Characterization of Unsteady Loading Due to Impeller-Diffuser Interaction in Centrifugal Compressors

by

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B.S., University of Missouri (2009)

Submitted to the School of Engineering in partial fulfillment of the requirements for the degree of Master of Science in Computation for Design and Optimization at the MASSACHUSETTS INSTITUTE OF TECHNOLOGY

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Abstract

Time dependent simulations are used to characterize the unsteady impeller blade loading due to impeller-diffuser interaction in centrifugal compressor stages. The capability of simulations are assessed by comparing results against unsteady pressure and velocity measurements in the vaneless space. Simulations are shown to be adequate for identifying the trends of unsteady impeller blade loading with operating and design parameters. However they are not sufficient for predicting the absolute magnitude of loading unsteadiness. Errors of up to 14% exist between absolute values of flow quantities. Evidence suggests that the \( k - \epsilon \) turbulence model used is inappropriate for centrifugal compressor flow and is the significant source of these errors.

The unsteady pressure profile on the blade surface is characterized as the sum of two superimposing pressure components. The first component varies monotonically along the blade chord. The second component can be interpreted as an acoustic wave propagating upstream. Both components fluctuate at the diffuser vane passing frequency, but at a different phase angle. The unsteady loading is the sum of the fluctuation amplitude of each component minus a value that is a function of the phase relationship between the pressure component fluctuations.

Simulation results for different compressor designs are compared. Differences observed are primarily attributed to the amplitude of pressure fluctuation on the pressure side of the blade and the wavelength of the pressure disturbance propagating upstream. Lower pressure side pressure fluctuations are associated with a weaker pressure non-uniformity at the diffuser inlet as a result of a lower incidence angle into the diffuser. The wavelength of the pressure disturbance propagating upstream sets the domain on the blade surface in which the phase relationship between pressure component fluctuations is favorable. A longer wavelength increases the domain over which this phase relationship is such that the amplitude of unsteadiness is reduced.

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Nomenclature

Abbreviations

CFD  Computational Fluid Dynamics
ETE  Euler Turbine Equation
HCF  High Cycle Fatigue
PIV  Particle Image Velocimetry
PS   Pressure Side
SS   Suction Side

Symbols

\( A \)  Area
\( c_p \)  Specific heat capacity at constant pressure
\( L \)  Impeller blade loading per unit area
\( \bar{L} \)  Time averaged component of loading fluctuation
\( L_f \)  Amplitude of loading fluctuation over time
\( L_\phi \)  Measure of the effect of phase angle on \( L_f \)
\( \bar{m}_{cor} \)  Corrected mass flow rate
\( N_d \)  Number of vanes in stationary (diffuser) row
\( N_i \)  Number of blades in rotating (impeller) row
\( P \)  Static pressure
\( \bar{P} \)  Time averaged component of static pressure
\( P_1 \)  Mechanism 1 static pressure component
\( P_2 \)  Mechanism 2 static pressure component \\
\( P_f \)  Amplitude of static pressure fluctuation over time \\
\( P^{ps} \)  Static pressure on pressure surface of impeller blade \\
\( P^{ss} \)  Static pressure on suction surface of impeller blade \\
\( q_i \)  Solution variable on \( i \) surface \\
\( r \)  Impeller radius \\
\( t \)  Time \\
\( T \)  Time period of fluctuation \\
\( T_{t1} \)  Stage inlet total temperature \\
\( u_\theta \)  Tangential velocity component \\

**Greek** 

\( \alpha \)  Flow angle \\
\( \alpha_v \)  Diffuser vane angle \\
\( \bar{\alpha}_i^l \)  Momentum averaged incidence angle \\
\( \lambda \)  Acoustic wavelength \\
\( \pi \)  Stage total to total pressure rise \\
\( \tau \)  Stage total to total temperature rise \\
\( \phi \)  Phase angle of fluctuation \\
\( \rho \)  Density \\
\( \omega \)  Diffuser vane passing radian frequency \\
\( \Omega \)  Rotation speed of rotor in radians per second
Chapter 1

Introduction

1.1 The Centrifugal Compressor

The centrifugal compressor is a mechanical device that increases the static pressure and stagnation enthalpy of a fluid stream. Figure 1-1 depicts the two main components of a centrifugal compressor: the rotating impeller and the stationary diffuser. Within the impeller passage the total pressure of the fluid is increased by two dominant means. A centrifugal force in the stream-wise (radial) direction increases the static pressure, and the flow is accelerated in the tangential direction. Downstream of the impeller is the diffuser which recovers additional static pressure from the high tangential velocity through a diffusion process.

The primary alternative to the centrifugal compressor is the axial compressor. Unlike the centrifugal compressor, the axial uses a diffusion process to achieve a static pressure rise in both its rotating and stationary components. This diffusion process is depicted in figure 1-2. In a reference frame locked to the blade row fluid is turned to a more axial direction. This is perceived by the fluid as an area increase and the static pressure rises through diffusion. The degree of diffusion that can be achieved in a single stage is limited by the flow separation on the suction side of the airfoil. Centrifugal stages can achieve higher pressure ratios per stage because the centrifugal impeller does not rely on diffusion alone, but also on a centrifugal force to increase static pressure. A high performance centrifugal machine can have
Figure 1-1: Schematic of typical centrifugal impeller and vaned diffuser from Krain [11].

pressure ratios as high as 10 to 1 [10, pg. 426] for a single stage, while a typical axial compressor would require around 9 stages to achieve the same pressure rise [6, pg. 48]. Drawbacks to the use of centrifugal compressors include an efficiency reduction of about 4-5% [10, pg. 425] and a larger frontal area which is undesirable in high performance aircraft applications in which it increases drag.

Centrifugal compressor designs are selected when cost is a key performance metric. With the ability to achieve high pressure ratios in single stages, centrifugal compressors can be designed with fewer stages than axial compressors for the same overall pressure ratio. Fewer stages in a design reduces complexity, part count, and cost. An inhibitor to low cost is high cycle fatigue (HCF) of compressor blades which constitutes a major concern to design engineers. The focus of the research described in this thesis is on the high frequency forcing function due to impeller-diffuser unsteady interactions that is the cause of HCF in impeller blades.
1.2 High Cycle Fatigue

High cycle fatigue is the reduction of mechanical strength due to a high number (greater than $10^3$) of stress cycles [3, pg. 265]. Centrifugal impeller blades experience fluctuating stresses at high frequencies during vibration. There are two types of blade vibration: flutter and forced. Flutter vibrations occur when there is a dynamic instability in the interaction between the flow field and blade displacement. Forced vibration is due to an unsteady flow field created by interaction between adjacent blade rows with relative motion. For example, wakes from an inlet guide vane are perceived as an unsteady inlet flow to a rotating impeller directly downstream. Another type of interaction occurs when adjacent rows are in close enough proximity to interact with each others potential field. This research will focus on force vibration due to interaction with the potential field from a downstream blade row. This forcing occurs at a frequency equal to the engine rotational frequency times an integer multiple of the blade count. This product can be on the order of $10^5$ per minute which
classifies this as HCF.

The amplitude of blade vibration can be greatly enhanced under resonant conditions. Any blade geometry has a set of modes each with a discrete natural frequency and displacement pattern. When a forcing frequency matches a particular natural frequency the corresponding mode can be excited and resonance can occur. The amplitude of resonant response will be set by the blade material damping and correlation between the mode shape and the forcing function shape. A standard design practice to avoid resonant response is to use a Campbell diagram shown in figure 1-3. The blue lines are the natural frequencies of the blade and the red lines are integer multiples (usually blade count) of the engine operating frequency. At operating speeds which these lines cross, resonance can occur. While it is useful to know where resonance may occur, it is often hard to avoid all crossings over the operating range of a compressor. It is not known which resonant crossings are most important to avoid because the Campbell diagram incorporates no information about the correlation between mode shape and forcing shape. This research focuses on characterizing the forcing function distribution and magnitude to develop an understanding of when aeromechanic difficulty may occur.

1.3 Computational Tools

Early studies conducted on the flow field within turbomachinery were mainly analytical or experimental and focused on specific regions of the compressor that could be measured. Recently more studies have employed computational fluid dynamics or CFD to solve complex models of compressor flow. Increased use of CFD is due to advances made in algorithms for solving partial differential equations and availability of computer resources; these allow large scale computations of adequate accuracy to be performed in reasonable time frames. These simulations compute detailed flow quantities throughout the entire compressor domain and allow an in-depth interrogation of flow processes of interest. While CFD is a powerful tool it is still based on many assumptions and therefore must be assessed against experimental data.
Figure 1-3: Typical Campbell diagram used to predict operating speeds where resonance may occur [13].

1.4 Previous Work

Many studies have been conducted in an effort to understand the flow field within the centrifugal compressor. The following studies are those that have provided a conceptual and experimental base for this research to build on.

Shum studied the effect of impeller-diffuser gap size on stage performance [12]. He showed that as gap size is reduced slip is reduced, flow blockage is reduced, and tip leakage mass flux is increased. Performance increases with a reduction in slip and flow blockage but decreases with an increase in tip leakage flow. This led to the conclusion that the trend of compressor performance with impeller-diffuser gap size is not monotonic and there is an optimum gap size that maximizes stage performance. This gap size was shown to be close to 9.2% of the impeller radius which is small enough that significant unsteady interaction will occur between the impeller blade trailing edge and the pressure field of the diffuser vane leading edge. The stages examined in the present study have impeller-diffuser gap sizes close to or smaller...
than 9.2% of the impeller radius.

Gould performed numerical experiments using the commercial CFD code, CFX, to study the effects of impeller-diffuser interaction on unsteady blade loading [9]. He studied a stage with a small impeller-diffuser gap of about 4% of the impeller radius. He determined that there were three controlling parameters that set the unsteady loading: impeller-diffuser gap size, relative passage Mach number, and stage loading. Impeller-diffuser gap size sets the peak amplitude of unsteady loading, relative passage Mach number sets the spatial distribution of the pressure wave along the impeller blade, and an increase in stage loading was shown to increase the distance the unsteadiness propagated upstream in the impeller passage.

Smythe and Villanueva examined two nearly identical centrifugal compressor stages with impeller-diffuser gap sizes that differed by .55% of the impeller radius [13, 14]. Experimental measurements performed showed that for this marginal difference in impeller-diffuser gap size the amplitude of fluctuating blade stress was nearly a factor of two greater in the stage with the smaller impeller-diffuser gap. It was observed by Villanueva that there was a correlation between the increase in unsteadiness at the diffuser leading edge and the unsteadiness at the impeller trailing edge when the impeller radius was increased.

Gallier and Cukurel acquired unsteady flow measurements of a centrifugal compressor stage [4, 7]. These measurements include the steady static pressure field on the shroud side of the impeller-diffuser gap, unsteady static pressure field on the hub side of the impeller-diffuser gap, Particle Image Velocimetry (PIV) flow velocity measurements in the impeller-diffuser gap and diffuser vane passage, and stage performance. These detailed flow measurements in the vaneless space are used to assess computational tools by comparing metrics that are closely related to unsteady blade loading.
1.5 Technical Objectives

The objective of this study is to improve understanding of how compressor design and operating parameters set the aerodynamic forcing function responsible for unsteady blade loading. The approach taken is to first assess the adequacy of numerical simulations by comparing them to experimental data. Next, the numerical data is interrogated to explain the unsteady loading phenomenon. The experimental data acquired by Gallier and Cukurel and computational data from Smythe have been made available for this research. Smythe’s computational data was generated using the research code TURBO. This computational data was not assessed against experimental data because there was insufficient experimental data available on Smythe’s compressor. This study will use experimental data from Gallier and Cukurel to assess both Smythe’s primary computational tool, TURBO, and the primary computational tool of this study, Fine Turbo. Given the resources available and the knowledge base built thus far, three specific technical objectives are put forth.

1. Assess the adequacy of two computational tools, TURBO and Fine Turbo to compute the flow features important to unsteady blade loading due to impeller-diffuser interaction.

2. Identify the fluid mechanic mechanisms which set the distribution and magnitude of unsteady loading and their relative significance.

3. Explain the trends of the most significant mechanisms identified with operating point and stage geometry.

1.6 Thesis Contributions

The specific contributions of this thesis fall under two categories. The first are those pertinent to computational modeling of unsteady loading.

- CFD data is generated using the commercial code, Fine Turbo, and compared against experimental data. The comparison shows that computations are capa-
ble of capturing the trends of the measured flow field, but errors of up to 14% exist between absolute values of flow quantities. Evidence is put forth that an inadequate turbulence model being used is the cause for the significant errors.

- CFD data generated using the commercial code, Fine Turbo, is compared against the research code, TURBO. Good agreement is demonstrated between codes.

The second category of contributions made are those that add to the conceptual understanding of unsteady loading.

- Computations on two different stages with significantly different unsteady loading characteristics are compared. The two most significant reasons for the observed differences in unsteady loading are identified. The first is due to a difference in the magnitude of pressure fluctuation on the pressure side of the blade. The second is due to a difference in the wavelength of the pressure disturbance propagating upstream in the impeller passage.

- A correlation is demonstrated between the magnitude of the pressure fluctuation on the main blade surface and the strength of the non-uniformity of the diffuser inlet potential field. The strength of diffuser inlet pressure non-uniformity is set by the diffuser vane incidence angle. This indicates that the magnitude of unsteady loading can be influenced by the design and operating parameters that set the vaneless space flow angle.

- Gould [9] identified a phase difference between the pressure fluctuation on the pressure and suction sides of the blade. In this work the effect of this phase difference is quantified and shown to be significant. It is shown that an increased wave length of pressure disturbances propagating upstream reduces the peak unsteady loading and moves the location of peak unsteadiness upstream.

1.7 Thesis Outline

This thesis is organized as follows.
Chapter 2:
This chapter describes the technical approach used to accomplish the research objectives. A description of the research articles and computational tools is presented along with the motivation for choosing them. Lastly, the design of experiment is presented.

Chapter 3:
In this chapter Fine Turbo and TURBO are assessed as computational tools. Fine Turbo results are compared to experimental measurements. TURBO results are then compared against Fine Turbo results on a different compressor. The capabilities and limitations of each code are identified.

Chapter 4:
This chapter compares the static pressure field of multiple different compressors. The unsteady loading is quantitatively characterized. The two most significant effects responsible for the differences in unsteady loading between compressor designs are identified.

Chapter 5:
This chapter addresses the two effects identified in chapter four. The flow processes responsible for the differences are correlated with compressor design parameters.

Chapter 6:
This chapter summarizes the key findings of this thesis and discusses recommendations for future work.
Chapter 2

Technical Approach

A computational approach is used in this study because of the difficulty in measuring detailed metrics that would characterize unsteady blade loading. To assess the adequacy of a computational approach simulations are compared against the vaneless space measurements acquired by Gallier and Cukurel. Computational data is then interrogated in detail to extract new information about unsteady blade loading.

2.1 Research Articles

This work investigates computational data from three different compressors. Each compressor has been studied in previous investigations of unsteady blade loading. This section describes relevant characteristics of each compressor.

2.1.1 Compressor A

The first compressor, referred to as compressor A, was studied by Gould. This compressor was initially studied because unsteadiness due to the impeller-diffuser interaction was detected as far upstream as the impeller blade leading edge. The impeller-diffuser gap of this stage is 15% of the diffuser passage width. This is the smallest impeller-diffuser gap of the compressors studied. The diffuser for this stage is a discrete passage diffuser. Ansys CFX was the computational tool used to study
this compressor. Computational results were compared against experimental data by Gould [9].

2.1.2 Compressor B

The next compressor was studied by Smythe and Villanueva. This compressor has two design variations which will be referred to as compressors B1 and B2. These compressors are of nearly identical design with the only difference being a marginally larger impeller radius in compressor B2. This compressor was studied to explain the high sensitivity of blade response to impeller diffuser gap size measured on a test rig. The gap to diffuser pitch ratio is about twice that of compressor A. The diffuser for these stages is a cambered vane. The research code TURBO was used to analyze these stages. In Smythe’s and Villanueva’s studies there was only a limited amount of test rig data available to compare with CFD results and thus it is still necessary to assess the adequacy of these computations before using them to draw conclusions on the trends of unsteady loading [13, 14].

2.1.3 Compressor C

The last stage to be interrogated is compressor C. This compressor is selected for study because unsteady experimental measurements were acquired in the vaneless space by Gallier and Cukurel. The source of the unsteady flow field is the interaction between the impeller and diffuser that occurs in the vaneless space. Availability of flow measurements in this region enables a direct assessment of the capability of computations to capture the unsteady flow field. In this work numerical computations are conducted on this stage using Numeca Fine Turbo. Compressor C has a vane wedge diffuser and a gap size similar to compressors B1 and B2 [5, 7]. Table 2.1 summarizes key design features of each compressor.
2.2 Numeca Fine Turbo

Numeca Fine Turbo is the primary computational tool used in this study. Fine Turbo solves the unsteady Reynolds-Averaged Navier-Stokes equations over a discretized approximation of the compressor domain. Fine Turbo uses a finite volume scheme that is 2nd order accurate in both space and time [1]. Turbulent flow is modeled with a k-ε turbulence model. This model is chosen to keep consistent with the model used by Smythe in her TURBO computations.

2.2.1 Phase-Lagged Boundary Condition

A particular advantage to using Fine Turbo is its ability to apply a phase-lagged boundary condition for circumferentially periodic geometries. The phase-lagged boundary condition assumes that all flow variables on the circumferential boundary of a blade passage are periodic with the passing of the adjacent blade row. This assumption allows for a significant reduction in computational effort to be achieved by only modeling one blade passage for each row. The phase-lagged boundary condition is depicted in figure 2-1 for two blade rows with spacing that differs by \( \frac{2\pi}{N_i} - \frac{2\pi}{N_d} \). \( N_i \) and \( N_d \) are the blade counts for the rotating and stationary blade rows respectively. Notice that the surface \( q_i \) at time \( t' \) is in the same position relative to the stationary blade row as surface \( q_{i+1} \) at time \( t' - \frac{1}{\Omega} \left( \frac{2\pi}{N_i} - \frac{2\pi}{N_d} \right) \). The relation for any flow variable on these surfaces can be written as:

\[
q_i(t') = q_{i+1} \left( t' - \frac{1}{\Omega} \left( \frac{2\pi}{N_i} - \frac{2\pi}{N_d} \right) \right) 
\] (2.1)
Figure 2-1: Depiction of phase-lag boundary condition. Surface $q_i$ at time $t'$ is set equal to surface $q_{i+1}$ at time $t' - \frac{1}{\Omega} (\frac{2\pi}{N_r} - \frac{2\pi}{N_d})$.

Chen [15] gives a general form of this relationship for both the rotating and stationary rows.

$$q_{i+m}^{\text{rotor}}(t') = q_i^{\text{rotor}} \left( t' + \frac{m2\pi}{|\Omega|N_r} + \frac{n2\pi}{|\Omega|N_d} \right)$$  \hspace{1cm} (2.2)

$$q_{i+m}^{\text{stator}}(t') = q_i^{\text{stator}} \left( t' - \frac{m2\pi}{|\Omega|N_r} + \frac{n2\pi}{|\Omega|N_d} \right)$$  \hspace{1cm} (2.3)

where $m$ and $n$ are integers of either sign. An implication of the phase-lag assumption is that flow features occurring at frequencies less than the blade passing frequency are not captured. Experimental data from Gallier shows that dominant frequencies are at or above the main blade passing frequency [7]. This indicates that the phase-lagged approximation is appropriate.

### 2.2.2 Computational Tool Selection

In this study computations are implemented on compressors C and B2, and the results are compared to computational data generated by previous researchers. While it
would be advantageous to use one of the same CFD tools used in previous studies. Fine Turbo is selected for use as a new code for several reasons. Fine Turbo is selected over CFX because CFX cannot implement the phase-lagged boundary condition\(^1\). Using CFX would either require modifying the compressor blade counts so that a periodic boundary condition could be used or resorting to a full annulus calculations. Fine Turbo was selected over TURBO due to fast convergence using a multigrid scheme and its far more robust operation. Appendix A gives additional details on why TURBO is not the primary computational tool of this study. Because results from different codes are compared, the method for assessing simulations must include a comparison between CFD codes.

### 2.3 Computational Domain

The computational grid of compressor B2 is the same grid used by Smythe. It is a structured H pattern grid with about 0.8 million nodes and 5 blocks. The computational grid used for compressor C is a structured H-O-H patterned grid with approximately 3.5 million nodes and 29 blocks. The grid structure of compressors B2 and C are shown in figures 2-2 and 2-3 respectively. The H-O-H pattern grid is comprised of O sections which are blocks with grids that are aligned with the blade surface and H sections which fill the rest of computational domain with a grid aligned with the flow path direction. The H-O-H grid structure allows better orthogonality at the blade surfaces where this is important to accurately capture boundary layers. A draw back to this grid structure is its complexity. H-O-H grids often contain a large number of blocks with non-uniform block orientation. This complicates the pre and post processing of computational data.

\(^{1}\)At the time of the CFD code selection the current CFX version was 13.0. In December 2011, CFX version 14.0 was released with a provision to apply phase-lagged boundary conditions [2].
Figure 2-2: Structured H grid of compressor B2.

Figure 2-3: Structured H-O-H grid of compressor C.
2.4 Computational Procedure

To initialize the unsteady computations, mixing-plane computations are performed first. The mixing plane approximation circumferentially mixes the flow exiting the impeller and then applies it as a circumferentially uniform inlet boundary condition to the diffuser [1]. This method reduces the computation to a steady one, and therefore requires less computational resources. Information about unsteady processes may not be gained from this type of simulation, however it is capable of approximating stage performance. The procedure for generating full unsteady solutions is to implement mixing-plane computations along a full speed line to compare with test rig data. Then a specific operating point of interest is chosen for implementing full unsteady calculations. The mixing plane solution at this operating point is used as an initial condition to the unsteady calculation.

2.5 Compressor C Experimental Data

The primary reasons for selecting compressor C as a research article is that there is a wealth of experimental data available in the vaneless space of the compressor. Data was collected by Gallier at three different operating points on two different speed lines (6 operating points total) plotted in figure 2-4. The 2 speed lines chosen are at 100% and 90% of design speed. The 90% speed line is chosen because a structural analysis shows a resonant crossing on the Campbell diagram at this speed [8]. Stage performance measurements include inlet and outlet total temperature, total pressure, and static pressure. Vaneless space measurements include steady pressure on the hub side and unsteady pressure measurements from the shroud side of the vaneless space. The steady and unsteady static pressure are measured via arrays of pressure transducers shown in figure 2-5. Particle Image Velocimetry (PIV) measurements were also acquired on the 90% speed line at the operating point indicated in figure 2-4. PIV measures the velocity field through an optical window on the shroud side of the compressor. Figure 2-5 shows the region that the velocity is measured. This data
Figure 2-4: Compressor C operating points where experimental data and computational data are available.

is available at 6 different spans and 10 different time delays during one blade passing.

2.6 Simulation Assessment

It is a technical objective of this research to assess both the TURBO computations performed by Smythe and the Fine Turbo computations performed on compressor C. An indirect method for assessing TURBO data is adopted because of the difficulty in implementing TURBO computations on compressor C. Fine Turbo computations are assessed against experimental data on compressor C. Once this comparison showed that Fine Turbo was capable of capturing features related to unsteady loading, Fine Turbo computations are run on compressor B2. TURBO results generated by Smythe are then assessed against Fine Turbo computations.

The comparison between Fine Turbo computations and experimental data is made at the operating point where PIV data is available. A series of mixing plane simulations are run to match the 90% speed line using the experimentally measured inlet
total pressure and temperature as boundary conditions to the computation. Each operating point along the speed line is achieved by adjusting the back pressure. Figure 2-4 shows that the computed pressure rise is about 10% greater than the measured value. An unsteady simulation is then run at an operating point midway between choke and stall to compare the unsteady flow in the vaneless space. This operating point is shown in figure 2-4. To compare computational and experimental operating points that have the same relative speed line locations (midway between choke and stall) a higher back pressure must be used in the simulation due to the 10% difference in the computed and measured pressure rise.

To assess TURBO computations on compressor B1 and B2 Fine Turbo is run at the near stall operating point labeled in figure 2-7 on compressor B2. This compressor and operating point is chosen because the unsteady loading is most significant.

## 2.7 Design of Experiment

The primary objective of this work will be to better understand the trends of unsteady loading with design parameters. When comparing two test cases it is desirable to only have one design parameter varied. With the need to assess computational tools with experimental data in this study, compressors were selected based on the availability of data and not their similarities or dissimilarities in design parameters. In addition to multiple design parameters being varied across compressors, data is only available at
operating points that differ across compressors. An argument must be made that the differences in unsteady loading characteristics across compressors are due to design parameters and not operating point. This is done by comparing unsteady solutions at multiple operating points on the same compressor. The features common across different operating points but different across stages are deemed the effects of design parameters. Reasoning is then put forth as to which design parameters are responsible for the results observed and why. Figure 2-6 shows the near stall operating point where unsteady data is shown for compressor A. Figure 2-7 plots the operating points where unsteady data is available on compressor B1 and B2.

2.8 Summary

The technical approach used to accomplish the research objectives stated in chapter 1 is described. To assess the ability of both Fine Turbo and TURBO to capture the flow features relevant to unsteady loading Fine Turbo computations are run on compressor C and compared against experimental measurements. Fine Turbo computations are
Figure 2-7: Operating points at which unsteady data is available in TURBO and Fine Turbo in compressors B1 and B2.

then conducted on compressor B2 and compare against TURBO computational data. Once the capabilities and limitations of CFD are determined the computational data from compressor A, B1, B2, and C are assessed to enable a general characterizations of unsteady loading.
Chapter 3

CFD Assessment

Two steps are taken to assess the capability of Fine Turbo and TURBO to capture the flow features relevant to unsteady loading. First, Fine Turbo unsteady results on compressor C are compared to the experimental data. Next, Fine Turbo and TURBO unsteady results are compared on compressor B2.

3.1 Fine Turbo Vs Experimental: Time Averaged

Stage performance metrics are compared to give a broad view of CFD’s capabilities and limitations in computing the flow field. This comparison is conducted at the data point on the 90% speed line where PIV data is available. Table 3.1 lists the time averaged total to total temperature ratio ($\tau$), total to total pressure ratio ($\pi$), and corrected mass flow ($\dot{m}_{cor}$). In each case the computed values are higher than the experimental. It is known that CFD is capable of computing stage temperature ratio accurately, therefore this error is interrogated further by comparing to the Euler Turbine equation (ETE). The ETE uses conservation of energy and angular momentum to relate the shaft work done by the compressor to the total temperature rise across the impeller [10].

$$\tau = 1 + \frac{\Omega r u_{\theta_2}}{c_p T_{r1}}$$

(3.1)
<table>
<thead>
<tr>
<th></th>
<th>Fine Turbo Unsteady</th>
<th>Experimental</th>
<th>% error</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\pi$</td>
<td>3.518</td>
<td>3.260</td>
<td>7.9%</td>
</tr>
<tr>
<td>$m_{cor}$</td>
<td>2.128 kg/s</td>
<td>1.864 kg/s</td>
<td>14.2%</td>
</tr>
<tr>
<td>$\tau$</td>
<td>1.569</td>
<td>1.489</td>
<td>5.4%</td>
</tr>
<tr>
<td>$u_{\theta 2}$</td>
<td>316.2 m/s</td>
<td>329.4 m/s</td>
<td>4.0%</td>
</tr>
<tr>
<td>$\tau$ from ETE</td>
<td>1.558</td>
<td>1.582</td>
<td>0.4%</td>
</tr>
<tr>
<td>% error from ETE</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.1: Computational and experimental measurements compared with Euler turbine equation.

$\Omega$ is the rotational speed of the impeller in radians per second, $u_{\theta 2}$ is the mass averaged tangential velocity at the impeller exit, $r$ is the radius at which $u_{\theta 2}$ is measured, $c_p$ is the specific heat at constant pressure, and $T_{t1}$ is the total temperature at the compressor inlet. To evaluate the ETE, knowledge of the vaneless space flow velocity is needed. Figures 3-1 and 3-2 are the Mach number and flow angle in the vaneless space from CFD and PIV data. $u_{\theta 2}$ can be computed directly from CFD data. For the experimental case, $u_{\theta 2}$ is estimated by taking the spanwise average of the flow quantities and using the mean acoustic speed of 411 m/s [8]. Table 3.1 shows values of $u_{\theta 2}$ and $\tau$ computed from the ETE for the CFD and experiment cases. The percent error between the CFD computed temperature ratio and the ETE equation is 0.7%, while the percent error between the experimentally measured temperature ratio and the ETE is 5.9%. This suggests error in either the outlet total temperature measurement or the PIV measurements. Decent agreement between the PIV measured quantities with computations in figures 3-1 and 3-2 gives evidence to the error being in the total temperature measurement.

Figure 3-3 compares time averaged static pressure at a distance 8% of the diffuser pitch (circumferential distance between diffuser vane leading edges) upstream of the diffuser inlet. The comparison shows that CFD captures the pressure distribution, but the computed performance is about 10% higher. The higher static pressure shown in figure 3-3 and the agreement in Mach number shown in figure 3-1 indicates a higher computed total pressure in the vaneless space and thus higher impeller efficiency.
Figure 3-1: Vaneless space Mach number versus span [8].

Figure 3-2: Vaneless space flow angle versus span [8].
3.2 Fine Turbo Vs Experimental: Unsteady

To determine the reason for the higher computed impeller efficiency, vaneless space Mach number is compared at discrete time instants. Figure 3-4 shows the measured and computed Mach contours at two different times. The jet-wake flow structure described by Dean [6] is seen from the PIV measurements. As flow enters the inducer, relative frame velocity is higher on the suction than the pressure side. The suction side high velocity fluid separates before exiting the impeller. Downstream of this separation the gradient of momentum across the impeller passage switches, with fluid momentum now being higher on the pressure side. When translating relative frame velocity to the absolute frame, the gradient of momentum across the impeller passage is reversed because the direction of blade backsweep is opposite to the direction of rotation. This results in the high momentum region leaving the suction side of the impeller in the absolute frame. While figure 3-4 indicates that the computations capture the jet-wake structure and the approximate Mach number magnitude, the distribution of Mach number is significantly different. Figure 3-5 plots the absolute
Figure 3-4: Comparison of absolute frame Mach contours between Purdue PIV data and simulation.

frame Mach number across the exit of one blade passage at one time instant. The mean Mach number is captured, however the gradient of Mach number from the suction to pressure side is significantly steeper in the experimental case. This gradient is set by the degree of separation that occurs on the suction side of the impeller blade. Flow in low momentum or separated regions can be highly sensitive to the turbulence model used. It is suggested that the k-ε turbulence model is under computing the degree of separation that occurs on the suction side of the impeller.

Additional evidence indicating the inadequacy of the k-ε turbulence model can be gained from figure 3-2. There is good agreement in flow angle near the hub surface, however toward the shroud the flow angle is lower in the experimental case. Lower flow angle on the shroud side is due to the separation of flow over the concave shroud surface. Computations under predict the degree of separation that occurs and thus compute a higher flow angle near the shroud.

From these observations it is suggested that there is significant error in the CFD solutions due to an inability to capture the turbulence sensitive flow processes that occur on the suction side of the impeller blade and shroud surface. Loss and the
The degree of separation is under predicted by CFD leading to the over prediction of vaneless space static pressure, stage pressure ratio, and corrected mass flow. Despite these significant offsets, the flow structure in the impeller passage and vaneless space is captured, thus the computational results are useful for determining the trends of unsteady loading with design and operating parameters.

### 3.3 TURBO Vs Fine Turbo Comparison

Fine Turbo computations are implemented at the near stall operating point of the B2 compressor. This operating point is chosen because the amplitude of unsteady loading is highest. Unlike the comparison with experimental data, flow quantities are available throughout the entire computational domain. Therefore a more direct comparison of the computation of unsteady loading can be made. The instantaneous loading is the difference between the pressure on the pressure and suction side of the blade. Figure 3-6 plots the instantaneous loading at each time step for Fine Turbo and TURBO and shows that there is good agreement between the two codes. Figure
3-7 plots the amplitude of the pressure fluctuation for each code on the pressure and suction side of the blade. Again, good agreement exists between the two codes. Fine Turbo computes a pressure side peak unsteadiness that is 3% of the dynamic head higher, however the distribution of unsteadiness is nearly identical. This result indicates that TURBO is subject to the same capabilities and limitations determined for Fine Turbo.

3.4 Summary

Fine Turbo and TURBO are assessed as computational tools for computing the flow features relevant to unsteady impeller blade loading. CFD is shown to over predict time averaged temperature ratio, pressure ratio, mass flow, and vaneless space static pressure by as high as 14%. The error between computed and measured temperature ratio is shown to be due to experimental error in the temperature measurement. An interrogation of instantaneous flow quantities gives evidence that the k-ε turbulence model currently used is responsible for the over prediction of stage pressure ratio, mass flow, and vaneless space static pressure. Despite the over prediction in absolute flow quantities the distribution of the flow is captured by CFD. It is thus concluded
that CFD is an adequate tool for computing relative differences in unsteady loading between different design and operating conditions. A comparison is made between TURBO and Fine Turbo that shows the near identical agreement. This indicates that TURBO is subject to the same capabilities and limitations determined for Fine Turbo.
Chapter 4

Characterization of Unsteady Loading

In chapter 3 it is shown that TURBO and Fine Turbo are adequate tools for computing relative differences in unsteady loading between different design and operating conditions. In this chapter the computed static pressure field of each stage is interrogated to characterize the unsteady blade loading. First the static pressure field is examined at discrete time instants to identify a set of unsteady loading mechanisms that are common to all stages. A quantitative analysis is then presented to measure the relative significance of each mechanism. Finally, the results of this analysis are compared across two of the compressors. The differences in unsteady loading between the two stages are explained in terms of the most significant mechanisms.

4.1 Unsteady Static Pressure Field

Figures 4-1, 4-2, and 4-3 are midspan static pressure contours at six different time instants for compressor A, B2, and C. T is the time period of one main blade passing. The static pressure field shown for compressor B2 is at the near stall operating point, however it is qualitatively representative to all other operating points on both B compressors. Static pressure fields for compressor B2 and C show there is a steady region far upstream in the impeller passage. The pressure field in this region is dom-
inated by the Coriolis force. The pressure varies with a near constant gradient in the circumferential direction with higher pressure on the pressure surface of the blade. Downstream of this is an unsteady region where the pressure distribution transitions from being dominated by the Coriolis force to the potential field of the downstream diffuser vanes. This is the region in which the flow mechanisms responsible for unsteady loading occur. This chapter will quantify the extent of this region and magnitude of unsteadiness on the blade surface. The static pressure distribution for compressor A differs from that of compressors B2 and C. There is no distinguishable steady region upstream (Gould shows there is significant unsteadiness all the way upstream to the blade leading edge [9]). In all compressors significant unsteadiness in pressure on the impeller blade tip is a result of the impeller blade passing through the diffuser vane potential field.

In examining figures 4-1, 4-2, and 4-3 two distinct mechanisms that contribute to unsteady loading can be identified.

1. Mechanism 1: As the blade sweeps through the non-uniform potential field of the diffuser, it experiences a high pressure when in close proximity to the diffuser leading edge stagnation region and a lower pressure when in between diffuser vanes.

2. Mechanism 2: As the blade passes the diffuser vane, pressure waves develop at the impeller blade trailing edge and propagate upstream.

Mechanism 1 is observable in all three compressors. This mechanism is elucidated in figure 4-2. When \( t = 5/12T \) the main impeller blade (marked by an M) is aligned with a diffuser vane and the static pressure around the impeller blade tip is relatively high. When \( t = 9/12T \) the impeller main blade is between diffuser vanes and the static pressure around the blade tip is relatively low. This observation suggests that the level of non-uniformity of the diffuser inlet potential field is proportional to the pressure unsteadiness on the blade surface.

Mechanism 2 is elucidated in figure 4-1. As the main blade passes by the diffuser vane leading edge at \( t = 5/12T \) a pressure wave forms. This pressure wave is marked
Figure 4-1: Static pressure contours at midspan of compressor A.
Figure 4-2: Static pressure contours at midspan of compressor B2.
Figure 4-3: Static pressure contours at midspan of compressor C.
by a black arrow and can be followed through each time instant as it propagates upstream along the blade surface. It is suggested that mechanism 2 superimposes on top of mechanism 1 as an acoustic wave would and increases the pressure unsteadiness on the blade surface. The strength of mechanism 2 appears to be different in each compressor. In compressor A wave structures are most visible, in compressor B2 they are visible but less significant, and in compressor C wave structures are almost indistinguishable.

4.2 Separation of Mechanism 1 and 2

Two mechanisms that contribute to unsteady loading are identified. The relative contribution that each mechanism makes to the overall magnitude of unsteady loading is now analyzed. This is useful for two reasons. First, the characteristics of different mechanisms are set by different design parameters. Knowledge of which mechanism is dominant indicates which design parameters the unsteady loading magnitude is most sensitive to. Second, the blade structural response is dependent on the correlation between the blade mode shape and the distribution of unsteady loading. Each mechanism has an inherently different distribution. Determining the dominant mechanism gives insight to the distribution of unsteady loading. In addition the effect of the phase difference between the fluctuation of these mechanisms on both the magnitude and distribution of unsteady loading will be considered. A process for separating these mechanisms based on their different spatial distributions will be demonstrated on compressor C. Results of this process will then be compared across different operating points and compressors.

The first step taken in delineating the contribution from each mechanism is to define the domain on the blade surface where the flow is unsteady. Pressure fluctuation, $P_f$, is the metric used to assess the magnitude of unsteadiness in the flow field and is defined as:

$$ P_f = \max_{time}(P) - \min_{time}(P) $$

(4.1)
Unsteady Region

Figure 4-4: Unsteady region is defined as area for which \( P_f \) is greater than 3% of the maximum \( P_f \) on that blade.

where \( P \) is the pressure at a point on the blade surface. Figure 4-4 is a representative plot of pressure fluctuation versus blade chord. The region of unsteadiness to be analyzed will be defined as the area of the blade surface for which \( P_f \) is greater than 3% of the peak \( P_f \) on that blade surface. The region of unsteadiness is highlighted in figure 4-4.

Next the pressure field is separated into two superimposed components that are both unsteady. One is the pressure field due to mechanism 1 and the other is the pressure field due to mechanism 2. A Fourier analysis is not suitable for separating these components because both mechanisms are associated with the same frequency (the diffuser vane passing frequency). Instead the spatial distribution associated with each mechanism must be utilized. The instantaneous mechanism 1 pressure distribution is monotonic along the blade chord. The instantaneous mechanism 2 pressure distribution can be characterized as spatially periodic along the blade chord. At each time instant the pressure over the unsteady region is fit to a monotonic function. This filters out the spatially periodic component of the pressure field and
is thus defined as the mechanism 1 pressure field or $P_1$. The difference between the actual instantaneous pressure and $P_1$ is the mechanism 2 pressure field or $P_2$. Figure 4-5 illustrates this method. The pressure profile along the blade chord is plotted at six different time instants. Over the region of unsteadiness the pressure due to each mechanism is plotted. Figure 4-5 shows that this method successfully extracts the upstream propagating pressure wave observed in figures 4-1 and 4-2 so that its contribution to unsteady loading can be measured individually.

The contribution of each mechanism to overall unsteady loading is measured by the peak to peak amplitude of fluctuation in $P_1$ and $P_2$. Each contribution can be measured on both the pressure and suction surface and is written as:

$$P_{1f}^{ps} = \max_{time}(P_1^{ps}) - \min_{time}(P_1^{ps})$$  \hspace{1cm} (4.2)

where the subscript denotes the mechanism and the superscript denotes which side of the blade. Gould showed that a phase difference between the pressure fluctuation on the pressure and suction side can significantly increase the loading fluctuation. The effect of this phase difference can be measured by noting that if $P_2^{ps}$ fluctuated exactly in phase with $P_1^{ps}$, and both $P_1^{ss}$ and $P_2^{ss}$ fluctuated exactly out phase with $P_1^{ps}$, the amplitude of unsteady loading would be the sum of amplitudes of the pressure fluctuation due to each mechanism. Therefore the difference between the sum of pressure fluctuations and the actual loading unsteadiness is the effect of the phase difference. This can be written as:

$$L_f = P_{1f}^{ps} + P_{2f}^{ps} + P_{1f}^{ss} + P_{2f}^{ss} - L_\phi$$  \hspace{1cm} (4.3)

where $L_f$ is the amplitude of unsteady loading and $L_\phi$ is a positive value that measures the effect of the phase difference. Figure 4-6 plots all the components of equation 4.3 in a layer plot for the compressor C main blade. The area above and below the x axis are $L_f$ and $L_\phi$ respectively. The four colored areas are the contributions of each mechanism to the overall loading unsteadiness.
Figure 4-5: Instantaneous pressure profile as a sum of mechanism 1 and mechanism 2 pressure fields.
Figure 4-6: Illustrative plot of loading unsteadiness as a sum of terms in equation 4.3.

4.3 Stage Comparison

For each compressor, the process described above is used to characterize the unsteady loading in terms of the individual contributions from each mechanism. The unsteady loading in compressor B2 at the near stall operating point is interrogated first to identify a set of baseline unsteady loading characteristics. Three comparisons are then made from this case. The first is a comparison with the near choke operating point to identify trends with stage loading. The second comparison is with the B1 compressor at the near stall operating point to determine trends with the marginal change in impeller radius. The final comparison made is with compressor C. Data sufficient to conduct this analysis is not available on compressor A, thus only cases from compressor B1, B2, and C are analyzed.

Figures 4-7 and 4-8 plot the unsteady loading as a sum of the terms in equation 4.3 at 10% (near hub) and 50% span of the B2 compressor at the near stall operating point. At 50% span the peak loading unsteadiness is 83% of the dynamic head. The peak loading occurs at 98% chord and the unsteadiness is attenuated by 80% chord. At decreased span the unsteadiness is significantly increased, but the distribution of unsteady loading is similar. Four important observations are true at both spanwise locations.
Compressor B2, Main Blade, Near Stall, 10% Span

Figure 4-7: Loading fluctuation as a sum of components in equation 4.3 for compressor B2 main blade at 10% span and near stall operating point.

Compressor B2, Main Blade, Near Stall, 50% Span

Figure 4-8: Loading fluctuation as a sum of components in equation 4.3 for compressor B2 main blade at 50% span and near stall operating point.
Figure 4-9: Loading fluctuation as a sum of components in equation 4.3 for compressor B2 main blade at 10% span and near choke operating point.

Figure 4-10: Loading fluctuation as a sum of components in equation 4.3 for compressor B2 Splitter blade at 50% span and near choke operating point.
Figure 4-11: Loading fluctuation as a sum of components in equation 4.3 for compressor B1 main blade at 10% span and near stall operating point.

Figure 4-12: Loading fluctuation as a sum of components in equation 4.3 for compressor B1 main blade at 50% span and near stall operating point.
Figure 4-13: Loading fluctuation as a sum of components in equation 4.3 for compressor C main blade at 10% span.

Figure 4-14: Loading fluctuation as a sum of components in equation 4.3 for compressor C main blade at 50% span.
1. The most significant contribution to unsteady loading is that from the pressure fluctuation components on the pressure side.

2. Of the 2 components of pressure fluctuation on the suction side, pressure fluctuation due to mechanism 2 is dominant.

3. Pressure fluctuation of each component is highest at the trailing edge and decreases monotonically upstream.

4. \( L_\phi \) is highest at the trailing edge and varies non-monotonically along the chord.

A comparison is now made between the stall and choke operating points. Figures 4-9 and 4-10 plot unsteady loading for the B2 compressor at the near choke operating point. At 50% span the peak loading unsteadiness for this case decreases to 67% of the dynamic head and occurs at 98% chord. With the decrease in magnitude of unsteady loading, the contribution from each mechanism decreases proportionally. Therefore the relative contribution each mechanism makes to the unsteady loading remains the same between the stall and choke cases. All the observations listed for the near stall operating point are also valid for the near choke operating point. This indicates that increasing stage loading increases the magnitude of unsteady loading, but the characteristics of unsteady loading do not change.

Next, a comparison is made between compressors B2 and B1. Figures 4-11 and 4-12 plot the unsteady loading for the B1 compressor at the near stall operating point. At 50% span peak loading unsteadiness is 60% of the dynamic head and occurs at 97% chord. Unlike the previous comparison some differences exist between the characteristics of unsteady loading. While the reduction in peak unsteadiness can be attributed to a reduction in contribution from all mechanisms, there is a more significant reduction in \( P'_{2f} \) and \( P''_{2f} \) at both spanwise locations. The first, third and fourth observations made on the B2 compressor still hold for the B1 compressor.

The final comparison made is between compressor B2 and C. Figures 4-13 and 4-14 plot the unsteady loading for compressor C. Observations 3 and 4 made on compressor B2 still hold for compressor C, but the magnitude of unsteady loading is
significantly lower. At 50% span peak loading unsteadiness is 32% of the dynamic head and occurs at 93% chord. Two reasons for this difference can be deduced from figures 4-14 and 4-8.

1. The pressure fluctuation on the pressure side, \( P_{1f}^{ps} \) and \( P_{2f}^{ps} \), are significantly lower in compressor C.

2. The length scale associated with the variation of \( L_\phi \) along the blade chord is longer.

The pressure fluctuations on the pressure side of compressor C are significantly lower than those in compressor B1 and B2 at both choke and stall operating points. This suggests that the difference is mostly due to compressor design and not operating point. At the blade trailing edge, both \( L_\phi \) and the contribution from all mechanisms are maximum. Of these two competing effects high \( L_\phi \) wins out. Moving upstream \( L_\phi \) decreases more rapidly than the contributions from each mechanism and the loading unsteadiness increases. The peak unsteadiness occurs at a location near the local minimum of \( L_\phi \) in most cases. Thus, a longer length scale of the variation of \( L_\phi \) moves the point of peak unsteadiness upstream and reduces the magnitude of peak unsteady loading. In the next chapter the two reasons identified for reduced unsteady loading in compressor C will be correlated with design parameters.

4.4 Summary

The unsteady static pressure field for three compressor stages, each of different design are studied. Two mechanisms which contribute to unsteady loading are identified. The first is the pressure fluctuation near the impeller blade trailing edge associated with the passing in and out of the high pressure region near the diffuser vane leading edge. The second mechanism is the pressure fluctuation due to a pressure wave initiated at the blade trailing edge propagating upstream along the blade surface. A method for measuring the significance of each mechanism is presented and used on compressors B1, B2 and C. It is shown that both mechanisms along with their
phase relationship make significant contributions to the overall loading unsteadiness. In the comparison of unsteady loading in each compressor, significant differences are observed between the B compressors and compressor C. These differences are attributed to a different overall pressure fluctuation on the pressure side of the blade and different length scales associated with the variation of the phase relationship between different mechanisms along the chord.
Chapter 5

Influence of Design Parameters on Unsteady Loading

In the previous chapter the unsteady loading was characterized as the sum of contributions from multiple mechanisms. The contribution from each mechanism was measured for compressor C, compressor B1, and two operating points on compressor B2. The comparison showed there is significantly lower unsteadiness in compressor C than all cases in compressor B. The difference is due to lower contributions from the pressure side unsteadiness and the longer length scale of $L_\phi$ variation. In this chapter the pressure side unsteadiness and the $L_\phi$ variation length scale are correlated with compressor design parameters.

5.1 Pressure Side Pressure Fluctuation

It was shown in the previous chapter that unsteady pressure on the impeller blade surface is due to the passing of the impeller blade through the non-uniform potential field of the diffuser inlet. In this section the diffuser inlet potential field is described and the strength of the non-uniformity at the diffuser inlet is quantified. A comparison is then made between the magnitude of pressure fluctuation on the pressure side and the strength of the diffuser inlet pressure non-uniformity.
5.1.1 The Vaned Diffuser

The purpose of a vaned diffuser is to recover static pressure from the tangential component of the velocity exiting the impeller. The diffuser vanes apply a torque in an opposite direction of the tangential velocity. This reduces the angular momentum of the fluid and increases the static pressure. Figure 5-1 shows the naming convention used for the diffuser pressure and suction surfaces and the sign convention for the inlet flow incidence angle. A positive incidence angle induces a higher pressure on the pressure surface and thus a torque opposite the direction of the tangential velocity. In this study the naming convention in figure 5-1 is used even when the pressure is higher on the suction side.

5.1.2 Diffuser Inlet Pressure Field

Figure 5-2 shows time averaged static pressure contours for compressor B and compressor C. The compressor B case is at the near stall operating point of the B2 variant, however the figure is qualitatively representative of both operating points on both variants. The pressure distribution associated with the compressor B and compressor C diffusers are significantly different. In compressor B there is high pressure on the pressure side of the diffuser leading edge and low pressure on the suction side. In compressor C the diffuser leading edge is loaded in an opposite orientation,
Figure 5-2: Time averaged static pressure at 50% span of the compressor B2 (top) and compressor C (bottom) diffusers

with higher static pressure on the suction side of the blade leading edge. To see the difference this makes on the strength of the pressure non-uniformity a line is drawn across the diffuser inlet. In both compressors this line intersects the stagnation point of the diffuser leading edge. However, only in compressor B does the line intersect the low pressure region on the suction side of the blade. This results in greater variation in static pressure across the diffuser inlet. This is shown in figure 5-3 where the time averaged static pressure across one diffuser passage inlet is plotted.

5.1.3 Diffuser Inlet Incidence Angle

A relationship has been shown between the pressure non-uniformity at the diffuser inlet and the loading on the vane leading edge. A connection is now made between the loading on the vane leading edge and the diffuser inlet incidence angle. Conserva-
tion of momentum states that the difference between the flux of angular momentum entering and leaving a control volume containing the diffuser leading edge is equal to the torque applied to the fluid by the diffuser vane leading edge. The inlet incidence angle sets the degree to which the flow direction or angular momentum is changed. Thus the loading on the diffuser vane leading edge is dependent on the inlet incidence angle. To compare the inlet incidence angle with vane loading it is appropriate to measure the incidence angle in a manner that conserves the angular momentum flux. This is done by computing the momentum averaged incidence angle, $\bar{\alpha}_i^d$, on the time averaged flow field.

$$
\bar{\alpha}_i^d = \frac{\int_A r \rho u_\theta^2 \tan(\alpha) dA}{\int_A r \rho u_\theta^2 dA} - \alpha_v
$$

(5.1)

$A$ is the area of the surface over which the flow angle is measured, $r$ is the radius, $\rho$ is the density, $u_\theta$ is the tangential velocity, $\alpha$ is the flow angle measured from the tangential direction, and $\alpha_v$ is the diffuser vane angle at the leading edge. The vane angle is measured from the suction surface of the vane because figure 5-2 shows that it

Figure 5-3: Time averaged static pressure across one diffuser passage inlet.
is the flow field on this side of the vane that interacts with the upstream impeller. For each case shown in figure 5-3 the momentum averaged incidence angle is computed on a surface midway between the impeller exit and diffuser inlet and plotted in figure 5-4. Comparing figures 5-3 and 5-4 shows that for the B compressors, as incidence flow angle increases, the minimum pressure near the suction side of the blade decreases and pressure variation increases. For compressor C the incidence angle and the pressure variation across the diffuser inlet are significantly lower. The same trend is seen between the diffuser inlet flow angle and the strength of the pressure non-uniformity at the diffuser inlet.

Figure 5-4 shows the inlet incidence angle is negative for all cases. This may suggest that both compressors should have negative loading at the diffuser leading edge. This is not the case for the B compressors because moving slightly downstream the diffuser vane becomes more radial and the incidence angle at this downstream
position effectively becomes positive. For reversed loading at the vane leading edge to occur the incidence angle must be substantially negative. Figure 5-5 shows a closer view of the time averaged pressure field at the diffuser leading edge in compressor B. It can be seen that very near the leading edge there is in fact a small region of reversed loading. The comparison of inlet incidence suggests that if the incidence were as negative in compressor B as it is in compressor C then compressor B would also have a substantial region of reversed loading at the diffuser leading edge.

5.1.4 Diffuser Inlet Pressure Non-uniformity Versus Blade Pressure Fluctuation

The strength of the pressure non-uniformity (the pressure variation) is measured from figure 5-3 by taking the difference between the maximum and minimum pressure across the diffuser inlet. This quantity is compared to the amplitude of pressure fluctuation on the pressure side of the blade surface. Figure 5-6 plots both the pres-
Figure 5-6: Correlation between pressure variation across diffuser inlet, time averaged, static pressure and amplitude of pressure fluctuation at 98% chord of impeller blade. Comparison is made at 50% span.

Pressure variation at the diffuser inlet and the amplitude of pressure fluctuation on the pressure side of the blade at a point corresponding to 50% span and 98% chord. This chordwise location is chosen because it is near the location where the peak unsteady loading occurs in all cases on compressor B. A strong correlation is shown between the two quantities. Table 5.1 shows that the ratio between the pressure fluctuation and pressure variation is between 0.34 and 0.51.

These results suggest that the pressure fluctuation on the pressure side of the blade in compressor C is lower due to a weaker pressure non-uniformity at the diffuser inlet. In turn this weaker pressure non-uniformity is due to a lower incidence angle into the diffuser. This observation links the loading unsteadiness with the operating and design parameters of the compressor that set the diffuser inlet incidence angle.
<table>
<thead>
<tr>
<th></th>
<th>Pressure Variation</th>
<th>Pressure Fluctuation</th>
<th>Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor C</td>
<td>0.912</td>
<td>0.308</td>
<td>0.338</td>
</tr>
<tr>
<td>Compressor B2 choke</td>
<td>1.229</td>
<td>0.546</td>
<td>0.444</td>
</tr>
<tr>
<td>Compressor B1 stall</td>
<td>1.218</td>
<td>0.550</td>
<td>0.452</td>
</tr>
<tr>
<td>Compressor B2 stall</td>
<td>1.283</td>
<td>0.658</td>
<td>0.513</td>
</tr>
</tbody>
</table>

Table 5.1: Pressure variation across diffuser inlet, time averaged, static pressure and amplitude of pressure fluctuation at 98% chord of impeller blade. Comparison is made at 50% span.

These parameters include the diffuser vane stagger angle at the leading edge, the impeller backsweep and the flow coefficient or stage loading. If these parameters are chosen such that the loading at the leading edge of the diffuser vane is low or negative the unsteady loading will be reduced.

5.2 Length Scale of $L_\phi$ Variation

In this section the difference in the length scale of the $L_\phi$ chordwise variation between compressors B and C is investigated. The contribution from each mechanism is examined at a single point on the blade to understand the effect of interaction between the fluctuations of each pressure component. Figure 5-7 plots each pressure component versus time for compressor C over one main blade passing at 50% span and 93% chord. The vertical black lines mark the time it takes for one diffuser vane to pass. In the previous chapter it was shown that the maximum unsteady loading and a local minimum of $L_\phi$ occur at 93% chord. Figure 5-7 shows that $P_{1}^{ps}$ and $P_{2}^{ps}$ fluctuate out of phase with $P_{1}^{ss}$ and $P_{2}^{ss}$ respectively. This explains the small value of $L_\phi$ at this chord location. The contribution of each phase difference is quantified by developing an analytical approximation for $L_\phi$.

Figure 5-7 shows that the fluctuation of each pressure component is periodic with the diffuser vane passing and can be closely approximated by a sinusoidal function. The pressure component fluctuation associated with mechanism 1 on the pressure and suction side at a given chord location are modeled as the real parts of the following
Figure 5-7: Pressure due to mechanism 1 (left) and 2 (right) versus time for each side of the impeller blade. Measured at 93% chord and 50% span.

expressions:

\[
P^{ps}_{1} = \bar{P}_{1} + P_{1f} \frac{1}{2} e^{i(\omega t + \phi_{1}^{ps})}
\]

\[
P^{ss}_{1} = \bar{P}_{1} + P_{1f} \frac{1}{2} e^{i(\omega t + \phi_{1}^{ss})}
\]

The pressure fluctuation due to mechanism 2 have approximately a zero mean so they can be modeled as the real parts of the following expressions:

\[
P^{ps}_{2} = P_{2f} \frac{1}{2} e^{i(\omega t + \phi_{2}^{ps})}
\]

\[
P^{ss}_{2} = P_{2f} \frac{1}{2} e^{i(\omega t + \phi_{2}^{ss})}
\]

The subscripts denote the mechanism, the superscripts denote the blade surface, and the over bar denotes the time mean component of pressure. \( \omega \) is the radian frequency of the diffuser vane passing and \( \phi \) represents the phase angle of each fluctuation. The instantaneous blade loading is the difference in pressure on the pressure and suction
The number of variables can be reduced by choosing \( \phi_1^{ps} \) as a reference phase angle and measuring all other phase angles from it.

\[
\phi_{\Delta 2}^{ps} = \phi_2^{ps} - \phi_1^{ps}
\]

\[
\phi_{\Delta 1}^{ss} = \phi_1^{ss} - \phi_1^{ps}
\]

\[
\phi_{\Delta 2}^{ss} = \phi_2^{ss} - \phi_1^{ps}
\]

Substituting in equations 5.8 to 5.10 and pulling out the time dependent terms gives the fluctuating component of loading.

\[
L = L + (P_{1f}^{ps} + P_{2f}^{ps} e^{i\Delta_2^{ps}} - P_{1f}^{ss} e^{i\Delta_1^{ss}} - P_{2f}^{ss} e^{i\Delta_2^{ss}}) \frac{1}{2} e^{i(\omega t + \phi_1^{ps})}
\]

Equation 5.11 shows that the amplitude of loading fluctuation is dependent on the amplitude of the pressure fluctuations on each side of the blade and their relative phase difference. Recalling from the previous chapter that \( L_\phi \) is the difference between the sum of the pressure component fluctuation amplitudes and the actual loading fluctuation amplitude, an analytical approximation for \( L_\phi \) is determined.

\[
L_\phi = P_{1f}^{ps} + P_{2f}^{ps} + P_{1f}^{ss} + P_{2f}^{ss} - L_f
\]

\[
L_\phi = P_{2f}^{ps} (1 - e^{i\Delta_2^{ps}}) + P_{1f}^{ss} (1 + e^{i\Delta_1^{ss}}) + P_{2f}^{ss} (1 + e^{i\Delta_2^{ss}})
\]

Equation 5.13 shows that \( L_\phi \) is the sum of the amplitudes of a set of three pressure fluctuations each multiplied by a factor. The factor is between zero and two and is a function only of the phase angle between that pressure component fluctuation and the reference pressure component fluctuation (\( P_1^{ps} \) in this case). This result can be
used to determine which term dominates the distribution of $L_\phi$.

Table 5.2 lists values for terms in equation 5.13 at the 93% and 98% chord in compressor C. These two locations correspond approximately to a local maximum and minimum of $L_\phi$. Figure 5-8 demonstrates the measurement of the pressure fluctuations and phase angles from the computational data. The difference between terms of equation 5.13 at the two chord locations indicates which mechanisms are responsible for setting the variation in $L_\phi$. The analytically approximated value of $L_\phi$ is shown to be close to the computed value. This indicates that the approximations are sufficient for explaining the variation in $L_\phi$ along the chord. The contribution to $L_\phi$ from the second term varies little between the two locations. The change in the value of this term is due only to a reduced contribution from mechanism 1 on the suction side because $\phi_{\Delta 1}^s$ is the same at both locations. Conversely, the difference in the first and third terms between the two locations is significant. This is mostly due to a change in phase angle of about 180° in both terms. This result shows that it is the variation of the relative phase angle of the mechanism 2 pressure components ($\phi_{\Delta 2}^p$ and $\phi_{\Delta 2}^s$) along the chord that is responsible for the $L_\phi$ variation.

A physical explanation for the variation of $L_\phi$ with chord is illustrated in figure 5-9. It was described in the previous chapter that the mechanism 2 pressure wave

<table>
<thead>
<tr>
<th></th>
<th>93% Chord</th>
<th>98% Chord</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{2f}^{p^s}$</td>
<td>.0795</td>
<td>.1241</td>
</tr>
<tr>
<td>$\phi_{\Delta 2}^s$</td>
<td>-48°</td>
<td>120°</td>
</tr>
<tr>
<td>$P_{2f}^{p^s}(1 - e^{i\phi_{\Delta 2}^s})$</td>
<td>.1263</td>
<td>.1862</td>
</tr>
<tr>
<td>$P_{2f}^{p^s}$</td>
<td>.0702</td>
<td>.0894</td>
</tr>
<tr>
<td>$\phi_{\Delta 1}^s$</td>
<td>120°</td>
<td>120°</td>
</tr>
<tr>
<td>$P_{1f}^{p^s}(1 + e^{i\phi_{\Delta 1}^s})$</td>
<td>.0351</td>
<td>.0447</td>
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<td>$P_{2f}^{p^s}$</td>
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</tr>
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<td>$\phi_{\Delta 2}^s$</td>
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<td>95°</td>
</tr>
<tr>
<td>$P_{2f}^{p^s}(1 + e^{i\phi_{\Delta 2}^s})$</td>
<td>.0015</td>
<td>.2027</td>
</tr>
<tr>
<td>$L_\phi$ analytically approximated</td>
<td>.0629</td>
<td>.4336</td>
</tr>
<tr>
<td>$L_\phi$ computed</td>
<td>.0878</td>
<td>.4082</td>
</tr>
</tbody>
</table>

Table 5.2: Comparison of terms from equation 5.13 between the 93% and 98% chord locations in compressor C. Measurements taken at 50% span.
propagates on top of the mechanism 1 pressure component. Figure 5-9 shows this condition at two different time instants half of a cycle apart. Over the half cycle the mechanism 1 pressure component goes from being at a maximum to a minimum value. In this same time the pressure wave propagates one half of a wavelength (λ) upstream because the frequency of both mechanisms are the same. Figure 5-9 marks two chord locations half a wavelength apart. At one location the mechanism 1 and 2 pressure components are in phase at all times and at the other they are out of phase at all times. Thus the length scale of the variation of $L_\phi$ along the chord is set by the length of the wave propagating upstream. This result is shown to be true for both compressor B and C in figures 5-10 and 5-11. The $L_\phi$ distribution is compared with the mechanism 2 pressure wave at one time instant. $L_\phi$ varies from a maximum to a minimum condition over one half of the wavelength for both stages.
5.3 Summary

Connections are drawn between compressor design parameters and the differences in unsteady loading observed in compressor B and C. The first significant cause of lower unsteady loading in compressor C is the lower pressure fluctuation on the pressure side of the blade. The amplitude of this pressure fluctuation is shown to correlate with the strength of the pressure non-uniformity at the diffuser inlet which is set by the flow incidence angle into the diffuser. This suggests that reducing the diffuser inlet incidence angle would result in lower blade loading unsteadiness. The second significant cause of lower unsteady loading in compressor C is due to the longer wave length of pressure waves propagating upstream in the impeller passage. These pressure waves were observed in both compressors to have a favorable phase angle for reducing the magnitude of unsteady loading at the blade trailing edge. The longer wavelength in compressor C resulted in a larger blade area over which this phase angle is favorable, and therefore a reduced unsteady loading.
Figure 5-10: Comparison of $L_\phi$ and the mechanism 2 pressure wave on the pressure surface at one time instant for compressor C.
Figure 5-11: Comparison of $L_\phi$ and the mechanism 2 pressure wave on the pressure surface at one time instant for compressor C.
Chapter 6

Summary and Conclusions

6.1 Summary

A numerical study is conducted to characterize the unsteady impeller blade loading due to the interaction between the impeller blade and the diffuser vane potential field in centrifugal compressor stages. Two computational tools, Fine-Turbo and TURBO are assessed to determine their capabilities and limitations. This is done through a comparison of computations with Particle Image Velocimetry (PIV) and static pressure measurements in the vaneless space. Computational results across different compressors and different operating points are compared to identify the trends of unsteady loading. The magnitude and distribution of unsteady loading are characterized as a sum of pressure fluctuation components on each side of the blade. The comparison of unsteady loading across different compressor designs reveals significant differences in the unsteady loading magnitude and distribution. The differences are explained in terms of changes in the sum and interaction of the pressure fluctuation components. Finally, design and operating point parameters are correlated with the contribution each pressure fluctuation component makes to the overall unsteady loading and magnitude.
6.2 Conclusions

The conclusions of this thesis fall under two categories, computational and fluid dynamic. The conclusions that contribute to the assessment of computational modeling of unsteady blade loading are as follows.

1. The assessment of Fine-Turbo against experimental data showed that Fine-Turbo was capable of capturing the trends of stage performance and vaneless space flow quantities. However there are significant errors in the computation of the absolute values of flow quantities. Therefore Fine-Turbo is an adequate tool for determining relative differences in unsteady loading between operating point and design, but it is not a reliable predictor of the absolute magnitude of unsteady loading. Evidence suggests that using a turbulence model more appropriate for flow in centrifugal compressors would reduce the observed differences.

2. A comparison between computational results from Fine-Turbo and TURBO shows that each code computes quantitatively similar unsteady loading. The capabilities and limitation identified for Fine-Turbo are true for TURBO as well. Despite similar results between codes TURBO is more difficult to implement. It is thus concluded that results from both codes are adequate for studying unsteady loading, however Fine-Turbo is the more appropriate tool for this research due to its ease of implementation.

The conclusions that contribute to the understanding of the fluid dynamic mechanisms responsible for unsteady blade loading are as follows.

1. The two most significant rationales for the observed differences in unsteady loading between two compressor stage designs are identified. The first is due to a difference in the magnitude of pressure fluctuation on the pressure side of the blade. The second is due to a difference in the length of acoustic waves propagating upstream in the impeller passage.
2. A correlation exists between the magnitude of the pressure fluctuation on the main blade surface and the strength of the non-uniformity of the diffuser inlet potential field. Increased incidence angle at the diffuser inlet increases the pressure non-uniformity. Thus the magnitude of unsteady loading can be influenced by the design and operating parameters that set the vaneless space flow angle.

3. The phase difference in pressure fluctuations due to pressure waves on the blade surface is shown to have a significant impact on setting the magnitude and distribution of unsteady loading. For cases in which the phase difference is favorable at the blade trailing edge, an increased wavelength reduces the magnitude of unsteady loading and moves the location of peak unsteady loading upstream.

6.3 Recommendations for Future Work

Three areas for future work on this topic are identified. The first area relates to computational aspects. The simulations performed in this study were shown to be adequate for making comparisons between different computational cases. However, comparison with experimental data shows that the absolute magnitude of unsteady loading is not accurately computed. Evidence is presented that the computation of unsteady loading is sensitive to the turbulence model used. This study was restricted to the one turbulence model available in TURBO. It would be beneficial to conduct a study to quantify the effect of turbulence modeling on the unsteady loading computation and to determine which models are most appropriate for capturing the flow processes responsible for this phenomenon.

The second area for future work relates to the correlation between design parameters and the unsteady forcing function. In this work two effects are identified and shown to be significant in setting the magnitude of unsteady blade loading. The design trade offs between these effects and other known effects is not addressed. An optimization study would be useful to determine the compressor design that minimizes blade loading unsteadiness given the presence of multiple effects.

The third area for future work is in defining the structural response of the impeller
blade to the forcing function characterized in this work. Cyclic blade stresses that lead to high cycle fatigue are the result of the blade vibratory response to the aerodynamic forcing function. To understand how design and operating parameters influence blade durability, a structural analysis of the blade response must be conducted.
Appendix A

TURBO as a Computational Tool

TURBO was initially selected as the primary research code for this study because it was used by Smythe to generate computational data on compressors B1 and B2. For her study there was insufficient experimental data available to assess her computations, so an attempt was made in this work to run TURBO computations on compressor C and compare against experimental data. Significant difficulties were encountered in implementing TURBO and eventually Fine Turbo was adopted as a primary CFD code. Because TURBO is a research code it lacks many of the features incorporated into commercially developed codes. Three specific challenges that were encountered in implementing TURBO are identified.

1. TURBO lacks the capability to read in generic file formats. The file formats read by the TURBO solver are not widely used among commercial software. This complicated transferring data between TURBO and pre/post-processing software. In several cases custom file converters were developed as a solution. This added both time and sources of error to the computational experiment.

2. TURBO provides a limited set of tools for input checking and convergence troubleshooting. Commercial solvers incorporate error messages or input checking routines to quickly guide users to the source of problems. While TURBO does incorporate some error messages, these were typically insufficient to identify why errors occurred. Finding the source of simple input errors often required
an investigation into the TURBO source code. This also added time and sources of error to the computational experiment.

3. The TURBO solver algorithm was observed to be less stable than Fine Turbo. All attempts to start simulations from uniform initial conditions in TURBO resulted in exploding solutions within only a few time steps. To initiate unsteady simulations in TURBO steady mixing plane simulations were generated in Fine Turbo and input as initial conditions. Unsteady simulations were eventually started, but convergence to a periodic state was never reached. Figure A-1 plots mass flow rate at the compressor inlet and exit versus main blade passings and shows a growing instability that eventually leads to back flow at the compressor exit. The unsteady pressure field after 10 main blade passings is examined to determine if the solution reached a state sufficient to compute unsteady loading. Figure A-2 compares the amplitude of unsteady loading of the preliminary TURBO solutions with the converged Fine-Turbo solutions. Differences in both the unsteady loading magnitude and distribution are significant.

Initiating unsteady computations was estimated to take 2 to 3 times longer in TURBO due to the first two challenges listed alone. Fine Turbo is determined to be the more appropriate tool for this work given the significantly higher effort required to use TURBO and the similar results produced by each code. The following are case in which TURBO should be considered as a suitable tool.

1. The source code to the TURBO solver was made available in this study. TURBO is an appropriate choice for work that requires modification of the source code solver.

2. TURBO computations on compressor B converged in Smythe’s work. While a specific reason for the instability in TURBO computations on compressor C is not identified, a significant difference between the computations on compressor B and C is the complexity of the grid. The H-O-H structured grid of compressor C may have contributed to the instability of TURBO computations. It is suggested that TURBO is more suitable for simpler H grid applications.
3. The k-ε turbulence model is the only turbulence model implemented by TURBO. TURBO should only be considered if this model is known to be adequate.
Figure A-2: Loading unsteadiness measured from preliminary TURBO solutions and converged Fine Turbo solutions.
Bibliography


