotor, which will also be time varying. This changes the optimum flow condition for the turbine stages resulting in reduced performance.

In addition, to aerodynamic loss generation, the tip leakage flows also unloads the blade tip, making it inefficient and unable to turn the incoming flow. Thus, there is a portion of the turbine blade (blade tip) which adds weight but does not do work. The blade unloading often is not confined only to the blade tip and may propagate to other parts of the blade. The propagation of unloading alters the work done by the turbine as there is less flow turning, and ultimately, affects the stall and surge margin of the component. The tip leakage flow study in reference [8] gives a detailed description of effect of tip clearance on stall margin of the compressor. Hence, it is worth noting that it is essential to understand the flow physics of tip leakage flows as it not only affects the efficiency, but also the performance of the engine.
1.1.3 Flow Unsteadiness Problem

Tip leakage flows are one of the main source of unsteadiness in the turbine flow field. As explained in reference [13], high level of turbulence such as 8% - 10% locally were found in the turbine rotor passage. The unsteadiness and turbulence generated in the turbine stage are actually a significant sources of energy loss, as the flow energy in unsteady/turbulent field is not used in power generation. Reference [A4] shows that the three-dimensionality aspect of the tip leakage flows are felt span-wise up to about 30% of the blade-span form the tip. The mixing process downstream of the blades is fast as shown in reference [14], where the 85% of mixing is done by 40% axial chord downstream of the trailing edge. The disturbance caused by the rotor propagates to the succeeding turbine stage and distorts the inlet condition for that stage. This propagation of aerothermal disturbance eventually results in creating unsteady pressure distribution, unsteady transition, unsteady boundary layer, and noise generation in the subsequent turbine stages. This unsteadiness drastically alters the flow condition, thus making the turbine stage less efficient as it is operating in the off-design condition. They are also sources of several mechanical problems in blades such as increased blade stress, vibration and flutter. In reference [15] it was found that the tip leakage vortex is the origin for the rotor instabilities which are responsible for rotor blade excitation and noise generation.

As described above the tip leakage flows are a major source of aerodynamic, structural and thermal problems in the turbine flow field. These negative effects related to tip leakage reduce the efficiency, lifetime, quality and performance of the machines.
1.2 Background and Need

1.2.1 Thermal losses in the turbine flow field

As it has been discussed, tip leakage flows imposes severe thermal penalty on the turbine blade tips and casing surface near the blade tips. There have been many approaches to prevent the turbine material to thermally fail, of those two approaches are used predominantly to solve the thermal problem. One is to cool the material surface for those structures affected by tip leakage flows (such as blade tips and its casing surface). And the other is to modify the blade or the casing structure to minimize the tip leakage flows, and hence its thermal effect. In modern gas turbine engines, both of these techniques are employed to reduce the thermal effect of tip leakage flows. The material cooling technique can be active or passive. In case of active blade tip cooling methods, the cold fluid flow is usually stolen from the compressor air which is then passed to the blade tip and casing to cool the surface. As the air is bled from the compressor, the efficiency and performance of the engines are generally deteriorated. In a study done by Alison on F100-PW-220 engines [16], it was found that for every 1% increase in extraction of the compressor bleed air, there is 1.5% increase in its Specific Fuel Consumption (SFC). Thus, cooling a turbine stage is an expensive and complicated process as it negatively affects the performance, reliability and efficiency of the turbines, in addition to adding weight to the system. Another way to alleviate the thermal effects of the tip leakage flows is to hinder its formation in the tip gap. Numerous concepts and ideas to modify the blade tips have been proposed in this area and many of them have been successfully employed to reduce the tip leakage flows. Concepts such as squealer tips, winglet tips and shroud tip have played a significant role in the
alleviation of tip leakage flows in the modern turbine stage. The design and function of these concepts are explained in chapter 2. In reference [17], by adding a winglet to the pressure side of a blade tip, it has been shown that a 30% reduction in the local heat transfer coefficient is possible.

1.2.2 Aerodynamic losses in the turbine flow field

In addition to eliminating thermal effects, several modifications were applied to the turbine blade tip region to reduce the aerodynamic loss in the turbine stage. To achieve this reduction, most of these designs have revolved around the goal of reducing the leakage flow mass, which in turn reduces the leakage flow effects. Two important concepts to reduce the tip leakage flows are squealer tips and the shrouded tip. There have been several research done on these techniques to improve the performance of the turbine stage such as references [6], [7] and [8]. Many of these designs have been successfully employed in the modern gas turbines. The recent study by Nicole L.Key [18] shows that the squealer tips have 10% less losses than the flat tips. A detailed investigation of these tip designs and their effects are provided in the next chapter.

1.2.2 Performance losses in the gas turbine engine

Even though the modern turbine blade uses optimized blade tip designs to mitigate the effect of tip leakage vortex, they still are not successful in efficiently eliminating them. As discussed before, losses due to tip leakage flows forms a significant portion of the
overall losses in the turbine stage. It is also a source of many aerodynamic, thermal and structural complications. It reduces the lifetime, efficiency and performance of the engines. As it causes multidisciplinary problems, finding an efficient way to successfully mitigate it would also have significant gains in those disciples. Moreover, any slight improvement in the efficiency or performance of the turbine stage would have great economic impact, as it is a global industry. Any improvement in the tip leakage area requires greater understanding of the fundamental but complex physics involved. The tip leakage effect should be subjected to extensive research to gain knowledge required for developing modern, efficient, and high performance gas turbine engines.

1.3 Purpose of Study

The purpose of this study was to analyze the effect of vortices created by the Tip Leakage Interrupters (TLI) on the tip leakage flow related vortices and also to investigate its impact on the measured aerodynamic losses and performance of the turbine blades. The experimental study was performed using the single stage Axial Flow Turbine Research Facility (AFTRF) at the Pennsylvania State University. The main goal of the project was to improve the turbine stage efficiency by finding ways to mitigate the adverse effect of tip leakage vortex.

A new turbine blade tip component or flow modifier called Tip Leakage Interrupter (TLI) was tested in the Axial Flow Turbine Research Facility (AFTRF) to find its effect on the turbine blade tip leakage flows. The basic principle of a TLI is to impose strategically designed vortices in the tip region of the blades, which would counter the tip leakage
vortices created, and thereby decreasing the adverse tip leakage effect. A fast response aerodynamic pressure transducer probe was used to collect total pressure data at 26% axial chord distance from trailing edge axially downstream of the turbine blades. The probe was also moved radially with 1% blade-span increments to collect span wise variation of the dynamic total pressure. Based on the results each blade tip design was assigned a “Figure of Merit” which then was used to find the efficiency of the design. Three parameter associated with the TLI such as mounting location of TLI on the blade tip, number of TLI attached on the blade tip and specific orientation of the TLIs were varied an attempt to optimize the TLI design.

The research effort have attempted to study the effect of TLIs on tip leakage flows. As a result of this study, novel turbine blade tip design with improved mitigation of tip leakage flows were expected. Another goal of this study was to understand the flow physics of the interaction between the newly introduced vortices and tip leakage vortices.
Chapter 2

Fundamentals of turbine fluid flow field

To study the tip leakage flows it is necessary to understand the basic working principles of the axial turbine stage. Axial flow turbines have been in use for centuries. Though the purpose of its usage have changed throughout the time, its basic function of extracting energy from fluid and converting it into mechanical work remains the same. In modern gas turbine engines, the turbines are used to generate power from a hot gas stream flowing through it. It has several components, and of those, two interesting ones are the stator and the rotor. The stator is the stationary component which turns and guides the incoming gas stream such that it is at desired angle to the rotor. The rotor is the rotating component which turns the flow and extracts the energy from the gas stream. Both the stator and rotor work based on the Newton’s 2nd and 3rd law. The stator as it is a fixed component, the tangential momentum of the fluid is changed by the stator blades and the force exerted by the gas stream onto the stator blades is reacted by the structure of the engine. In the case of the rotor, as it is a rotating component, the tangential momentum of the fluid is changed by the rotor blades and the force exerted by the gas stream onto the blades provides a moment in the rotor blade row to rotate and thus extracting energy (rotational work) from the fluid.

2.1 Turbine working process

The working process of turbine stage is described in reference [3] and shown in Figure 2.1. The inlet to the stator, the exit of the stator and the exit of rotor is represented
by 1, 2, and 3 respectively. The gas stream is guided by a stator and in the process also slightly expanded. There is a drop in pressure between the inlet and the exit of the stator. Even though there is no work extraction in the stator, as the process is non-isentropic, there is also slight drop in the total pressure. The expansion process in the rotor blades is represented in the curve 2-3 and there is a drop in the total pressure at the exit of the rotor blade. The point 3ss represents the isentropic process in both the stator and the rotor. The efficiency of the stage is given by equation 2.1

$$\eta_t = \frac{Actual \ work \ output}{Ideal \ work \ output}$$

$$\eta_t = \frac{T_{01} - T_{03}}{T_{01} - T_{03ss}}$$

$$\eta_t = \frac{1 - \frac{T_{03}}{T_{01}}}{1 - \left(\frac{P_{03}}{P_{01}}\right)^{\gamma-1}}$$

---Equation 2.1

### 2.2 Turbine flow field

Different types of flow such as highly three dimensional viscous laminar, transient and turbulent flows are present in a turbine flow field. Due to the variety of flows and its constant interaction with each other, the flow field in turbine blade is one of the most complex flow fields to analyze. The core of the flow in the blade passage, away from end-walls and blade surface has limited viscous effects. It is generally approximated to be two
dimensional inviscid flow and hence it is the simplest flow to analyze in the turbine flow field. The fluid near the blade surface is highly viscous and develops a three dimensional boundary layer. The boundary layer is dependent on the blade profile and develops into the blade wake on exiting the stage. The flow near the end-walls are the most complex flows in the turbine flow field. There are two end-walls in the turbine stage one is the inner end-wall due to the hub of the rotor and the other is the outer end-wall due to the casing surface. Because of the turning of this end-wall boundary layer, secondary flows arise in the stage section. It can analytically be shown that the turning of a momentum deficit carrying end-wall boundary layer in the turbine passages results in significant stream wise vorticity defining the secondary flows. Another flow which plays a significant role in the turbine flow field analysis, and the focal point for this study, is the tip leakage flows. The tip leakage flows arises due to the pressure driven viscous flows in the gap between the rotor blade tip and the turbine casing. Both the secondary flows and the tip leakage flows are highly complex flows with several vortical structures in them. These two flows combined account for more than 60% of overall losses in the turbine stage. As they play significant role in loss generation in the turbine flow field, numerous experimental and numerical studies were performed on these fields to understand their physics. The current knowledge of these flows is given below.

### 2.2.1 Secondary flow

The inlet flow to the turbine stage is a non-uniform flow. The non-uniformity arises due to viscous effects related to hub and end-wall boundary layers, temperature gradient
from the combustion chamber, and rotational effects. Whenever this non-uniform flow with normal vorticity component is turned in a duct, a secondary flow is generated. The secondary flow developed by this mechanism gives rise to a secondary vortex called passage vortex. Another mechanism in which the secondary flows are produced is by the viscous effect of the end wall. This viscous effect creates a three-dimensional boundary layer which rolls up into a horse shoe vortex. A detailed review of secondary flow is given in references [19] and [4]. The secondary flow is a complex system with many vortices of different strength. Of those two important ones are the passage vortex and the horse shoe vortex. The formation of passage vortex taken from reference [20] is shown in the Figure 2.2. The horseshoe vortex and the passage vortex usually rolls up into one vortical structure as shown in Figure 2.3 [21].

**Passage Vortex**

Formation of passage vortex in a turbine rotor blade passage is taken from reference [3] and shown in Figure 2.4. The inlet velocity has gradient in the \( n \) direction. Point A is in the uniform flow region and AAA is its corresponding streamline. Point B is in shear region and BBB is its corresponding streamline. If the flow is assumed to be incompressible and steady then at point A the pressure gradient in \( n \) direction is balanced by the centripetal acceleration along the streamline AAA as shown by the equation 2.2.
Figure 2.1. Fluid working process in turbine stage, [3].

Figure 2.2. Passage vortex in the secondary flow, [20].
Figure 2.3. Illustration of secondary vortical structures, [21].

Figure 2.4. Formation of Passage vortex in the turbine blade passage, [3].
\[
\frac{\partial p}{\partial n}_A = \frac{\rho u_A^2}{R_A} = \frac{\rho u_B^2}{R_B}
\]

--- Equation 2.2

Both the streamlines AAA and BBB are subjected to the same normal pressure gradient and get equation 2.3

\[
\frac{\partial p}{\partial n}_A = \frac{\partial p}{\partial n}_B = \frac{\rho u_A^2}{R_A} = \frac{\rho u_B^2}{R_B}
\]

--- Equation 2.3

In this assumption \( R_A = R_B \) is invalid, as \( u_A > u_B \) there is an imbalance in the equation. The only way to balance this equation (ie. Balance the normal pressure gradient and the centripetal acceleration) is to have \( R_A > R_B \), which implies that the slower moving fluid follows tighter radius of curvature. Hence that the flow is deflected inward towards the suction side of the nearby blade. This means that there exists a cross-flow in the flow field, which is called the secondary flow. As the flow near the blade surface move from the pressure side to suction side, to preserve the mass some of the flow moves from suction side to pressure side creating a vortical structure at blade exist. This vortical structure is called the passage vortex. The strength of the passage vortex is determined by the velocity gradient near the blade surface and the magnitude of flow deflection in the field.

**Horse Shoe Vortex**

The horse shoe vortex in the blade passage is developed from the hub end-wall boundary layer as shown in Figure 2.3. The inlet flow of the blade experiences a deceleration due to the physical presence of the blade. The end-wall boundary of the inlet
flow experiences a strong adverse pressure gradient as it nears the corner of the blade leading edge and the end-wall. The inlet boundary layer grows rapidly due to the presence of an adverse pressure gradient and separates near the blade leading edge to form a horseshoe vortex. The horseshoe vortex wraps itself around the blade leading edge with its two legs surrounding the pressure side and suction side of the blade. The suction side leg of the vortex travels with a radially outward component along the suction side of the blade surface, while the pressure side leg of the vortex crosses the passage and moves towards the suction side of the adjacent blade. At this location, the vortex merges with the passage vortex and forms a single passage/pressure leg vortex system.

**Other Vortex**

Similar to the hub end-wall, the casing end-wall also creates a vortex system. Due to presence of a gap near the casing, this vortex is generally absorbed into the high strength leakage vortex and will be discussed in the next section. Apart from the discussed vortical structures, there are several other vortices developed in the secondary flow which are usually over shadowed or absorbed by the horse shoe vortex and passage vortex.

### 2.2.2 Tip Leakage flow

For mechanical reasons, there is always a gap between the rotor blade tip and the casing surface. The pressure gradient at the blade tip forces the flow to leak from pressure side to suction side. These leakage flows have not been involved in the energy conversion
process. The main flow, on the other hand, have lost some of its momentum during its interaction with the rotor. The leakage flows have different whirl velocity than the core flow, resulting in mixing of the two flows. This intense mixing results in a vortex called leakage vortex.

When the flow enters from the pressure side to the tip gap, it sees a sharp corner and separates. The behavior of the flow inside the tip gap depends on the local thickness of the blade. If the blade is thick enough locally, then the separated flow is reattached inside the gap forming a bubble. The minimum ratio of gap height to the local blade thickness required for the bubble to form was found to be 6 as per reference [22], while reference [23] predicted it to be 4. In case of locally thin blade tip, the flow does not reattach inside the tip gap. As there is no bubble formed in this region, there is also no pressure recovery. The discharge coefficient in the gap is also reduced. The schematic of the flow through the tip gap region for both locally thick and thin blades is taken from reference [23] and shown in figure 2.5.

A detailed study on the formation of a bubble in a locally thick blade and their effects is given by references [24, 25]. The thickness of the blade tip surface continuously varies from leading edge to trailing edge. The pressure distribution on the leakage surface at three locations along the chord is given in Figure 2.6. The initial peak at 4mm from the pressure side in Figure 2.6 signifies the reattachment of the flow to form a bubble in the tip gap. From figure 2.7 it can be seen that the location of reattachment (size of the bubble) depends on the tip gap height.

The presence of a bubble in the tip gap near the pressure side facilitates the leakage flow with rounded inlet geometry. As shown in Figure 2.8 taken from reference [26], the
discharge coefficient \((C_d)\) of such flow increases with increase in inlet corner radius \((r)\). This elevated discharge coefficient results in increased aerodynamic loss. The bubble formed in the tip gap region can be eliminated by having a blade design with corner radius in the pressure side.

A study performed on a low speed research turbine rig [24] showed that the bubble in the tip gap can be eliminated by having a corner radius of 2.5 times the tip gap height on the pressure side. Even though the bubble is eliminated, the stage loss in the turbine is increased due to presence of rounded corner. The mathematical model developed in reference [27] predicted that there is a 12% increase in the discharge coefficient for a blade with a corner radius of 0.3% of the blade chord when compared to sharp corner blade. A schematic representation of the bubble formation on the pressure side corner is taken from reference [23] and shown in Figure 2.9.

The casing has a significant influence on the leakage flow. Figure 2.10 taken from reference [28] shows a possible casing effect on the tip leakage process. The leakage jet in the tip gap is sheared by the casing with velocity \(U_{casing}\). As the flow very near the casing is in the opposite direction of the leakage flow there exists a stagnation line \(A_1 - A_2\) in the tip gap region. There is a high production of turbulent kinetic energy in the region between the stagnation line and the casing surface. Because of this shearing effect the mass flow and strength of the leakage vortex is reduced.
Figure 2.5. Flow in the tip gap region for locally thick and thin blades, [23].

Figure 2.6. The pressure distribution on the leakage surface at three location along the chord, [25].
Figure 2.7. The pressure distribution for three tip clearance, [25].

Figure 2.8. The variation of discharge coefficient with inlet corner radius, [28].
Figure 2.9. Several effects of the separation bubble in the tip clearance region, [23].

Figure 2.10. Casing effect on tip leakage flows, [28].
An interesting numerical study on the effect of casing motion on the tip leakage flows was done in reference [28]. The results showed that due to the casing motion, the strength of the leakage vortex was reduced by a third and a reduction in the tip leakage mass flow rate was observed. Similar effects have been observed in reference [29] in a water analog rig, where the rotational effect of the casing reduces the tip leakage flow. In addition to casing motion, blade loading at the blade tip also plays a critical role in the mass flow of the tip leakage flows. As the blade loading increases, the mass flow of the leakage flow increases, thus increasing the aerodynamic loss in the system.

2.2.3 Radial effects in the Turbine flows

In turbomachinery, most of the time the fluid flow follows an annular path. Because of this annular path, the passage flow experiences two main radial effects. One is the radial pressure gradient and the other is related to the centrifugal force generated by the blade. The radial pressure gradient exists because of the higher blade velocity at the tip than at the hub. Because of this gradient, the low energy fluid in the blade boundary surface tends to move towards the hub. This effect is predominant near the suction side trailing edge as it has thick boundary layer. Because of this effect the loss distribution is altered with greater loss near the hub and lesser loss near the casing. The centrifugal force on the other hand pushes the flow radially outward and is greater than the radially inward pressure gradient. Hence the net result of these two radial effect is that the low energy fluid in the blade boundary layer flows from hub to casing aiding the tip leakage flows.
2.2.4 Interaction between the secondary flow and tip leakage flow

The interaction between the passage vortex and tip leakage vortex is an interesting phenomenon which is yet to be fully answered. Several studies were completed done on this field with few having conflicting observations. The most popular observation are that the passage vortex and leakage vortex counter rotate inside the passage flow field. The interaction of these vortex and effects on streak lines was studied in reference [30]. The streak lines in the turbine blade passage is shown in Figure 2.11. It was found that the passage vortex moves towards the suction surface and below the leakage vortex where it intensely interacts with the leakage vortex. As the leakage vortex and secondary vortex are counter rotating, the turbine flow field with smaller tip vortex will have larger secondary vortex and vice versa. The opposing nature of the leakage flow and the secondary flow is clearly demonstrated in reference [31]. An experimental study on a turbine blade with varying tip clearance in a linear cascade test section showed that as the tip clearance increases the tip vortex also increases. And the tip leakage vortex with increased strength reduces the passage vortex to a small portion and occupies the whole passage width.

2.2.5 Loss Generation

As discussed before the end-wall and the blade tip gap creates vortical structures in the turbomachinery flow field which are the primary source of aerodynamic loss. The three main vortices are the passage vortex, horse shoe vortex and the leakage vortex. The passage vortex and horse shoe vortex usually merge together into a single vortex by the end of blade passage and the aerodynamic loss they create is called secondary loss.
The leakage vortex results in leakage loss. In addition to the end-wall and blade tip gap, the blade profile and the boundary layer on its surface are also a source of aerodynamic loss called profile loss. The loss generation mechanism of the secondary loss, leakage loss and profile loss are dependent on each other and sometimes it is difficult to differentiate. Typically, each loss accounts for one third of the overall aerodynamic loss in a turbine stage, but their relative sizes can be varied by the design of the turbine blade.

The loss which is studied in this research is the leakage loss. It can be split into three parts. First is the loss generated due to secondary losses and end wall effects near the blade tip, second is the loss generated in the tip gap region due to the formation of a shear layer in the gap, and the third is the loss generated due to mixing of the leakage flow and the mainstream flow. The losses due to secondary and end-wall near the blade tip is very small compared to other losses and is usually studied along the tip gap losses. In the tip gap region, the flow separates on the pressure side corner and reattaches back in the tip gap forming a bubble as shown in Figure 2.5. The fluid between the shear layers behind the bubble and the rest of leakage flow mixes in the boundary layer of the solid surface. Extend of fluid mixing depends on the ratio of the gap height to the blade height. Hence the loss generated in this region strongly depends on the tip gap height. The significant portion of the losses in the leakage flow is due to the turbulent mixing of the leakage flow and the mainstream flow. This mixing extracts energy from the mainstream as the energy is transferred from the mainstream flow field to the turbulent flow field which is then dissipated through viscous effects. An interesting study on the distribution of the origin of these three losses along the blade chord was studied in reference [25]. The distribution of the three tip leakage losses along the blade chord is shown in Figure 2.12. The loss due to
the shear in gap region increases drastically after mid chord while the loss due to mixing increases after 80% of the chord. Most of the tip leakage loss generated in the turbine blade are near the trailing edge.

2.3 Leakage Loss Quantification

Quantification of leakage loss is a hard to define since the tip leakage flow is a complex flow field. One of the simplest models of the leakage loss was proposed in reference [23]. The loss model was developed by treating the mixing of leakage flow and the main flow as a cross flow in jet stream. The cross flow is the leakage flow and the main flow is the jet stream. The local mass flow rate of the leakage flow depends on the pressure difference in the blade tip and the discharge coefficient based on pressure side corner. The leakage flow arriving at the suction side corner of the blade is assumed to have the pressure side velocity and is mixed with the main flow with suction side velocity as shown in Figure 2.13. The loss coefficient derived by Denton in reference [23] is given in equation 2.4. Information about the discharge coefficient, pressure side and suction side velocity are needed for this calculation. Even though this model is basic and cannot be used to find the absolute loss value of tip leakage flow, it is very helpful in determining the trends of the leakage loss.

\[
\zeta = \frac{2C_d}{\cos(\beta_2)} \left( \frac{g}{h} \right) \left( \frac{c}{s} \right) \int_0^1 \left( \frac{V_s}{V_2} \right)^3 \left( 1 - \frac{V_p}{V_s} \right) \sqrt{1 - \left( \frac{V_p}{V_s} \right)^2} \left( \frac{dz}{c} \right)
\]

--- Equation 2.4
Figure 2.11. Streak lines in the turbine blade passage, [30].

Figure 2.12. Distribution of origin of leakage loss along the blade chord, [25].
2.3.1 Leakage flow loss reduction methods

A summary of theoretical ways to reduce the leakage flow loss based on equation 2.4 is given by reference [32]. For a given gap height \((g/h)\) the leakage loss in a turbine stage can be reduced by several methods such as reducing the discharge coefficient \((C_d)\), increasing the pressure side velocity \((V_p)\) decreasing the suction side velocity \((V_s)\), decreasing the \(\beta_2\), increasing the \(V_2\) and modify the pitch to chord ratio \((s/c)\) in the turbine stage.

Reducing the discharge coefficient in the tip gap is one of the best ways of reducing the leakage loss. A detailed investigation on the tip geometries and their effect on the leakage loss was carried out by Booth in reference [33] and it was found that the tip design with lowest \(C_d\) had the lowest aerodynamic loss confirming the above observation. As we have discussed before the counter-productive of having lower \(C_d\) is that the formation of bubble in the tip gap. When a bubble is formed it reduces the area of the tip gap, as the flow passes above the bubble the area of the flow path is reduced similar to convergent duct. This is called vena contraction and it is a source of aerodynamic loss due to mixing of fluid behind it. In addition to aerodynamic loss, the vena contraction also increases local heat transfer rates and results in shock formation if the exit Mach number after the contraction is high enough. Though reducing the \(C_d\) have negative effect on the turbine performance, there have been several different tip geometry designed to reduce \(C_d\) and have been successfully employed in the gas turbine engines.

From equation 2.4, increasing the local velocity at tip \((V_2)\) should result in a reduced loss. This doesn’t make a physical sense as increasing the local inlet velocity
would increase the loading at the blade tip which in turn increases the aerodynamic losses due to leakage flows. On close examination it can be seen that increasing $V_2$ would also increase $V_s$ and an increased $V_s$ results in increased leakage loss. The concept of offloading blade tip to reduce leakage loss was studied in reference [35]. 3-D blade design was used to off load the blade at tip and resulted in reduction of leakage loss.

According to equation 2.4 reducing the pitch to chord ratio ($s/c$) reduces the leakage loss. The effect of changing this ratio on the leakage flow is studied in reference [32]. Another advantage of reducing the $s/c$ ratio is that it reduces the suction side velocity ($V_s$) which also reduces the leakage loss. But this is not a simple solution as chord to pitch ratio is a critical parameter in turbine stage design. Altering it would put turbine blades in an off design condition and requires extensive turbine design modification.

### 2.4 Blade Tip designs

#### 2.4.1 Shroud blade tip

A turbine stage with zero tip clearance is the best aerodynamic turbine blade design, as it completely eliminates the tip leakage flows. To achieve this design, the casing should be attached to the blade tip and rotate along with the blades. This is the basic idea behind the shrouded blades where the shroud is attached at the blade tip and acts as a rotating casing. The tip leakage flows are reduced but not completely eliminated in this design as the pressure drop across the turbine stage drives the leakage flow between the rotating shroud and stationary casing. This leakage flow is further reduced by adding fins (as shown
in Figure 2.14) in the shroud and the leakage mass flow rate depends on the number of fins attached to the shroud. Clear aerodynamic advantage of using a shrouded blade can be seen from Figure 2.15. The flow field in the shrouded turbine blades are very complex and several studies have been done by references [36], [37] and [38] to understand the secondary flow structures in these flow fields. A detailed study on fins and their effect on the discharge coefficient in the leakage flow was done by reference [39].

Even though the shrouded blades have a better aerodynamic performance than the shroud-less blades, the addition of shroud adds to the weight of the turbine stage. As they are attached at the tip of the blade, they increase the mechanical stress on the blade root and the hub disc. The turbine blade are already placed in high thermal stress environment and addition of shroud to the blade tip worsens the overall stress experienced by the blades. The shroud itself also experiences bending stress due to rotation and thermal stress due to high inlet temperature. They also have to be cooled thus increasing the mass flow rate of bleed air stolen from the compressor and reducing the overall efficiency of the system.

2.4.2 Squealer blade tip

As discussed before the vena contraction in the tip gap region is a significant source of leakage loss. The effect of vena contraction can be reduced by having a cavity in the blade tip surface. Squealer blade tips are obtained by extending the pressure and suction surface shell on the blade tip radially outward. Single side squealer blade tips are obtained by extending either pressure or the suction surface shell radially outward. The squealer tips reduces the tip gap flow and also protects the blade from incidental rub against the casing.
Conventional squealer blade tip have a fully surrounded cavity was first patented by R.H Anderson in 1979 [9]. Several modification to the squealer tips have been developed, with the recent designs having a cut in the cavity at the trailing edge allowing the some of the leakage flow to reach the main flow without having to turn around the suction side corner.

The effect of single sided squealer tip was studied and compared to the flat tip blades in reference [27]. The Leakage flow mass was reduced by both the squealer blade tips and the suction side squealer was found to have improved loss characteristics and the results are shown in Figure 2.16. A similar result was found in reference [40] on studying the effect of partial squealer tip, here squealer tips are extended on a specific portion of the blade tip. The suction side squealer tips were more effective in reducing the leakage flow across the blade tip. This is because when the leakage flow meets with the suction side squealer in the tip gap it rolls up into vortex on the central portion of the tip. This vortex hinders the entry of the leakage flow into the tip gap effectively reducing the leakage mass flow rate.

2.4.3 Winglet or Partial shroud blade tip

Similar to airplane wings where the winglets are used to reduce the effect of wing tip vortices, the winglets in turbine blades can be used reduce the effect of tip leakage vortices. The winglet on the pressure side of the blade tip reduces the discharge coefficient of the leakage flow and winglet on the suction side reduced the driving differential pressure in the blade tip both of which helps in reducing the tip leakage flows. The first study on
winglet blade design was done in reference [41]. Using winglets in the blade tip increased the aerodynamic efficiency of the turbine stage was by 1.2%.

Several other designs of winglet was studied in reference [33] and concluded that the good aerodynamic performance was given by double winglet design. The winglet is also called partial shroud as it is similar to shroud but less than half the size of the shroud. It eliminates the drawbacks in the shrouded blade design such as high bending stress and high cooling requirement while improving the aerodynamic efficiency of the system. A computational study done in reference [32] showed that the winglet would result in 31% reduction of tip loss exchange rate and improve the efficiency of the turbine stage by 1.2%-1.8%. An experimental study on the effect of suction side and pressure side winglets on the tip leakage flows was done in reference [42]. An interesting observation of suction side winglet was noted where the suction side winglets are not as effective as the pressure side winglets. The pressure side winglet reduced the mass flow rate of the tip leakage flows while the suction side winglet doesn’t play a significant role in reduction of the leakage flow. This is because the extensions in pressure side acts as a barrier to the flow leaking to the tip gap region effectively reducing the mass flow rate of the leakage flow. The minimum total pressure value as a function of tip gap for flat, pressure side and suction side winglet blade tips are shown in Figure 2.17.
Figure 2.13. Mixing of leakage flow and the main flow, [23].

Figure 2.14. Fences and fins in the shroud of the turbine blade, [1].
Figure 2.15. Turbine Stage efficiency comparison of Shrouded and Shroud less blade, [1].

Figure 2.16. The Leakage loss and discharge coefficient of three different configuration, [27].
Figure 2.17. The Leakage loss and discharge coefficient of three different configuration, [42].
2.4.4 Blade tip fluid injection

The turbine blade tips experience high thermal stress and require effective cooling system design. The coolant in this region is usually injected radially outward from the blade tip and doesn’t involve in any work generation. The coolant air in this system is stolen from the high-pressure compressor stage and results in loss of overall efficiency of the turbomachinery. If the coolant air is strategically injected in the blade tip section it helps in reducing the total pressure loss and improve the efficiency of the system.

The high momentum coolant jet blocks the entry of leakage flow and reduces its mass flow rate. Resulting tip leakage flow is equivalent to the one due to reduced tip clearance. A numerical study on the effect of coolant flow in the tip leakage flow was done in reference [43] and found that the behavior of the coolant jet and its interaction with leakage flow is very sensitive to coolant slot location and width. An experimental study on coolant jet and leakage flow interaction was performed by reference [44] on the low speed rotating rig and found that the coolant flow injected from the blade tip to have beneficial effect on the leakage flow. A similar experiment was conducted on the same facility by references [45] and [46] and studied the effect of coolant mass flow rate and the coolant slot location on the leakage flow. The coolant slot located at 81% of the chord gave a better loss reduction as it the more successful in imparting high momentum to the pressure deficit vortex core. Thus optimizing the coolant jet in the blade tip region have significant benefits in the turbomachinery system.
2.4.5 Casing Treatment

The leakage flow in the blade tip gap can be reduced by modifying both the blade tip and the casing surface near the blade tip. With all the concepts described above (such as shroud, squealer, winglet and blade tip fluid injection) the modifications were made in the blade tip. A different and passive approach to control the tip leakage loss is to make modification in the casing surface near the blade tip. One of the benefits of this approach is that the casing is stationary and doesn’t experience the rotational force as opposed to rotors. Modifying the casing surface to improve performance is already in use in the compressor where grooves in the compressor casing helps to improve its stall margin but not has been extensively studied for casing.

A detailed experimental study on the casing grooves was done in a turbine rig by reference [47]. The results showed that tip leakage loss in the system was better reduced by having curved patterns similar to the airfoil shape on the casing surface. A similar numerical study on the circumferential grooves in the casing was studied by reference [48] and found that the deep circumferential grooves in the casing surface have better aerodynamic performance than the shallow circumferential grooves.

2.5 Tip Leakage Interrupter (TLI)

The blade tip design modification called Tip Leakage Interrupter (TLI) as shown in Figure 2.18 is examined in this study. TLIs are basically distributed small vortex generators. The basic concept behind this design is to generate a vortex or a set of vortices that is counter rotating to tip leakage vortex near the suction side of the blade tip. The
generation of vortical near wall flow structure is studied in reference [49]. Figure 2.19 illustrates a typical vortex generated by a TLI. The flow passing over the inclined edge of the TLI sees a low pressure in the immediate downstream of the TLI. When this flow crosses angle with the clean undisturbed flow above, it forms a clock-wise rotating vortices. The generated vortex as it is counter rotating to leakage vortex is expected to reduce the vortical strength of leakage flow. Any surface modifications on the blade tip especially on the suction surface should result in increased local total pressure loss. But as the turbine blade tip region is already somewhat handicapped in terms of work generation, the proposed TLI designs should not introduce significant penalty in the work output of the turbine. It is likely that the damaging aerodynamic loss generation character of the tip vortex flow system can be positively influenced by carefully sized and distributed vortex generators named TLIs.

If the TLIs are successful in reducing the effect of tip leakage vortex the net effect of the design would be, a reduction in total pressure loss and as a result increase in the efficiency of the turbine system. The concept of generating additional vortices to enhance the flow mixing were used in various other fields. They are used to control the boundary layer separation in the lifting surfaces of highly swept wings. In case of turbine blades they are used to prevent the boundary layer separation on the suction side of the blade as shown by references [50] and [51]. Similar vortex generation technique is also used for effective gas turbine blade cooling. In reference [52] adding vortex generator increases the heat transfer inside the turbine blade by 27%. But this is the first study to investigate the effect of strategically introduced counter-acting vortices on the tip leakage flows.
Figure 2.18. Tip Leakage Interrupter (TLI) blade design.

Figure 2.19. Vortex generated by TLI.
Chapter 3

Experimental methods and facility

The Axial Flow Turbine Research Facility at the Pennsylvania State University was used to study the effect of TLI on the leakage flow. The description of the facility and the experiment setup is provided in this chapter. More information about the design, construction and operation of this high pressure turbine stage can be found in references [47], [53], [54], [55], [56], [57] and [58].

3.1 Axial Flow Turbine Research Facility (AFTRF)

The Axial Flow Turbine Research Facility (AFTRF) at the Pennsylvania State University is a low speed, large scale turbine rig (as shown in Figure 3.1) running at approximately 1330 RPM. A detailed report about this “cold flow” facility is presented in reference [59]. This open circuit facility with single turbine stage is driven by four stage axial flow fan system that is connected in series. The four-stage axial flow fan system works in a suction blower mode to induce an ambient temperature flow from an atmospheric inlet to turbine exit. It is considered as a cold flow aerodynamic research facility with an HP stage including an annular nozzle guide vane NGV assembly and a rotor. There is about 5-7 °C temperature drop across the rotor of the facility due to work extraction from the ambient fluid at the inlet. This large-scale and low-speed research turbine stage was designed by GE aviation, Cincinnati. The loading distributions of the stage simulate the $E^3$ design of NASA (Energy Efficient Engine).
The inlet to this facility is in the shape of a large bell-mouth followed by the turbine test section consisting of a nozzle guide vane row and a rotor row. There is an optional turbulence generating grid section in-between the inlet and the test section. The turbulence grid is not used in the present investigation. The inlet to this facility is in the shape of a large bell-mouth followed by the turbine test section consisting of a nozzle guide vane row and a rotor row. An Eddy Current Brake (ECB) is used to absorb the power generated in the turbine stage, which is also used to control the rotational speed of the rotor. The mechanical power generated by this research turbine could be anywhere from 25 HP to 120 HP depending on the operating conditions of the test system. The mechanical (shaft) power is absorbed in the eddy current brake by keeping the RPM by the brake is transferred to water in order to keep the ECB temperature constant at operable levels. A chilled water-system or tap water can passed through the ECB during the turbine operation. Using ECB the rotor RPM can be kept constant in an RPM range from 175 and 2000 and can be adjusted within ±1 RPM.

The diameter of the facility is 91.4 cm with the hub to tip radius ratio of 0.73. It has E³ advanced axial turbine blading configuration designed by GE Aviation. The E3 blade design is considered to be a NASA product. However GE Aviation implemented general E³ blade design features in the design of this large-scale, low- cold flow turbine for aero-thermal basic design studies. For this experiment there are 23 nozzle guide vanes and 29 rotor blades in this facility with provision to change vane-blade axial spacing from 20% to 50% of the chord. Under normal operating conditions the blower system can generate a mass flow rate of 10.4 cubic meter per second with 40 inches of static pressure drop at the stage exit. The inlet has atmospheric flow conditions. At the downstream end of the facility
there is an aerodynamically designed throttle to control the mass flow rate in the test section.

Both stationary and rotating frame measurements can be taken in AFTRF. In the stationary frame of reference, all of the measurements are taken from the casing window (240mm by 530mm) located by the side of the test section in the rig. Figure 3.2 shows the inner surface of the window which is curved according to the casing curvature and fits in flush. Particle Image Velocimetry, Laser Doppler Velocimetry and other aero-thermal measurements are also taken from the same window. In case of rotational frame measurements, an instrumented drum is present at the center of the rig to facilitate rotor loading and end-wall static pressure measurements. The data collected in the rotating frame using the rotating drum is then transferred to the stationary frame using a 150 channel slip ring type data transfer system. For the experimental effort presented in this thesis, only the stationary frame dynamic total pressure measurements are taken using a phase-locked and time accurate manner at rotor exit.

3.2 Turbine Stage

The nominal operating speed of the rotor for this experiment is 1330 RPM. Total-total isentropic efficiency of the turbine stage was calculated to be 0.893. The turbine stage design characteristics are given in table 3.1. Depending on the present day weather condition the inlet temperature and pressure typically varies from 297 K and 961 mbar to 300 K and 967 mbar. The turbine stage has nozzle guide vanes (stator) and rotor row as shown in Figure 3.3. The design parameters for the nozzle guide vanes are provided in
table 3.2. The flow Reynolds number in the turbine stage is representative of a modern high pressure stage. The stator blades deflects the flow by 70° and guides it to the rotor blades. The rotor blades have a hub-tip ratio of 0.7317 which is similar for the high-pressure turbine stage. The absolute flow at the exit of the rotor is aligned with respect to the axial direction at an angle of 25.16° at the tip and 35.13° at the hub. The rotor blade velocity at hub, mid span and tip are given in Figure 3.4. The rotor design parameters are summarized in table 3.2. The airfoil shape of the rotor blade tip is given in Figure 3.5 and the details of the airfoil coordinates are presented in Appendix A. layers were to reduce the tip clearance of some of the blades with large clearance. The minimum shim thickness of 0.001 inch corresponds to a tip clearance increment of % 0.02 of blade height. It should be noted that the two blades having a tip clearance specification within +/- % 0.05 are considered to have the same tip clearance. For example the four individual blades having the tip clearances of 0.68%, 0.70%, 0.70% and 0.74% are considered to have the same overall tip clearance, as shown in Figure 4.12. This approach using additional thin shim layers near the tip platform is effective in assigning and controlling the specific tip clearances of selected blades in the rotor assembly.
Figure 3.1. Cross sectional drawing of AFTRF, [53].

Figure 3.2. AFTRF Measurement Casing window.
3.3 Instrumentation

A pair of thermocouples are used to measure the stage inlet and outlet total temperatures. The pitot static probes are used to measure the inlet and exit velocity at the blade mid-span. The static probes attached along the casing surface provides the pressure drop across the turbine stage. The optical encoder located on the main shaft reads the rotational speed of the rig while a linear accelerometer magnetically attached on the exterior of the rig provides the vibrational level of the facility. The encoder also provides information about the rotational position of the rotor within \( \pm 360/6000 = \pm 0.06 \) degrees. Other than the basic instrumentation provided above there are provisions for detailed instrumentation on the nozzle vane passages, rotor blades, nozzle casing, rotor hub and nozzle hub.

For this experiment the aerodynamic data is collected only in the stationary frame of reference. The probe traverser moves radially at 30% axial chord downstream of the rotor blades using a high resolution (Vexta) stepper motor attached to linear traversers (Velmex Unislide). Stepper motor controller (Velmex VXM) controlled by computer is used to drive the motors. The motors with their micro-stepping ability has better probe positional accuracy. Its quick response and fast movement also reduces the time required to move to specific measurement location. The radial traverser used in this experiment is mounted on the precision built non-intrusive window located on the side of the test section.
Figure 3.3. Turbine stage with outer casing removed.

Figure 3.4. The rotor blade velocity triangles at hub, mid span and tip section.
Figure 3.5. The airfoil shape of the rotor blade tip.

Figure 3.6. Fast Response Aerodynamic Probe (FRAP).
Table 3.1. AFTRF Turbine stage design characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor hub-tip ratio</td>
<td>0.7317</td>
</tr>
<tr>
<td>Mass flow rate, $\dot{m}$ (kg/sec)</td>
<td>10.12</td>
</tr>
<tr>
<td>Rotational speed, $N$ (rpm)</td>
<td>1330</td>
</tr>
<tr>
<td>Total-Total isentropic efficiency, $\eta_{tt}$</td>
<td>0.893</td>
</tr>
</tbody>
</table>

Table 3.2. AFTRF Stator and rotor design specification

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Stator</th>
<th>Rotor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of blades</td>
<td>23</td>
<td>29</td>
</tr>
<tr>
<td>Zweifel Coefficient</td>
<td>0.7247</td>
<td>0.9759</td>
</tr>
<tr>
<td>Relative Mach number</td>
<td>---</td>
<td>0.24</td>
</tr>
<tr>
<td>Blade Height, $h_b$ (m)</td>
<td>---</td>
<td>0.1229</td>
</tr>
<tr>
<td>Tip clearance (mm)</td>
<td>---</td>
<td>0.9</td>
</tr>
<tr>
<td>Turning Angle, Tip/Hub</td>
<td>$70^\circ$</td>
<td>$95.42^\circ$ / $125.69^\circ$</td>
</tr>
<tr>
<td>Chord (m)</td>
<td>0.1768</td>
<td>0.1287</td>
</tr>
<tr>
<td>Axial Tip chord (m)</td>
<td>---</td>
<td>0.084</td>
</tr>
<tr>
<td>Spacing (m)</td>
<td>0.1308</td>
<td>0.1028</td>
</tr>
<tr>
<td>Maximum Thickness (mm)</td>
<td>38.81</td>
<td>22</td>
</tr>
<tr>
<td>Reynolds Number ($\times 10^5$) inlet/exit</td>
<td>3<del>4 / 9</del>10</td>
<td>2.5<del>4.5 / 5</del>7</td>
</tr>
<tr>
<td>Rotor efficiency, $\eta$</td>
<td>0.994</td>
<td>0.8815</td>
</tr>
</tbody>
</table>
Table 3.3. Operational Characteristics of Endevco Transducer 8507C-1, [54]

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Range</td>
<td>±1 psig</td>
</tr>
<tr>
<td>Positive Sensitive</td>
<td>200 ± 50 mV/psi</td>
</tr>
<tr>
<td>Combined: Non-Linearity, Non-Repeatability, Pressure Hysteresis</td>
<td>1.5 % FSO RSS Max</td>
</tr>
<tr>
<td>Non-linearity, Independent</td>
<td>1.5 % FSO Typical</td>
</tr>
<tr>
<td>Non-Repeatability</td>
<td>0.2 % FSO Typical</td>
</tr>
<tr>
<td>Pressure Hysteresis</td>
<td>0.2 % FSO Typical</td>
</tr>
<tr>
<td>Zero Measured Output</td>
<td>±10 mV Max</td>
</tr>
<tr>
<td>Zero Shift After 3X Range</td>
<td>0.2 (0.02) ±% 3X FSO Max</td>
</tr>
<tr>
<td>Thermal Zero Shift from 0°F to 200°F</td>
<td>3 ±% FSO Max</td>
</tr>
<tr>
<td>Thermal Zero Shift from 40°F to 140°F</td>
<td>3 ±% FSO Max</td>
</tr>
<tr>
<td>Thermal Sensitive Shift from 0°F to 200°F</td>
<td>4 ±% Max</td>
</tr>
<tr>
<td>Sensitivity Shift from 40°F to 140°F</td>
<td>4 ±% Max</td>
</tr>
<tr>
<td>Resonance Frequency</td>
<td>55000 Hz</td>
</tr>
<tr>
<td>Non-Linearity at 3X Range</td>
<td>2.5 % 3X FSO</td>
</tr>
<tr>
<td>Thermal Transient Response</td>
<td>0.003 psi/°F</td>
</tr>
<tr>
<td>Photoflash Response</td>
<td>0.01 Equiv. psi</td>
</tr>
<tr>
<td>Warm-Up Time</td>
<td>1 ms</td>
</tr>
<tr>
<td>Acceleration Sensitivity</td>
<td>0.0002 Equiv. psi/g</td>
</tr>
<tr>
<td>Burst Pressure</td>
<td>20/20 psi Min</td>
</tr>
</tbody>
</table>
Table 3.4. Operational Characteristics of Endevco Amplifier Model 136, [54]

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Impedance</td>
<td>±1 MOhm min</td>
</tr>
<tr>
<td>Input Range: Differential</td>
<td>0 to ±10 VDC</td>
</tr>
<tr>
<td>Input Common Mode</td>
<td>±10 VDC</td>
</tr>
<tr>
<td>Input Common Mode Rejection - Noise</td>
<td>70 dB</td>
</tr>
<tr>
<td>Input Common Mode Rejection – Impedance</td>
<td>200 Ω</td>
</tr>
<tr>
<td>Output Impedance</td>
<td>0.2 Ω Max</td>
</tr>
<tr>
<td>Linear Output</td>
<td>10 V peak</td>
</tr>
<tr>
<td>Current Output</td>
<td>10 mA, min</td>
</tr>
<tr>
<td>Excitation Voltage Accuracy</td>
<td>±1 %</td>
</tr>
<tr>
<td>Excitation Current, Short Circuit Protected</td>
<td>30 mA, Max</td>
</tr>
<tr>
<td>Noise and Ripple, 10 Hz to 50 kHz, 1 kOhm Load</td>
<td>1 mV rms Max</td>
</tr>
<tr>
<td>Transfer Characteristics - Gain Accuracy of Full Scale Max, DC to 1kHz, Filters Disabled</td>
<td>±0.5 %</td>
</tr>
<tr>
<td>Transfer Characteristics - Gain Linearity of Full Scale, Best Fit Straight Line at 1kHz Reference</td>
<td>0.1 %</td>
</tr>
<tr>
<td>Transfer Characteristics - Gang Stability of Full Scale, 0°C to +50°C</td>
<td>±0.2 %</td>
</tr>
<tr>
<td>Broadband Frequency Response, DC to 200 kHz, Referenced to 1 kHz</td>
<td>−3 dB</td>
</tr>
<tr>
<td>Filter Characteristic/Type</td>
<td>4-pole Buttersworth</td>
</tr>
<tr>
<td>Crosstalk Between Channels</td>
<td>80 dB RTI</td>
</tr>
<tr>
<td>Input Imbalance Adjustment</td>
<td>±100 mVDC</td>
</tr>
<tr>
<td></td>
<td>±1 VDC</td>
</tr>
<tr>
<td></td>
<td>±10 VDC</td>
</tr>
<tr>
<td>Output DC Bias Stability Temp</td>
<td>±5 μV/°C</td>
</tr>
<tr>
<td></td>
<td>±0.1 mV/°C</td>
</tr>
<tr>
<td>Output DC Bias Stability Time After 1 Hour Warmup</td>
<td>±20 μV/°C</td>
</tr>
<tr>
<td></td>
<td>±5 mV/°C</td>
</tr>
</tbody>
</table>
Figure 3.6 shows the Fast Response Aerodynamic Probe (FRAP) which is developed in-house at Penn State. The FRAP probe has a differential dynamic pressure transducer which has been modified to suit our experimental needs. The response time of this transducer ranges from 50 kHz to 150 kHz. A protective mesh is used to cover the piezo-resistive transducer and it is attached to the stem using glue and epoxy. The signal from this probe is amplified, low pass filtered and then collected by National Instruments PCI 6110-E multipurpose I/O board.

3.4 Data Acquisition

During the AFTRF operation a pair of thermocouples, three pitot probes, a fast response aerodynamic probe (FRAP) and several static ports are used to find the characteristics of the flow in the facility. The data measured from FRAP probe is phase-locked using a MIL-SPEC grade precision encoder. The encoder has a rotating disk with 6000 equidistant precision holes etched around its circumference and is attached to the rotating shaft. These holes are then used to create 6000 TTL/CMOS pulses per revolution. These pulses are then used as a trigger to collect the data. As they are directly attached to the rotating shaft any changes in the RPM is taken care by the pulses of this 6000 pulse encoder. The data are always collected in the bins that are always $360^\circ/6000 = 0.06$ degrees in circumferential length.

The raw data from FRAP is amplified and conditioned using an Endevco model 136 amplifier. Validyne variable reluctance transducers are used to measure the steady-state inlet static pressure, inlet total pressure, outlet static pressure, outlet total pressure and
the static wall pressure drop across the turbine stage. The data from these transducers are also conditioned by Validyne Demodulator units. External conditioners are not required for the data from the thermocouples measuring inlet total temperature and outlet total temperature. The data collected from these sensors are then processed via LABVIEW to calculate the flow properties and rotor performance. The final results from the LABVIEW is stored and also displayed on the GUI screen. The data processing system is completely automated with an option of all the data being processed online. A detailed description on the data acquisition system used here is given by Jason in reference [54]. Table 3.3 and 3.4 obtained from reference [54] gives the operational characteristics of the Endevco transducer and amplifier. The technical illustration of the Endevco Amplifier model 136 obtained from reference [54] is shown in figure 3.7

The radial probe traversing resolution is programmable and can be adaptive and can provide the required resolution in the tip leakage region. The grid with the current system can be refined much faster for better radial resolution for any interesting flow structures.
Figure 3.7. Technical Illustrations of Endevco Amplifier Model 136, [54].

Figure 3.8. Convergence analysis for Total pressure at different span location.

Indicates the converged results
3.5 Experimental Procedure

The blade tips in the turbine facility were retrofitted with different TLI arrangements on various blade tip locations as shown in figure 3.9. Once the modified blade tips are installed on the selected blades of the rotor, the access window is fastened. The facility is checked for any loose items that may be left in the main flow passage and interferences before running AFTRF. During the running process the power generated from turbine is absorbed by a water-cooled eddy current brake. The initial inlet temperature and atmospheric pressure is recorded. The turbine stage is adjusted to a rotational speed of 1330 RPM for this research and made to run initially for 40 mins to get a stable temperature reading in the inlet and exit of the turbine stage. After stabilizing the temperature, the VELMEX stepper motor is used to precisely move the FRAP radially inward positioned at the 30% axial chord after the rotor blades. The FRAP is moved from 97% blade span to 25% blade span with 1% span increment. The FRAP collects 6000 phase-locked data points per revolution for 400 revolutions per span using the encoder system.

An initial investigation on the statistical stability of FRAP probe results (As Shown in figure 3.9) indicate that a total of 400 ensembles is more than enough for obtaining stable results of the measured rotor exit total pressure measurements. During total pressure measurements, data at 73 individual span wise locations are obtained with 2.4 million readings per span at 6000 measurements per revolution along 400 consecutive rotations. The collected data is amplified, filtered and send to the LABVIEW system, where it is stored for further post processing.
Figure 3.9. Measurement plane location of FRAP probe at the blade tip.

Figure 3.10. Ten equal segments of axial chord.
Figure 3.11 (a). Schematic drawing of Tip leakage Interrupter (TLI).

Figure 3.11 (b). Solid model of Tip Leakage Interrupter (TLI).
Figure 3.12 (a). Tip Leakage Interrupter (TLI) blade design.
Figure 3.12 (b). Reverse Tip Leakage Interrupter (reverse TLI) blade design.
3.6 Experimental Arrangement

The schematic of TLI and reverse TLIs are shown in Figure 3.11 and 3.12. The TLIs are mounted at 45 deg to the normal line of the blade tip drawn at the mounting location. The dashed line which is almost normal to the tip platform surface is the reference line for the placement of the TLI as shown in Figure 3.12. Possible mounting locations of TLIs on the suction side of blade tips are shown in Figure 3.10. Three different sets of experiments called A, B and C were performed with the TLIs. Experiment A was designed to analyze the effect of TLI mounting position on the leakage flows. All of the data analyzed in this experiment were collected from single experimental run of the turbine rig. In this experiment blade a1, a2 and a3 (blade 6, 8 and 14 respectively) with similar tip clearances were chosen to attach the TLIs on the suction side of the blade tips at three different locations along the blade axial chord. The fourth blade (20) was also selected to act as a reference blade to compare the modified blades. The locations of the TLIs on blade tips for the three blades are given in table 3.5 and also shown in Figure 3.13, 3.14 and 3.15.

Experiment B was performed in the turbine facility to analyze the effect of number and mounting location of TLIs on the leakage flows. Three different experimental runs were performed to collect the required aerodynamic data. In the first run, only one TLI was attached on blade b1 tip (blade 6). In second experimental run, three TLIs were attached on the Suction side of blade b2 (blade 6). Five TLIs were attached on blade b3 tip (blade 6). The location of the TLIs on the blade tip for three experimental runs are given in table 3.6 and also shown in Figure 3.16, 3.17 and 3.18. For all the three runs blade 20 was selected as a reference blade.
The influence of TLI orientation on the leakage flow was studied in Experiment C. Reverse TLIs are used only in experiment C. All the data was collected from single experimental turbine run. Two blades c1 and c2 (blade 6 &12) were modified for this experiment. Five TLIs and five reverse TLIs as shown in Figure 3.11 were attached at five locations on blade c1 and c2 respectively. Table 3.7 and Figures 3.19 and 3.20 gives the mounting location of TLIs and reverse TLIs on blade tips.
**Table 3.5: The TLI mounting locations on the three chosen blades for experiment A**

<table>
<thead>
<tr>
<th>Experiment A</th>
<th>Blade a1 (Blade 6)</th>
<th>Blade a2 (Blade 10)</th>
<th>Blade a3 (Blade 14)</th>
<th>Reference (Blade 20)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Position dependent</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tip Clearance (in % of blade span)</td>
<td>0.7</td>
<td>0.76</td>
<td>0.79</td>
<td>0.74</td>
</tr>
<tr>
<td>Location of TLI on Suction side of the blade tip (in % of axial chord)</td>
<td>40</td>
<td>65</td>
<td>75</td>
<td>---</td>
</tr>
</tbody>
</table>

**Table 3.6: The TLI mounting locations on the three chosen blades for experiment B**

<table>
<thead>
<tr>
<th>Experiment B</th>
<th>Blade b1 (Blade 6)</th>
<th>Blade b2 (Blade 6)</th>
<th>Blade b3 (Blade 6)</th>
<th>Reference (Blade 20)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Number dependent</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of TLIs</td>
<td>1</td>
<td>3</td>
<td>5</td>
<td>0</td>
</tr>
<tr>
<td>Location of TLI on Suction side of the blade tip (in % of axial chord)</td>
<td>40</td>
<td>40, 65, 75</td>
<td>40, 55, 65</td>
<td>70, 75</td>
</tr>
</tbody>
</table>

**Table 3.7: The TLI mounting locations on the three chosen blades for experiment C**

<table>
<thead>
<tr>
<th>Experiment C</th>
<th>Blade c1 (Blade 6)</th>
<th>Blade c2 (Blade 12)</th>
<th>Reference (Blade 20)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Orientation dependent</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tip Clearance (in % of blade span)</td>
<td>0.7</td>
<td>0.76</td>
<td>0.74</td>
</tr>
<tr>
<td>Location of TLI on Suction side of the blade tip (in % of axial chord)</td>
<td>40, 55, 65</td>
<td>40, 55, 65</td>
<td>70, 75</td>
</tr>
</tbody>
</table>
Figure 3.13. Blade a1 with TLI at 40% of axial chord.

Figure 3.14. Blade a2 with TLI at 65% of axial chord.
Figure 3.15. Blade a3 with TLI at 75% of axial chord.

Figure 3.16. Blade b1 with TLI at 40% of axial chord.
Figure 3.17. Blade b2 with TLIs at 40%, 65% and 75% of axial chord.

Figure 3.18. Blade b3 with TLIs at 40%, 55%, 65%, 70% and 75% of axial chord.
Figure 3.19. Blade c1 with TLIs at 40%, 55%, 65%, 70% and 75% of axial chord.

Figure 3.20. Blade c2 with reverse TLIs at 40%, 55%, 65%, 70% and 75% of axial chord.
Chapter 4

Facility Adjustment and Baseline Flow Field at Turbine Stage Exit

4.1 Flow Coefficient

The first step in preparing the facility was to adjust and quantify the flow coefficient for the experiments. The flow coefficient is originally a non-dimensional number given by equation 4.1. Dimensional analysis results in a non-dimensional definition of the flow coefficient. However, most turbine designers also employ dimensional forms of it as it is defined in equation 4.1. It depends on the mass flow rate in the system, inlet total temperature and inlet total pressure. The mass flow rate in the system can be modified by varying the back pressure of the turbine rig. The multi-stage axial flow fan performance working in the suction blower mode is not altered during the current research effort. It should be noted that the suction-blower fan performance can be altered via fan blade pitch and rpm adjustments.

The back pressure of the four-stage axial flow fan system in the turbine rig was varied by axially adjusting the throttle valve at the end of the facility. Varying the effective flow area of the throttle valve has an effect on the mass flow rate passed in AFTRF. The pressure drop across the stage is unique for a given mass flow rate. The mass flow rate in the turbine stage was calculated by integrating the inlet velocity profile using the trapezoidal rule. The exact inlet velocity profile in turn was obtained by multiplying the “high-resolution non-dimensional” inlet velocity profile by the measured mid span inlet velocity magnitude. The “high-resolution non-dimensional” profile was borrowed from
[51] and shown in figure 4.1. The past investigations in AFTRF proved that this “high-resolution non-dimensional” inlet velocity profile is invariant in the mass flow range of interest for this facility.

\[
\Phi = \frac{\dot{m}\sqrt{T_{in}}}{P_{o,in}} \left( m \cdot s \cdot \sqrt{K} \right) \text{ or } \left( \frac{lb \cdot \sqrt{\sigma R}}{psi} \right)
\]

---Equation 4.1

The inlet velocity profile is experimentally obtained using a probe traverser system that has a high resolution in span-wise direction. No-slip condition is used for the wall point velocity determination. The span-wise inlet velocity profile is obtained using a conventional Pitot velocity probe at one NGV chord upstream of the NGV leading edge. The pressure drop across the stage, mass flow rate and flow coefficient were calculated for eight different throttle settings and shown in Table 4.1.

From Figure 4.2, it can be seen that the flow coefficient is around 0.0013 for 24 mm and 33 mm of the throttle valve setting. There is a jump in flow coefficient from 0.001375 to 0.001663 when throttle valve setting is increased from 33mm to 36.5mm. This jump in flow coefficient is possibly due to easing of choking of the flow in this range of throttle setting. In Figure 4.2 after 36.5mm of throttle setting the flow coefficient of the system doesn’t change much. The throttle setting represents the axial gap existing at the throttle valve exit of the AFTRF. The throttle setting with maximum pressure drop across the turbine stage which gives the maximum stage pressure drop is ideal for the experiment. The variation of pressure drop across the turbine stage with respect to the mass flow rate in the turbine stage is shown in Figure 4.3.
Figure 4.1: Non dimensional inlet velocity profile measured in AFTRF facility, [54].

Figure 4.2. The flow coefficient in the turbine stage for various throttle gap setting.
Figure 4.3. The pressure drop across the turbine stage for different mass flow rate in the system.

Table 4.1: Flow coefficient of the flow at various throttle valve setting

<table>
<thead>
<tr>
<th>Throttle (mm)</th>
<th>Mass flow rate (kg/s)</th>
<th>$\Delta P = (P_{inlet} - P_{exit})$ kPa</th>
<th>Flow coefficient (SI units)</th>
</tr>
</thead>
<tbody>
<tr>
<td>24</td>
<td>7.32</td>
<td>3.94</td>
<td>0.001308</td>
</tr>
<tr>
<td>33</td>
<td>7.78</td>
<td>4.6</td>
<td>0.001375</td>
</tr>
<tr>
<td>36.5</td>
<td>9.30</td>
<td>6.32</td>
<td>0.001663</td>
</tr>
<tr>
<td>44</td>
<td>9.50</td>
<td>6.68</td>
<td>0.001682</td>
</tr>
<tr>
<td>52</td>
<td>9.63</td>
<td>6.75</td>
<td>0.001720</td>
</tr>
<tr>
<td>62</td>
<td>9.97</td>
<td>7.18</td>
<td>0.001767</td>
</tr>
<tr>
<td>76</td>
<td>10.10</td>
<td>7.42</td>
<td>0.001773</td>
</tr>
<tr>
<td>88</td>
<td>10.12</td>
<td>7.47</td>
<td>0.001779</td>
</tr>
</tbody>
</table>
Table 4.2: The tip clearance values of all the blades in the baseline experiment

Blade span, h = 123mm

<table>
<thead>
<tr>
<th>Blade No</th>
<th>Tip clearance (%) of blade-span (t/h)</th>
<th>Blade No</th>
<th>Tip clearance (%) of blade-span (t/h)</th>
<th>Blade No</th>
<th>Tip clearance (%) of blade-span (t/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.97</td>
<td>12</td>
<td>0.76</td>
<td>23</td>
<td>0.58</td>
</tr>
<tr>
<td>2</td>
<td>0.94</td>
<td>13</td>
<td>0.78</td>
<td>24</td>
<td>0.65</td>
</tr>
<tr>
<td>3</td>
<td>0.92</td>
<td>14</td>
<td>0.79</td>
<td>25</td>
<td>0.68</td>
</tr>
<tr>
<td>4</td>
<td>0.87</td>
<td>15</td>
<td>1.07</td>
<td>26</td>
<td>0.85</td>
</tr>
<tr>
<td>5</td>
<td>0.68</td>
<td>16</td>
<td>1.17</td>
<td>27</td>
<td>0.65</td>
</tr>
<tr>
<td>6</td>
<td>0.7</td>
<td>17</td>
<td>1.04</td>
<td>28</td>
<td>0.72</td>
</tr>
<tr>
<td>7</td>
<td>0.7</td>
<td>18</td>
<td>0.96</td>
<td>29</td>
<td>0.59</td>
</tr>
<tr>
<td>8</td>
<td>0.74</td>
<td>19</td>
<td>0.74</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>0.74</td>
<td>20</td>
<td>0.74</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>0.76</td>
<td>21</td>
<td>0.83</td>
<td></td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>0.78</td>
<td>22</td>
<td>0.65</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
From Table 4.1 and Figure 4.3 it can be seen that changing the throttle valve from 66mm to 88mm doesn’t significantly change the mass flow rate in the turbine system and the pressure drop across the turbine stage. Hence 88 mm of throttle setting which has the maximum stage pressure drop is selected for this experiment. It has the flow coefficient of 0.001779 and mass flow rate of 10.18 kg/s in the turbine stage.

4.2 Baseline Fluid Flow

In baseline turbine flow field all of the 29 blades were geometrically similar with slightly varying tip clearances. The minimum and maximum value of the tip clearance were 0.59% and 1.17% of the blade span respectively. No blades were treated with TLIs in this experiment. There were 29 blades with each assigned a blade number from 1 to 29. The passage right after the blade was assigned the same numerical value as the blade. FRAP (Fast Response Aerodynamic Probe) probe collected the total pressure value at the measurement plan. The experiment was conducted from 97% of the blade-span to 25% with 1% increment. The non-dimensional blade-span is measured from a cylindrical hub surface (25%) to 97% near the outer casing inner surface. The $C_p$ value is calculated using the equation 4.2 and the $U_m$ is the blade velocity at mid span ($\omega r_m$). The nominator of equation 4.2 is generated by the fast response differential pressure transducer imbedded in FRAP probe. The non-dimensional pressure as a pressure coefficient is as follows.

\[ C_p = \frac{P_{o,exit} - P_{atm}}{\frac{1}{2} \rho U_m^2} \]

---Equation 4.2
4.3 Ensemble Averaging from FRAP Unsteady Total Pressure Output

After fixing the flow coefficient to be 0.00178, a sample experiment was conducted to find sufficient number of revolutions required to get a statistically stable result in terms of the unsteady pressure measurement at each circumferential point, at a selected span-wise position. As previously discussed, there are 6000 circumferential measurement points or bins around the circumference of the rotor as defined by the optical encoder of the turbine rotor. The number of revolutions needed for a statistically stable ensemble average is found from the convergence analysis. The convergence analysis for five different span locations is shown in Figure 4.4. The required number of revolutions varies depending on the span position. Ensemble averaging process effectively removes the random dynamic pressure fluctuations occurring around a deterministic wave form of the dynamic pressure. Turbulent flow, electrical noise, probe vibration related random oscillations can be effectively removed by ensemble averaging. However, passage to passage variations and spatial variations of the dynamic pressure signal are preserved as they are much needed in turbomachinery aerodynamics research. The stars in the Figure 4.4 indicate the number of revolutions required to get statistically stable results corresponding to the blade-span. It can be seen from Figure 4.4 that near the blade tip, a higher number of revolutions (300) is needed for stability while near the core flow and hub region, a relatively smaller number of revolutions (200) is needed to get stable results. As the focus of this experiment was to analyze the tip leakage flows, 400 revolutions were used for the ensemble average. This number was set after a comprehensive experimental evaluation of ensemble averaging process in the beginning of experiments.
Figure 4.4. Convergence analysis for Total pressure at different span location.

Figure 4.5: Correlation between Passage averaged $C_p_{97\%to70\%}$ and the Tip clearance for all the passages.
4.4 Effect of Tip Clearance in the Turbine Flow Field

The passage averaged $C_p$ value of a given blade passage is calculated by taking an average of measured $C_p$ at all the angular positions within a blade passage for given span-wise position (equation 4.3). The data in all measurement bins along one blade pitch are arithmetically averaged in a selected passage. Since the shaft encoder used divides one rotation into 6000 measurement bins, each blade passage in the rotor assembly contains $(6000/29)$ bins along the pitch distance. Hence on passage averaging $C_p$ in each passage will have passage averaged $C_p$ at each span-wise point. Although the circumferential flow variation related pressures are somewhat smoothed out along one selected passage pitch, a good indication of the average level of dynamic total is obvious in these passage averaged representations. This is a good way of comparing two selected passages that may have two different tip clearance gaps or tip designs. It should be noted that FRAP probe due to its high sensor response time (around 100 KHz) can map the unsteady total pressure at rotor exit in a time accurate way. More information on the performance of this probe from past AFTRF research is given in [45], [51], and [52]. Based on the existing tip leakage vortex knowledge, the tip leakage vortex has greater influence on the flow field from the blade tip to 70% of blade-span after which a secondary flow related vortex would be dominant. There are usually situations that a horseshoe vortex may be combined/mixed inside a conventional passage vortex.

The Passage averaged $C_{p_{97\% \text{ to } 70\%}}$ of the blade-span is obtained from taking average of the passage averaged $C_p$ value from 97% of the blade-span to 70%. As we have seen from chapter 2 the more the tip clearance value the more the leakage related
aerodynamic losses in the system. In general, the higher the leakage loss lesser the Passage averaged $C_{p_{97\% \text{ to } 70\%}}$ value. Figure 4.5 shows the correlation between the tip clearance and the passage averaged $C_p$ from 97% to 70% of blade-span for all blade passages. Except a few passages 7, 8, 9, 18 and 20 all other blade passages show the trend of increased loss with increased tip clearance value.

*Passage Averaged $C_p$ of $i^{th}$ passage*

$$= \text{Average of } C_p \text{ of all the angular position in the } i^{th} \text{ passage}$$

---Equation 4.3

### 4.5 Baseline wake plots from FRAP based total pressure measurements

The dynamic total pressure measurements around one complete rotation in their ensemble averaged form are called “wake plots”. The term “wake plot” is related to the fact that the influence of the blade wake and clean core flow is exhibited very clearly in each passage at a fixed spanwise (radial) position. Of course, other flow influences like secondary flows, tip vortices and boundary layers can also be displayed by a complete wake plot along 360 degrees or in one complete rotation of the rotor. The wake plots around the circumference at a selected radial position are shown in figure 4.6. For this 29 bladed rotor, wakes of the 29 individual passages are displayed one after the other. The first negative peak in $C_p$ value for 95% blade-span at 4 degree is from blade 17, every negative peak after that is due to the next blade and ending with blade 16 at 352 degree. The amplitude of oscillation is higher at 95% blade-span due to the presence of the leakage vortex.
Figure 4.6: Wake plot at 95% and 25% of blade-span for all 29 passages.

Figure 4.7: Wake plot at 80% and 25% of blade-span for all 29 passages.
As shown in Figure 4.7 the amplitude of oscillation is lower for the 80% blade-span indicating fewer vortical structures in the area. Then at the 60% of the blade-span the amplitude is greater with almost perfect sinusoidal wave (as shown in Figure 4.8), this is due to regular alternation of core flow and wake structure at that span.

A comparison of wake plots at four different location of (95%, 80%, 60% and 25%) blade-span for passage 6 is shown in Figure 4.9. All of the plots have the same frequency of oscillation, due to fixed frequency of blade passing in front of FRAP probe. Each blade of the 29 bladed rotor assembly generates a wake and core flow related total pressure signal. The amplitude is greater at 60% of blade-span and lower for the 25% of blade-span. Hence from Figure 4.9 the flow near the 60% of the blade-span has higher total pressure. This is due to the presence of a core flow in that region and the flow near the 95% blade-span has more loss because of the leakage vortex in that region. In general the core flow is free from disturbances from a tip leakage vortex, secondary flows and end-wall boundary layers. The viscous flow near the casing suffer from the energy dissipation of end-wall boundary layers and tip vortices. Unsteady flow interactions also exists in the near casing region.

### 4.6 Contour Plots Obtained from Time Accurate and Ensemble Averaged FRAP Based Total Pressure Measurements

The data acquisition from the FRAP probe is phase-locked with the rotor RPM. This phase-locking is controlled by using optical shaft encoder on the turbine shaft. The encoder output has two control signals, the first is one pulse per revolution and the second one is 6000 pulses per revolution. The one pulse per revolution triggers and initiate the FRAP to collect data and 6000 pulses per revolution controls the data acquisition. It marks
Figure 4.8: Wake plot at 60% and 25% of blade-span for all 29 passages.

Figure 4.9: Wake plot at 95%, 80%, 60% and 25% of blade-span for passage 6.
the start of data acquisition. The working of this phase locked data acquisition system is been explained in detail in [45], [51], [52].

In general, the total pressure in an axial flow turbine drops continuously as the passage flow progresses in axial direction. The specific passage plane almost normal to overall stream-wise direction has a somewhat lower total pressure from its upstream counterpart due to work extraction going on in the passage. As the main flow turns and progresses in the rotor passages, the total pressure is significantly reduced because of work extraction and because of the loss of mean kinetic energy of the flow due to viscous dissipation. In a modern high pressure turbine stage the total pressure at the exit of the stage could easily be half of its stage entry value that is determined by the combustor exit flow conditions. As the flow progresses in stream-wise direction if one observes the local flow features at a selected stream-wise position, the passage flow area normal to stream-wise direction shows tremendous local total pressure variations. The same is true for the stage exit plane that is defined as a plane normal to axis of rotation as shown in figure 4.10. In general, the highest local total pressure is experienced in the core flow near 50-60 percent span where a flow system free from passage vortices, tip vortices and boundary layers exists. This flow actually represents the part of the flow that does most of the work. The “reddish” total pressure zones of figure 4.10 at a $C_p$ level of about -5 belong to that high work extraction zone near mid-span. The phase-locked unsteady total pressure image of figure 4.10 also shows other zones that may contain blade wakes (light green, $C_p = -4.6$), tip vortices (light blue $C_p=-5.281$) and secondary flows. Hub boundary layers are not shown in figure 4.10 since the probe is intentionally stopped near 25 % span measured
from the hub surface. FRAP probe generated contour plots show a great comparison of
different aerodynamic loss areas existing in AFTRF rotor exit plane. This approach allows
the researcher to compare each one of the 29 passages effectively. For example, the tip
vortex image of passage 23 (t/h=0.58%) indicates a local $C_p$ level of -4.6. t/h=0.58 is almost
the minimum tip gap in this rotor. Passage 16 contains the tip vortex originating from the
blade with the highest tip clearance value of t/h=1.15 %. The influence of tip gap height
on the local total pressure is obvious. The minimum $C_p$ value in all passages of figure 4.10
is measured in passage 16 since it has the highest tip gap height of t/h=1.15 %. Obtaining
a rotor exit instantaneous $C_p$ contour map is an effective way of visualizing and quantifying
passage to passage variations for the stage. If one installs a blade with a very large clearance
compared to the rest of the blades, its influence on $C_p$ is clearly observable. This research
effort uses this significant property of the current FRAP operation in an effort to quantify
the positive or detrimental influence of new blade tip designs.

Unsteady total pressure information about blade passage, core flows, blade wakes
and leakage vortices and end-wall boundary layers can be seen clearly in a contour plot as
shown in figure 10. Lower the negative value of the $C_p$ lesser the loss at that region when
a passage is compared to another one. The orange/red color in the spectrum represent most
efficient region while the blue color in the spectrum represents the most lossy region in the
flow field. The radial black lines represent the start and end of a passage. The start and end
points are set by the user. The dotted radial line in the inner circle indicates the mid line of
the corresponding passage where t/h=0.50. The outer dotted circle represents the circle
traversed by blade tip and the inner circle indicates the root of the blade attached to the
hub. There is no data from the inner circle (hub) to 25% of blade-span and 97% of blade span to the outer circle. The number in black color near the outer circle represents the blade passage number and the red number represents its corresponding tip clearance value. The blade passage number is the same as the blade number assigned in the beginning of study.

As explained in chapter 2, the most energy efficient region in the turbine flow field is the core flow and the least efficient region is near the blade tip region. The overall aero loss level existing in the wake region and secondary flow zones are intermediate levels between the least efficient (tip vortex) level and the most efficient (core flow) levels. The core flow in figure 4.10 is shown in reddish/orange color. All of the passages have a blue circular region near the blade tip, this is due to the presence of tip leakage vortex and is a source of loss in the turbine field. The level of loss or color in this region directly correlates with the tip gap height. The blades with larger tip clearance generate larger leakage vortex (as seen in figure 4.10) with more momentum deficit in this highly re-circulatory and turbulent region in turn generate more loss in the system.

4.7 Passage Averaged Delta $C_p$ Plots Obtained from FRAP Probe

As explained before, the passage averaged $C_p$ value of a given blade passage is calculated by taking an arithmetic average of $C_p$ value at all the angular position within a blade passage for given span. The average is obtained along the pitch line of the specific passage chosen. The passage averaged $C_p$ plot of passage 6 is shown in Figure 4.11. The passage averaged $C_p$ plot increases linearly from 25% to 57% of the blade-span. The passage averaged $C_p$ reaches a maximum value in between 57% to 60% of the blade-span,
after which it decreases linearly till 70% blade-span. The behavior of plot above 70% blade-span is the main focus of the study here. The region between 70% and 97% of blade-span can be split into four sub-regions. Passage averaged \( C_p \) decreases in the sub-region from 97% to 92% of blade-span and increases in the sub-region from 92% to 85% of blade-span. The curve is nonlinear in the sub-region 85% to 75% of blade-span and the non-linearity depends on the blade modifications. The curve is again linear in the sub-region 75% to 70% of blade-span. The slope of all these linear lines depends on the blade modifications.

The relatively lower value of passage averaged \( C_p \) near 25% blade-span is due to lower flow velocity in this region, as this region is near the hub. In addition, this region is also affected by the hub end-wall boundary layer. As explained in chapter 2, due to the rotation of the hub, end-wall the boundary layer is skewed which in-turn creates three dimensional and rotating boundary layer systems. The axial length of the hub end-wall is 86.4 mm. After the flow passes the rotating end-wall, it experiences the stationary end-wall. The linear increase in passage-averaged \( C_p \) plot from 25% to 57% of blade-span in Figure 4.11 can also be correlated with the linear increase in distribution of orange color from 25% to 57% of blade-span in the Figure 4.10. The maximum passage average \( C_p \) value is around 57% to 60% of blade-span as this region has the core of the flow. After 60% of blade-span, passage averaged \( C_p \) decreases linearly due to the secondary vortex. The decrease in passage averaged \( C_p \) near the blade tip is due to tip leakage vortex and the non-linearity in the passage averaged \( C_p \) distribution is due to the mixing of secondary vortex and tip leakage vortex.
Figure 4.10: Total pressure contour plot of 29 passages in the baseline flow field.
Near the blade tip the lowest passage average $C_p$ is at the 92% of the blade-span. This is because the core of the leakage vortex is in this region. The passage averaged $C_p$ at the very end of the blade tip increases due to the shearing effect of the casing surface in the casing boundary layers.

4.8 Elimination of Bias Error from the Passage Averaged Total Pressure Coefficient

The passage averaged $C_p$ distribution from four different passages (figure 4.12) has an observable level of data scatter with respect to each other (red circle) which makes it difficult to compare with the reference blade passage. The $C_p$ data near 25-30 % span have this data scatter in this region. These slight passage to passage variations can be effectively removed with the approach described in this section. This approach is based on the observation that any modification near blade tip should not significantly affect the flow field near the hub. Hence the passage averaged $C_p$ distribution near the hub should have very similar magnitude. If there is some data scatter near the hub, this is because of passage to passage variation of overall blade passage geometry. The data as shown in figure 4.12 is obtained from passages 5, 6, 7 and 8 of one AFTRF experiment. Based on this assumption the data scatter between the passages can be somewhat reduced or even eliminated by subtracting a reference pressure obtained from the passage of interest. This reference pressure could be taken as passage averaged $C_p$ value from 25% of the blade-span (closest data point with respect to hub).
Figure 4.11: Passage averaged $C_p$ span wise distribution for passage 6.

Figure 4.12: Passage averaged $C_p$ span wise distribution for passage 5, 6, 7 and 8.
Figure 4.13: Passage averaged \((C_p - C_p^{25\%})\) span wise distribution for passage 5, 6, 7 and 8.

Figure 4.14: Span wise distribution of delta \(C_p\) for passage 5, 6, 7 and 8.
Even the passage averaged $C_p$ values of the next position (26%) varies significantly with respect to each other. This approach tends to remove passage to passage variations from the $C_p$ data effectively as shown in Figure 4.13. The passages 5, 6, 7 and 8 have all very close tip clearance values. The tip clearance values of 5, 6, 7 and 8 are all carefully adjusted to 0.68 %, 0.7 %, 0.7 % and 0.74 % for these experiments. Figure 4.13 clearly shows the removal of passage to passage variations from the passage averaged $C_p$ data.

Another method is to take an arithmetic average of 10 data points near the hub (i.e. from 25% to 34% of blade-span) and subtract the passage average $C_p$ distribution of the passage with this averaged value. This approach is essentially the same approach as described in the previous paragraph. However, this approach provides a more repeatable result as the passage averaged $C_p$ distribution of the passage is shifted by a value obtained from a collection of 10 data points and not from a single data point. This type of averaging improves the removal of data scatter from the passage averaged data as shown in figure 4.14. The $C_p$ distributions obtained from the four individual passages having almost the same tip clearances now show almost identical $C_p$ distributions in all span-wise locations. The shifted passage averaged $C_p$ obtained by this definition is called the delta $C_p$ given by equation 2. Almost the same distribution of delta $C_p$ value can be seen near the 25% of the blade-span and are overlapping each other in the 25% to 50% blade-span region.

**Delta $C_p$ of passage $i$**

\[
\Delta C_p = (\text{passage average } C_p \text{ of passage } i) - \text{avg } (\text{passage averaged } C_p_{25\%} \text{ to } C_p_{34\%}) \text{ of passage } i
\]

--- Equation 4.2
4.9 Figure of Merit (FOM)

The Figure of Merit (FOM) of a blade is calculated by using the delta $C_p$ span-wise distribution of an individual blade. This delta $C_p$ distribution is then compared to the reference blade of the same experimental run. The reference blade usually carries a standard flat tip design. In general, FOM is useful in comparing different (novel) blade tip designs running at the same clearance in the same turbine run. FOM is effective in quantifying the aerodynamic effectiveness or ineffectiveness of a (novel) blade tip design from a blade tip having a conventional flat tip running at the same effective tip clearance.

In the delta $C_p$ span-wise distribution plot a vertical line called Reference line is drawn at the maximum point (indicated by the star in Figure 4.15) of the reference blade passage (around 65% span) as shown in Figure 4.15. The area between the delta $C_p$ curve of the reference curve and the reference line from 97% to 70% of the blade-span is the Reference Area, $A_B$. Similarly the area between the delta $C_p$ curve of the desired passage and the reference line from 97% span to 70% span is Area, $A_i$. The reference area is used as a merit figure which is used to compare the area of the desired passage. The Figure of Merit (FOM) definition is given as a quantity which is normalized by the reference area, and multiplied by 100. The figure of merit for passage $i$ is given by equation 4.3. The figure of merit of passage 4 with respect to passage 6 is shown in Figure 4.16. As the blade 4 have higher tip clearance than the blade 6 the passage 4 has more loss than the passage 6 near the blade tip region. This can be seen from the negative value of FOM (-2.282) obtained for passage 4 with respect to passage 6.
\[ FOM_i \equiv \frac{A_i - A_B}{A_B} \times 100 \]

--- Equation 4.3
Figure 4.15: Figure of merit, FOM for passage i.

Figure 4.16: Figure of merit, FOM for passage 4 with passage 6 as reference passage.
Chapter 5

Results and Discussion

Three different experimental studies are completed on recently introduced TLIs that are mounted in the rotor of the AFTRF. The first study was performed to analyze the effect of TLI mounting location on the suction side of blade tips. The second experimental study involved varying the number of TLIs attached onto the suction side of the blade tip and analyzing its effect on the overall leakage flow system. The effect of mounting orientation of TLIs on the leakage losses also was studied in the third experimental study.

5.1 Study on Mounting Location of TLI near Blade Tip Region (Experiment A)

In this experimental investigation the variation of leakage loss depending on the mounting location of TLIs onto the suction side of the blade tip was determined. Four different blades with similar non-dimensional tip clearance within $\pm$ 0.05 % (t/h) were chosen for this study and their details are given in table 3.3. As explained before, two blades with tip clearances within $\pm 0.05\%$ of blade-span are considered to have same tip clearance, as this change is easily within the measurement and installation uncertainty. On converting 0.05% of the blade-span to measurement units gives 0.061 mm or 2.4 mils. Hence any two blades with tip clearance difference of $\pm 2.4$ mils is considered to have same tip clearance. For example the reference blade (blade 20) which has the tip clearance of 0.74% of blade-span and the blade a1 (blade 6) which has the tip clearance of 0.7% of blade-span are considered to have same tip clearance.
The contour plot obtained from the FRAP (Fast Response Aerodynamic Probe) based total pressure measurement for all 29 blade passages is given in Figure 5.1. The passages under detailed study are indicated by the red box in Figure 5.1. The shades of blue indicate lower local total pressure in the region while shades of orange indicated higher local total pressure in the region. The ensemble averaged wake plot obtained from the FRAP probe measurement for all the blades at 92%, 60% and 25% of blade-span is given in Figure 5.2. 400 ensembles of the measured total pressure around the rotor circumference is used to obtain a statistically stable rotor exit total pressure field. This pressure field is phase-locked and represents the field for one selected position of the rotor when observed from the stationary frame of reference. The green line in the Figure 5.1 represents the once per revolution signal (OPR) from the optical encoder, which initiates the data acquisition process from the FRAP probe. This line is located at 85% of the passage 16, and represents zeroth angular location in the measurement frame.

The angular location of the blades in the measurement frame is given in Figure 5.1. Based on these angular locations it can be seen that the first peak of the 92% of blade-span wake plot in Figure 5.2 belongs to passage 17. A typical wake plot is shown for a selected span-wise position. The ensemble averaged data using 400 ensembles is presented around the circumference. There are 6000 equally spaced data points around the circumference in one complete rotation. It is also said that there are 6000 data bins in one complete rotation of the AFTRF rotor. The data in each bin is the arithmetic average of the data from 400 subsequent rotations.
Figure 5.1: Total pressure contour plot of 29 passages in the Experiment A.

Figure 5.2: Wake plot of all the passages at 92%, 60% and 25% of blade-span.
Figure 5.2 contains the ensemble averaged total pressure data from all 29 passages at a selected span-wise position. Each scatter plot containing 6000 data points have passage flow features from all 29 blades. The 60% of blade-span data as shown with red open symbols have 29 peak points (highest total pressure, approximately -6300 Pa) and also 29 minimums (lowest total pressure around -7300Pa). The inlet total pressure is at atmospheric level in AFTRF at 0 Pa. The measurements presented in Figures 5.1 and 5.2 are all differential pressure measurements with respect to the inlet atmospheric level.

The highest total pressure level (29 peak points) around -6300 Pa belongs to the core flow area near 60% of blade-span where there are no flow complications like wakes, secondary flows, boundary layers or tip vortices. The lowest total pressure of about -7300 Pa belongs to the light green wake region as shown in the contour plot of Figure 5.1. The influence of the tip vortex system is minimal at 60% of blade-span and the variation of the signal observed at 60% of blade-span is almost periodic from one passage to another. Very slight passage to passage variations of this signal from this measurement is due to minimal passage to passage geometrical/flow variations. This plot from the mid-span region near 60% of blade-span is the cleanest flow zone minimal aerodynamic losses.

The total pressure data measured at 92% of blade-span contains the major influence of the tip vortices. The scatter plot with the blue symbols show 29 distinct maximums at around -6900 Pa. This peak level is sampled in the zones where there is no tip vortex dominance a (92% of blade-span). However, when the FRAP probe samples from the area with tip vortex influence near the suction side of each blade, the minimum total pressure level in the passage is encountered because of the high levels of the dissipation of mean kinetic energy near the core of the individual tip vortex system. The minimum total
pressure measured in the core of the tip vortex dominated zone is about \(-7900\text{Pa}\) on the average. The minimum level of blue symbols measured in each passage show a good level of passage to passage variation. This is mainly the influence of the local tip clearance existing on each individual blade tip. The exact level of the total pressure minimum directly correlates with the tip clearance height of individual blade.

At 25\% of blade-span the total pressure data of all the passages obtained from FRAP probe shows that there is no significant variation in the signal among the passages and the signal is almost periodic. This is because there is no significant passage to passage flow variation and the variation in the tip region due to tip clearance is no longer felt in this region. The amplitude of the signal obtained in this region is also small compared to 60\% and 92\% of blade-span with a maximum peak value around \(-7300\text{ Pa}\). The minimum value of this signal is around \(-7400\text{ Pa}\) which is smaller than the minimum value of the signal from 92\% of blade-span, indicating that this region has lower losses in the flow field compared to the 92\% of blade-span.

One interesting feature of the current experiments is that the current measured FRAP data from AFTRF has great sensitivity to the tip clearance height and design. A slight tip clearance modification or any new blade tip design is immediately captured on the current contoured plots and wake plots. This ability to differentiate the flow physics related variations from one passage to another makes this system of operation very effective in the conceptual design of new tip mitigation schemes.
5.1.1 Characteristics of Blade a1

The contour plot obtained from the FRAP based total pressure measurement for the blade a1 and reference blade is given in Figure 5.3. The leakage vortex present in both passages are indicated by shades of blue. The wake profile obtained from the FRAP probe for the blade a1 and the reference blade at 92%, 60% and 25% of the blade-span is given in Figure 5.4. From the angular measurement in Figure 5.1, the reference passage is from 34 to 46 deg and the passage a1 is from 220 to 233 deg. From the Figure 5.4 it can be seen that there is not much difference in wake profile of the reference passage and passage a1 at 60% and 25% of the blade-span, but for passage a1 at 92% of the blade-span the initial oscillation slightly flattens out. The improvement in performance of the blades modified with TLI tips can be clearly seen in the span wise variation of passage averaged delta $C_p$ plot for the passage a1 (passage 6) and reference passage as shown in figure 5.5.

As explained in chapter 4, the figure of merit (FOM) for the passage a1 with respect to reference passage was calculated and found to be 3.6. As the FOM is positive the blade a1 has better performance than the reference blade. While comparing passage a1 to reference passage using Figure 5.5 it can be seen that any rightward shift of the passage averaged Delta $C_p$ plot for passage a1 with respect to the reference passage is beneficial as the average Delta $C_p$ value increases. Similarly any leftward shift of the averaged delta $C_p$ plot for passage a1 with respect to the reference passage shows more loss as the average delta $C_p$ value decreases. The span-wise comparison of the passage a1 to the reference passage can be obtained by drawing lines between the passage averaged delta $C_p$ plot for passage a1 and reference passage for each span-wise location.
Figure 5.3: Total pressure contour plot of the passage a1 (passage 6) and reference passage (passage 20).

Figure 5.4: Wake plot of the passage a1 (passage 6) and reference passage (passage 20) at 92%, 60% and 25% of blade-span.
The green lines in figure 5.5 indicated the rightward shift of Delta $C_p$ graph of the blade $a1$ and the orange line in figure 5.5 indicate the leftward shift. The lengthier the green line or the orange line more beneficial or more loss the blade design is at that span-wise location. In Figure 5.5 the length of green line is greater near 97% of blade span indicating the most beneficial place because of this concept. The length of the green line decreases from 97% to 93% of blade span. From blade span 92% to 85% the length of the green line decreases again. The curves are nonlinear in the sub-region from 85% to 74%. The length of the green line remains almost constant from 75% to 70%. This shows that by attaching TLIs to the blade tips, the TLIs have a significant beneficial effect near the blade tip and its effect decreases from the blade tip to the blade hub. After 70% of the blade-span the TLIs have very little effect and the performance of the blade $a1$ is similar to the performance of the reference blade.

Figure 5.6 provides an explanation for these results. As explained before in chapter 3, when a flow is passed over a TLI, it generates vortical structures in the downstream. The direction of rotation of these vertical structure depends on the local angle of incidence of the flow and the TLI. If it is mounted at 45 deg with respect to the line, then it creates a vortex system in the clockwise direction. The leakage vortex generated by the tip leakage flow are usually in the counter-clockwise direction. As these two vortices are in opposite direction, the induced TLI vortex system on interaction with the tip leakage vortex reduces the overall effect of the leakage vortex. The influence of the newly introduced vortex system is only felt near the blade tip region.
Figure 5.5: Span-wise distribution of the Passage averaged Delta $C_p$ plot for passage a1 and reference passage.
Figure 5.6: Interaction of the Tip leakage vortex and the TLI induced vortex in the blade.
5.1.2 Characteristics of Blade a2

In blade a2 the TLI is attached at 65% of the axial chord in the blade tip. The contour plot indicating the variation of total pressure inside the passage a2 and the reference passage is shown in Figure 5.7. The angular range of the passage a2 (passage 10) is from 273 deg to 285 deg (as shown in Figure 5.1). The wake plot of the 92%, 60% and 25% of the blade-span for the passage a2 and the reference passage is shown in Figure 5.8. It can be seen that the minimum peak pressure value of the passage a2 at 92% of the blade-span is -7750 while the same for reference passage is -7950. This indicates that the passage a2 has better performance than the reference passage at 92% of the blade-span as lower the pressure value indicate the more loss in the system. Passage a2 total pressure level at 92% span is measurably higher than its reference counterpart. This can also be seen in Figure 5.9 which shows the span-wise variation of the passage averaged delta $C_p$ value of the passage a2 and the reference passage.

In Figure 5.9 the FOM of the passage a2 is 2.3 and since it is a positive value the blade a2 have better performance than the reference blade. The passage a2 exhibit similar span-wise delta $C_p$ distribution as the blade a1 in the sub-region 97% to 93% of the blade-span. But it has lengthier green line at 91% of the blade-span than blade A1 and the length of the green line decreases linearly in its length till 84% of the blade-span. In the sub-region 85% to 75% of the blade-span the blade a2 has poor performance in comparison to the reference blade and it is indicated by the orange line in Figure 5.9. The lengthier the orange line, more poor the performance of the blade a2 compared to the reference blade. The varia-
Figure 5.7: Total pressure contour plot of the passage a2 (passage 10) and reference passage (passage 20).

Figure 5.8: Wake plot of the passage a2 (passage 10) and reference passage (passage 20) at 92%, 60% and 25% of blade-span.
Figure 5.9: Span-wise distribution of the Passage averaged Delta $C_p$ plot for passage a2 and reference passage.
Figure 5.10: Interaction of the Tip leakage vortex and the TLI induced vortex in the blade a2.
tion of the length is non-linear in this region. The plot has a small green line at 74% of the blade-span but back to orange line in the sub-region 73% to 70% of the blade-span. In a general sense the performance of this blade can be split into two regions from 97% to 84% of the blade-span the attachment of the TLI at 65% of the blade axial chord at the blade tip is beneficial while from 85% to 70% of the blade-span it is detrimental to the performance of the blade.

Figure 5.10 provides an explanation for the reduced FOM of blade a2 compared to blade a1. Like the TLI in blade a1 the TLI in blade a2 also generates vortex system. But the TLI in blade a2 are mounted at 65% of the axial chord while in blade a1 they are mounted at 45% of the axial chord. Hence the vortex formed from the blade a2 is smaller compared to the vortex generated by blade a1 and have lower effect in reducing the leakage vortex.

5.1.3 Characteristics of Blade a3

The TLI on blade a3 is attached at 75% of the axial chord on the blade tip. The contour plot indicating the variation of the total pressure is obtained from time accurate FRAP probe. Figure 5.11 shows the total pressure contour plot for the passage a3 and the reference passage. Passage a3 (Passage 14) in Figure 5.1 has the angular range from 321 to 33 deg. The wake plot for passage a3 and reference passage at 92%, 60% and 25% of the blade-span is provided in the Figure 5.12. The maximum peak pressure value at 60% of the blade-span for the passage a3 is -6450 while the same for reference passage is -6300. In general the lower the negative pressure in the flow field more favorable it is for the
performance of the blades. This indicates that the blade a3 has worse performance than the reference blade at 60% of the blade-span. Figure 5.13 also shows the poor performance of the blade a3 compared to reference blade at 60% of the blade-span.

FOM of blade a3 is -3.0 as shown in Figure 5.13, as the FOM is a negative value the blade a3 have poor performance than the reference blade. Except the regions from 91% to 80% of the blade-span indicated by the green lines, the introduction of TLI at 75% of the axial chord is less beneficial from 97% to 70% of the blade-span. The length of the orange line is maximum at 92% signifying the worst performance location of the blade a3 compared to the reference blade. The curve and the length of the lines are nonlinear from 84% to 77% of the blade-span. Orange line increasing in length from 75% to 70% of the blade-span indication detrimental effect of the blade in comparison with the reference blade.

Hence mounting TLI at 75% of the axial chord degrades the performance of the blade. In Figure 5.14 it can be seen that the before the TLI can induce a clockwise rotating vortex in the flow field, the leakage flow overshadows the TLI and hinders the possibility of generating a vortex. The TLI on the blade tip without generating a vortex system to counteracts the leakage vortex acts as a barrier to the flow and introduces loss in the system. This is one of the reason the FOM of the blade b3 is negative.
Figure 5.11: Total pressure contour plot of the passage a3 (passage 14) and reference passage (passage 20).

Figure 5.12: Wake pot of the passage a3 (passage 14) and reference passage (passage 20) at 92%, 60% and 25% of blade-span.
Figure 5.13: Span-wise distribution of the Passage averaged Delta $C_p$ plot for passage a3 and reference passage.
Figure 5.14: Interaction of the Tip leakage vortex and the TLI induced vortex in the blade a3.
5.1.4 Comparison of the Three TLI Modified Blades

The mounting of TLI in the blade tip at 40\% and 65\% of blade axial chord in blade a1 and blade a2 respectively reduces the strength of the tip leakage vortex, this can be seen from Figures 5.5 and 5.9 where the passage averaged Delta $C_p$ plot is shifted towards right. This shows that the vortex created by the TLI counteracts with the tip leakage vortex and reduces its effective strength. On the other hand mounting TLI at 75\% of the blade axial chord in blade a3 have detrimental effect on the passage flow field. The vortex created by the TLI here aids the leakage flow in creating loss in the system. On comparing results from Figures 5.5, 5.9 and 5.13 it can be seen that the blade a1 have superior performance than the blade a2 and a3 in all the region from 97\% to 70\% of the blade-span, except 92\% to 89\% of the blade-span where blade a2 have the best performance. This shows that the effect of TLI on the leakage flow is dependent on its mounting position in the blade tip. Figure 5.15 shows the FOM value obtained based on the mounting location of the TLIs on the blade tip. Blade A1 gives 36\% higher FOM than blade A2. While Blade A3 have increased the losses in the flow field. Hence among the three axial chord location on the blade tip, mounting TLI at 40\% of the blade axial chord have the greatest effect in reducing the loss in the passage flow field.
Figure 5.15: Variation of FOM with the mounting location of the TLI.
5.2 Study on the Effect of Number of TLI Mounted in Blade Tip (Experiment B)

The effect of number of TLIs mounted on the blade tip over the leakage loss is analyzed in this study. Three different experimental runs (B1, B2 and B3) with 1, 3 and 5 individual TLIs mounted on the blade tip respectively were performed to understand the influence of TLIs over the leakage flow. In all the three experiments blade 6 was modified with TLI blade tips and blade 20 was selected as reference blade. The details of the number and mounting location of TLIs on-to the blade tips for three different experiments are given in the Table 3.4.

5.2.1 Characteristics of Blade b1

In blade b1 only one TLI is attached at 40% of the axial chord on the blade tip. The total pressure contour plot obtained from time accurate FRAP probe for all the 29 passages is given in the Figure 5.16. The passages of interest are the passage b1 (passage 6) and reference passage (passage 20), both are marked by red boxes in the Figure 5.16. The angular location of all the passages in the measurement frame is also given in Figure 5.16. Figure 5.17 shows the total pressure contour plot of the passage b1 and the reference passage. The patches of blue region near the 92% of the blade-span in Figure 5.17 are the tip leakage vortex generated due to blade b1 and reference blade. The wake plot obtained from the total pressure data measured by FRAP probe for all the passages at 92%, 60% and 25% of the blade-span is given Figure 5.18. From Figure 5.16 it can be seen that the angular location of the passage b1 (passage 6) is from 223 to 235 deg, while for the reference passage (passage 20) it is from 34 to 46 deg.
Figure 5.16: Total pressure contour plot of 29 passages in the Experiment B1.

Figure 5.17: Total pressure contour plot of the passage b1 and reference passage 20.
Figure 5.18: Wake plot of all the passages at 92%, 60% and 25% of blade-span in Experiment B1.

Figure 5.19: Wake plot of the passage b1 (passage 6) and reference passage (passage 20) at 92%, 60% and 25% of blade-span in Experiment B1.
Two sections of Figure 5.18 which represents the wake plot of passage b1 and reference passage are shown Figure 5.19. The wake plot at 92% of the blade-span of both the reference blade and the blade b1 are similar except at the maximum peak of the plot where the pressure distribution is flat in blade b1 compared to clear peak in reference blade.

The blade span-wise distribution of the delta $C_p$ plot for the passage b1 and reference passage is shown in Figure 5.20. The FOM for passage b1 is 3.7 showing that the blade b1 has better aerodynamic performance than the reference blade. The green lines in Figure 5.20 shows the rightward shift of the delta $C_p$ plot of the passage b1 compared to the reference passage and in turn indicates the better performance regions in passage b1 due to the attachment of the TLIs. The rightward shift is greater in the region 93% to 84% of the blade-span signifying the most beneficial region from the TLIs. The TLI worsens the performance of the blade close to blade tip (97% to 95% of the blade-span). This can be seen from the orange lines in those region. There is a slight improvement in blade performance in the region 85% to 75% of the blade-span. The TLIs does not have much influence in the region 74% to 70% of the blade-span as seen from alternating small green and orange lines. Even though there is a slight performance deterioration in the blade tip (as shown by orange line in Figure 5.20), it is overcome by the improvement in performance at all other regions of the blade span (as shown by green lines in Figure 5.20). Hence the net result of attaching a TLI on to the blade tip is positive (FOM is 3.7) and it improves the performance of the blade. The flow physics behind the improvement in the performance of the blade by attaching the TLI at 40% of the axial chord of the blade tip is explained before and shown in Figure 5.15.
Figure 5.20: Span-wise distribution of the Passage averaged Delta $C_p$ plot for passage b1 and reference passage in Experiment B1.
5.2.2 Characteristics of Blade b2

Three TLIs are attached on the blade tip of blade b2 at 40%, 65% and 75% of the axial chord. Figure 5.21 shows the contour plot of the total pressure of all the passages along with their angular location in the measurement plane. The passage b2 and the reference passage are marked with red boxes in the Figure 5.21. From Figure 5.21 the detailed total pressure contour plot of the passage b2 and the reference passage are shown in Figure 5.22. From the total pressure data obtained by the time accurate FRAP probe the wake plot of all the passages have been shown in Figures 5.23. Figure 5.24 shows the wake plot for the passage b2 and reference passage. The amplitude of pressure variation at 25% of the blade-span is small compared to the pressure variation at 60% and 92% of the blade-span, indicating fewer vertical structure in that region.

Figure 5.25 shows the blade span-wise distribution of the delta $C_p$ plot for the passage b2 and the reference passage. The figure of Merit obtained for the passage b2 is 4.3, which is higher than the figure of merit obtained for the blade b1 (FOM is 3.7). The effect of three TLIs on the leakage flow can be clearly here. The maximum delta $C_p$ rightward shift for the blade b2 in the figure 5.25 is at 90% of the blade-span. The green line at this span has the highest length indicating the most benefited blade span due to TLIs. Orange lines are present near the blade tip from 97% to 95% of the blade-span, hence have more loss in this region compared to the reference blade. From 96% to 85% of the blade-span the curve is linear with blade b2 having better performance than the reference blade. The length of the green line varies non-linearly in the region 86% to 75% of the blade-span.
Figure 5.21: Total pressure contour plot of 29 passages in the Experiment B2.

Figure 5.22: Total pressure contour plot of the passage b2 and reference passage 20.
Figure 5.23: Wake plot of all the passages at 92%, 60% and 25% of blade-span in Experiment B2.

Figure 5.24: Wake plot of the passage b2 (passage 6) and reference passage (passage 20) at 92%, 60% and 25% of blade-span in Experiment B2.
Figure 5.25: Span-wise distribution of the Passage averaged Delta $C_p$ plot for passage b2 and reference passage in Experiment B2.
Figure 5.26: Interaction of the Tip leakage vortex and the TLI induced vortex in the blade b2.
The effect of mounted TLIs are not seen in the region 74% to 70% of the blade-span, as there are alternating green and orange lines with smaller length. Similar to blade b1 the TLIs results in more loss near the blade tip but improves the overall performance of the blade.

The passage b2 has greater FOM than the passage b1. By increasing the number of TLIs attached on the blade tip from one to three the FOM of the blade increased by 16.2%. This is because adding additional TLIs on the blade tip introduces more TLI induced vortices in the flow field (as shown in Figure 5.26). This increases the strength of the overall TLI induced vertical structure and are more effective in minimizing the effect of the leakage vortices, as the leakage vortex and the TLI induced vortex are counter rotating.

5.2.3 Characteristics of Blade b3

The blade b3 have five TLIs attached on the suction side of the blade tip at 40%, 55%, 65%, 70% and 75% of the blade axial chord. The total pressure contour plot of all the passages obtained from the time accurate FRAP probe is given in Figure 5.27 with passage b3 and reference passage marked in red boxes. Figure 5.28 shows the total pressure contour plot of the passage b3 and the reference passage. The blue patches near the 95% of the blade-span in the Figure 5.28 indicates the leakage vortex generated in the turbine flow field. The range of $C_p$ in the contour plot is from (-5.3 to -3.9). The wake plot of all the passage at 92%, 60% and 25% of the blade-span is provide in Figure 5.29. Figure 5.30 shows the wake plot of the passage b3 and the reference passage.
Figure 5.27: Total pressure contour plot of 29 passages in the Experiment B3.

Figure 5.28: Total pressure Contour plot of the passage b3 and reference passage.
Figure 5.29: Wake plot of all the passages at 92%, 60% and 25% of blade-span in Experiment B3.

Figure 5.30: Wake plot of the passage b3 (passage 6) and reference passage (passage 20) at 92%, 60% and 25% of blade-span in Experiment B3.
The maximum peak in the wake plot in 60% of the blade-span in Figure 5.30 represents the most efficient region in the passage flow field. The delta $C_p$ plot for the passage b3 and reference passage in span-wise distribution is shown in Figure 5.31. The FOM obtained for the passage b3 is 5.0. From Figure 5.31 it can be seen that attachment of five TLIs on the blade tip negatively affects the tip leakage flow near the blade tip (97% to 95% of the blade-span) after which it reduces the leakage loss. The TLIs improve the performance of the blade from 96% to 71% of the blade-span. The lengthiest green line is at 89% of the blade-span indicating the most beneficial region from the TLIs. The delta $C_p$ curve is linear from 92% to 86% of the blade-span. The length of the green line varies nonlinearly from 85% to 75% of the blade-span. The effect of TLIs in the turbine flow decreases from 75% to 70% of the blade-span. After 70% of the blade-span the TLIs have very little effect over the turbine flow, as it can be seen from linearity of the delta $C_p$ curve of bladeb3 and the reference blade. The overall effect of having five TLIs in blade tip positive as the FOM of blade b3 is positive.

Increasing the number of TLIs mounted on the blade tip from three to five, increased the FOM of the blade by 16.3%. As shown in Figure 5.32 adding more TLIs onto the suction side of the blade tip increases the overall strength of the clockwise rotating TLI induced vortex, thus effectively reducing the counter clockwise rotating leakage vortex.
Figure 5.31: Span-wise distribution of the Passage averaged Delta $C_p$ plot for passage b3 and reference passage in Experiment B3.
Figure 5.32: Interaction of the Tip leakage vortex and the TLI induced vortex in the blade b3.
5.2.4 Comparison of the Three TLI Modified Blades

Increasing the number of TLI mounted on the suction side of the turbine blade tip has a clear benefit on the tip region of the turbine flows. This can be clearly seen in Figures 5.20, 5.25 and 5.31 where adding TLIs onto the surface of the blade tip results in rightward shift of the delta $C_p$ plot. Due to this shift the FOM values of the three different blade tip configurations (blade b1, blade b2 and blade b3) obtained from experiment B shows that the FOM value linearly increases with the number of TLI mounted on the blade tip surface (as shown in Figure 5.33). Even though addition of TLI onto the blade tip induces loss near the blade tip region (97% to 95% of the blade-span) the overall effect of the TLIs is to decrease the leakage loss in the flow filed. Blade b3 has five TLIs mounted on it, highest TLIs mounted among the three blade tip configurations hence have highest FOM and the most beneficial blade among the three blade tip configurations. It also should be noted that this study on the effect of number of TLIs mounted onto the leakage flow also depends on the position on which the TLIs are mounted. Mounting the TLIs after 75% of the blade axial chord would have detrimental effect on the leakage flow.
Figure 5.33: Variation of FOM with the number of TLIs attached in blade tip.
5.3 Study on Effect of Orientation of TLI Mounted in Blade Tip (Experiment C)

The mounting orientation of the TLI plays a significant role in the way the TLI induced vortices affect the leakage vortex. In this experimental study the blade tip performance was analyzed for two different orientations of the TLI. Three different blades with similar tip clearance (blade 6, blade 12 and blade 20) out of 29 turbine blades are selected for this study. Table 3.5 gives the detail of the mounting location and the mounting orientation of TLIs on the blade tip. The total pressure contour plot on the measurement plane obtained using the FRAP probe is given in Figure 5.34. The angular location of the blades in the measurement plane is given in Figure 5.34 and the blades under study are marked with red boxes. Figure 5.35 shows the wake plot of all the blades at 92%, 60% and 25% of the blade-span extracted from the total pressure data obtained from the time accurate FRAP probe. From Figure 5.34 it can be seen the zero angular location in the measurement plane is at 90% of the passage 16. The first maximum peak in pressure at 92% of the blade-span in Figure 5.35 is due to passage 17.

5.3.1 Characteristics of Blade c1

Five TLIs are attached at five different axial chord location on the suction side blade tip of the blade c1 (blade 6). The contour plot of the total pressure obtained from the FRAP probe for the passage c1 (passage 6) and the reference passage (passage 20) is shown in Figure 5.36. Figure 5.37 shows the corresponding wake plot of the passage c1 and the reference passage at 92%, 60% and 25% of the blade-span.
Figure 5.34: Total pressure contour plot of 29 passages in the Experiment C.

Figure 5.35: Wake plot of all the passages at 92%, 60% and 25% of blade-span.
Figure 5.36: Total pressure contour plot of the passage c1 and reference passage.

Figure 5.37: Wake plot of the passage c1 (passage 6) and reference passage (passage 20) at 92%, 60% and 25% of blade-span in Experiment C.
Figure 5.38: Span-wise distribution of the Passage averaged Delta $C_p$ plot for passage c1 and reference passage in Experiment C.
The span-wise distribution of the passage averaged delta $C_p$ of the passage $c1$ and the reference passage is given in Figure 5.38. The maximum shift in the delta $C_p$ for blade $c1$ is at 80% of the blade-span as shown by green line in Figure 5.38. There is a significant aerodynamic loss reduction in the region 93% to 75% of the blade-span. The TLIs have negative effect in the region 97% to 95% of the blade-span and increases the leakage loss in this region. The length of the green line remains relatively same from the region 92% to 94% of the blade-span. The delta $C_p$ curve of passage $c1$ in Figure 5.38 is nonlinear from 85% to 75% of the blade-span. The length of the green line also varies non-linearly in this region, but the aerodynamic loss in this region is significantly reduced by the TLIs. From 74% to 70% of the blade-span the influence of TLI decreases. The delta $C_p$ curves for both the blade $c1$ and the reference blade is linear from 70% to 62% of the blade-span and overlap each other. The FOM of the blade $c1$ is 5.5 as the FOM is positive the net effect of have TLIs in the blade tip is positive and results in reduced leakage loss. This is due to the induced vortex generated by TLIs which counters and reduces the leakage flow (as shown in Figure 5.32).

### 5.3.2 Characteristics of Blade $c2$

Five reverse TLIs (as shown in Figure 3.10) are attached on the blade tip of the blade $c2$ (blade 12). Due to the orientation of the reverse TLIs on the blade tip, the vortex generated by them would have counter clockwise rotation. The total pressure contour plot of the passage $c2$ (passage 12) and reference passage (passage 20) is shown in Figure 5.39.
Figure 5.39: Total pressure contour plot of the passage c2 and reference passage.

Figure 5.40: Wake plot of the passage c2 (passage 12) and reference passage (passage 20) at 92%, 60% and 25% of blade-span in Experiment C.
The patches of blue near the 92% of the blade-span indicates the leakage vortex present in the passage c2 and reference passage. In Figure 5.39 the leakage vortex in passage c2 is bigger than the reference passage, indicating more loss in the tip region of the passage c2. The wake plot at 92%, 60% and 25% of the blade-span for the passage c2 and reference passage is shown in Figure 5.40.

Figure 5.41 shows the span-wise distribution of the pressure averaged delta $C_p$ of the passage c2 and the reference passage. The FOM of the blade c2 is -4.0 and hence the blade c2 have more aerodynamic loss than the reference blade. This is clearly visible in Figure 5.41 from the leftward shift of the delta $C_p$ curve of blade c2 from 85% to 70% of the blade-span. The most loss part of the blade is around 76% of the blade-span where the length of the orange line is maximum. The blade c2 have better performance near the blade tip (97% to 95% of the blade-span) where the blade c1 performs poorly. The delta $C_p$ curves of passage c2 and reference passage are linear and lie on top of each other from 92% to 86% of the blade-span. From 86% to 80% of the blade-span the delta $C_p$ curve are linear and the length of orange line is relatively small. But from 81% to 70% of the blade-span the length of the orange line is large and remains relatively same.

This is the region where the reverse TLIs have deteriorating effect on the performance of the blade c2. The overall effect of the blade c2 is that it results in increased aerodynamic loss. The poor performance of blade c2 is mainly due to the effect of vortex generated by the reverse TLIs attached on the blade tip. Since they are mounted at an opposite angle to TLIs, the vortex induced by the reverse TLIs are counter clockwise as shown in Figure 5.42. As both the TLI induced vortices and the leakage vortices are both
rotating counter clockwise, the TLI induced vortex aides the leakage vortex in creating losses in the turbine flow field.

### 5.3.3 Comparison of the Two TLI Modified Blades

The mounting orientation of the TLI onto the blade tip surface significantly affects the performance of the blades. From Figure 5.38 and 5.41 it can be seen that the TLIs on the blade tip improve the performance of the blade while the reverse TLI on the blade tip degrades the performance of the blade. Figure 5.43 shows the FOM of the two different blade tip configuration, where the passage c1 has a positive number (5.5) while the passage c2 has a negative number (-4.0), indicating that the passage c1 has less losses in the turbine flow field than the passage c2. This can also be seen from the Figure 5.44 which compares the total pressure contour plot of the passage c1 and passage c2. The leakage vortex generated by the blade c1 in passage c1 is smaller than the leakage vortex generated by blade c2 in passage c2. This is mainly due to change in the orientation of the vortex induced by the reverse TLI which aids the leakage vortex as oppose to the vortex induced by TLI which opposes the leakage vortex.
Figure 5.41: Span-wise distribution of the Passage averaged Delta $C_p$ plot for passage c2 and reference passage in Experiment C.
Figure 5.42: Vortex induced by TLI and reverse TLI in blade c1 and c2 respectively.
Figure 5.43: Variation of FOM with the number of the TLIs attached in blade tip.

Figure 5.44: Total pressure contour plot of the passage c1 and passage c2.
Chapter 6

Conclusions

Conventional vortex generators designed and implemented in the form of Tip Leakage Interrupters (TLI) can be instrumental to reduce aerodynamic losses originating from conventional tip leakage vortices in axial turbines.

The aerodynamic loss mitigation performance of the blades with newly introduced TLI blade tips depends on the mounting location, number and orientation of the TLIs mounted on the suction side of blade tips. The mounting location of the TLI in the blade tip determines the effectiveness of the TLI in reducing the leakage flows. In a comparative study of the three TLI mounting locations, it is determined that the TLI mounted at 40% of the axial chord (blade a1) provides the best tip leakage mitigation; the TLI mounted at 65% of the axial chord (blade a2) provides moderate tip leakage mitigation, and the TLI mounted at 75% of axial chord (blade a3) worsens tip leakage loss. Among the three configurations, a negative FOM is obtained for blade a3, whereas FOM is highest for blade a1 (36% higher than blade a2). Hence, the configuration of blade a1 reduces tip leakage loss most effectively, when a single TLI is considered on the suction side region very close to the tip platform. The effectiveness of the TLI improves when the “only” TLI is located very near the minimum pressure point (40% axial chord) of the current tip airfoil. The effectiveness monotonically deteriorates when the TLI is moved away from the minimum pressure point towards the trailing edge.

The performance of the blade is directly influenced by the number of TLIs simultaneously mounted on the blade tip. Each TLI mounted on the blade tip induces a
vortical structure in the flow field to counteract the leakage vortex. Hence mounting a greater number of TLIs on the blade tip before 75% of the axial chord would improve the performance of the blade. Mounting five TLIs on the blade tip (blade b3) mitigates the leakage vortex more effectively than mounting three TLIs (blade b2) or mounting one TLI on the blade tip (blade b1). While calculating the FOM of the three configurations a linear correlation is obtained between the FOM of the blade and the number of the TLIs mounted on it with blade b3 giving the best result. The corrective effect of TLIs are accumulative up to five combined TLIs. However caution needs to be exercised on deciding how many TLIs can be used without starting to induce higher aerodynamic losses.

The mounting angle of TLI on the blade tip plays a significant role in the attenuation of the tip leakage vortex. Among the two different configurations (TLI in blade c1 and reverse TLI in blade c2) the reverse TLI have adverse effect over the leakage loss in the turbine flow field as it creates a vortical system in aiding the leakage flows. This augments the strength of the leakage vortex and in-turn the leakage losses in the form of local total pressure degradation in the turbine tip flow field. TLI (blade c1) on the other hand creates a vortical system counteracting the leakage vortex. Hence mounting TLI on the blade tip reduces the strength of the leakage vortex and this in-turn reduces the leakage losses in the turbine flow field.

6.1 Future Work

The Tip Leakage Interrupter (TLI) is a new blade tip design concept to mitigate the effect of tip leakage vortex. This experimental study is a proof of concept for the TLI blade
tip design, and from the results obtained it can be seen that the TLI reduces the effect of the leakage vortex. Hence the next step in this study is to numerically analyze the turbine flow field with TLI mounted on the turbine blade tips. CFD study in a linear cascade domain would give a better understanding of the interaction between the TLI induced vortex and the tip leakage vortex. A similar CFD study will be done in a rotating turbine rotor to include the rotational effect in the tip leakage flows and determine the total-to-total efficiency changes in full 3D turbine stage. With better understanding of the flow physics involved with the interaction of vortical structures, further optimization study will be done on the existing TLI design. Based on the results obtained advanced TLI geometries will be designed for less leakage losses in the system.
REFERENCES


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Appendix A

Uncertainty Analysis

Uncertainty estimates of the measured and derived parameters are shown in this section. Every measurement system has an error associated with it. The measurement error associated with the sensors used in this experiment is provided in the table A-1. All of the measurements were taken in the stationary frame of reference. The steady state pressure measurements in the experiment were taken by Validyne type variable reluctance transducers and the dynamic pressure measurements was taken by the Endevco unsteady dynamic pressure transducer based total pressure FRAP probe. K-type thermocouples were used to measure the temperature reading in the test facility. The optical encoder attached to the shaft of the turbine stage gives the RPM of the rotor. Kline and McClintock [56] method of uncertainty estimation is used to calculate the measurement uncertainty in this experiment. A sample calculation of the uncertainty estimate is shown in this section. The calculation of coefficient of pressure, (obtained from the total pressure value from the endevco probe) is shown in equation A-1. The uncertainty associated with this derived parameter is calculated by equation A-2.

\[ C_p = \frac{\Delta P_0}{\frac{1}{2} \rho U_m^2} \]

--- Equation A-1

where \( \Delta P_0 \) is the total pressure reading obtained from the endevco probe
Table A.1: Measurement Uncertainty Values of the experimental parameters

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<th>Parameter</th>
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Table A.2: Measurement Uncertainty Values of the derived parameters

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\[ \frac{\delta_{CP}}{C_p} = \sqrt{\left(\frac{\delta_{\Delta P_0}}{\Delta P_0}\right)^2 + \left(\frac{\delta_{\rho}}{\rho}\right)^2 + \left(\frac{2\delta_{U_m}}{U_m}\right)^2} \]

--- Equation A-2

where \( \delta \) is the measurement uncertainty of the parameter in the square brackets.

The uncertainty estimates of the derived parameter used in this experiment is shown in table A-2. The details of the uncertainty values of the air density (\( \rho \)), blade speed at mid span (\( U_m \)) are also calculated with similar method from Kline and McClintock [53] but are not shown here.

**Appendix B**

**Rotor Blade Tip Profile**

Figure B.1 and Table B.1 shows the rotor blade tip profile used in the TLI study.

![Figure B.1. Turbine rotor blade tip profile](image)
Table B.1: The Coordinates of the Rotor Blade Tip Profile

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