EFFECT OF UPSTREAM UNSTEADY FLOW CONDITIONS ON ROTOR TIP LEAKAGE FLOW

by

BORISLAV TODOROV SIRAKOV

B.Sc. Aeronautical Engineering
United States Air Force Academy, Colorado Springs, Colorado, 1999

Submitted to the Department of Aeronautics and Astronautics
in partial fulfillment of the requirements for the degree of

Master Of Science In Aeronautics and Astronautics

at the

Massachusetts Institute Of Technology

September 2001

Author: ____________________
Borislav T. Sirakov
Department of Aeronautics and Astronautics
August 23, 2001

Certified by: ____________________
Choon S. Tan, Ph.D.
Senior Research Engineer
Thesis Supervisor

Accepted by: ____________________
Professor Wallace E. VanderVelde
Professor of Aeronautics and Astronautics
Chair, Committee on Graduate Students

Effect of Upstream Unsteady Flow Conditions on Rotor Tip Leakage Flow

By

Borislav T. Sirakov

Submitted to the Department of Aeronautics and Astronautics
on Aug 20, 2001 in Partial Fulfillment of the
Requirement for the Degree of
Master of Science

Abstract

A study has been conducted, using unsteady three-dimensional Reynolds-averaged Navier-Stokes simulations to determine the impact on rotor performance of the interaction between the upstream stator wakes and rotor tip clearance flow. The key effects of this interaction are: (1) a decrease in loss and blockage associated with tip clearance flow; and (2) an increase in passage static pressure rise. Performance benefit is seen in the whole operability range of interest, from near design to high loading. The benefit is modest near design and increases with loading. The highest calculated benefit is a 5 % increase in the passage static pressure rise coefficient, a 27 % decrease in tip region blockage and, a 40 % decrease in tip region loss coefficient in the time-average unsteady case relative to the steady case. These significant beneficial changes occur when the phenomenon of tip clearance flow double-leakage is present. Double-leakage occurs when the tip clearance flow passes through the tip gap of the neighboring blade. Double-leakage typically takes place at high loading but can be present at design condition, as well. A benefit due to unsteady interaction is also observed in the operability range of the rotor and is estimated to be a 2.8 % decrease in the corrected mass flow coefficient from that of the steady flow situation.

A new generic causal mechanism is proposed to explain the observed changes in performance. It identifies the interaction between the tip clearance flow and the pressure pulses, induced on the rotor blade pressure surface by the upstream wakes, as the cause for the observed effects. The direct effect of the interaction is a decrease in the time-average double-leakage flow through the tip clearance gap. The reduction of double-leakage flow means that a smaller amount of low relative stagnation pressure fluid exits the tip clearance gap. Thus, the stream-wise defect of the exiting tip flow is lower with respect to the main flow. A lower defect leads to a decrease in loss and blockage generation and hence an enhanced performance compared to that in the steady situation. The performance benefits increase monotonically with loading and scale linearly with upstream wake velocity defect.

Thesis Supervisor: Choon S. Tan, Ph.D.
Title: Senior Research Engineer, Gas Turbine Lab,
Massachusetts Institute of Technology

ACKNOWLEDGEMENTS

I would like to express my deep gratitude to the people who made it possible for me to attend MIT and accomplish this research degree. First, I would like to thank my thesis supervisor, Dr. Choon Tan for his technical advice and guidance, support, and many useful discussions on both technical and every-day issues.

I owe special thanks to my Air Force Academy mentor, Dr. Bill Heiser, and to Dr. David Quick, Manager at Rolls Royce, for their continuous help and encouragement. I am proud to acknowledge the honor and generous financial support that I received from Rolls Royce and MIT through the Sir Frank Whittle Graduate Fellowship.

My research was also made possible through the financial support from Mitsubishi Heavy Industries. The help and support received from Dr. Eng. Sunao Aoki, General Manager at MHI, and Mr. Hikaro Tashiro, Manager at MHI, is gratefully acknowledged. MHI also provided geometry and data from their low-speed research compressor for which I am indebted.

This research was also funded by the Advanced Gas Turbine System Research (AGTSR), subcontract # 96-01-SR-045, DOE Morgantown, contract # DE-FC21-92MC24061. This support is gratefully acknowledged.

I would like to thank Prof. Edward Greitzer, Prof. Frank Marble, and Prof. Bill Dawes for meeting with me and providing me with very useful advises. I thank Prof. John Denton for allowing me to use his CFD code and for the helpful discussions. I owe thanks to Dr. Choon Tan, Prof. Edward Greitzer, Prof. Alan Epstein, Prof. Ian Waitz, and Prof. Jack Kerrebrock for teaching me so many things about turbomachinery and engineering.

I would also like to thank the Aero faculty at the Air Force Academy, especially Gen. Smith, Col. Haven, Col. Chen, Col. Pluntze, and Dr. Bertin for their great teaching, help, and encouragement.

I am very grateful to my friends at Rolls Royce, Dr. Hernando Munevar, Dr. Bob Delaney, Dr. Ed Hall, Dr. Steve Gegg, Dr. Kurt Webber, Dr. Steve Wellborn, Dr. Shyam Neerarambam and Mr. David Hinko, for their help and advise.

I thank the AFOSR for allowing me to use the computational resources at GTL acquired with their support.

I also would like to thank Lori Martinez, Holly Anderson, Paul Warren, Marie Stupard, and Diana Park for their assistance and help in GTL and the department.

Of course, this research would have been much harder without the every-day help from my GTL friends. Especially, I would like to thank Duc, Taek, and Yifang for providing research guidance on daily basis, and for being my best friends here, in GTL. I thank Tony, Patrick, Jeremy, Hyung-Soo, Dongwon, Sumita, Steve, Geoff, Luis, Jinwoo, Jin-Wook, Keith, Brenda, Niall, Aurelie and all GTL students for their friendship and help.

Zdrasti ! – to my best friends in Bulgaria – Javor, Plamen, and Rossen who always remind me that there are other dimensions to life besides school, labs, and textbooks, and always cheer me up.

I would like to express my gratitude to my girlfriend, Anastassia, for being my loving and caring best friend. I also need to thank her for the understanding when I had to spend many nights and weekends in the lab.

Finally, I would like to acknowledge my parents, Todor and Jivka, and my sister, Diana. Their unconditional love and faith in me have always been giving me strength and encouragement.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

SYMBOLS

Ab	Blocked Area
A_p	Total Pressure peak defect
Atot	Total Area
A_v	Velocity peak defect
b	Distance between tip vortex and image across the casing
B	Blockage
C_p	Static Pressure Rise Coefficient
e	Energy
_e	Referring to rotor trailing edge exit plane
_in	Referring to rotor domain inlet plane
m dot	Mass flow
p	Pressure
Pt	Total Pressure
R_c	Radius of vortex core
S	Entropy
t	Time
T	Time Period
Tip	Rotor Tip Region
T_p	Total Pressure 99 % defect thickness
Tt	Total Temperature
T_v	Velocity 99 % defect thickness

u	Axial velocity
Utip	Rotor Tip Velocity
v	Circumferential velocity
w	Radial velocity
Γ	Circulation
η	Efficiency
θ	Inclination angle of vortex motion plane
ρ	Density
ω	Total Pressure Loss Coefficient

ACRONYMS

CV	Control Volume
CFD	Computational Fluid Dynamics
LE	Blade Leading Edge
TE	Blade Trailing Edge
PS	Pressure Surface
SS	Suction Surface
STD	Steady
UNS	Unsteady
IDH	Inlet Dynamic Head

CHAPTER 1 INTRODUCTION

1.1 BACKGROUND

The economic stimulus for the development of affordable and efficient gas turbine engines with acceptable reliability and durability continues to pose challenges to the researchers in the turbomachinery field. There have been continuous efforts for the development of compressors with less number of stages, with more aggressive blade design, and with higher efficiency and pressure rise. Traditionally, compressor design is based on boundary-layer/streamline curvature methods with extensive use of correlations to account for viscous and three-dimensional effects [1]. In recent years, Computational Fluid Dynamics (CFD) is providing designers and researchers with means to investigate the compressor flow field in detail. It is the richness of the axial compressor flow field, which allowed the turbomachinery research to continue for so many years and bring the discovery of so many fluid mechanical mechanisms. However, there are still fluid mechanics phenomena that are not well understood. The effects of tip clearance flow in a multi-blade environment on performance, for example, have not yet been delineated on a quantitative basis.

The overall goal of this thesis is to quantify the effects of tip clearance flow, specifically those associated with its interaction with upstream stator wakes (upstream unsteadiness), on performance.

Tip clearance is the space between the tip of the rotor blades and the engine casing for unshrouded rotors. The pressure difference across the rotor blades drives fluid from the pressure side towards the suction side of the blade, through the tip clearance gap (Figure 1.1). This phenomenon is known as tip clearance flow. Tip clearance flow has profound effects on compressor performance and stability (Wisler [2], Smith [3], Koch [4], among others). Tip clearance flow generates blockage and loss within the blade passage, which reduces the pressure rise capability of the rotor, as well as the component efficiency.

Unsteadiness, on the other hand, is inherent to turbomachines. There is relative motion between the rotor and stator blades and any flow non-uniformities created in a blade row will be rotating with respect to the following or preceding blade row. Thus, the stator generated wakes appear as normal jets directed away from the following rotor blade and convect downstream in the rotor relative frame introducing unsteadiness in the rotor flow field (Figure 1.2). The pressure field of a downstream stator causes pressure fluctuations at the trailing edge of an upstream rotor. Shock waves in transonic machines affect the neighboring blade rows in an unsteady manner. The above examples show that unsteadiness can manifest itself through vortical (e.g. wakes, tip vortices), potential (e.g. blade pressure field), and shock interactions.

As alluded to in the above, this thesis investigates the interaction between stator wakes and downstream rotor tip clearance flow and its relative importance to other fluid mechanical events in axial compressors in affecting the time average performance of the rotor.

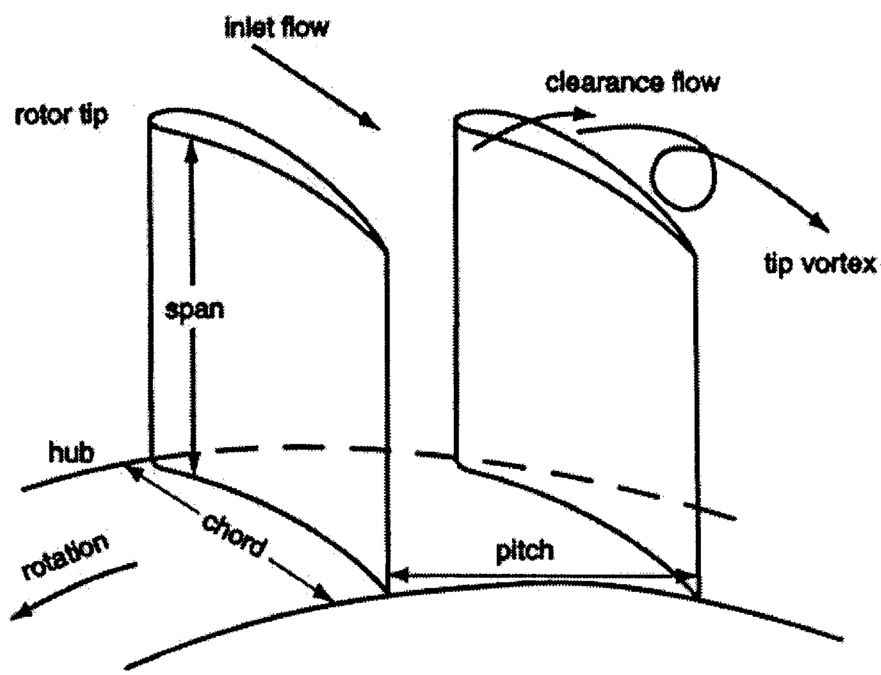


Figure 1.1 Schematic Representation of Tip Clearance Flow in a Compressor Rotor [5].

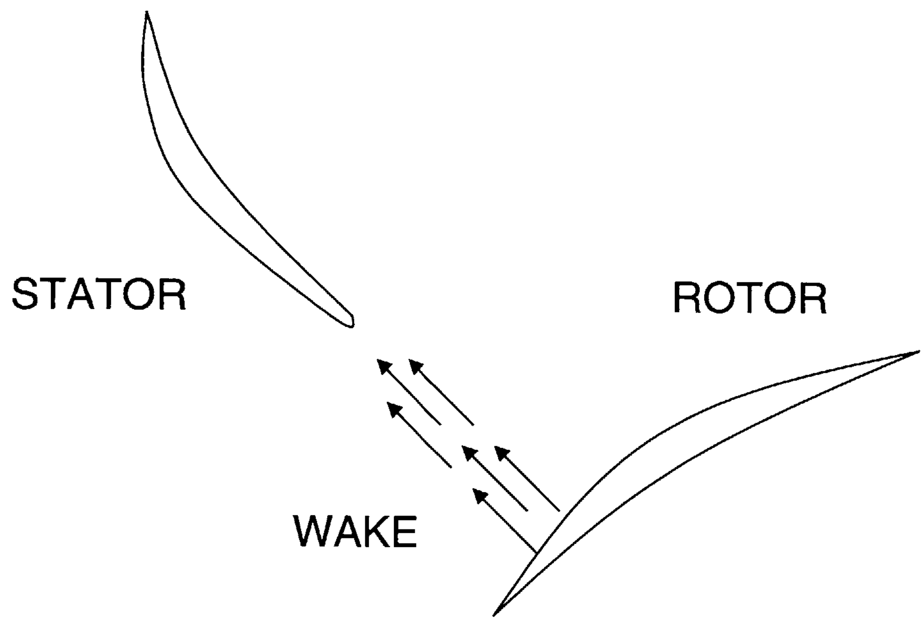


Figure 1.2 Stator Wake Appears as a Normal Jet Directed Away from the Rotor Suction Side in the Rotor Relative Frame.

1.2 PREVIOUS WORK

Previous work pertinent to tip clearance flow and flow unsteadiness is briefly reviewed in the following to provide the context for the present work: The structure and physics of tip clearance flow have been investigated by Rains [6], Hunter and Cumpsty [7], Chen et al. [8], Storer and Cumpsty [9], Khalid [10], and Nikolaoue et al. [11] among others. Rains [6] proposed an inviscid model for the leakage flow velocity by relating the tip leakage flow to the blade static pressure difference. He assumed that the kinetic energy of the leakage flow velocity component normal to the chord cannot be recovered and he calculated the loss in efficiency, based on this assumption.

Storer and Cumpsty [9] developed a simple control volume analysis model for calculation of tip clearance flow loss using ideas from Rains' model. They view the mixing process of the leakage jet and the main stream as the major loss mechanism.

Khalid [10] provided a rational methodology for quantifying the flow blockage associated with tip clearance flow. He also developed a simple model using a description of the growth of a two-dimensional wake in an adverse pressure gradient, which provides insights to the important processes associated with blockage growth.

Khalsa [12] reported the phenomenon of double-leakage (the passage of tip clearance fluid through the neighboring tip clearance gap) and the invalidity of one of the assumptions in Storer's loss prediction model in such situations.

Smith [13] experimentally investigated the effect of changing the axial spacing between blade rows on compressor performance changes. He obtained a one-point efficiency gain and a two to four percent stage pressure rise increase in a low-speed research compressor, by reducing blade row spacing from 0.37 to 0.07 chords. Mikolajczak [14] reported similar results. Hetherington and Moritz [15], however, achieved a 2-point efficiency gain by increasing the spacing between the blade rows in a multistage compressor. Such findings suggest the existence of more than one mechanisms affecting compressor performance. It is believed that unsteady mechanisms due to the relative motion of the blade rows are responsible for the observed changes.

Valkov [1] investigated the effect of upstream wakes and tip clearance vortices on stator time-average performance. He identified two generic causal mechanisms with significant impact on performance. These are reversible recovery of energy in the disturbances (beneficial) and non-transitional boundary layer response (detrimental). The net effect is beneficial. Valkov reports 0.2 points efficiency gain for design spacing (0.37 chords) between the stator and the rotor and 0.6 points efficiency gain for the close spacing (0.07 chords) case with stronger unsteadiness.

Graf [5] and Tzeng [16] investigated the effects of the downstream stator pressure field on the time-average performance of the upstream rotor. Graf found that the back pressure fluctuations are important for blade passage performance and tip clearance flow development. He reported 10 to 50 % decrease in tip clearance loss and 20 to 40 % increase in overall loss. Graf also observed inherent unsteadiness of the tip clearance vortex. Presently, it is believed that this may be similar to the unsteadiness observed in the tip region by Mailach [17] and Bae [18] in experiments, and by Vo [19] in his calculations. Bae hypothesized that the observed vortex unsteadiness could be associated with Crow instability observed in wing trailing vortex pairs and showed that the frequency of the instability scales with the passage flow-through time.

In a more recent study, Tzeng [16] showed that the effect of back pressure on upstream rotor time-average performance may be negligible. He deduced this as follows: he performed a rotor-stator calculation and then compared it to another calculation where distributed body forces represent the stator in the stage. The use of distributed body force stator representation provides a mean to calculate rotor-stator flow on a steady basis. The effects of unsteadiness are extracted by comparing the rotor-stator calculation to the rotor – stator body force calculation. Body forces are adjusted until time averaged radial profiles of mass and entropy flux, pressure, etc. are matched between the two calculations in a plane between the rotor and stator. The results of Tzeng's work, appear to show that unsteadiness from downstream may have negligible effect on rotor performance. Thus, it is suggested that upstream unsteadiness could be responsible for a portion of the experimentally observed performance changes that could not be solely accounted for based on reversible recovery of energy in the wakes and tip vortices [1, 13].

1.3 TECHNICAL OBJECTIVES

The goal of this effort is to investigate the effect of upstream unsteadiness on rotor tip clearance flow in terms of performance. The specific research objectives are as follows:

- Quantify response of rotor tip leakage flow to upstream unsteady flow conditions set by stator in terms of a change in time-average performance.

- Understand the fluid mechanics phenomena responsible for the change in performance, if any.

- Provide design guidelines

To address the above objectives the following research questions are posed:

1. What constitutes an adequate comparison between steady and unsteady flow?
2. How can the effect of unsteadiness on tip leakage flow be isolated from the effect of unsteadiness on blade boundary layer behavior [1]?
3. What numerical experiments are required to clearly identify the unsteady aspects of tip leakage flow that have a time-average influence on rotor performance?
4. What is the effect of unsteadiness on tip leakage passage loss, blockage, and overall pressure rise?
5. What is the sequence of fluid mechanical events leading to the observed effects? What is the cause and effect relation?
6. What improvements can be made to the design of compressors based on the new findings?
7. What aspects of this research should be further investigated?

1.4 APPROACH AND NUMERICAL TOOLS

1.4.1 Approach

The approach of this research is to implement a set of numerical experiments for identifying the cause-and-effect in the interaction between upstream wakes and rotor tip clearance flow. Time-accurate, Reynolds-averaged, Navier-Stokes simulations are carried out to obtain unsteady solutions at different operating conditions. Averaged boundary conditions are extracted from the unsteady calculations and steady solutions are obtained at the same mass flow, with the same boundary conditions, for a consistent back-to-back comparison. Iterations on the back pressure boundary condition are required to match mass flow between unsteady and steady solution at each operating condition. Detailed description of a consistent back-to-back method for assessing time-average effects of unsteady flow against an equivalent steady flow is presented in Chapter 2. Four important cases are selected for investigation to delineate the parametric dependence of the effect of unsteadiness on blade row axial spacing and rotor blade loading. The four cases are strong wakes (50% velocity defect in stator frame, corresponding to reduced spacing between the blade rows) at high loading and design conditions and typical wakes (27% velocity defect in stator frame, corresponding to nominal spacing between the blade rows) at high loading and design conditions. Shear stresses are turned-off on the casing wall, hub wall, and blade surfaces to exclude the effects of wake-blade boundary layer interaction [1]. Instead of carrying-out full stator-rotor calculations a stator wake is prescribed upstream of the rotor. The goal of this study is to assess the effect of the stator wakes on rotor tip clearance flow and therefore a full interaction is deemed unnecessary. As mentioned before, the unsteady interactions can be of vortical, potential, and shock wave nature. For low-speed axial compressors shocks are not present, the potential field of the stator will not have appreciable effect on the rotor (most of the pressure field redistribution due to the stator presence appears close to the stator leading edge, away from the rotor) and the only interaction of interest is the one due to the upstream wakes or vortical disturbances. Therefore, the substitution of the stator row with stator wakes is an approximation that will capture the important aspects of the interaction between a real stator and rotor in terms of changes in rotor tip region time-average performance.

1.4.2 Numerical Tools

The Computational Fluid Dynamics code selected for this investigation is Denton's UNSTREST solver [20]. This solver has been validated and used extensively at Cambridge University, and in industry [21]. It is a 3D, unsteady, viscous code employing an explicit, second-order-accurate scheme to solve the unsteady continuity, momentum, and energy equations. The code uses a simple mixing length turbulence model. UNSTREST requires a simple H-mesh grid and is able to run both multiple blade-row and multiple blade-passage calculations.

The grid for this study consists of two computational domains. Three different perspectives of the grid are shown in Figures 1.3, 1.4 and 1.5 to give a general idea of the computational domain and the locations selected for performance calculations. The first domain is a stationary duct (36 axial x 49 circumferential x 37 radial). A moving rotor domain follows it (126 axial x 49 circumferential x 37 radial). Performance is calculated from averaged values in the inlet plane of the rotor domain (Station 1) and in the blade trailing edge plane (Station 2). The exact location of the planes is designated in Figure 1.3. Radial profiles for boundary conditions are presented for the inlet plane of the rotor domain (Station 1) . The shear stress is turned-off on the casing, hub, and blade surfaces in the numerical experiments, which allows for the construction of uniform grid in the circumferential direction. The purpose of the short duct upstream of the rotor is to release wakes in the stationary frame so as to simulate the effect of an upstream stator wake. The wakes are rotating with respect to the rotor domain. The circumferentially uniform grid mitigates the numerical dispersion of the wakes traveling through the rotor domain. Denton [22] recommends a 100,000-point grid for a grid independent rotor solution. The current grid has 293,706 points to ensure grid independence of the results without performing extensive grid-independence studies. The code is written in FORTRAN and uses single precision calculations. Results are calculated from several different cycles from a converged periodic solution. When the computed pressure rise, loss, and blockage results from the different cycles are compared, they show uncertainty in the fourth significant figure. Results are reported to within this accuracy. The results are obtained with the UNSTREST solver and post-processed with MATLAB tools. There are five points in the tip clearance region, which is sufficient to capture the inviscid nature of the tip clearance flow passing through the clearance gap (Dawes [9]). Calculations by Dawes [9] show that tip clearance flow experiences negligible total pressure loss when passing through the clearance gap, except for cases when it reattaches to the blade tip. In this study the tip leakage flow does not reattach because the thickness of the blade at the tip is set to zero (Figure 1.5). At the same time the shear stress on the casing is turned-off. Therefore, the tip clearance fluid passes through the tip gap in an inviscid manner and refinement of the grid there is deemed unnecessary.

The above facts show that the numerical tools employed in the current study can capture changes in performance with sufficient accuracy in the comparison between steady and unsteady flow.

1.5 CONTRIBUTION OF THESIS

The key contributions of this thesis can be summarized as follows:

- The effect of upstream wakes on rotor tip clearance flow loss, blockage, and pressure rise has been quantified. Rotor static pressure rise coefficient increases by up to 5 % of the steady value (0.52 steady vs. 0.55 time-average unsteady), tip region loss coefficient decreases by up to 40 % of the steady value (0.174 steady vs. 0.103 time-average unsteady), and tip region blockage generation decreases by up to 27 % of the steady value (0.064 steady vs. 0.047 time-average unsteady),.

- The parametric dependence of the effect of wake-tip clearance flow interaction on rotor loading and upstream stator wake defect has been established. The beneficial effects increase monotonically with loading. The effects also scale linearly with upstream wake defect strength

- A generic fluid mechanism explaining the changes caused by the stator wake-rotor tip clearance flow interaction is proposed. It states that the pressure pulses on the pressure side of the rotor blade, caused by the stagnation of the upstream stator wake jet on the rotor blade, decrease the time-average double-leakage flow through the tip clearance gap. Thus, the relative total pressure defect of the tip clearance fluid exiting the tip gap is lower in the time-average unsteady case compared to the steady case, and consequently, the stream-wise velocity defect is lower in the time-average unsteady case leading to enhanced performance. A simple control volume mixing analysis, based on the proposed mechanism, estimates the performance to within 15 % of the calculated values.

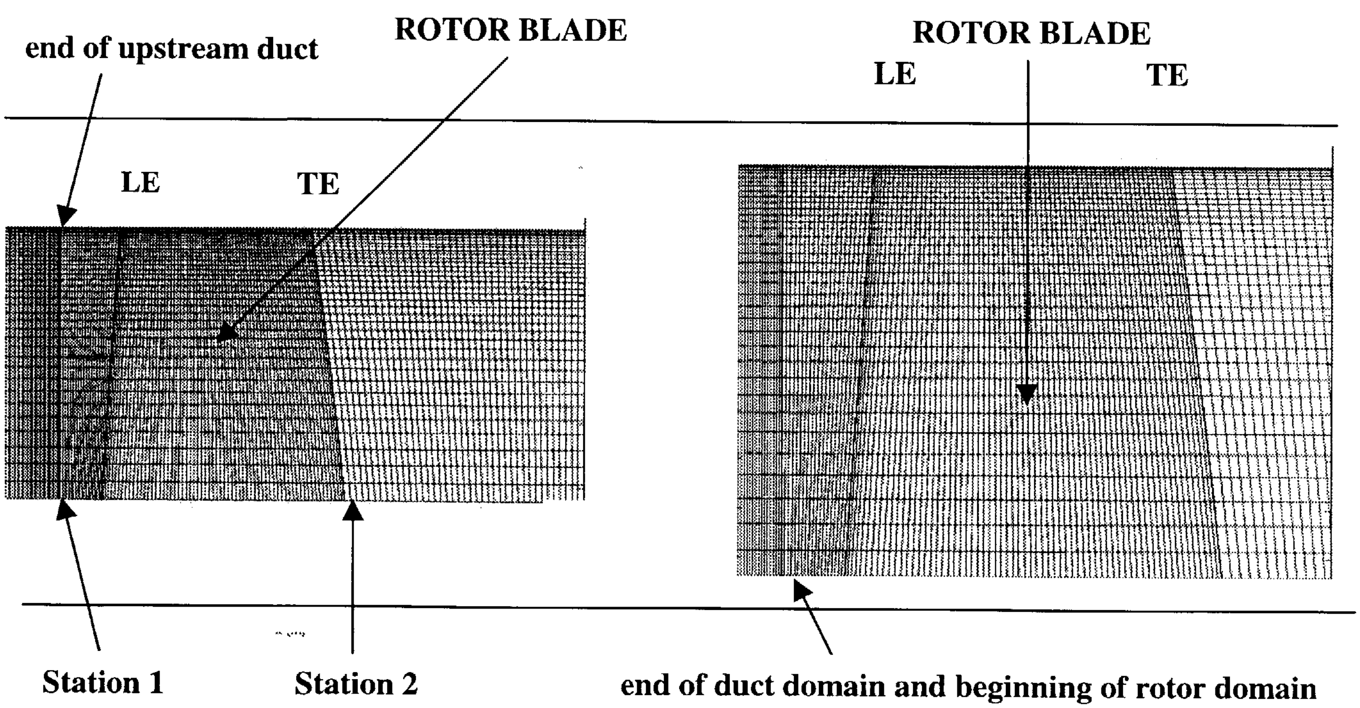


Figure 1.3 Computational Grid Plane at 50 % Pitch Showing the Rotor Blade and Locations of Planes Used for Performance and Boundary Conditions Calculations. Station 1 is Rotor Inlet Plane and Station 2 is Blade Trailing Edge Plane. Station 1 is Located 0.15 chords Upstream of the Blade LE.

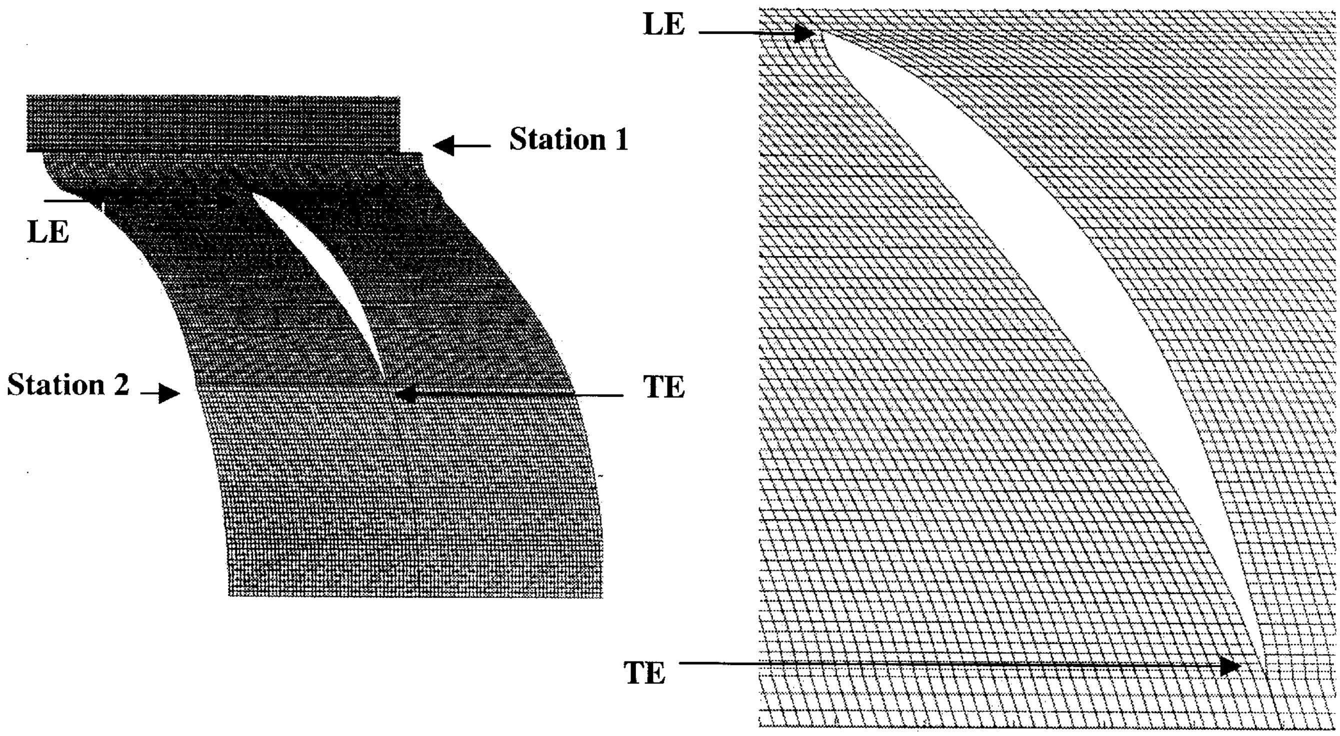


Figure 1.4 Computational Grid Plane at 25 % Span Showing the Rotor Blade and Locations of Planes Used for Performance and Boundary Conditions Calculations. Station 1 is Rotor Inlet Plane and Station 2 is Blade Trailing Edge Plane. Station 1 is Located 0.15 chords Upstream of the Blade LE.

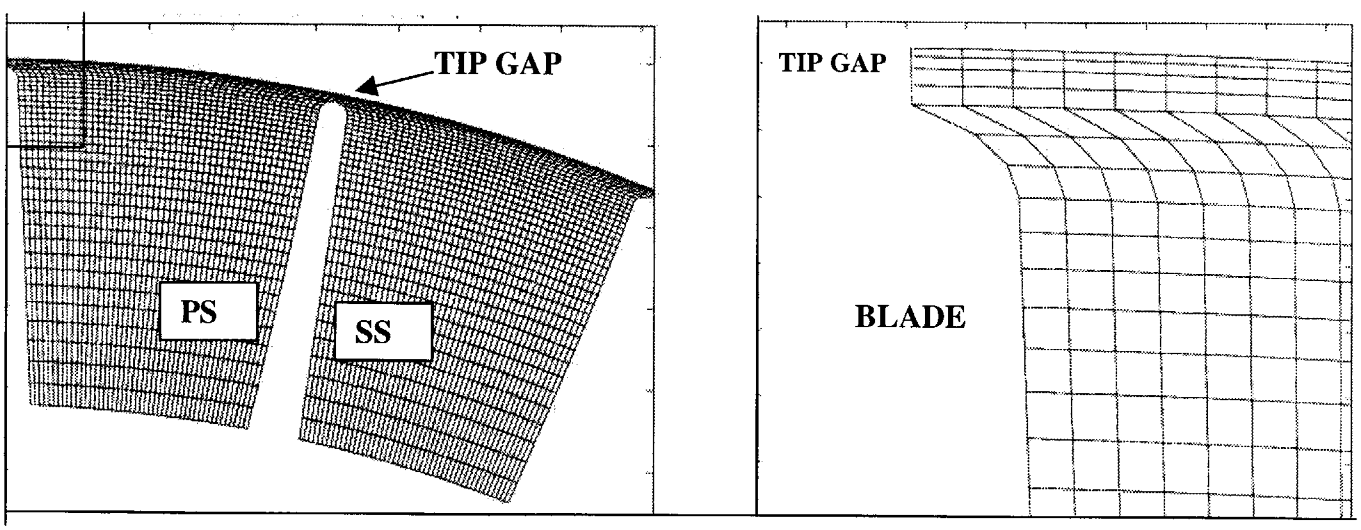


Figure 1.5 Axial Computational Grid Plane at 30 % Chord From LE Showing the Rotor Blade and the Rotor Tip Clearance Gap. Blade Thickness is Zero at the Blade Tip.

1.6 ORGANIZATION OF THESIS

This thesis is arranged as follows: Chapter 2 describes the methodology used to design steady calculations from unsteady solutions and two validation runs of the method. Chapter 3 presents the key results from this investigation. A discussion of the results and delineation of a fluid mechanical causal mechanism follows in Chapter 4. Conclusions and recommendations for future work are given in Chapter 5.

CHAPTER 2 METHODOLOGY FOR COMPARING STEADY AND UNSTEADY FLOW

2.1 DEFINITION OF UPSTREAM UNSTEADINESS AND DESCRIPTION OF UPSTREAM WAKES

Before constructing a method for comparison between unsteady and steady flows, the source of upstream unsteadiness needs to be delineated. Turbomachinery flows are inherently unsteady because of the relative motion of the blade rows. For this investigation a total pressure defect is prescribed at the inlet of the duct, upstream of the rotor, as a boundary condition. This defect simulates a wake coming from upstream stator and convects downstream through the rotor blade passage. There is relative motion between the defect and the rotor blades. At any particular point in the rotor blade passage, the flow field changes periodically with time due to this rotating upstream flow defect. These temporal changes in the flow-field will be referred to as upstream unsteadiness in the rotor blade passage.

The wake at the inlet of the duct is prescribed as a total pressure defect. The defect is constructed as a Gaussian profile because the objective here is to elucidate the key role of the relative defect and any non-symmetry measured in real wakes is ignored. The wake is characterized by a peak defect A_p and by a 99% defect thickness, T_p. However, once the wake is generated, it is accepted to describe it with a peak velocity defect A_v and a 99% velocity defect thickness, T_v. The strongest wake achieved for this study is a wake with $A_v = 50\%$. The corresponding thickness for such a wake strength should be $T_v = 0.12$ (chords) based on measurements in a low speed axial compressor (Stauter [23]) . For detailed description of wake shape, development, and typical values for strong and design conditions, see Valkov's Doctoral Thesis (Appendix G.2.) [1]. However, 0.12 chords is too small of a thickness to be supported by a realistic size grid. A very strong and thin wake dissipates and disperses very quickly before reaching the rotor. For this reason the peak defect is set to the original maximum of 50% but the thickness is increased by 2.8 times the measured value to alleviate this problem. Because the increase in thickness is the same for the strong and typical wake cases the relative changes between the two cases remain the same and the performance trend is preserved. Figure 2.1 and Table 2.1 describe the wakes chosen for the strong and nominal design interaction cases:

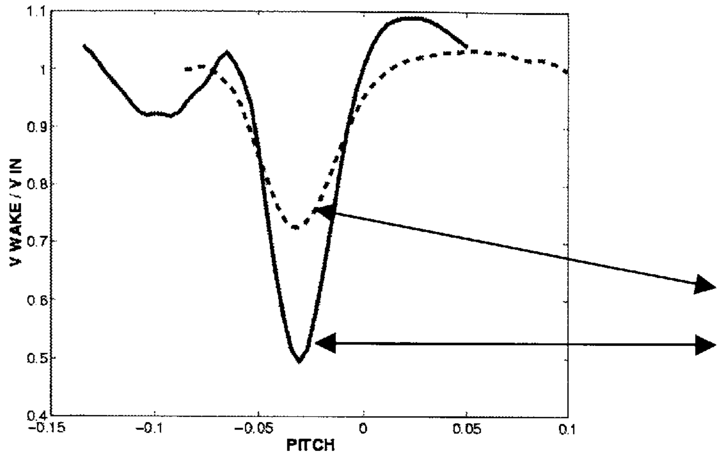


Table 2.1 Upstream Wake Description.

UPSTREAM WAKE DESCRIPTION		
	Av (%)	Tv (chords)
TYPICAL	27	0.56
STRONG	50	0.34

——— **Strong Wake, Av = 50 %** — — — **Typical Wake, Av = 27 % (rotor inlet plane)**

Figure 2.1 Strong and Typical Wakes. The Strong Wake Represents Strong Blade Row Interaction Case With 0.07 Chords Axial Spacing Between the Stator and the Rotor. The Typical Wake Represents Typical Interaction Case with 0.37 Chords Axial Spacing Between the Blade Rows.

2.2 METHOD FOR DESIGNING A STEADY CALCULATION

2.2.1 Introduction

The issue of designing equivalent steady flow from given unsteady flow field to achieve a consistent back-to-back comparison is conceptually non-trivial. One of the major difficulties is that physically, such equivalence does not exist. For example, if each variable of interest is averaged separately in the unsteady flow field, the resulting averaged steady flow field is inconsistent. The average total pressure from the unsteady solution will be different from the total pressure calculated from the averaged Mach number and averaged static pressure. Therefore, the unsteady solution should be brought to a steady state through a physically meaningful process, despite the changes in the thermodynamic state of the fluid that will inevitably occur. To obtain a consistent steady solution only five quantities can be conserved because this is the number of the primitive variables (pressure, density, and three velocities) in the flow field. Only five equations can be used to solve for five unknowns (if energy is introduced as a separate independent variable the equation of state is needed to close the system). Therefore, five averaged quantities, which will remain the same from the unsteady to the steady solution, must be selected.

A possible physical situation is to allow the flow to mix-out completely before reaching the rotor. It is a common industry practice to use mixing planes between the blade rows [1]. The flow is brought to a steady (or circumferentially uniform) state through conservation of axial impulse, angular and radial momentum, mass and energy. These are also the quantities selected for conservation in this research. Since there are six unknowns appearing in the equations to be solved, the equation of state is added as a sixth equation to close the system.

2.2.2 Methodology

Based on the arguments presented in section 2.2.1, the following method is used to design steady calculations for comparison with the unsteady solutions:

1. Obtain a time accurate solution for a given operating condition.

2. Time average in a cycle the five selected quantities at each grid point:
   - mass
   - energy
   - axial impulse
   - angular momentum
   - radial momentum

3. Construct the steady flow field at a chosen inlet plane by solving the following six equations for energy, pressure, density, and three velocities. (in addition to the five equations, the equation of state is included as well):

$$A = \overline{\rho u} = \frac{1}{T}\int_0^T \rho u dt$$

$$B = \overline{p} + \overline{\rho u^2} = \frac{1}{T}\int_0^T (p + \rho u^2) dt$$

$$C = \overline{\rho uv} = \frac{1}{T}\int_0^T \rho uv dt$$

$$D = \overline{\rho uw} = \frac{1}{T}\int_0^T \rho uw dt$$

$$E = \left(\overline{\rho e} + \overline{p} + \overline{\rho}\frac{\overline{u^2} + \overline{v^2} + \overline{w^2}}{2} \right)\overline{u} = \frac{1}{T}\int_0^T \left(\rho e + p + \rho\left(\frac{u^2 + v^2 + w^2}{2}\right)\right) u dt$$

and

$$\overline{p} = \overline{\rho e}(\gamma - 1)$$

By substituting the remaining equations into the energy equation, a quadratic equation in P can be obtained:

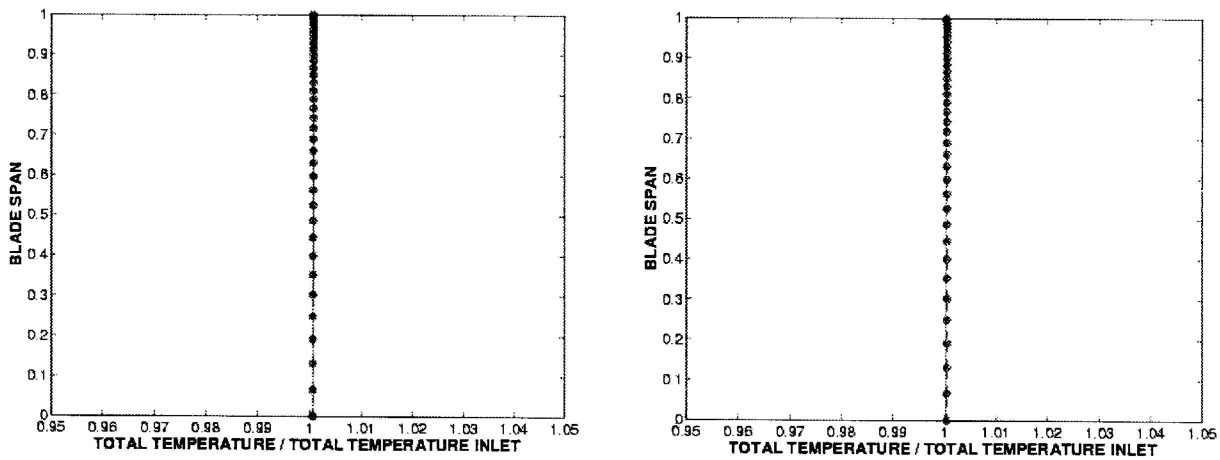
$$\left(\frac{-1}{\gamma - 1} - \frac{1}{2}\right)\overline{p}^2 + B\left(\frac{1}{\gamma - 1}\right)\overline{p} + \frac{1}{2}(B^2 + C^2 + D^2) - EA = 0$$

The root corresponding to an increase in entropy is selected.

4. Extract the inlet averaged boundary conditions (Pt, Tt, and absolute angle) required as input to the CFD code from the above averaged flow field.
5. Obtain the steady solution without upstream wakes with the extracted boundary conditions and at the same time-average mass flow. Iterations on the exit pressure boundary condition are required to match the mass flow in both cases. Obtain the solution using the same time-accurate code, same grid, and same post-processors.
6. Construct the steady flow fields at the selected inlet and exit planes in the rotor domain, as described in step 3, for both the unsteady and steady solutions. Since the steady solution is also obtained in a time accurate mode for consistency, it is subjected to the same averaging procedure.
7. Once both cases are brought to steady state, perform the appropriate spatial averages at the inlet and exit planes to obtain one-dimensional figures of merit, such as area-average static pressure and mass average total pressure.

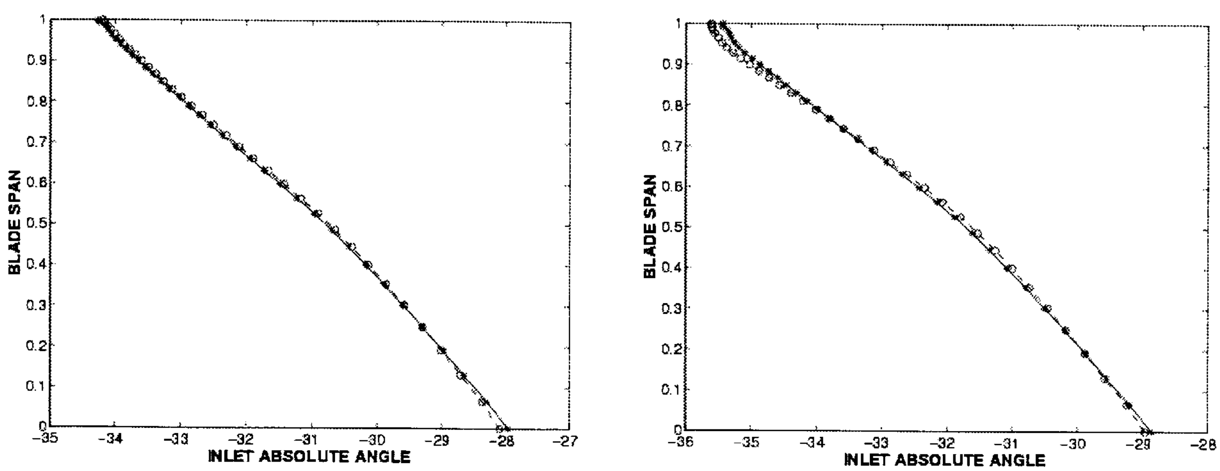
8. Compare the unsteady performance of the rotor to the steady one. The "steady" flow in this study is a flow field obtained at the same time averaged mass flow with the same mixed-out inlet boundary conditions as the unsteady one

Figures 2.2, 2.3 and 2.4 show the agreement of averaged boundary conditions between the unsteady and steady solutions for the strong wake cases. The boundary conditions are calculated at Station 1 – the inlet of the rotor domain. The results in these figures show that the time-average unsteady boundary conditions are the same as those used for the steady calculations.



design condition **high loading condition**

Figure 2.2 Inlet Absolute Total Temperature for Time-Average Unsteady Solution (stars) and for Steady Solution (circles). The Inlet Total Temperature Radial Profiles are Uniform and They are the Same for Both Cases.



design condition **high loading condition**

Figure 2.3 Inlet Absolute Angle for Time-Average Unsteady Solution (stars) and for Steady Solution (circles). The Inlet Angle Radial Profiles Agree Very Well between the Steady and The Time-Average Unsteady Solutions.

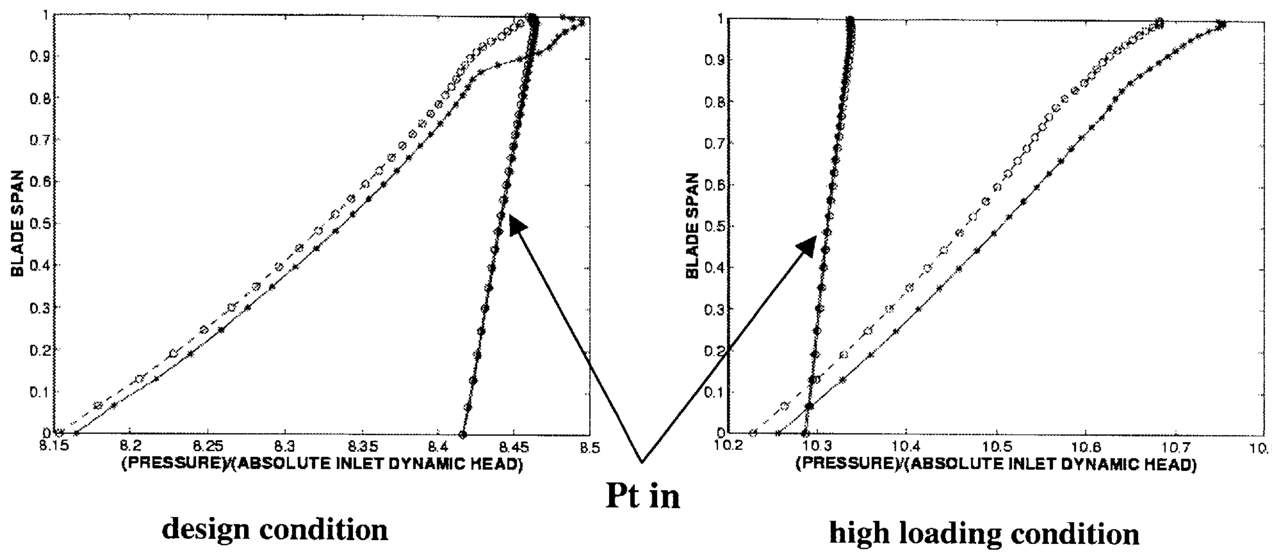


Figure 2.4 Inlet Absolute Total Pressure and Exit Static Pressure for the Time Average Unsteady Solution and the Steady Solution. The Inlet Absolute Total Pressure Shows Very Good Agreement as Required by the Comparison Procedure and the Exit Static Pressure Profiles Show the Effect of the Strong Interaction (unsteadiness) that Results in Higher Pressure Rise.

LEGEND: ★ -time average unsteady o – steady

2.3 METHOD ASSESSMENT AND VALIDATION

A two-dimensional case was selected to assess the comparison method between the time-average unsteady and the steady solutions. The mid-span section of the rotor blade was solved for a strong upstream wake case. The shear stresses were turned-off at the "tip" and "hub" walls of the computational domain, as well as, on the blade surfaces. There are no boundary layers, nor tip clearance flow in the solution. Therefore, there must not be significant changes between the unsteady and steady cases. The only changes observed in the comparison were increase in loss due to the diffusion of the strong wake in the rotor blade passage and the corresponding pressure rise loss with respect to the steady case. The important aspect of this comparison is that the observed changes due to the unsteadiness are all detrimental. This fact will be used later to show that the observed changes in the actual three-dimensional study and the reported interaction effects are not caused by the comparison method and are physical.

An alternative opportunity to assess the comparison method was to run a two-dimensional case with boundary layers on the blade surfaces and quantify the effect of upstream wake velocity defect on boundary layer loss. Such a study by Valkov [24] showed that the loss increases linearly with upstream wake velocity defect. Computations were carried-out for three different wake sizes and the results are in accord with Valkov's findings (Figure 2.5), which were based on using a high order numerical scheme, namely, the spectral element method.

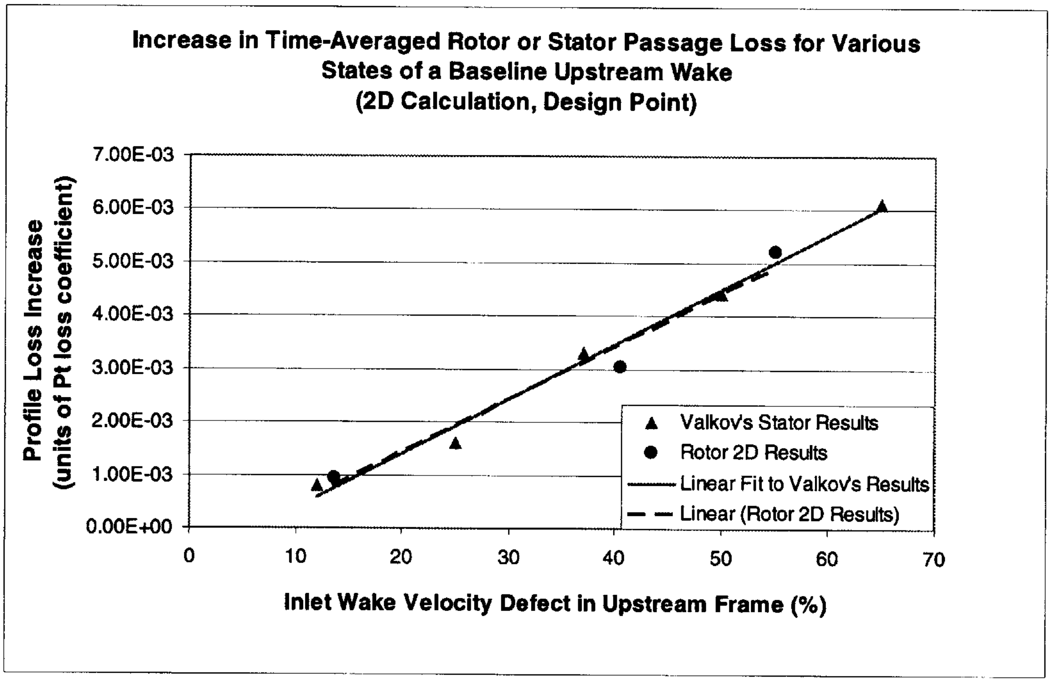


Figure 2.5 The Linear Dependence Between Loss and Wake Velocity Defect Calculated by Valkov for a 2-D E^3 Stator Section is Confirmed for a 2-D Mid-span Section of the Present Rotor.

2.4 SUMMARY

A method for achieving a consistent comparison between time-average of unsteady and steady rotor performance is defined and described. The method conserves axial impulse, angular and radial momentum, mass and energy from the unsteady to the steady flow field. This process corresponds to the physical situation of allowing the unsteady flow to mix-out completely to a steady state. Such an approximation is common in industry practice. The conclusions from such comparison will provide physical understanding of the effects of unsteadiness on rotor performance and also will be directly applicable to any industry practice employing the mixing-plane approximation. The results will give information to the designer about the corrections required to a design or performance prediction based on a mixing-plane steady approximation.

The method was applied to two two-dimensional situations. The first one involved a rotor section without shear stresses on the blade and showed detrimental changes, based on mass-average figures of merit, due to the diffusion of the strong wake within the rotor passage. The second situation involved a rotor section with boundary layers. The calculated effect of the wake-boundary layer interaction is in accord with previous results presented by Valkov [1]. The results from the two assessment runs give confidence that the method can be applied in direct comparisons between unsteady and steady flows.

CHAPTER 3 EFFECT OF UPSTREAM UNSTEADINESS ON ROTOR TIP CLEARANCE FLOW TIME-AVERAGE PERFORMANCE

3.1 INTRODUCTION

This chapter presents the main results of this study. The time-average effect of the unsteady interaction between upstream wakes and rotor tip clearance flow in terms of tip clearance flow loss and blockage, and blade passage pressure rise, is quantified. The focus is on results from four key computational experiments. To extract the effects of unsteadiness and to assess their dependence on rotor operating conditions (loading) and stage design choices (axial spacing between blade rows) the following experiments are selected:

1. Nominal Spacing Between Blade Rows Near Design Condition (Effect of Typical Wake Near Design).
2. Nominal Spacing Between Blade Rows at High Loading Condition (Effect of Typical Wake at High Loading).
3. Reduced Spacing Between Blade Rows Near Design Condition (Effect of Strong Wake Near Design).
4. Reduced Spacing Between Blade Rows at High Loading Condition (Effect of Strong Wake at High Loading).

Comparisons between these four experiments can elucidate the effects of upstream unsteadiness on rotor performance and the effect's dependence on design choices and operating conditions.

3.2 PERFORMANCE RESULTS

The overall effect of upstream unsteadiness on rotor total-to-static pressure rise can be seen in figures 3.1 and 3.2. The two speed-lines show that upstream unsteadiness has a beneficial effect on rotor performance. It can be inferred from the results that the effect on total-to static pressure rise coefficient is modest close to design but becomes significant at high-loading with a change of up to 19 % from the computed value in the steady calculation (0.17 steady vs. 0.20 time-average unsteady). Speed lines are shown for strong and typical wake cases.

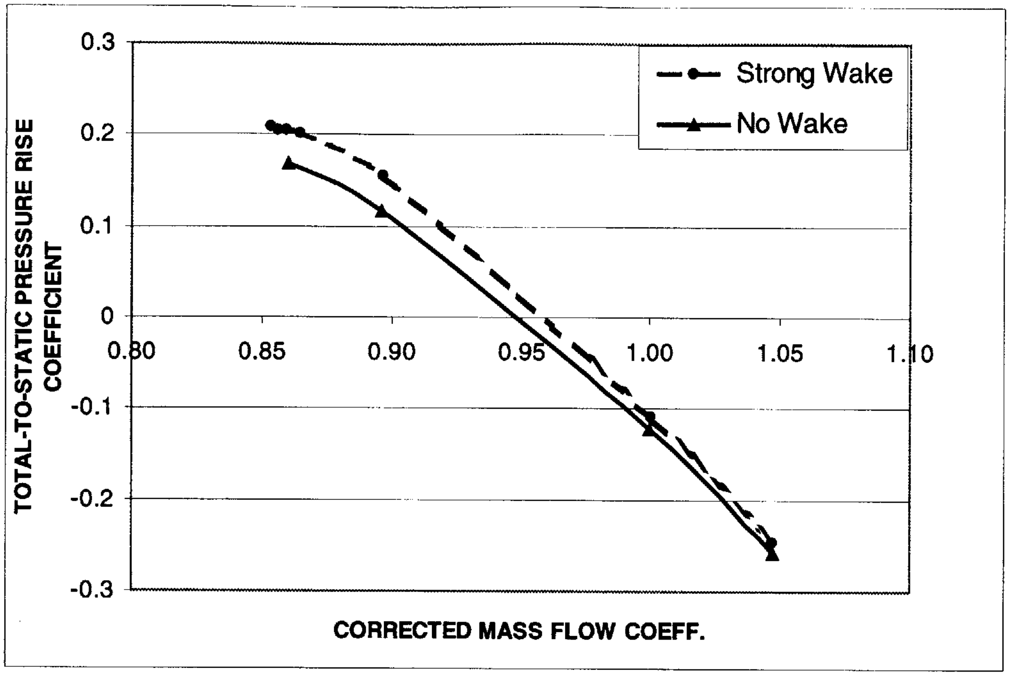


Figure 3.1 Effect of Strong Upstream Wake on Rotor Total-to-Static Pressure Rise Coefficient Showing the Benefit of Upstream Stator Wake-Rotor Tip Clearance Flow Interaction on Time-Average Performance.

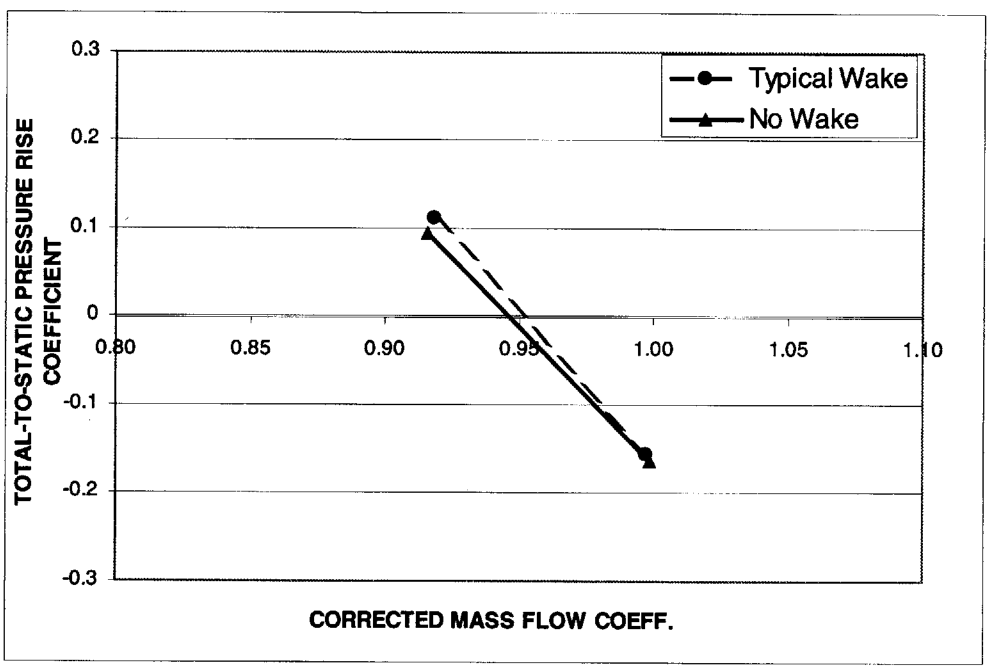


Figure 3.2 Effect of Typical Upstream Wake on Rotor Total-to-Static Pressure Rise Coefficient Showing the Benefit of Upstream Stator Wake-Rotor Tip Clearance Flow Interaction on Time-Average Performance. The Typical Wake Interaction Brings Less Beneficial Change than the Strong Wake Interaction.

Figure 3.1 also suggests that there could be some benefit in the operability range of the rotor due to the unsteady interaction. Smith [13] previously observed such a benefit in his experiments. A discussion of this issue will follow in Chapter 5. Figure 3.2 shows that the performance benefit caused by the typical wake unsteady interaction with the rotor tip clearance flow is lower than the performance benefit in the strong unsteady interaction case.

Figures 3.3 through 3.5 present the effect of upstream wakes on rotor static pressure rise, tip region loss, and blockage. Figure 3.3 shows the benefit from the unsteady interaction between upstream wakes and rotor tip clearance flow on rotor static pressure rise. Changes in static pressure rise are caused by both changes in blade passage loss and blockage. Therefore, the effect on static pressure rise can be viewed as the net effect on performance. To check the consistency of the solution, the change in static pressure rise is estimated from the changes in loss and blockage and compared to the calculated change. Influence coefficients derived by Shapiro [25] provide a convenient way to perform such a comparison. The results from the CFD calculation and the estimated results based on the method of influence coefficients are in good agreement (within 4% from the CFD results). It is also of interest to note that the beneficial effect from the interaction increases linearly with loading.

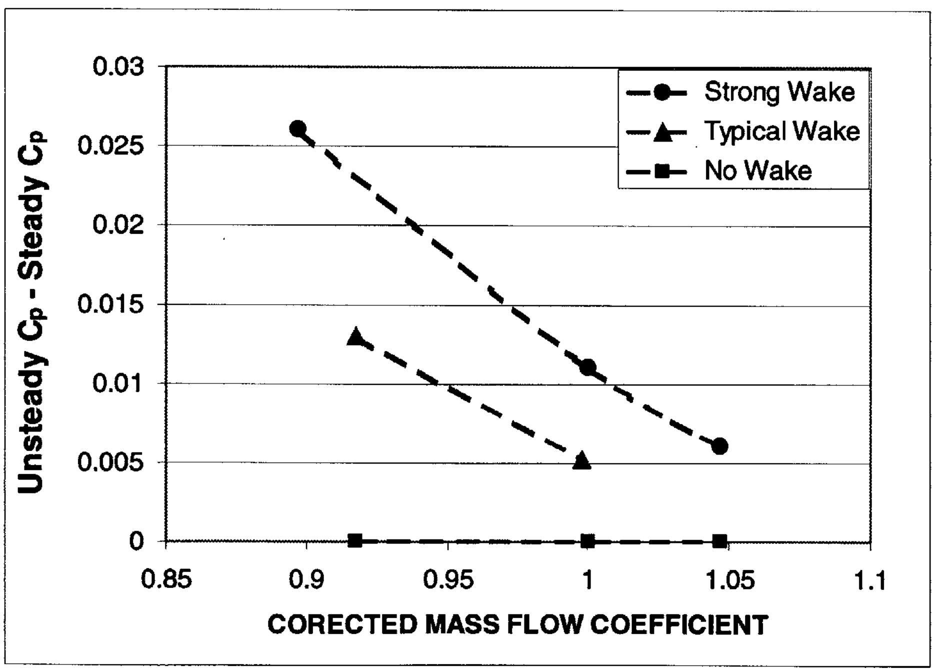


Figure 3.3 Beneficial Effect of Upstream Stator Wake – Rotor Tip Clearance Flow Interaction on Rotor Static Pressure Rise. The Benefit Increases With Loading and With Upstream Wake Velocity Defect.

Figures 3.4 and 3.5 show the effects of upstream wakes on tip region loss and blockage generation. Loss is calculated from the difference between blade inlet and exit time averaged entropy flux and blockage is calculated using Khalid's procedure [10]. For detailed description of Khalid's procedure the reader is referred to Khalid's Doctoral Thesis [10]. The benefit in blockage is up to a 27 % decrease from the computed value in the steady case (0.064 steady vs. 0.047 time-average unsteady) and the benefit in loss coefficient is up to a 40 % decrease from the computed value in the steady case (0.174 steady vs 0.103 time-average unsteady). Figure 3.4 shows that there is benefit in blockage generation due to the upstream stator wake – rotor tip clearance flow interaction and the effect increases with loading and upstream wake velocity defect. The same conclusions can be drawn from Figure 3.5 about the effect of unsteadiness on tip region loss generation.

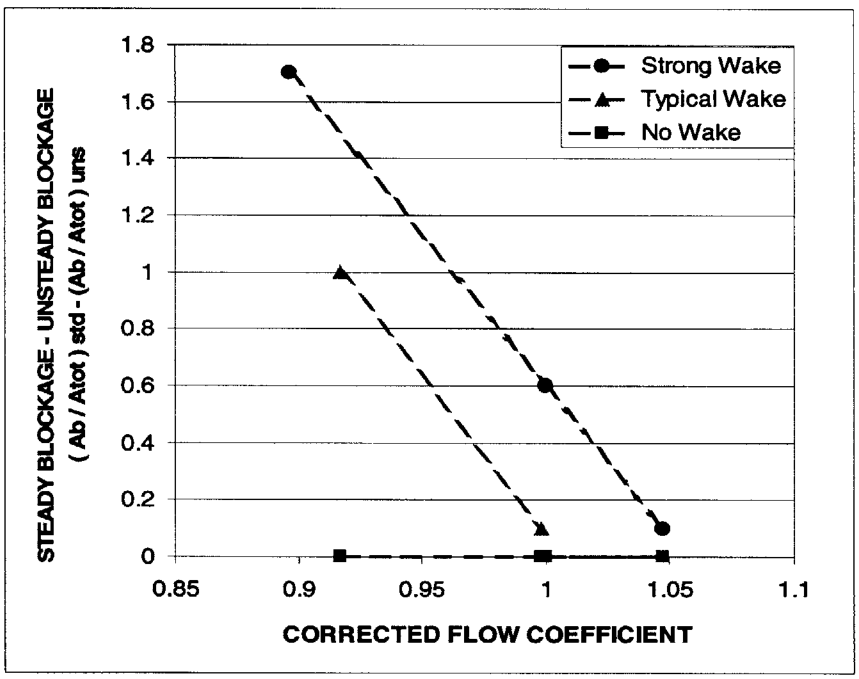


Figure 3.4 Beneficial Effect of Upstream Stator Wake – Rotor Tip Clearance Flow Interaction on Tip Region Blockage Generation. The Benefit Increases With Loading and With Upstream Stator Wake Velocity Defect.

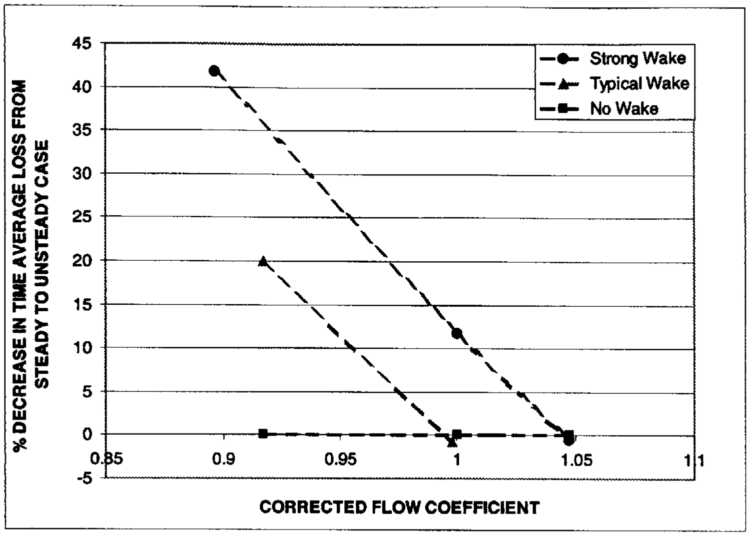
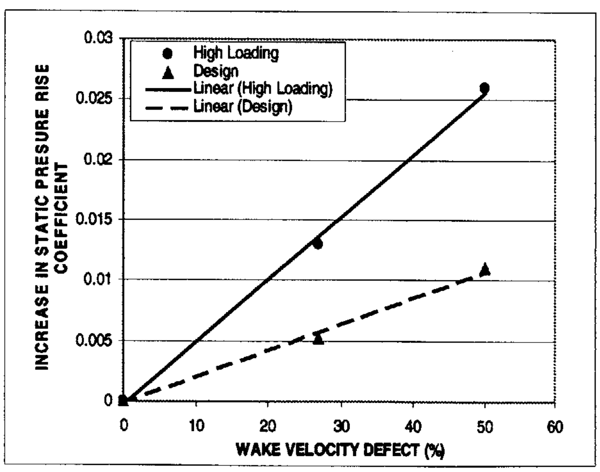
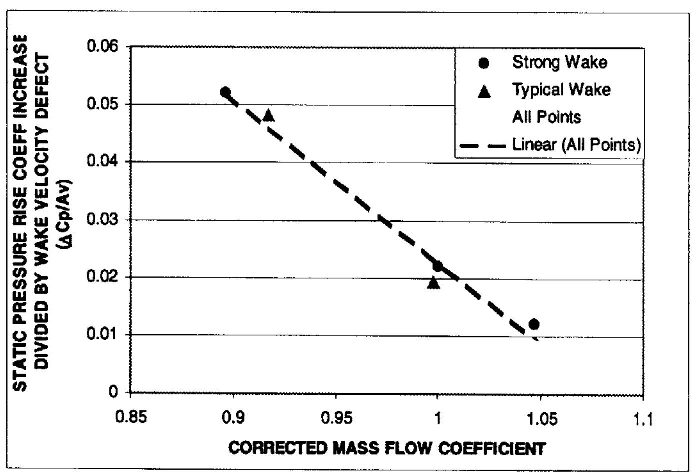


Figure 3.5 Beneficial Effect of Upstream Stator Wake – Rotor Tip Clearance Flow Interaction on Tip Region Loss Generation. The Benefit Increases With Loading and With Upstream Stator Wake Velocity Defect.

The increase in loss reduction and blockage reduction with loading is also linear as can be seen in figures 3.4 and 3.5. Another observation is that the beneficial effect on pressure rise, blockage, and loss depends on the strength of the incoming wake defect. Figure 3.6a shows that the beneficial effect from the unsteady interaction scales linearly with the upstream wake velocity defect. Thus, when the observed changes in static pressure rise are normalized by the wake velocity defect, the data collapses on the same linear curve, as shown in Figure 3.6b. This information will allow the designer to estimate the aerodynamic benefit from closing the spacing between the blade rows and correct any results obtained with a mixing-plane steady approximation.



(a)



(b)

Figure 3.6 The Beneficial Effect of Upstream Unsteadiness Increases Monotonically with Upstream Wake Defect. Static Pressure Rise Coefficient is Normalized by Wake Velocity Defect to Obtain a Single Linear Dependence on Operating Condition for All Wakes.

Figures 3.7 through 3.12 show the calculated performance data used in the quantification of the effect of upstream unsteadiness on rotor tip clearance flow. Each set of figures presents data for the strong wake and typical wake cases. Pressure rise coefficient data is shown in Figures 3.7 and 3.8, followed by blockage data in Figures 3.9 and 3.10, and loss data in Figures 3.11 and 3.12.

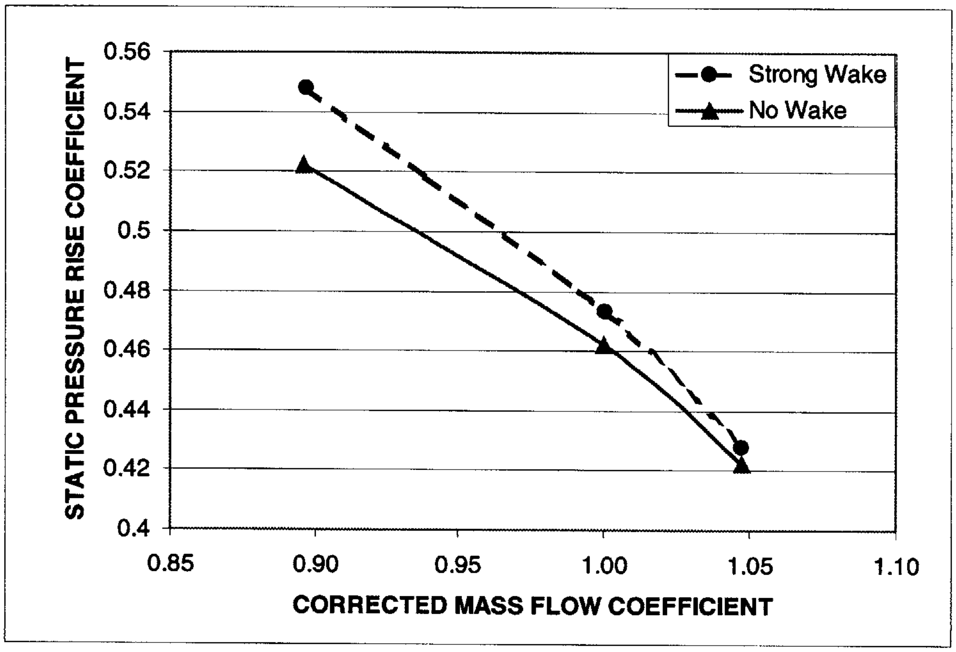


Figure 3.7 Beneficial Effect from Strong Upstream Stator Wake – Rotor Tip Clearance Flow Interaction on Rotor Static Pressure Rise Coefficient. The Benefit Increases with Loading.

One could infer from Figure 3.7 that there is benefit due to the unsteady interaction in the whole range of interest, from near design to high loading condition. The beneficial effect is modest near design but increases with loading. For the strong unsteady interaction case the benefit in rotor static pressure rise coefficient, Cp, is up to 5 % from the steady calculated value (0.52 steady vs. 0.55 time-average unsteady). Figure 3.8 shows the benefit in static pressure rise for the typical wake unsteady interaction case. The dependence of the benefit on loading is similar to the dependence in the strong unsteady interaction case but the highest benefit calculated for the typical unsteady interaction case is 2.8 % increase in Cp from the steady calculated value. This shows that the benefit depends on the strength of the upstream wake velocity defect.

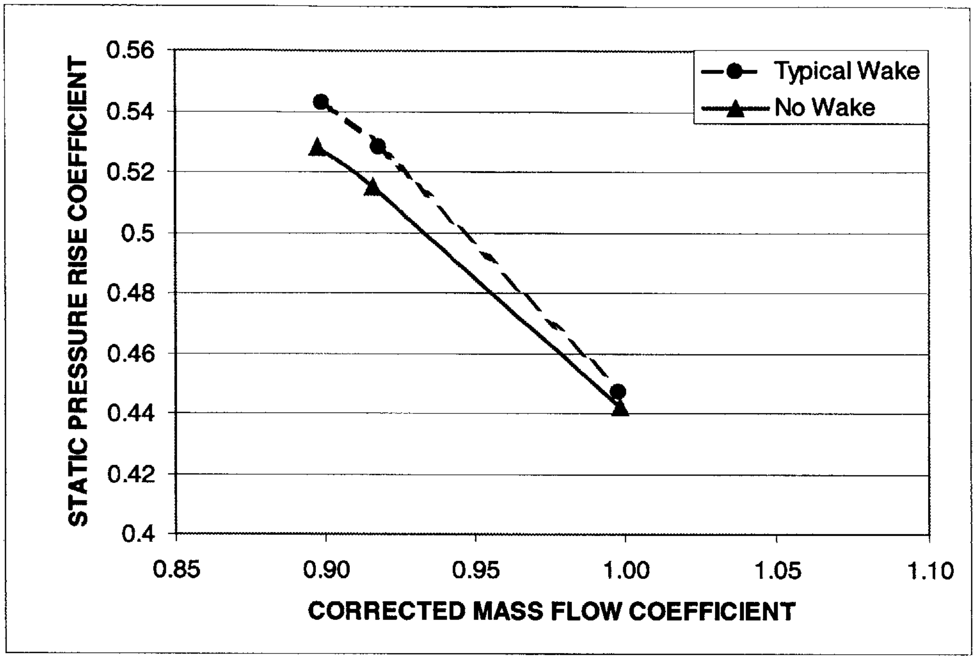


Figure 3.8 Beneficial Effect from Typical Upstream Stator Wake – Rotor Tip Clearance Flow Interaction on Rotor Static Pressure Rise Coefficient. The Benefit Increases with Loading. The Performance Benefit From The Typical Wake Interaction is Lower than the Performance Benefit From the Strong Wake Interaction.

Figure 3.9 shows the reduction in tip region blockage generation for the strong wake unsteady interaction case. Again, the benefit is modest near design and increases with loading. The highest calculated benefit is 27 % decrease in tip region blockage from the steady calculated value (0.064 steady vs. 0.047 time-average unsteady). Figure 3.10 shows the effect of typical unsteady interaction on tip region blockage generation and its dependence on loading. The highest calculated benefit for the nominal wake unsteady interaction case is lower than the highest benefit calculated for the strong unsteady interaction case, which is similar to the static pressure rise coefficient results.

Figures 3.11 and 3.12 show the benefit from unsteady blade row interaction on tip region loss generation. The highest benefit occurs at high loading and for the strong unsteady interaction case. The tip region loss coefficient decreases with 40 % from the calculated steady value (0.174 steady vs. 0.103 time-average unsteady).

From the results presented in this section it can be inferred that the benefit in blade passage static pressure rise, tip region loss generation, and tip region blockage generation increases with loading and with upstream stator wake velocity defect.

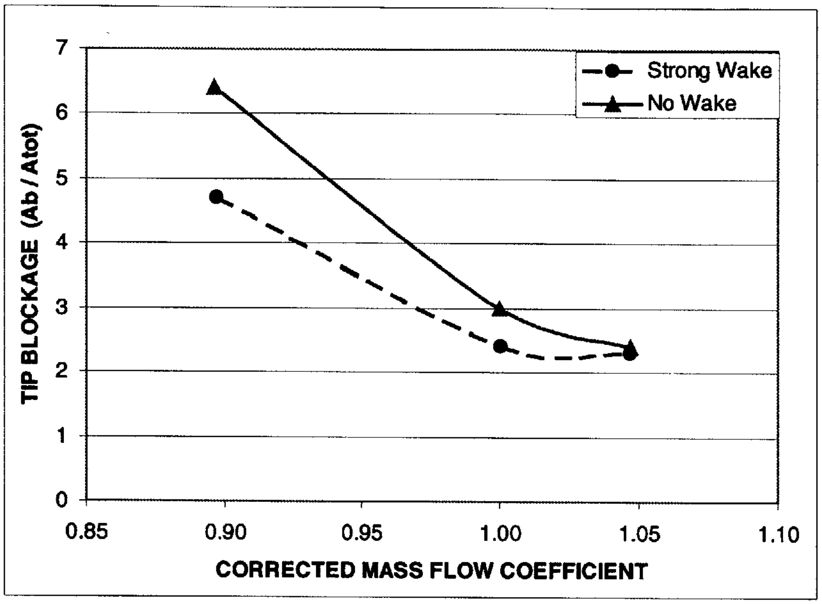


Figure 3.9 Beneficial Effect from Strong Upstream Stator Wake – Rotor Tip Clearance Flow Interaction on Tip Region Blockage Generation. The Benefit Increases with Loading.

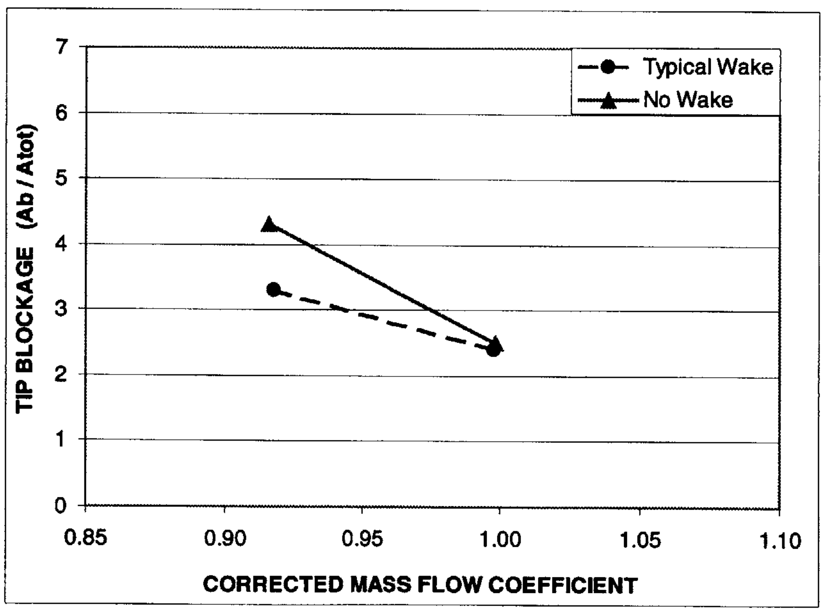


Figure 3.10 Beneficial Effect from Typical Upstream Stator Wake – Rotor Tip Clearance Flow Interaction on Rotor Static Pressure Rise Coefficient. The Benefit Increases with Loading. The Performance Benefit From The Typical Wake Interaction is Lower than the Performance Benefit From the Strong Wake Interaction.

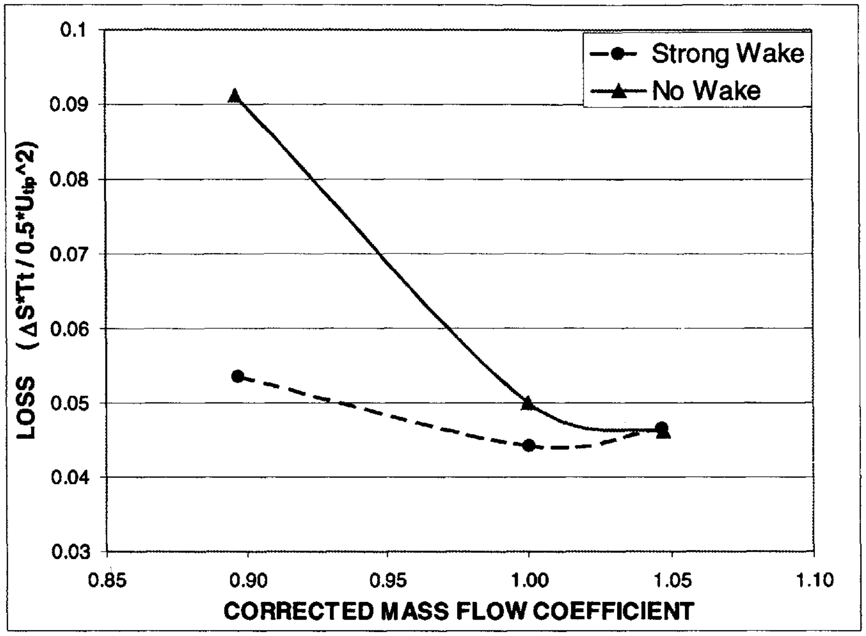


Figure 3.11 Beneficial Effect from Strong Upstream Stator Wake – Rotor Tip Clearance Flow Interaction on Tip Region Loss Generation. The Benefit Increases with Loading.

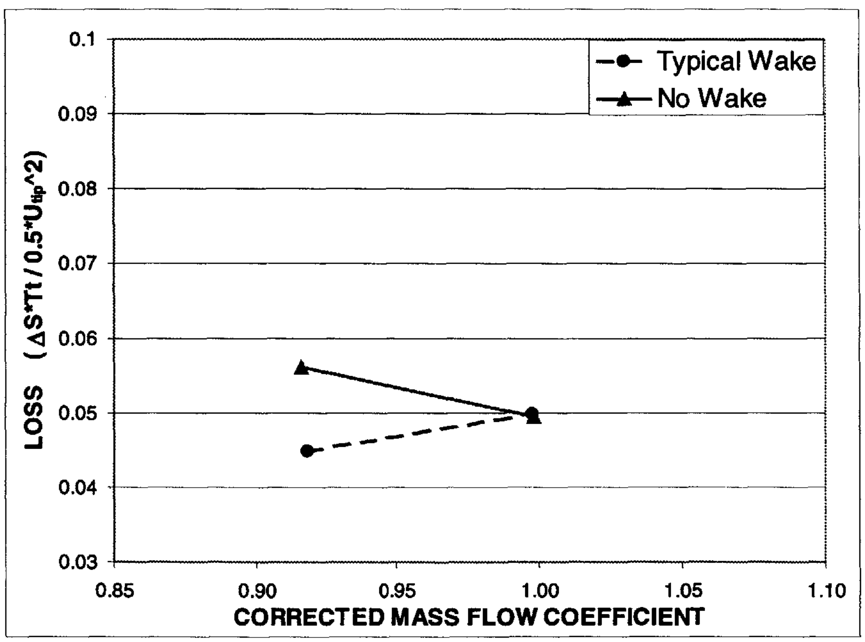


Figure 3.12 Beneficial Effect from Typical Upstream Stator Wake – Rotor Tip Clearance Flow Interaction on Rotor Static Pressure Rise Coefficient. The Benefit Increases with Loading. The Performance Benefit From The Typical Wake Interaction is Lower than the Performance Benefit From the Strong Wake Interaction.

3.3 COMPARISON TO PREVIOUS WORK

Smith's experiment [13] in 1970 on a low speed axial compressor demonstrated a measured benefit in both pressure rise and efficiency by reducing the spacing between the blade rows from 0.37 to 0.07 blade chords. The effect that he measured has been attributed to the increased unsteadiness in the blade rows. However, the complete set of fluid mechanical mechanisms responsible for the observed changes has not been quantitatively clarified nor identified.

Some of the effect is attributed to wake recovery through inviscid stretching based on Kelvin's theorem (Smith [13]). Valkov [1] performed a numerical study and quantified the effect of upstream wakes on stator performance. He identified two causal mechanisms (reversible recovery of wake energy and non-transitional boundary layer response) leading to changes in performance and quantified the net benefit. The results in the current study exclude the effects of upstream wakes on blade boundary layers and the effects of wake stretching. The focus of the present work is on the effect of unsteadiness in the tip region. The design characteristics of the blade geometry used in the previous work done by Smith [13] and Valkov [1] and the blade geometry used in the present study are not the same but many of the important geometric and flow parameters are similar (Table 3.1). Based on the observation that the design characteristics of these three blade rows are similar on an approximate basis, the results from the current study may be combined with Valkov's findings to assess against Smith's experimental data on the relative importance of the new mechanism to the measured beneficial changes. Table 3.2 presents the comparison.

	LAR ROTOR (SMITH)	E3 STATOR (VALKOV)	ROTOR IN PRESENT STUDY
Mach Number	N/A	0.55	0.50
Reynolds Number	178,000	247,000	355,000
Aspect Ratio	1.96	1.37	0.96
Solidity	1.09	1.67	1.19
Stagger	34.1	32	39.4

Table 3.1 Rotor Geometry for the LAR Rotor as Described by Smith [13], GE E^3 Blade Geometry as Described by Wisler [26] and Present Study Rotor Blade Geometry.

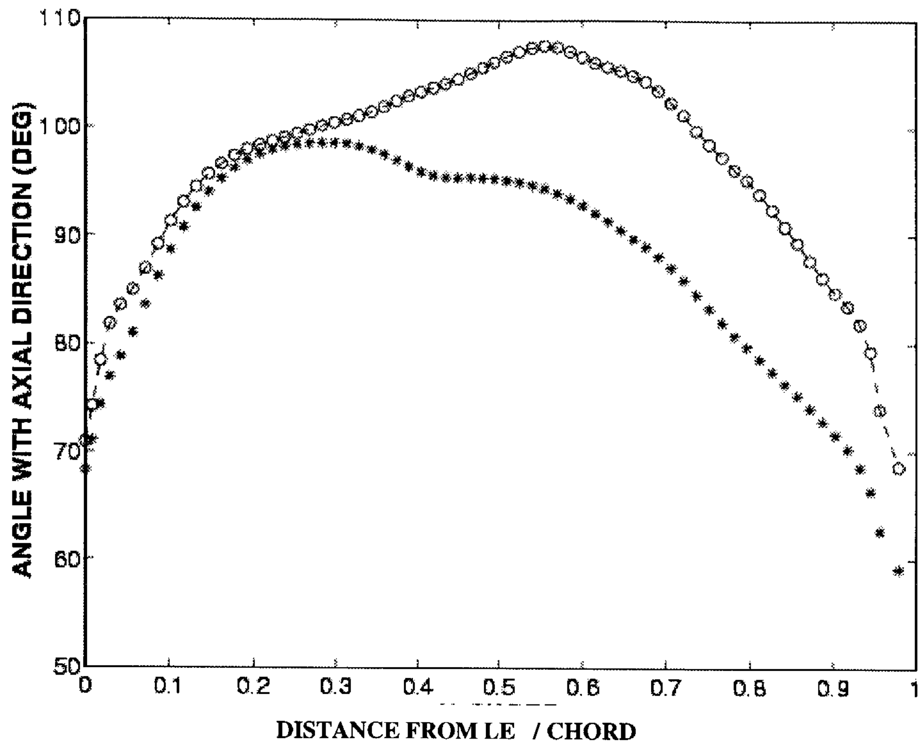
BENEFIT FROM INCREASING UPSTREAM UNSTEADINESS (REDUCTION OF BLADE ROW AXIAL SPACING FROM 0.37 TO 0.07 CHORDS)		
	PRESSURE RISE	EFFICIENCY
VALKOV (net effect from reversible recovery of wake energy and non-transitional boundary layer response)	N/A	0.40 pts +
PRESENT STUDY (effect of wake pressure pulse – tip clearance flow interaction)	1.1 % +	0.38 pts +
NET EFFECT (Valkov + Present Study)	**1.1 % +**	**0.78 pts +**
SMITH (Experimental Data)	**2~4 % +**	**1.00 pts +**

TABLE 3.2 The Benefit From Upstream Stator Wake-Rotor Tip Clearance Flow Interaction Combined with the Net Benefit From Reversible Recovery of Wake Energy and Non-Transitional Boundary Layer Response Appears to Account for 80% of Smith's Experimental Findings.

The results in Table 3.2 show that the beneficial effect observed in the current study is comparable to the net beneficial effect calculated by Valkov [1], and both combined effects can explain a significant part (about 80 %) of Smith's experimental findings.

3.4 EFFECT OF UNSTEADY INTERACTION ON TIP CLARANCE FLOW INTERFACE ANGLE

It was observed that the angle of the interface between the incoming flow and the tip clearance flow was different in the steady and time-average unsteady cases. For the strong interaction case the average interface angle in the unsteady case is 3 degrees lower than the interface angle in the steady case. This is a consequence of the difference in the tip clearance flow exit stream-wise velocity. This difference leads to a difference in the tip clearance fluid exit angle (Figure 3.13) which consequently causes a difference in the interface angle between the time averaged unsteady and steady cases. Possible benefits due to this change in interface angle are discussed in Chapter 5.



★ (stars – time-average unsteady) o (circles – steady)

Figure 3.13 Angle Between Tip Clearance Flow Exit Direction and Axial Direction. The Tip Clearance Flow from the Time-Average Unsteady Solution Exits the Tip Gap at a Lower Angle with Respect to The Blade and to the Axial Direction than the Tip Clearance Flow in the Steady Calculation.

3.5 SUMMARY

The results presented in section 3.2 show that upstream unsteadiness has a beneficial effect on rotor tip clearance flow. The strong interaction case showed tip region loss coefficient which is up to 40 % lower than the calculated value in the steady case (0.174 steady vs. 0.103 time-average unsteady), tip region blockage which is up to 27 % lower than the one calculated in the steady case (0.064 steady vs. 0.047 time-average unsteady), and overall static pressure rise coefficient which is up to 5 % higher than the pressure rise coefficient calculated in the steady case (0.52 steady vs. 0.55 time-average unsteady). The beneficial effect increases monotonically with loading and increases linearly with the strength of the velocity defect in the upstream wake. Therefore, reduced spacing between the blade rows may lead to an enhancement in aerodynamic performance. The knowledge of the linear dependence provides designers with means to estimate the effects of varying the spacing between the blade rows on rotor tip region performance. Based on the results presented in this chapter, it is suggested that designers may want to propose a more aggressive rotor blade design in terms of loading because the beneficial effect increases with loading.

When the benefit in performance associated with the time-average effect of stator wakes-rotor tip clearance flow interaction is combined with the net benefit from reversible recovery of wake energy and non-transitional boundary layer response reported by Valkov [1], the sum appears to account for 80 % of the Smith's experimental findings [13].

As discussed in chapter 2, the two-dimensional assessment case for the comparison method between the time-average unsteady and the steady cases showed detrimental effects on performance caused by the unsteady interaction. This is in contrast to the beneficial changes presented in this chapter for the three-dimensional study. Therefore, the comparison methodology itself does not lead to calculation of beneficial changes and is not the cause for the observed beneficial changes. There is a physical cause for the calculated benefit.

CHAPTER 4 DISCUSSION OF RESULTS AND ESTABLISHMENT OF CAUSE AND EFFECT RELATION

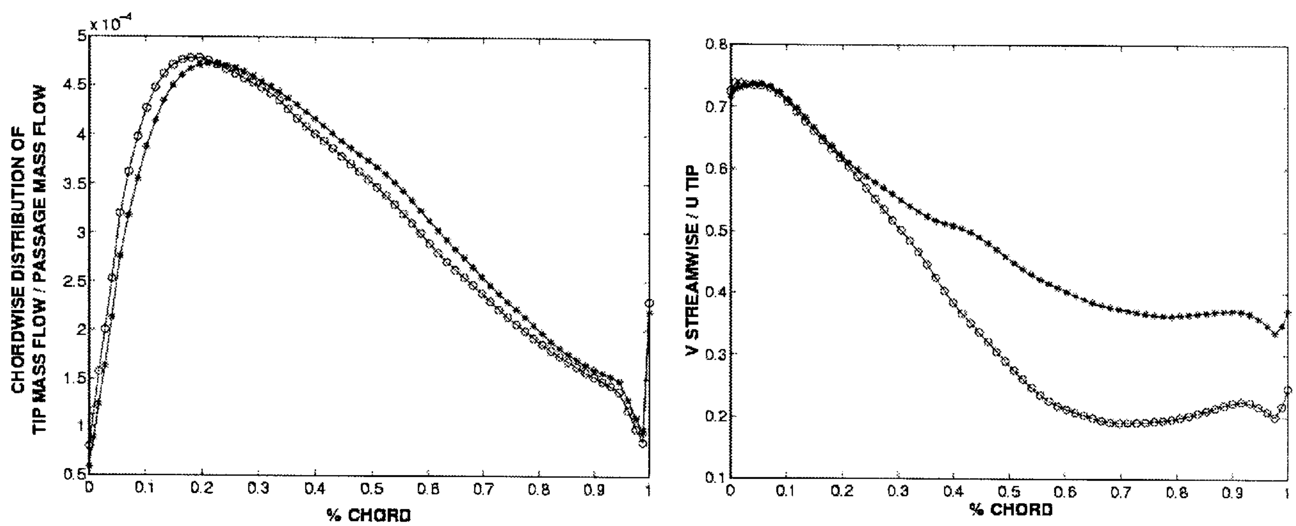
4.1 INTRODUCTION

The results from Chapter 3 indicate that the interaction between upstream stator wakes and rotor tip clearance flow can be beneficial, especially at high loading. The increased interaction causes less blockage and loss, and consequently an enhanced pressure rise. This chapter investigates the cause for the computed changes and proposes a causal mechanism to explain the observations. The relevance of the proposed mechanism to the observed effects is evaluated with the use of simple control volume mixing analysis based on Storer and Cumpsty's model for tip clearance loss estimation [9]. The comparison between the results from the estimation and the calculations show that the proposed mechanism is responsible for the identified changes in performance.

4.2 EXPLANATION OF CAUSE AND EFFECT RELATION

After calculating a benefit in loss due to the unsteady interaction between upstream stator wakes and rotor tip clearance flow it was of interest to calculate the mixed-out loss for the tip region. This will reveal whether the difference in loss between the steady and time-average unsteady cases is caused by a difference in the tip clearance fluid exiting the tip gap (if the mixed-out loss is different for the steady and the time-average unsteady cases) or the benefit comes from less mixing locally in the tip region of the blade passage (if the mixed-out loss is the same for the steady and time-averaged unsteady cases). The mixed-out loss in the unsteady case is 27 % lower than the value calculated in the steady case. This indicates that the tip flow in the unsteady case exits the tip clearance region with significantly smaller defect with respect to the main flow, than the tip flow in the steady case.

Graf [5] reported similar behavior in his study of the effect of back pressure fluctuation due to downstream blade potential field on rotor performance. Graf explained the difference with the fact that the time-average mass flow through the tip clearance in the case with unsteadiness is 6% less than that in the steady case. In other words, the effect of the fluctuating downstream pressure is to decrease the tip flow through the tip clearance, leading to less loss and blockage. However, this is not the case for the effect of upstream unsteadiness, as can be seen in Figure 4.1a.

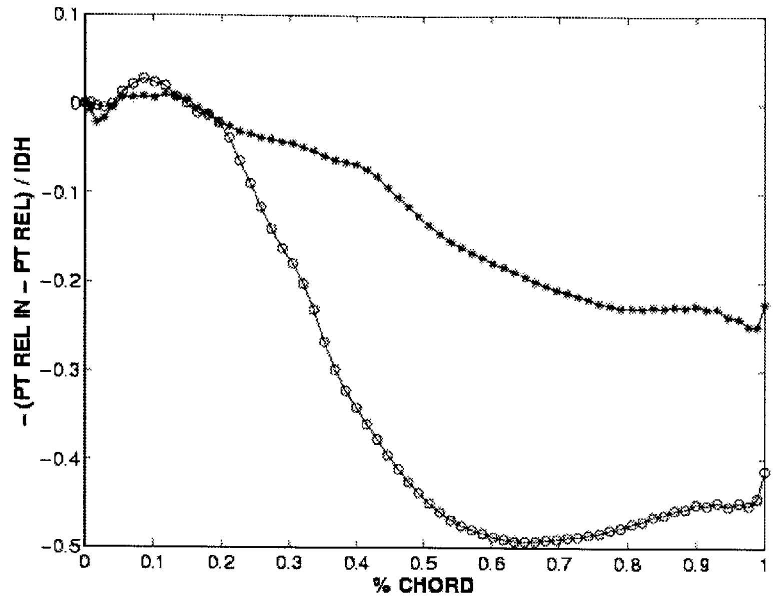


★ - time-average unsteady o - steady

(a) (b)

Figure 4.1 Tip Clearance Mass Flow and Stream-wise Velocity for Steady and Unsteady Cases. Total Tip Clearance Mass Flow is 2% of Blade Passage Mass Flow.

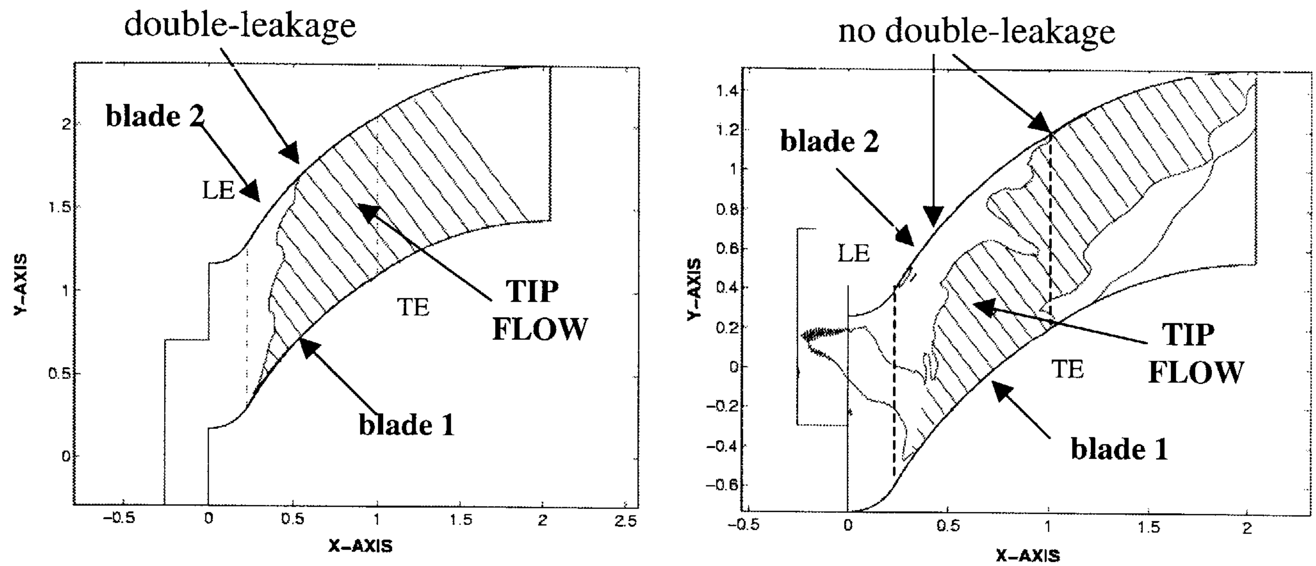
The average normal components of velocity through the tip clearance flow (which determine the mass flow) are the same for the steady and unsteady cases (Figure 4.1a). However, the stream-wise components of the tip flow exit velocity are different between the time-average of the unsteady calculation and the steady calculation. This is the reason for the difference in the exit defect of the tip flow in the steady and time-average unsteady cases. The average stream-wise component of the tip flow exit velocity normalized by the rotor tip velocity in the unsteady case is 0.50, which is 25 % higher than that in the steady case, 0.40 (Figure 4.1b). To examine further this change in tip clearance flow behavior the rotary stagnation pressure in the tip clearance gap is calculated and shown in Figure 4.2. Rotary stagnation pressure is chosen because it can show where the tip clearance fluid is coming from. The main flow and the wakes have high relative stagnation pressure while the tip clearance flow has a relative total pressure defect. Entropy is not suitable to differentiate between wake and tip clearance fluid because both of them have high levels of entropy. It can be clearly seen from the figure that on a time-average basis the tip flow exiting the tip gap in the steady case has much higher rotary stagnation pressure defect than the one in the unsteady case.



★ - unsteady o - steady

Figure 4.2 Rotary Stagnation Pressure of Tip Clearance Fluid Exiting the Tip Gap. On a Time-Average Basis Unsteady Tip Clearance Flow Exits the Tip Gap with Less Rotary Stagnation Pressure Defect.

The reason for this behavior can be seen in Figure 4.3. The tip clearance flow in the steady case double-leaks (passes through the tip gap of the neighboring blade) through a significant portion of the next blade. Double-leakage means that the pressure difference across the blade drives through the tip gap the low total pressure fluid originating from the previous tip clearance instead of the high total pressure fluid from the main flow and the wakes. Wakes in the relative frame have high total pressure. Double-leakage happens at every instant of time for the steady case, which is also run in a time-accurate mode for consistent comparison. The low stagnation pressure tip fluid exits the tip gap with low stream-wise velocity component. In the unsteady case, however, double-leakage occurs only at certain selected instants of time. Figures 4.3 and 4.4 show instants of time when there is absolutely no double-leakage in the unsteady case. Therefore, only some of the fluid that exits the tip clearance gap in the unsteady case comes from the previous tip clearance and has high loss and low stagnation pressure. Thus, on a time average basis, the rotary stagnation pressure of the fluid exiting the tip clearance is higher in the unsteady case. Consequently, the stream-wise velocity component of the exiting tip flow is higher in the unsteady case and the defect with respect to the main flow is smaller. The smaller defect generates less loss and less blockage in the blade passage.



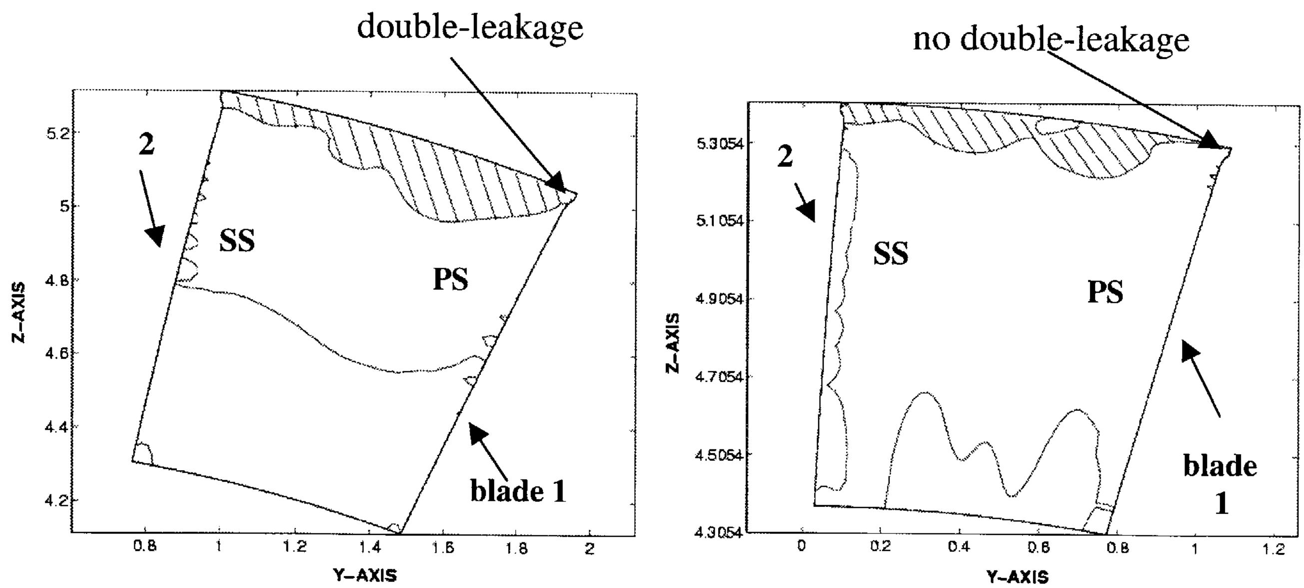
steady

(tip clearance fluid from blade 1
passes through tip gap of blade 2)

instantaneous unsteady

(NO tip clearance fluid from blade 1
passes through tip gap of blade 2)

Figure 4.3 Tip Clearance Flow Behavior in Steady and Unsteady Environment (98 % cut).



(tip clearance fluid from blade 1
passes through tip gap of blade 2)

(NO tip clearance fluid from blade 1
passes through tip gap of blade 2)

Figure 4.4 Tip Clearance Flow Behavior in Steady and Unsteady Environment (70 % cut from LE). Double-leakage is Present at All Times in a Steady Environment and Only at Certain Instants of Time in an Unsteady Environment. The Figures on the Right Show Instants of Time in the Unsteady Environment When Double-Leakage is Completely Absent.

To address the question on why at high loading the tip clearance flow double-leaks constantly in a steady environment and double-leaks only in certain instants of time in an unsteady environment, the effect of the upstream wakes needs to be examined. In the rotor relative frame the upstream wakes appear as jets normal to the blade chord directed away from the blade suction surface. This behavior is sketched in Figure 4.5.

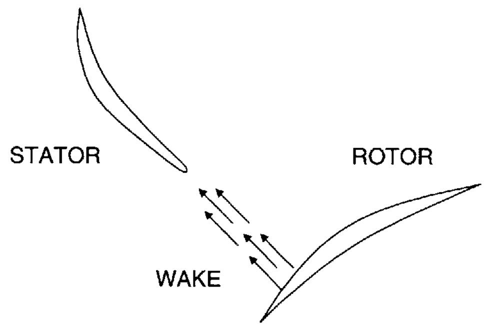


Figure 4.5 Upstream Wakes Appear as Normal Jets Directed Away from the Rotor Suction Side in the Rotor Relative Frame.

The normal jet in the rotor frame impinges on the pressure surface of the blades and a local stagnation point appears (Figure 4.6). This stagnation of the fluid produces strong pressure pulses on the blades (Figure 4.7). The strength of these pressure pulses reaches values of up to 50 % of the inlet relative dynamic head, which is significant. Such pressure pulses have been previously reported by Valkov [1].

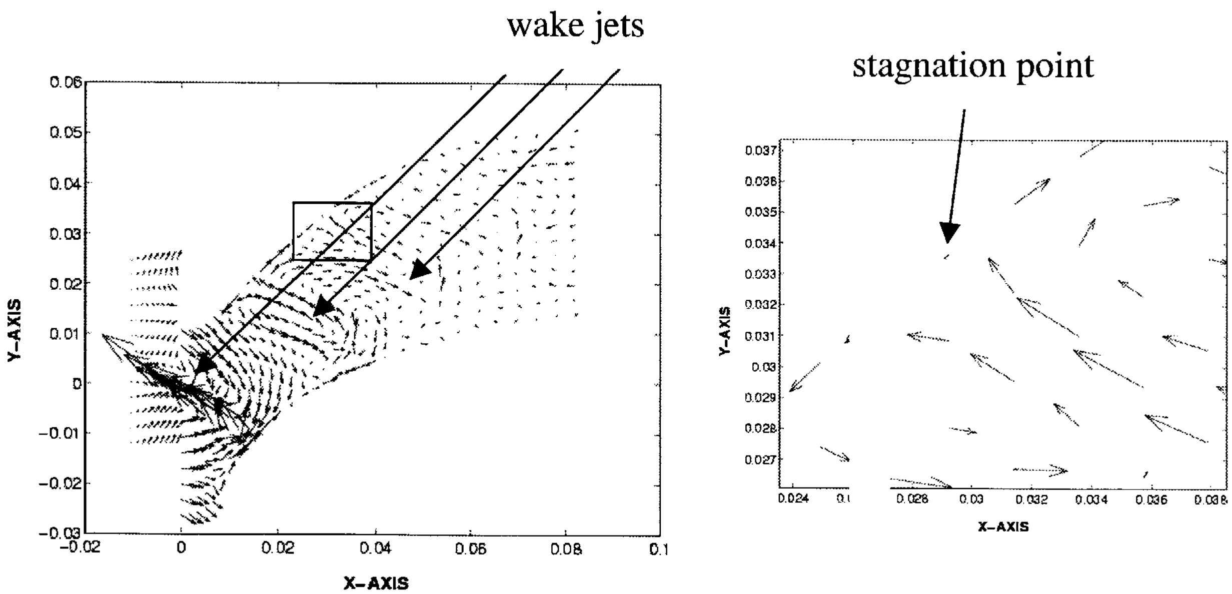


Figure 4.6 Instantaneous Disturbance Velocity Field in the Rotor (50 % cut). The Upstream Wakes Impinge on the Pressure Side and Stagnation Points Appear.

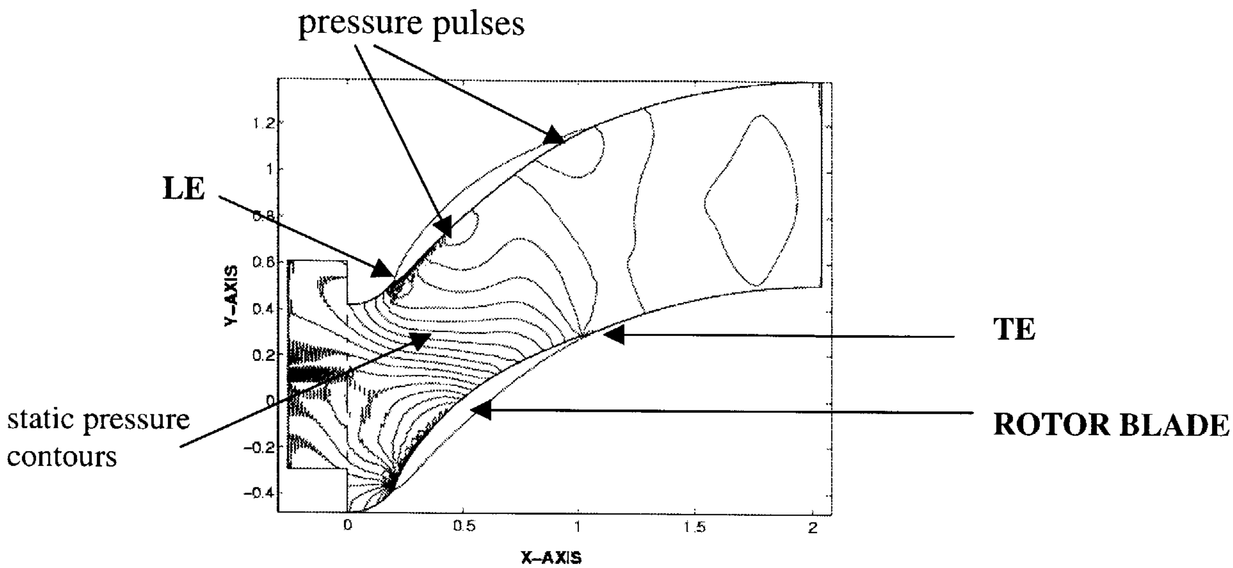
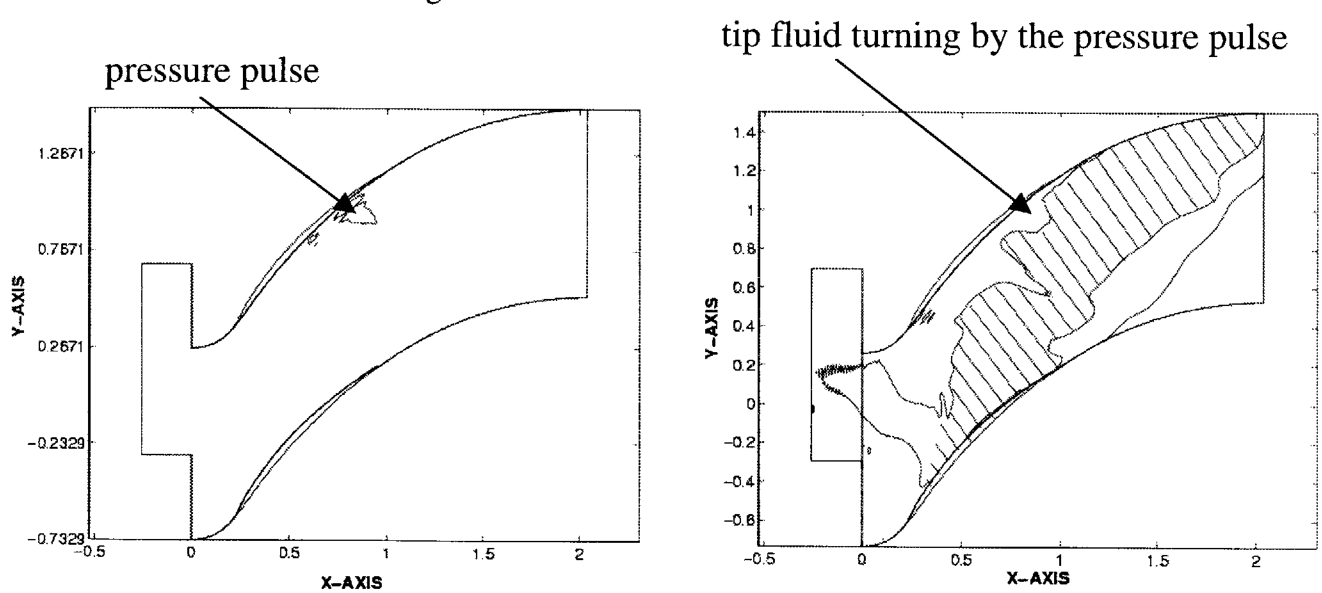


Figure 4.7 Instantaneous Position of Pressure Pulses in the Rotor Passage (50% Cut). Pressure Pulses Appear as a Result of the Wake Jet Stagnation on the Pressure Surface.

The following two figures elucidate the effect of the pressure pulses on the rotor tip clearance flow. Figure 4.8a identifies the location of a pressure pulse using a single pressure contour at a given instant of time. Figure 4.8b shows that at this exact location at the same instant of time, the tip clearance fluid is directed away from the pressure pulse. The pressure pulse creates a locally strong pressure gradient, which tends to deflect the clearance fluid away, thus reducing the ammount of double-leakage flow.



single static pressure contour

single relative stagnation pressure contour showing the border between tip and main fluid

Figure 4.8 Location of Isolated Pressure Pulse and Its Turning Effect on Tip Clearance Flow (98% Span Cuts).

Therefore, whenever a pressure pulse passes through the region of double-leakage, it turns the tip clearance fluid away from the blade and prevents the double-leakage. Since the upstream wakes causing the pressure pulses pass periodically through the rotor domain, the result is a tip clearance flow, which oscillates back and forth between double-leakage and no double-leakage. This mechanism is described in Figure 4.9. The oscillatory change in the direction of the tip clearance flow leads to a significant reduction of the time-average double-leakage flow in the unsteady case. This explains the smaller extent of the defect with which the tip clearance flow exits the tip gap in the unsteady case and consequently, the improvement in performance.

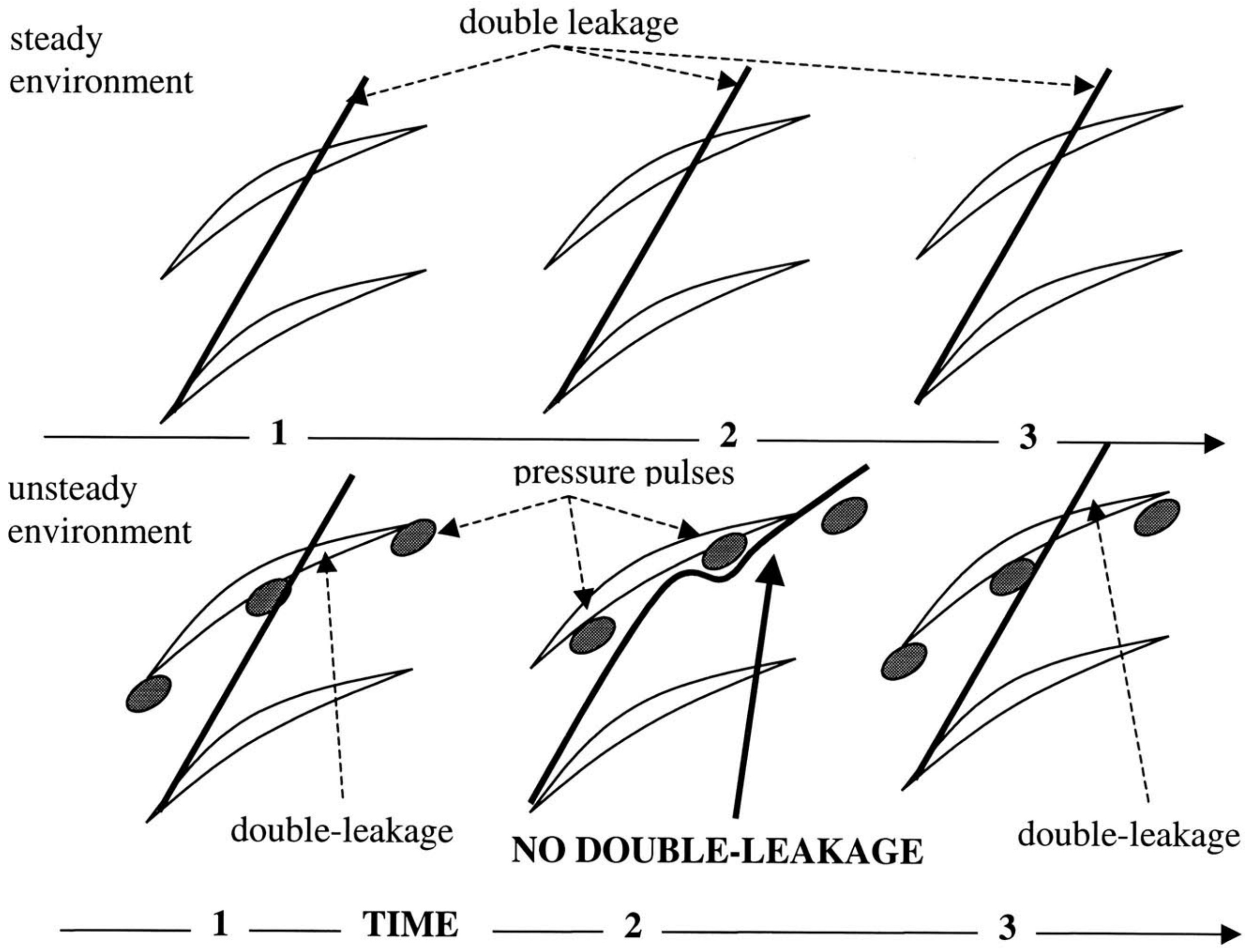
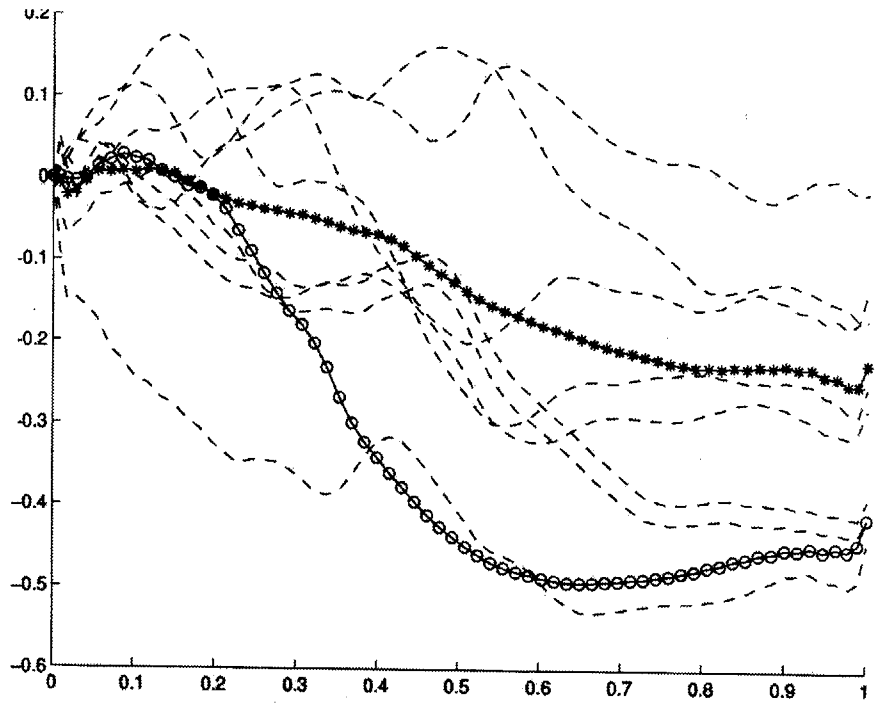


Figure 4.9 Fluid Scenario to Explain the Reduction of Tip Clearance Fluid Double-Leakage and Enhancement of Performance. Pressure Pulses Prevent Double-Leakage during Selected Instants of Time in a Cycle.

To confirm the periodic changes in the direction of the tip clearance flow and the resulting periodic changes of the rotary stagnation pressure of the fluid passing through the tip gap, Figure 4.10 is presented. It shows the rotary stagnation pressure at different instants of time and the average values for the unsteady and steady cases. It can be observed that in the unsteady case both low stagnation pressure tip fluid and high stagnation pressure main flow and wake fluid pass through the tip gap at different times.

(-) Rotary Stagnation Pressure Loss



DISTANCE FROM LE / CHORD

★ - time-average unsteady　　o – steady　　- - - - instantaneous

Figure 4.10 Wake Pressure Pulses Change the Tip Fluid Direction Close to the Blade Pressure Surface and Decrease Time-Average Double-Leakage. Rotary Stagnation Pressure Defect is Shown with Dotted Lines in The Tip Gap for Different Instants of Time in a Cycle. High Rotary Stagnation Pressure Defect Fluid Passes Through the Tip Gap When Double-Leakage is Present and Low Rotary Stagnation Pressure Defect Fluid Passes when Double Leakage is Prevented by the Pressure Pulses. Average Steady and Unsteady Values are Shown with Solid Lines.

4.3 RELEVANCE OF PROPOSED MECHANISM TO OBSERVED PERFORMANCE CHANGES

To evaluate and quantify the relevance of the proposed mechanism to the observed performance changes a simple control volume mixing analysis is performed. The analysis uses Storer and

Cumpsty's [9] idea but all necessary input parameters are extracted from the computed solutions instead of being estimated with the assumptions in the original model. The inlet and exit areas, densities, and the inlet stream-wise velocities of the tip fluid and the main fluid are needed for the control volume calculation to obtain a mixed-out loss. The analysis uses conservation of mass and momentum in the stream-wise direction and assumes mixing at constant pressure. A schematic of the control volume can be seen in Figure 4.11.

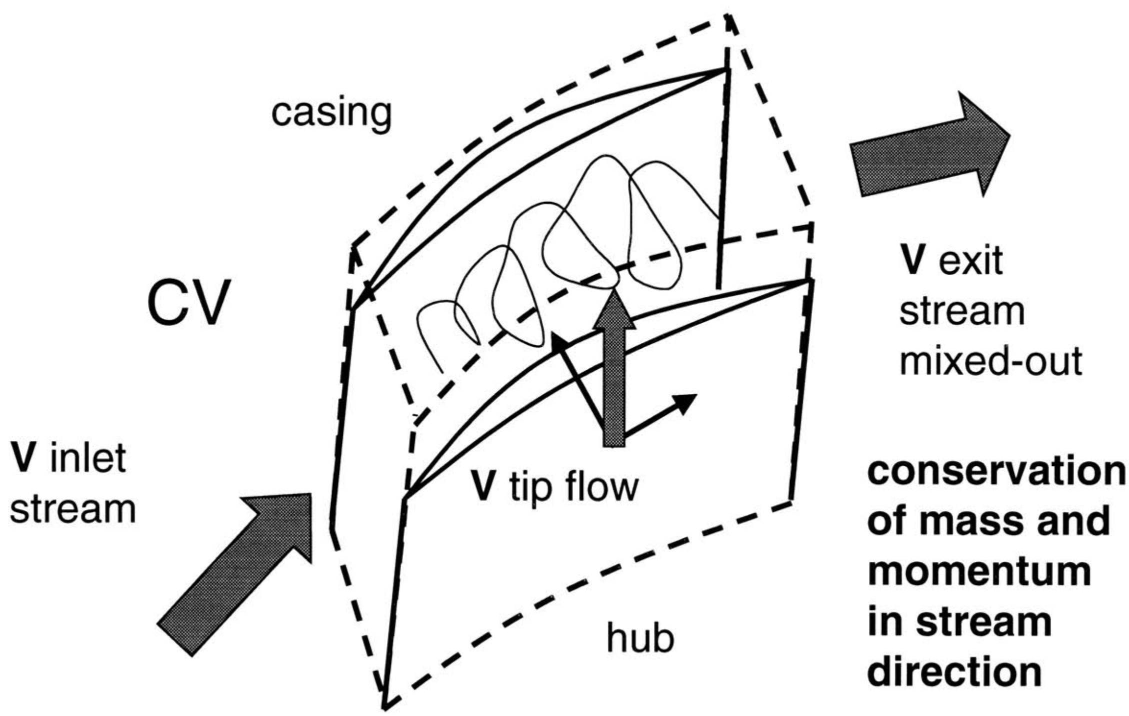


Figure 4.11 Control Volume Mixing Analysis for Prediction of Tip Clearance Loss.

The assumption of Storer and Cumpsty in their original model that the tip fluid stagnation pressure is equal to the stagnation pressure of the upstream main flow breaks down as soon as double-leakage occurs. Double-leakage changes significantly the tip clearance fluid stagnation pressure as shown in the previous figures (Figures 4.2 and 4.10). This fact was previously reported by Khalsa [12]. Therefore, the original model cannot be applied to situations with double-leakage. It will underestimate the losses by up to 60 % as shown in the following two tables. Table 4.1 compares the loss generated in a steady environment to the one generated in an unsteady environment with strong wakes interaction. Loss is estimated using the original model in the first row, the model using input parameters extracted from the solution in the second row, and the CFD results in the third row. Table 4.2 compares the loss generated in a steady environment to the loss generated in an unsteady environment with typical wakes interaction.

LOSS (TtΔS / 0.5Utip^2)			
	No Wake	Strong Wake	Difference
Storer's Mixing Model	5.06 e-3	5.06 e-3	0%
Storer's Model (Corrected Pt Tip Fluid)	11.89 e-3	9.09 e-3	23.50%
CFD Calculation	12.06 e-3	7.87 e-3	34.70%

Table 4.1 Original Loss Prediction Model by Storer and Cumpsty [9] Underpredicts Loss by Up to 60 % (first row) of CFD Results (third row) for Both Steady and Strong Wake Unsteady Environment When Double-Leakage is Present. If the Correct Relative Total Pressure is Used in the Prediction Model, Loss is Reasonably Estimated (second row) for Steady Environment and Unsteady Environment with Strong Wakes and the Effect of Unsteadiness is Captured (column "Difference").

LOSS (TtΔS / 0.5Utip^2)			
	No Wake	Typical Wake	Difference
Storer's Mixing Model	2.58 e-3	2.58 e-3	0%
Storer's Model (Corrected Pt Tip Fluid)	4.66 e-3	3.74 e-3	19.70%
CFD Calculation	4.90 e-3	4.18 e-3	14.70%

Table 4.2 Original Loss Prediction Model by Storer and Cumpsty [9] Underpredicts Loss by Up to 47 % (first row) of CFD Results (third row) for Both Steady and Typical Wake Unsteady Environment When Double-Leakage is Present. If the Correct Relative Total Pressure is Used in the Prediction Model, Loss is Reasonably Estimated (second row) for Steady Environment and Unsteady Environment with Typical Wakes and the Effect of Unsteadiness is Captured (column "Difference").

The results above show that when Storer and Cumpsty's model is corrected for the stagnation pressure of the tip clearance flow it can give a reasonable estimate for the tip clearance loss for both steady and unsteady configurations. As soon as double-leakage occurs, the relative stagnation pressure of the tip clearance fluid for both steady and unsteady cases has to be adjusted accordingly. It will not be the same with the stagnation pressure of the main flow.

4.4 SUMMARY

This chapter presented a fluid mechanical mechanism of generic nature to explain the observed performance changes. This mechanism will take effect whenever double-leakage is present, regardless of the operating condition, rotor geometry, or flow regime. Double-leakage is a function of loading and there will be an increase in double-leakage at high-loading operating conditions for the same geometry. For different geometry, a higher solidity and a lower aspect ratio will lead to an increase in double-leakage flow, because geometrically, the tip clearance flow will arrive at the neighboring blade tip gap earlier. Double-leakage is an unwanted phenomenon. However, in the flow regime where double-leakage occurs, its effect on performance can be mitigated through the upstream unsteadiness manifested in the wake induced pressure pulses on the pressure surface. The wake-induced pressure pulses on the pressure surface significantly decrease the time-average amount of double-leakage through the rotor tip clearance and thus improve performance. As shown in this chapter, designers can use Storer and Cumpsty's model to estimate the tip region performance, as long as they have knowledge of the correct average relative stagnation pressure of the fluid exiting the tip clearance. A model for the prediction of the correct average relative stagnation pressure of the tip fluid has not been developed in this study. The proposed mechanism leads to significant changes in rotor performance and the discussed effects should be considered in multiple blade-row compressor design and performance prediction tools.

CHAPTER 5 CONCLUSIONS AND RECOMMENDATIONS

5.1 SUMMARY OF RESULTS AND CONCLUSIONS

This study investigated the effect of interaction between upstream wakes and rotor tip clearance flow in terms of time average performance. It was found that strong interaction can decrease the tip region loss coefficient by up to 40 % with respect to the steady case value (0.174 steady vs. 0.103 time-average unsteady), decrease tip region blockage by up to 27 % with respect to the steady case value (0.064 steady vs. 0.047 time-average unsteady), and increase rotor passage static pressure rise coefficient by up to 5 % with respect to the steady case value (0.52 steady vs. 0.55 time-average unsteady). The following conclusions can be stated:

1. Interaction with upstream wakes has a beneficial effect on rotor tip flow behavior resulting in enhanced rotor performance. At high loading the effect becomes significant. Significant beneficial changes occur only when double-leakage of tip clearance fluid is present. This phenomenon typically occurs at high loading but can be present at design condition. The beneficial effect increases monotonically with loading and scales linearly with upstream wake amplitude measured in terms of velocity defect.

2. The cause for the observed performance changes is the ***wake induced pressure pulse – tip flow*** interaction. It decreases the amount of double-leakage on a time-average basis and thus, decreases the stream-wise defect of the tip clearance fluid exiting the tip gap.

3. The beneficial effect of upstream wake – tip flow interaction, combined with the net beneficial effect of isentropic wake recovery and non-transitional boundary layer response, accounts for 80 % of the benefit observed in Smith's experiment.

4. The effect on loss can be reasonably estimated with Storer and Cumpsty's model if the correct tip clearance fluid relative stagnation pressure is used. The original model tends to underestimate loss by up to 60 % of the calculated value.

5. A design with reduced spacing between the blade rows can benefit from the higher upstream unsteadiness. The beneficial response of the rotor tip leakage flow to upstream stator wakes implies that a more aggressive rotor blade design in terms of loading can be sought. The effect of unsteadiness from upstream wakes should be included in design and performance prediction tools.

5.2 RECOMMENDATIONS FOR FUTURE WORK

There are two important aspects of this research that are worthy of further investigation. One is the effect of upstream wakes on rotor operability range. The other issue is the effect of resonant behavior of the tip clearance vortex. Comments about these topics are made in the following two sections:

5.2.1 Effect of Upstream Wakes on Rotor Operability Range

As mentioned in Chapter 3, the interaction between upstream wakes and tip clearance flow may also have a beneficial effect on rotor operability range. The speed-lines calculated for the strong interaction case suggested such a possibility. There is a difference between the strong interaction and the steady case speed-lines indicating that stall occurs at a lower mass flow for the strong interaction case. An emerging criteria for tip clearance flow field breakdown states that a necessary condition for the tip region flow field breakdown is the spilling of the interface between the main flow and the tip clearance flow in front of the leading edge of the neighboring blade (Vo [19]). This happens when the interface angle with the axial direction reaches 90 degrees. It was observed in this study that the time average interface angle in the strong interaction case is 3 degrees lower than that in the steady case, due to the difference in the tip fluid exit velocities. Therefore, the interface in the steady case may approach rotor blade leading edge plane before the interface in the corresponding situation that involves wake-tip clearance flow interaction. If the rate of change in interface angle with mass flow is estimated from the speed line calculations, the benefit in operability range can be estimated for the strong interaction case. Such estimation is performed and the result is shown in Figure 5.1. The estimation shows that the strong interaction case can achieve a 2.8 % lower corrected mass flow coefficient than the steady case. A detailed study is required to confirm or invalidate this idea.

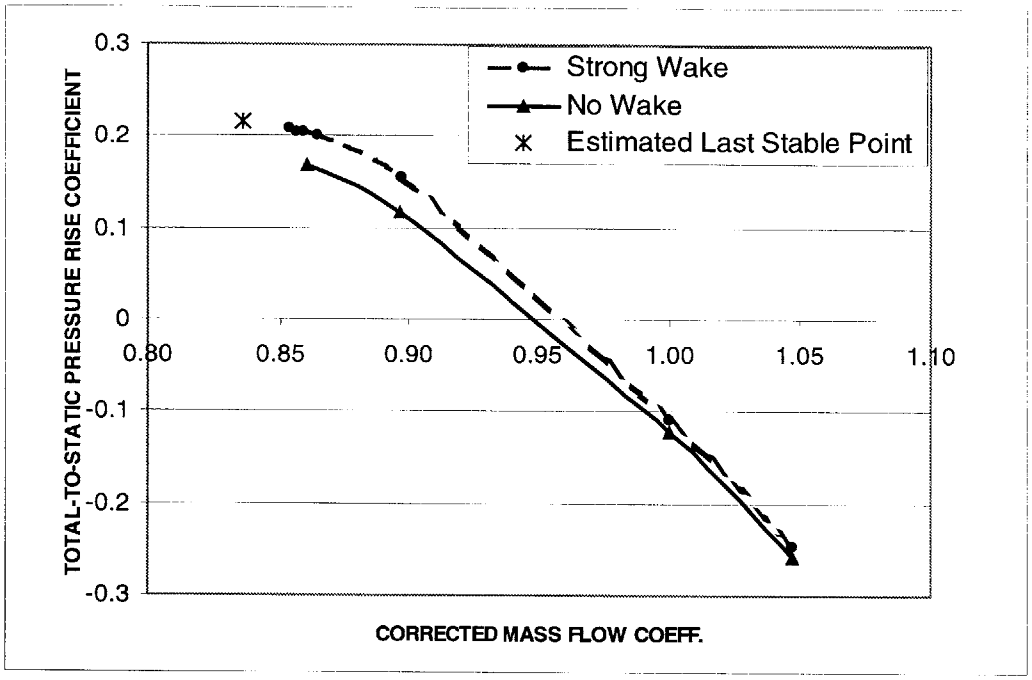


Figure 5.1 Estimation of Last Stable Point for Strong Interaction Case Speed Line. Strong Upstream Unsteadiness May Enhance the Operability Range of the Rotor.

5.2.2 Effect of Tip Clearance Vortex Resonance

Bae [18] proposed that the rotor tip clearance vortex and its image in the casing exhibit an instability behavior, which can be linked to the Crow's vortex instability theory (Figure 5.2). Bae showed that if the tip clearance vortex is excited at its natural frequency, significant changes could take place in the flow field. Such vortex instability was first reported by Graf [5], and then confirmed by Bae [18], Mailach [17], and Vo [19], among others. The tip clearance vortex can be excited if the resonant frequency is introduced into the flow field. One way of achieving this effect is through the upstream stator wakes. If the number of upstream stators is appropriately selected, the stator wake passing frequency in the rotor frame can be used. However, it is challenging to perform a numerical experiment with different numbers of rotor and stator blades because of the large computational resources needed. A possible way of introducing the resonant frequency is to sinusoidally vary the amplitude of the passing wakes.

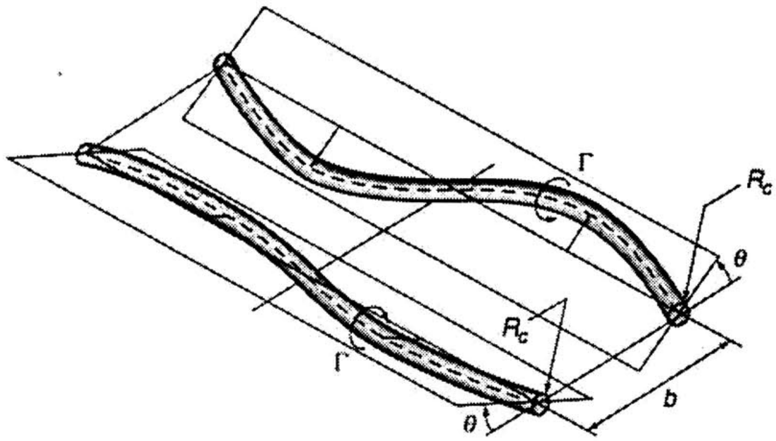


Figure 5.2. Vortex Pair Instability.

If the tip clearance vortex is driven into a resonant behavior, it is suggested that the tip vortex may respond in a manner that enhances the mixing of the tip clearance fluid with the main flow. This will prevent the growth of the tip clearance fluid defect in the adverse pressure gradient and may decrease blockage significantly. The resonant behavior of the tip vortex may also act to prevent double-leakage from taking place. The consequence of this could be an enhancement in the aerodynamic performance of the rotor in the compressor.

REFERENCES

[1] Valkov, T. V., "The Effect of Upstream Rotor Vortical Disturbances on the Time-Average Performance of Axial Compressor Stators", Ph.D. thesis, MIT, August 1997.

[2] Wisler, D.C., "Loss Reduction in Axial-Flow Compressors Through Low-Speed Model Testing", ASME J. Eng. Gas Turbines and Power, 1985, vol.107, pp.354-363

[3] Smith, L. H., Jr., "The Effect of Tip Clearance on Peak Pressure Rise of Axial-flow Fans and Compressors", ASME Symposium on Stall, 1958, pp.149-152.

[4] Koch, C. C., "Stalling Pressure Rise Capability of Axial Flow Compressors", ASME J. Eng. Power, 1981, Vol.103, pp.645-656.

[5] Graf, M. B., "Effects of Stator Pressure Field on Upstream Rotor Performance", Ph.D. thesis, MIT, June 1996.

[6] Rains, D. A., "Tip Clearance Flows in Axial Flow Compressors and Pumps", California Institute of Technology, Hydrodynamics and Mechanical Engineering Laboratories, Report No. 5. 1954.

[7] Hunter, I. H., and Cumpsty, N. A., "Casing Wall Boundary-Layer Development Through an Isolated Compressor Rotor" , ASME J. Turbomachinery, 1982, Vol. 104.

[8] Chen, G. T., Greitzer, E. M., Tan, C. S., and Marble, F. E., "Similarity Analysis of Compressor Tip Clearance Flow Structure", ASME Paper No. 90-GT-153.

[9] Storer, J. A., and Cumpsty, N. A., "An Approximate Analysis and Prediction Method for Tip Clearance Loss in Axial Compressors", ASME Paper No. 93-GT-140.

[10] Khalid, S. A., "The Effects of Tip Clearance on Axial Compressor Pressure Rise", Ph.D. thesis, MIT, February 1995.

[11] Nikolaou, I. G., Giannakoglou K.C., Papailiou K.D., "Study of a Radial Tip Clearance Effects in a Low-Speed Axial Compressor Rotor" ASME Paper No. 96-GT-37.

[12] Khalsa, A. S., "Endwall Blockage in Axial Compressors", Ph.D. thesis, MIT, June 1996.

[13] Smith, L.H., Jr., "Casing Boundary Layers in Multistage Axial Flow Compressors", Flow Research on Blading, L.S. Dzung ed., Elsevier Publishing Company, 1970.

[14] Mikolajczack, A. A., "The Practical Importance of Unsteady Flow", *Unsteady Phenomena in Turbomachinery.*, AGARD CP-144, North Atlantic Treaty Organization, 1977.

[15] Hetherington R., and Morritz R. R., "The Influence of Unsteady Flow Phenomena on the Design and Operation of Aero Engines", *Unsteady Phenomena in Turbomachinery.* AGARD CP-144, North Atlantic Treaty Organization, 1977.

[16] Tzeng, Y. S., "The Effect of Rotor-Stator Interaction on Rotor Performance', GTL Report (unpublished), MIT Gas Turbine Laboratory, December 2000.

[17] Mailach, R., Sauer, H., and Vogeler, K., "The Periodic Interaction of the Tip Clearance Flow in the Blade Rows of Axial Compressors", to appear in Proceedings of ASME Turbo Expo, New Orleans, June 2001.

[18] Bae, J., "Active Control of Tip Clearance Flow in Axial Compressors", Ph.D. thesis, MIT, June 2001.

[19] Vo, H. D., "Role of Tip Clearance Flow on Compressor Stability ", Ph.D. thesis, MIT, September 2001 (expected).

[20] Denton, J. D., PROGRAM UNSTREST, Version UNSTSS13, June 2000.

[21] Denton, J. D., "The use of a distributed body force to simulate viscous effects in 3D flow calculations", ASME Paper No. 86-GT-144.

[22] Denton, J. D., and Dawes, W. N., "Computational Fluid Dynamics for Turbomachinery Design", IMechE paper, Vol 213, Part C.

[23] Stauter, R. C., Dring, R. P., and Carta F.O., "Temporally and Spatially Resolved Flow in a Two-Stage Axial Flow Compressor", ASME J. Turbomachinery, April 1991, Vol. 113.

[24] Valkov, T. V., and Tan, C. S., "Effect of Upstream Rotor Vortical Disturbances on the Time-Averaged Performance of Axial Compressor Stators: Framework of Technical Approach and Wake-Stator Blade Interactions", ASME J. Turbomachinery, July 1999, Vol. 121.

[25] Shapiro, A., *The Dynamics and Thermodynamics of Compressible Fluid Flow.* The Ronald Press Company, New York, USA, 1953

[26] Wisler, D. C., "Core Compressor Exit Stage Study", Volume IV – Data and Performance Report for the Best Stage Configuration, NASA CR-165357, NASA LEWIS Research Center, 1981.