EXPERIMENTAL AND NUMERICAL STUDY OF BLADE FORCED RESPONSE IN A FULL-SCALE ROTATING TURBINE

A Thesis

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By

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ABSTRACT

The forced response of aircraft engine turbine blades has been studied with a careful combination of numerical predictions and experimentation to provide a data set and accompanying analysis of the coupled unsteady aerodynamics and structural response. The numerical component encompasses both aerodynamic and structural predictions of blade structural response due to unsteady flows that exist in a turbomachine. The unsteady aerodynamics through a turbine stage are modeled with a Quasi-3D Reynolds-averaged Navier-Stokes CFD solver (UNSFLO) and the structural response of the rotor blade is modeled with the 3D finite element commercial code ANSYS. The experimental component of the study involves full-scale rotating turbine tests of the actual engine hardware from a jet engine that has experienced forced response problems in service. A turbine rig is utilized in a shock tunnel facility where it is exposed to representative high temperature and pressure conditions while rotating at design corrected speeds. Blade flush-mounted surface transducers with high frequency response are used to record both the unsteady pressure loading and the structural response. Experiments are performed with a nominal spacing between the stator and the rotor and an increased gap between the components to analyze the change in the forcing function. Both aerodynamic and piezoelectric excitation techniques are used to study the
blade vibratory response with and without aerodynamic loading. This provides a measurement of the total damping and an accurate estimate of the contributions from structural and aerodynamic sources.

The results provide for the first time the coupled measurement of unsteady pressure and vibratory response of a high pressure turbine rotor blade due to the high pressure turbine vane flow interaction. Initial comparisons of the CFD predictions and the time-averaged pressure data showed close agreement, assuring that the overall flow conditions are met in the experiment. Similar comparisons of the pressure harmonic amplitudes showed good agreement at the midspan and moderate agreement near the tip of the blade. The results indicate a large decrease in the amplitude of the pressure harmonic when the distance between the rotor and the stator is increased by 10% axially, especially near the tip and on the suction surface.

The unsteady pressure field was analyzed prior to and during resonance. It was observed that during resonance, the vibration acts as a destructive interference with the vane wake forcing function. By comparing the results of the experiments with and without aerodynamic excitation, it was observed that for this turbine the aerodynamic damping was seen to be a large component of the total damping. Without the aerodynamic excitation around the blade the damping level decreased by as much as 10 times. In addition, a comprehensive study of the vibratory response and damping in vacuum from speeds of 0 to 20,000 RPM showed the significant effect of increased speed on the response. Overall the results provided a much needed data set for the study of turbine blade forced response that can be used to save the engine community development costs and reduce safety concerns in future aircraft engine designs.
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Above all else I would like to thank my family for undivided love and support throughout all my endeavors in life.
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NOMENCLATURE

Symbols

\( A \)  Amplitude of Displacement
\( a/a_0 \)  Percentage of Peak Response
\( \alpha_{\text{exit}} \)  Blade Exit Angle
\( \alpha_{\text{inlet}} \)  Blade Inlet Angle
\( c \)  Modal Damping
\( c_c \)  Critical Modal Damping
\( \zeta \)  Nondimensional Parameter for Peak Response Calculation
\( F \)  Amplitude of Sinusoidal Forcing Function
\( \phi \)  Phase Angle Between Excitation and Response
\( F_c \)  Centrifugal Force
\( i \)  Incidence Angle
\( k \)  Modal Stiffness
\( m \)  Modal Mass
\( m_b \)  Blade Mass
\( P \)  Measured Surface Pressure
\( P_{\text{inlet}} \)  Measured Static Inlet Pressure
\( \pi_i \)  Static to Static Pressure Ratio Across Turbine Stage
\( r_{\text{effective}} \)  Radius to Blade Center of Mass
\( t \)  Time
\( T_{\text{inlet}} \)  Total Inlet Temperature to Turbine Stage
\( \omega \)  Circular Frequency of Blade Oscillation (rad/s)
\( \Omega \)  Acceleration of Turbine Disk
\( \omega_r \)  Resonant Frequency (Hz)
\( x \)  Modal Displacement
\( \zeta \)  Critical Damping Ratio (c/c_0)
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>DAS</td>
<td>Data Acquisition System</td>
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<tr>
<td>EO</td>
<td>Engine Order Excitation</td>
</tr>
<tr>
<td>FEA</td>
<td>Finite Element Analysis</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
</tr>
<tr>
<td>HAR</td>
<td>High Aspect Ratio</td>
</tr>
<tr>
<td>HCF</td>
<td>High Cycle Fatigue</td>
</tr>
<tr>
<td>HPT</td>
<td>High Pressure Turbine</td>
</tr>
<tr>
<td>LAR</td>
<td>Low Aspect Ratio</td>
</tr>
<tr>
<td>LE</td>
<td>Leading Edge of Blade</td>
</tr>
<tr>
<td>PF</td>
<td>Flow Path Pressure Transducer</td>
</tr>
<tr>
<td>PR</td>
<td>Blade Surface Pressure Transducer</td>
</tr>
<tr>
<td>PS</td>
<td>Pressure Surface of Blade</td>
</tr>
<tr>
<td>SDOF</td>
<td>Single Degree of Freedom</td>
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<tr>
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<td>Strain Gage</td>
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CHAPTER 1

INTRODUCTION

1.1 General Background

The structural integrity of turbine blades has long been a major concern in the design of modern gas turbine engines. Since these blades operate at high rotational speeds in high pressure and temperature environments, the limit of material stresses in these small, lightweight components have contributed to the design limitations of these aircraft engines. The centrifugal stresses have been understood and predicted well, but there is still the lack of a thorough understanding for the multiple sources of unsteady forces on the blade. Forced vibration caused by unsteady blade loading when combined with the high steady loading over the course of operation can cause the blades to crack and fail structurally. The mechanism for these aerodynamically induced failures is called high cycle fatigue (HCF). This problem is not limited to gas turbine engines and can occur in any structural application with unsteady loading. A better understanding of HCF in gas turbine engines is needed to reduce development costs, increase operational readiness, and lower maintenance fees. This is especially true in modern day designs since aircraft manufacturers desire engines that are cost effective and will last a long time. Only in the past 10 years has significant attention been paid to investigate the
physical nature of HCF, as evidenced by several research programs that have been initiated, including the current study.

The problem of HCF is both a safety and economic concern that is mostly (but not limited to) aircraft engines, due to the high loading and low weight requirements. Particularly in single engine military aircraft the loss of a blade can mean the complete loss of the aircraft. Safety is slightly less of a concern in commercial aircraft, since they have multiple engines and a blade loss is designed to be contained within the casing, allowing for a secure shutdown of the affected engine. On the economic side it has been estimated that 90% of all HCF problems are uncovered in the development process, but the remaining problems have accounted for 25% of all engine distress events (El-Aini et al., 1996). Another study showed that the average engine design would exhibit 2.5 serious HCF problems during its lifetime. It was also discovered that maintenance costs for HCF related problems have accounted for 5% of the total maintenance bill. If a better understanding of HCF for aircraft applications is not soon met, its cost could be over 2 billion US dollars in the next 20 years (Kielb, 1999).

In order to begin to understand forced response excitation, the nature of the forcing function and the system damping needs to be identified. The forcing function can be from an upstream disturbance, as in vane or strut wakes, or inlet distortion, from wind gusts or flight maneuvers. The frequency of unsteady pressure loading on the blade may match a blade natural frequency and forced vibration can occur.

The current study focuses on the unsteady pressure loading on the rotor blade due to the preceding vane wake. Both the frequency and amplitude of the forcing function are needed to consider the importance of a particular excitation. The frequency of the
wake excitation is a multiple of the rotational speed of the engine. Consequently, many of the forcing frequencies of importance can occur at very high values due to the high rotational speeds of turbine disks. The damping is made up of both aerodynamic (coming from the motion induced unsteady pressure around the blade) and structural (due to both the frictional contacts and inherent material properties). The material or hysteretic damping is often times ignored because of its relatively low value when compared to the other forms. Once the physical nature of the forced vibration is understood, a prediction of a turbine blade structural response to the unsteady pressure loading can be obtained.

1.2 Methods for Solving Forced Response Problems

In the past, possible forced response problems have been identified by using the Campbell Diagram approach. Since a forcing function becomes important when its frequency approaches one of the natural resonant frequencies of the blade, a crossing of the two values can indicate a possible HCF problem. The Campbell Diagram is created when all the possible forcing frequencies are plotted as a function of engine rotational speed, an example of which is shown in Figure 1.1. A designer can run a free vibration finite element analysis to determine the structural modes of the blade as a function of centrifugal loading. The engine order (EO) excitation is simply the conversion of the number of occurrences per revolution into frequency. For instance, the 41 EO is attributed to the excitation coming from the 41 vane wakes. The designer can then tailor the crossings so that they occur at low speed or outside of the operating range.
Either the blade resonant frequency or magnitude of the excitation can be adjusted to fix this type of HCF failure. To change the modal frequency, the designer can alter the blade shape (thickness, chord, etc.) or add another constraint in the form of a shroud. The magnitude of the excitation can be adjusted by changing the geometry of the blade or the blade row configuration, such as increasing the axial gap between the vane and the blade. These types of changes usually come at the penalty to the aerodynamic design, but are necessary for a structurally sound engine design.
1.3 The Need for Experimental Data

The Campbell Diagram approach is the most commonly used method and is widely understood, but it only indicates the likelihood of a crossing. The magnitude of the excitation can not easily be seen without a better understanding of the fluid-structure interaction. With modern computing abilities significant work has recently been put into modeling the forced response of turbine blades due to the unsteady loading. Chiang and Kielb (1993) suggested a system of considering the forcing function and aerodynamic damping in two separate categories. This requires accurate prediction of the flow field and blade structural response. Only in the past 10 years have the computational models been able to capture the subtle effects needed to model the highly three-dimensional unsteady flow in a turbomachine. Recent progress in these computational fluid dynamic (CFD) codes has allowed for a better estimation of the flow field, including Abhari, et al. (1995) and Giles (1993). Obviously to verify the validity of the computational tools and obtain a better view of the physical process there is a strong need for experimental data. This is quite evident in recent experimental and computational studies by Dunn, et al. (1992), Abhari, et al. (1993), Weaver and Fleeter (1994), and Manwaring and Wisler (1993).

To extend beyond the standard Campbell diagram approach and to support modern analysis systems experimental data on the coupled unsteady loading and blade structural response is much needed. Recent advances in state of the art instrumentation allow for the accurate measurement of the unsteady flow and structural response. The problem with obtaining code validation is that experimental programs are very expensive, preventing each engine manufacturer from analyzing every turbine design.
experimentally. Much of the cost for experimental data is reduced by using a short
duration facility to perform experiments on a relatively inexpensive turbine stage, when
compared to a full-scale engine test.

There have been a few experimental studies dealing with forced response of
compressors and fans, as evidenced by the study by Manwaring et al. (1996), but there is
little data available for turbines. This is partially due to the fact that traditionally
compressors and fans have exhibited more severe structural response problems due to
the thinner and longer designs, or low aspect ratio (LAR) blading. There is clear
evidence that turbine concerns are becoming just as important with the advent of modern
LAR turbine blade design and the possibility of more sources of excitation.
Experimental work on the prediction of aerodynamic damping has been even more
limited. As far as the author is aware, there has not been a reported experimental study
of detail on the forced response of a turbine blade in a rotating turbine at matched engine
conditions, which is completed in the current investigation.

1.4 Scope of the Current Study

The current study encompasses a full forced response numerical prediction and
supporting experimental data both in structural and aerodynamic form. The numerical
prediction involves unsteady stage CFD results with associated finite element
predictions to obtain a forced response analysis. To obtain experimental data, a turbine
rig containing a full high pressure stage of 360° vanes and blades are put in a
representative engine environment of high pressure, temperature, and rotational speed.
Consequently the main focus of the experiment is to study the high pressure turbine
(HPT) vane wake loading and excitation of the HPT blade. The experimental portion of the program is performed in a short duration, or shock tunnel, facility located at The Ohio State University Gas Turbine Laboratory. Crawley (1981) pioneered the use of a short duration facility for similar types of studies in aerodynamic damping. The current facility has been used for 2 decades, garnering experience in using a shock tunnel facility as a source of heated and pressurized air for gas turbine engine studies, including Dunn (1992) and Dunn and Hause (1982).

Actual engine hardware, consisting of turbine vane and blade rows are used to make the experiments as realistic as possible. The turbine rig selected for these experiments is the high pressure turbine (HPT) stage of an AlliedSignal TFE 731-2, a small turbofan engine commonly found in Learjet aircraft. This particular turbine is an ideal candidate for this type of study due to the occurrence of many problems when the engine was put into service. It is speculated that these problems were attributed to HCF coming from the forcing function of the HPT vane wake, which excited the 2nd Torsion structural mode.

To complete the study a careful combination of experimentation and analysis are performed with the intent of achieving two main goals:

- develop an understanding of the forcing function, aerodynamic, and structural damping
- create a unique code validation database for aerodynamic and structural prediction tools.

In order to do this, two similar but separate sets of experiments are performed. The first is to measure pressure and structural response to an aerodynamic excitation. State-of-
the-art high frequency response transducers are used to record blade surface pressures and strain gages are used to record the blade vibration. The short duration facility is operated such that the matching of engine design conditions is obtained. This allows for the measurement of blade vibratory response at engine conditions for computation of the total blade damping. The second set of experiments is performed by spinning the rotor in an evacuated tank with excitation from piezoelectric crystals. This allows for the computation of the structural damping by itself, since the aerodynamic forces have been removed. The influence of an increased axial gap between the rotor and stator is also investigated by performing the aerodynamic excitation experiments twice (once at a nominal spacing and again at a 10% increase in spacing).

The results of this research program are intended to provide the gas turbine industry with a much needed data set for unsteady aerodynamics and vibratory response of a turbine at actual engine conditions. The use of common computational tools and realistic engine hardware is intended to make the study as accurate as possible in applying the results to real and important problems encountered in the design of gas turbine engines.
CHAPTER 2

NUMERICAL INVESTIGATION

An extensive forced response study would not be complete without both a computational prediction, using the latest methods and software tools, and the supporting experimental data. The numerical component of this study was performed not only to obtain a matching set of predictions to compare with the turbine rig results, but also to guide the implementation of the sensors for the experimental study.

Since the aeromechanical study involves the unsteady fluid dynamics around the blade coupled with its structural response, the analysis has to be composed of two separate investigations. In the first component a Quasi-3D Reynolds-averaged, Navier-Stokes CFD solver (UNSFLO) is used to study unsteady fluid dynamic interaction. In the second component a 3D finite element program (ANSYS) is used to study structural characteristics.

2.1 Computational Fluid Dynamics

A brief background on the particular solver used in this study is necessary to understand how the model replicates the experiments. Both the steady and unsteady results are shown for selected conditions.
2.1.1 Goals of the Prediction

The main goal for this prediction is to obtain the unsteady surface pressures on the blade, which allows for the identification of the forcing function. It is necessary to first obtain the steady pressure distributions. The computational prediction includes an investigation into how the axial vane/blade spacing affects the amplitude and phase of the time-dependant pressure and a study of the span-wise pressure distribution. Another goal is to achieve an understanding of the physical processes that contribute to the pressure distribution on the blade. This is possible by looking at the flow vectors and Mach number contours.

2.1.2 Background of Solver

UNSFLO is a quasi-3D, Reynolds-Averaged, unsteady multi-blade row Navier-Stokes code originally developed by Gies (1985). The code uses a two-dimensional model that combines both an inviscid and viscous grid for the computation of flow fields in both the inner fitted boundary region and the outer region. The third (span-wise) dimension is accounted for by the specification of a streamtube height thickness. The shear computations in the thin boundary layer around the airfoil are solved using an alternating direction implicit (ADI) method for the Navier-Stokes equations and an O-type grid. Likewise, the outer flow field is solved using Ni's Lax-Wendroff algorithm, Ni (1981), for the Euler equations and an H-type grid. The Baldwin-Lomax (1978) algebraic turbulence model is implemented in the viscous solution with the boundary layer transition specified by the user. The full stage calculations combine vane and blade grids conservatively through a line of cells where a special inviscid algorithm is
performed. For the unsteady computations, an innovative "time-tilting" routine is implemented without adding considerable computation time to perform the space-time transformation. Many more details on this code and its verification of use in similar applications can be found in Giles and Haimes (1993), Abhari et al. (1993), and Abhari and Giles (1995).

A typical solution process begins with the input of the blade and vane external coordinates, and the corresponding stream-tube height thickness', for the two-dimensional section of interest. Typically around 400 coordinate points make-up a complete model of one airfoil. The outer region is meshed with a combination of quadrilateral and triangular elements, while the inner region creates quadrilateral elements to encapsulate the entire boundary layer. Both inviscid and viscous solutions are run for the vane and blade individually to model each entity by itself. The two grids are then combined in a conservative manner by the use of shearing cells to form a steady, viscous solution of the entire stage. The final part of the simulation is to perform the viscous, unsteady, stage solution.

2.1.3 Computational Model

Before explaining the grid itself, a brief explanation of the turbine stage and run conditions is needed. The HPT stage includes 41 trailing edge cooled stator vanes and 78 un-cooled rotor blades. A summary of the turbine stage geometry can be seen in Table 2.1, showing that it is of a relatively small size and classified as a high aspect ratio design. The stage geometry was measured from the engine hardware and the flow angles were calculated using the CFD model.
<table>
<thead>
<tr>
<th>Number of Airfoils</th>
<th>Stator</th>
<th>Rotor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chord (in) 50% Span</td>
<td>0.82</td>
<td>0.61</td>
</tr>
<tr>
<td>Pitch (in) 50% Span</td>
<td>0.80</td>
<td>0.42</td>
</tr>
<tr>
<td>Aspect Ratio</td>
<td>1.69</td>
<td>2.04</td>
</tr>
<tr>
<td>Tip Radius (in)</td>
<td>5.5</td>
<td>5.5</td>
</tr>
<tr>
<td>(\phi_{\text{Outlet}}) (°) 50% Span</td>
<td>0.0</td>
<td>28.0°</td>
</tr>
<tr>
<td>(\phi_{\text{Outlet}}) (°) 85% Span</td>
<td>0.0</td>
<td>3.0°</td>
</tr>
<tr>
<td>(\alpha_{\text{Inlet}}) (°) 50% Span</td>
<td>67.0</td>
<td>-60.0°</td>
</tr>
<tr>
<td>(\alpha_{\text{Inlet}}) (°) 85% Span</td>
<td>67.0</td>
<td>-60.0°</td>
</tr>
</tbody>
</table>

*Angles are in the relative frame of reference.

Table 2.1 Turbine Stage Geometry

A grid refinement study was performed and it was determined that 100 grid points were required in the axial (horizontal) direction, while 30 were required in the radial (vertical) direction for each airfoil to achieve grid independence. For the boundary layer, 15 grid points through the thickness were used. This accounts for a total grid of around 15,000 elements for the full stage model. The boundary layer transition was set far upstream of the vane leading edge, making the calculation fully turbulent. Figure 2.1 shows the model with both the viscous O-grid and the inviscid H-grid used in the current study.
The numerical simulations were performed on a Silicon Graphics ONYX R8000 processor. The time needed to solve each step varied with the complexity of the calculation leading up to the full unsteady stage solution. Unsteady simulations typically took about 3 hours to solve, requiring around 10 periods and 15600 iterations to obtain a residual of $1 \times 10^{-4}$.

2.1.4 Results

As previously discussed, the numerical simulation provides a large sum of information on the flow through the turbine stage. Of primary interest in this study is the steady and unsteady pressure distributions on the blade.
Figure 2.2 shows the average pressure distribution at mid-span on the rotor blade for two different axial gaps between the vane and the blade. This prediction comes as expected, with the only disagreement coming near the leading and trailing edges where the grid resolution is not optimized. When the two axial spacings are compared it can be seen that the average pressure difference varies by less than 5% on the suction surface and 1% on the pressure surface. This agrees with the findings of Dunn et al. (1992) showing negligible differences both computationally and experimentally for similar axial gap comparisons. The steady pressure distributions on the vane show similar comparisons between the two axial spacings (Figure 2.3).

![Graph showing pressure distribution](image)

**Figure 2.2 Comparison of Rotor Steady Surface Pressure Distributions for Two Different Axial Spacings**
Figure 2.3 Comparison of Vane Steady Surface Pressure Distributions for Two Different Axial Spacings

The CFD simulations were also performed on a blade section near the tip at the 85% span location. UNSFLO is a quasi-3D code, so the solutions do not take into account the 3D flow fields near the end wall region. The results indicate a larger increase on the pressure surface than on the suction surface for the tip location.
Figure 2.4 Comparison of Blade Steady Surface Pressure Distributions for Two Different Span Locations

A more important comparison is made by looking at the harmonic surface pressure distributions. This is done by taking a Fast Fourier Transform (FFT) of the unsteady pressure calculations. Figure 2.5 shows the first harmonic pressure distribution at midspan for both the nominal and larger axial spacings. The larger amplitude of pressure for the closer spacing is clearly seen, since the decay in the wake for the larger spacing reduces the loading.
Figure 2.5 Comparison of First Harmonic Pressure Distribution for Two Different Axial Spacings at 50% Span

Another goal of the CFD prediction was to obtain a visual representation of the entire flow field. The flow vectors provide an indication of the directional and speed content of the flow for different inlet flow angles to the blade, as seen in Figure 2.6. The three different conditions shown represent the flow encountered during the test matrix of the experimental work. The areas of re-circulating flow on the pressure surface are seen for the negative incidence case of Figure 2.6b. There is a small area of flow re-circulation near the leading edge on the suction surface for the slight positive incidence case in Figure 2.6c. These flow vector plots provide useful information when analyzing the experimental results for different inlet conditions.
Figure 2.6 Flow Vectors for a) $0^\circ$, b) $-50^\circ$, and c) $+10^\circ$ Incidence on the Blade
Since a large portion of this forced response study involves the wake propagation from the vane to the blade it is necessary to visualize its propagation through the turbine stage. Figure 2.7 shows several passages of the blade passing through the vane wake in the form of entropy contours. It is evident that the blade "chops" through the wake with the leading edge of the suction surface receiving the impact. The wake can also clearly be seen in Figure 2.8, which shows the Mach number distribution of the steady-state, viscous solution for the larger axial spacing case. Figure 2.9 shows the steady-state pressure distribution for the same case.

![Image](image_url)

Figure 2.7 Entropy Contours Showing Wake Convection Through the Rotor at 50% Span with No Incidence and 50% Axial Spacing
Figure 2.8 Predicted Steady-State Mach Number Contours (increments of 0.05) at 50% Span with No Incidence and 50% Axial Spacing

Figure 2.9 Predicted Steady-State Pressure Contours (increments of 0.02) at 50% Span with No Incidence and 50% Axial Spacing
2.2 Finite Element Analysis

To complement the aerodynamic predictions it is necessary to also study the structural response of the rotor blade. This section is intended to focus only on the steady centrifugal stresses and free vibration response. This same convention used in the explanation of the CFD is used here, starting with the goals of this computation, then a background of the solver and computational model, and finally leading to the results.

2.2.1 Goals of the Computation

The finite element prediction is used as a tool to provide a comprehensive study of a variety of structural characteristics of a turbine blade. The first task is to examine the steady stresses due to the centrifugal loading. The second task is to perform a free vibration analysis, allowing for the classification of every resonant mode within the frequency range of interest. This computation also provides information about areas of high stress and how the relative magnitude of these stresses varies with frequency.

2.2.2 Background of the Finite Element Method

Industry uses finite element codes almost exclusively in the design process of analyzing turbine blades. The most common approach is to model only one blade (or a few blades) and assume that the rest of them are identical. Recently, many reduced order models have been developed to analyze the small geometric differences in blade manufacturing, called mistuning, but they were not available for the current study. It is usually too complicated to model the highly 3D shape of typical turbine blades with purely analytical results such as energy methods or plate/shell theory. The finite element method is relatively quick with the advent of today's fast computing ability and has been

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shown to provide accurate results in the past. Also, creating meshes has advanced rapidly with the solid model of a blade being readily available to the designer. For all of these reasons a single blade finite element analysis was chosen.

The finite element solver used for these computations was the common commercial code ANSYS. Version 5.3 and 5.4 were both used in combination, with no variation in the results.

2.2.3 Computational Model

The finite element solid model and mesh were originally created at AlliedSignal Engines. Figure 2.10 shows both the isometric view and top view of the entire blade model, including the airfoil, platform, and firtree attachment. It is composed of 3D structural solid elements (SOLID45 in ANSYS) with 8 nodes per element and 6 degrees of freedom per node. The entire model is made up of 6981 nodes and 4644 elements. All displacement directions are constrained at the top surfaces of the firtree attachment to simulate the disk mounting. The degree of freedom coupling between the airfoil and the platform is modeled with constraint equations (included as part of the AlliedSignal finite element model) to provide moment transfer capability in order to accurately replicate the platform movement with the airfoil. It is also important to note that the stress stiffening effect is included for the modal analysis so that the in-plane and transverse displacements are coupled. This is necessary with a thin structure since the bending stiffness is very small compared to the axial stiffness. The blade is made from a nickel-based super alloy.
As with the CFD simulations, the finite element simulations were conducted on a Silicon Graphics ONYX R8000 processor. The static solution, only analyzing the centrifugal and thermal stresses, took less the 2 minutes, while the free vibration solution took about 3 minutes to solve and extract the element results.

Figure 2.10 Finite Element Solid Blade Model with Mesh Lines
2.2.4 Results

Since a turbine blade has two distinct types of loading cycles when operating at engine conditions (centrifugal loading with thermal conditions and alternating vibratory loading), both steady and unsteady stress solutions are needed. The main concern for this particular study is the vibratory response, so the steady results are only presented for the sake of completeness. Also, the steady loading of turbine blades has been solved thoroughly in the past and is well known. Most of the discussion in this section focuses on the unsteady blade response.

2.2.4.1 Steady Condition

The stresses due to the centrifugal loading at high rotational speed and the thermal loading from the high temperature gases leaving the combustor are the primary boundary conditions used for the solution. The results of the analysis also showed the highest stresses in the firtree (about 100 ksi) where the blade is constrained. This is typical for a blade, since the steady stresses need to be lowest in the airfoil for a structurally sound design. Figure 2.11 shows the stress contours in the airfoil for a rotational speed of 26000 RPM and inlet temperature of 1265 °F. There is a relatively even distribution throughout the airfoil, which is also a typical design choice. A uniformly thin cantilevered beam under centrifugal loading has its highest stress near the support, varying parabolically to zero at the free edge. To minimize this stress distribution blades are designed to be thicker near the platform and thinner near the tip. Therefore, the relatively even stress distribution, with the maximum occurring at the base is logical.
2.2.4.2 Vibratory Condition

The steady stresses are very important, but in a high cycle fatigue study the alternating (or vibratory) stresses are the driving factor. These stresses, over a number of cycles, can cause fatigue failures when combined with high steady stresses. To get an impression of the vibratory response, a free vibration modal analysis is needed (without a forcing function). This entails calculating the natural frequencies and mode shapes of the blade.
Figure 2.12 Resonant Frequencies and Displacement Mode Shapes of the Free Vibration Simulation at 29,000 RPM

Figure 2.12 summarizes the results of the free vibration simulation at a rotational speed of 29,000 RPM and inlet temperature of 2600 °F. It can be seen that there are 5 modes in the frequency range of interest (1 to 20 kHz). Because this blade has a high aspect ratio geometry the modes are not closely packed, making each particular mode easier to define. Several of the modes exhibit tip flapping, having the nodal line near the trailing edge and maximum displacement at the tip. In particular, mode 5 (2nd torsion) shows a potential tip flap condition, meaning that the predictions indicate that a crossing with this mode on the Campbell Diagram should to be avoided.

To determine if a possible crossing is imminent, the initial conditions can be adjusted to create a computational Campbell Diagram. This can be done by applying the appropriate inlet temperature and rotational speed in a parametric study. The temperature is found by assuming a constant corrected speed (\( N_{\text{phy}}/\sqrt{T_o/T_{\text{ref}}} \)) in the
operating range of interest. It is important to determine how much of an effect the high temperature conditions have on the vibratory response. In the current experiments the blade temperature remains close to ambient even for runs with high inlet air temperatures (since data is taken for very short durations). Consequently it is necessary to show that the high temperature effects on the blade are small so that the experiment provides valid results. The fully computational Campbell Diagram with the appropriate inlet temperatures is shown in Figure 2.13.

![Figure 2.13 Computational Campbell Diagram from Finite Element Results](image)

Three engine order sources of excitation are plotted on the Campbell diagram to try and analyze where a potential hazardous crossing is located. Of crucial importance is the 41 EO corresponding to the HPT vane count excitation. This source of excitation
crosses modes 2 through 5, with a particularly important crossing occurring at a high speed point of around 27,000 RPM. Since this crossing occurs at high speed (or high stress) and mode 5 has the tip flap modal pattern, it would indicate a high likelihood of a fatigue problem. The 13 EO due to the 13 vane sectors that make up the entire 360° vane ring, along with the 3 EO from the support struts are also shown.

Another point to notice about the computational Campbell Diagram is the slope of the modal lines. When a turbine blade is exposed to the high temperature air, the material heats up over the course of operation, imposing thermal stresses in the blade. The resultant modal frequencies are consequently softened at higher speeds. The current simulation seems to indicate that the rotational stiffening is the more dominant force, allowing the frequencies to slightly increase with rotational speed. It can be concluded that the thermal softening effects are minimal and the ambient blade temperatures that are present in the experiment do not cause discrepancies with realistic conditions. It should be noted though that the author is hesitant to put complete faith in the ability of the finite element program to accurately predict the thermal softening with the imposed boundary conditions. A better analysis of the thermal effects could have been obtained by applying more accurate thermal boundary conditions, but this was felt to be unnecessary for the current study.

To analyze how different initial conditions affect the modal response, several adjustments to the boundary conditions were analyzed. The first parameter investigated was the constraint at the firtree attachment. Since this a complicated stick and slip friction contact, there are many ways that this constraint can affect the response. Although the nonlinear friction contact was not modeled, the constraints on the firtree
were adjusted. The two possible constraint situations which could occur, in addition to having all four firtree surfaces in contact as in the original model, are shown in Figure 2.14.

![Figure 2.14 Diagram of the Two Alternate Constraint Locations](image)

The first alternate constraint is to assume only opposite firtree surfaces are in contact. This is simulated by removing the constraints on the opposite firtree surfaces. The results indicate that the resonant frequency drops by around 5%. A second alternate constraint is to assume that only the top firtree surfaces are in contact. By removing the constraints on the bottom firtree this situation was simulated. Results for this case show a smaller resonant frequency drop of around 1%. Therefore, the simple simulations of different constraining situations indicate a relatively small change in the resonant response. However, the computations provide a range of expected resonant frequencies that can be used to determine if the experimental results fit within the boundaries of the predictions.
CHAPTER 3

EXPERIMENTAL SETUP

The experimental component of the current study combines well established methods with novel techniques to create a unique code validation database for aerodynamic and structural prediction tools. This chapter intends to explain the details of the how data was acquired and how the measurement program is applied to analyze forced response in a rotating turbine.

3.1 Overview of Experimental Methods

Since the goal of this study is to analyze the forced response of a turbine blade, a realistic simulation can only be obtained if actual engine hardware in a full-scale turbine rig is exposed to the proper aerodynamic and structural conditions seen during operation of the engine. By using a shock tunnel facility the representative temperatures, pressures, and rotational speeds can be replicated for a steady-state time frame of around 50 to 100 milliseconds. This allows for the acquisition of blade loading, vibratory response, and total damping (including aerodynamic and structural). To quantify the amount of blade damping it is necessary to separate the effect of the aerodynamic and structural components. The same facility can be used to spin the rotor in vacuum while providing another form of blade actuation, which in this case uses piezoelectric
ceramics. Without the aerodynamic forces around the blade, it is possible to quantify the amount of structural damping by itself at engine speeds. This second set of experiments provides the vibratory response and damping (only structural) to a prescribed excitation. By combining data from both sets of experiments a coupled fluid and structural interaction database for the analysis of forced response is obtained, as diagrammed in Figure 3.1.

![Diagram of Approach to Experimental Method](image)

Figure 3.1 Diagram of Approach to Experimental Method

3.2 Transducer Selection and Implementation

Three types of transducers were used in this experiment, including unsteady pressure sensors, strain gages for the structural dynamic measurement, and piezoelectric ceramics for blade actuation.

3.2.1 Measurement of Unsteady Pressure

There is a need for pressure transducers both on the surface of the blade and in the flow path to investigate the blade loading and monitor the axial pressure distribution.
through the turbine rig. The pressure sensors need to have a high frequency response to capture the unsteadiness and be able to operate under harsh conditions. Piezoresistive silicon wafer transducers made by Kulite Semiconductor Products, Inc. were chosen due to their superior characteristics and reliability in similar experiments. The surface mounted gages were model XCQ-062-100A gages with a 100 psi nominal range and a transducer frequency response of 650 kHz. In similar experiments, the induced strain in the wafer due to mechanical vibration has been shown to add minimal interfering signals (less than 2%) to the unsteady flow measurement, Manwaring et al. (1996). After having pockets and wire channels electro-discharge machined (EDM) into the blade surface, these gages were custom-built into each airfoil by Kulite. Given the small size of the blade (axial chord length of 0.8 inches) the installation of the sensors was challenging. A photo of the surface mounted pressure transducers with corresponding dimensions is shown in Figure 3.2.
In order to determine where the gages should be optimally installed a combination of the CFD results and the blade geometry was used. It was determined that there should be 2 radial locations to examine the three-dimensional variation in pressure. Two passages were created (one at 50% span and the other at 85% span) with gages on both the pressure and suction surface of neighboring blades. The chordwise distribution of gages was chosen by analyzing the surface pressure distribution results from the CFD predictions. The time-averaged distribution of Figure 2.2 on page 14 shows a relatively constant variation with chord except near the leading and trailing edges. The first harmonic pressure distribution of Figure 2.5 on page 17 similarly shows a steeper slope near the edges and a more constant variation in between the edges. With this distribution in mind and since the gages are relatively large compared to the blade chord, a relatively equal gage distribution was obtained with a concentration near the edges. A total of 18 pressure transducers flush-mounted over a sum of 4 blades completed the matrix of surface pressure measurements. Figure 3.3 shows the gage locations on the blades which are next to each other in the disk mount, creating two passages at different spans. Table 3.1 lists the corresponding measurement locations as a percent of the axial blade chord and the percent wetted distance at the corresponding span.
Figure 3.3 Pressure Gage Locations on the Rotor Blades at a) the 50% span passage and b) the 85% span passage

<table>
<thead>
<tr>
<th>Gage Number</th>
<th>% Axial Chord*</th>
<th>% Wetted Distance*</th>
<th>Gage Number</th>
<th>% Axial Chord*</th>
<th>% Wetted Distance*</th>
</tr>
</thead>
<tbody>
<tr>
<td>9</td>
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<td>-45%</td>
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<td>-95%</td>
<td>-90%</td>
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<td></td>
<td></td>
</tr>
</tbody>
</table>

*leading edge is 0% and negative percentages are pressure surface

Table 3.1 Pressure Gage Numerical Locations as a Percent of the Blade Axial Chord

The flow path transducers were model XCQ-062-600A with a 600 psi nominal range and a transducer frequency response of 1300 kHz. These gages came in a standard
0.064 inch diameter metal tubing that was 0.375 inches long. The distribution of these gages was determined such that the inlet, exit, and shroud pressures could be captured. Figure 3.4 shows a diagram of the turbine rig with the axial distribution of the static pressure sensors. The details of this rig are explained later in Section 3.5.

![Diagram of turbine rig with static pressure sensors](image)

Figure 3.4 Axial Distribution of Static Flow Path Pressure Sensors in Turbine Rig

### 3.2.2 Measurement of Structural Dynamics

To determine the blade response to both forms of excitation, precision metal foil resistive strain gages were necessary. Micro-Measurements model WK-06-062AP-350 gages were chosen for reliability and low cost. To determine the location of where the gages should be placed, a combination of the finite element results and blade geometry were used. It is desirable to place the gages in a location of high strain to obtain a clear signal. The mode shapes from the free vibration simulation (Figure 2.12 on page 26) showed a large deflection in the trailing edge tip for most modes. Also, placing gages at
the mid-span and base helps to confirm each of the particular mode shapes of interest. Ultimately, the curvature of the blade limited the application of strain gages to only the trailing edge, since there is too much curvature near the leading edge. A diagram of the strain gages on the blades with the locations as a percent of the entire span can be seen in Figure 3.5. The circumferential distribution of strain gages, along with the other instrumentation can be seen in Figure 3.6. Strain gages are labeled as SG31-SG39 (only the second numerical digit of this label is shown in the Figure 3.6). Similarly the blade-mounted pressure sensors are labeled PR1-PR18 (only the digit of this label is shown in Figure 3.6). The blade numbers correspond to the AlliedSignal Engines identification scheme and do not reflect the numerical location around the disk.

Figure 3.5 Diagram of Strain Gage Locations (Measured as Percent Span)
A total of 9 strain gages were used over the sum of 6 blades. Four of these blades had additional instrumentation, while the other two only had strain gages. This was done to quantify the change in response due to the addition of the other instrumentation. The distribution also allowed for a sampling of how different blades respond to the same actuation, or in other words a mistuning study.

3.2.3 Dynamic Actuation

In the absence of the aerodynamic excitation, it becomes necessary to utilize another technique that would allow the blades to be excited under vacuum. Without the air surrounding the blades the aerodynamic damping goes to zero, allowing for only the structural and material damping to limit the blade response. Typical forms of actuators used in experimental modal analysis include shaker systems, audio speakers, and impact hammers. These standard systems were ruled out because they could not easily be
implemented on a rotating turbine blade without interfering with the rest of the experiment. Recently, with a surge in high cycle fatigue concerns a new generation of experimental blade excitation studies of high frequency modes has emerged. Some possible alternatives that have recently been investigated are piezoelectric ceramics, magnetic exciters, or air flow interrupters. Flush-mounted piezoelectric ceramic actuators provide the best means for exciting the blades for the current study because of their functionality and the well-controlled frequency output over a wide range.

Early use of piezoelectric actuators began with Fabumni (1980) attaching them to blades for shroud interface studies. Rapid progress has recently been made in the manufacturing of this special material, allowing for a larger deflection per applied voltage. In fact, after the current study had begun, new actuators that provided almost twice the output per voltage went into production. Since they were not available at the inception of this research effort, standard piezoelectric wafers manufactured by Aura Ceramics, Inc. were customized for use in actuating a turbine blade.

Many of the current problems with using piezoelectric actuators stem from the existence of cross-talk between the high voltage excitation signals and the low voltage strain gage signals. In the past, experimentalists have powered the actuator, then shut off the power to watch the decay in the response, as evidenced by the recent work of Gordon and Holkamp (1998). In the current study these actuators were used to simultaneously monitor the response during actuation. It is based off the Master of Science Thesis work of a concurrent study at the same laboratory by Jeffers (1999). More details on the piezoelectric actuation technique can also be found in Jeffers, Kielb, and Abhari (2000).
Initially the placement of the actuators in the current study was intended to allow for a replication of a particular mode shape. This would have entailed placing the piezo in an area of high displacement. Since the blade is very small, thin, and highly curved in certain areas the placement became restricted to a relatively flat surface near the base of the trailing edge. In subsequent bench tests it was determined that placing the actuators at this location provided sufficient actuation for the study. As with the pressure transducers, EDM pockets were necessary for the piezo and lead wires. The gage was then constructed and glued flush to the surface of the blade as shown in Figure 3.7. Only 2 actuators were used, but the piezos are intended to also excite neighboring blades by vibration coupling through the disk.

Figure 3.7 Photo of Piezo Actuators and Strain Gages Mounted on Blades
3.2.4 Miscellaneous Other Transducers

Several other transducers were needed to ensure safe operation and make sure that the correct flow conditions were met. This includes sensors to record the tunnel and dump tank pressure and temperatures, and a 360 pulse per revolution encoder to monitor the speed of the turbine. Some other transducers were used more for monitoring the rig during the experiment rather than to record data during the run. This includes a standard piezoelectric accelerometer, mounted near the rear bearing housing, that was used to make sure the system vibrations remained under control. Also, a surface mounted RTD was installed near the rear bearing to study the bearing temperature.

3.2.5 Using the Piezoelectric Ceramics as Vibratory Response Sensors

The piezoelectric actuators that were originally intended to only provide the forcing function to the blades during the spin tests in vacuum can also be used as sensors. Since the ceramic creates a voltage when displaced they can theoretically be used as extra "strain" sensors during the aerodynamic excitation experiments. The problem with using them is that extensive bench studies would be needed to calibrate each sensor to determine how much voltage is created at a particular strain level. Since the primary concern in the current study is to analyze the damping level (which only requires the knowledge of the resonant frequency and a clean response), this problem does not take away from the usefulness of the sensor. The data obtained from the piezoelectric sensors is not presented in the current document and is left as a future task.
3.3 Overview of Data Acquisition Components

Due to the different types of excitation systems required in this study there are two separate schemes used to obtain data. For the piezoelectric excitation experiments, only the strain gage and accelerometer signals were recorded. The aerodynamic excitation method required the acquisition of the response from all types of sensors.

3.3.1 Piezoelectric Excitation and Strain Gage Acquisition System

Experimental control and data acquisition was accomplished through the use of an 18-bit Siglab data acquisition system from DSP Technology, Inc. The Siglab allowed for 16 channels of input, 8 channels of output, and a frequency range of 20 kHz. Figure 3.8 gives an overview of the data acquisition system. The software for operating the Siglab was written by DSP Technology, Inc. for the PC version of Matlab. To acquire data the software would activate an output signal of a user-defined amplitude and frequency. The resulting signal was then amplified and used to power the piezoelectric actuator mounted on the turbine blade. The vibratory response was sensed by a strain gage and passed through a preamplifier that served to complete the Wheatstone bridge and amplify the signal. The conditioned signal was then fed into the Siglab input channel where it was converted from an analog signal to a digital signal for storage in the computer.
3.3.1.1 Actuation Signal Techniques

The piezoelectric actuators can be powered with one of two forms of excitation. The first is to use a swept sine routine that provides an AC signal that can vary from 1-20 kHz with a wide range of resolutions. The signal-processing unit can simultaneously record and compute the transfer function of the blade response. This technique is useful for obtaining very accurate data in high noise environments, but is slow. For a 12.5 Hz resolution sweep from 1-20 kHz the run time is about 15 minutes. It is preferable to avoid long acquisition times because maintaining a high rotational speed for a long time period permits the bearings to reach excessive temperatures causing failure. A much quicker alternate method is to use a chirp excitation that provides a burst of all frequencies within a specified range. This method is slightly less accurate in the presence of high noise, but allows for quick data acquisition at high speeds. For a 6.25 Hz resolution chirp from 0-20 kHz the run time is about 2 minutes. Therefore, the chirp
excitation allows for the same data acquisition in less than 10 times the amount of time that the sweep method provides. Consequently, chirp excitations were used for almost all of the runs, although some runs were repeated with the sweep method to show that both forms of excitation indeed provide the same results.

3.3.2 Aerodynamic Excitation Acquisition System

The data acquisition system used for collecting data during the aerodynamic excitation experiments is very similar to that used for the spin tests in vacuum. The schematic shown in Figure 3.8 is the same, except that the data is collect through the fully programmable, computer controlled model 1402E amplifier, from DSP Technology, Inc. Data is digitally converted and stored on a Sun SPARCstation 5 at a sampling rate of 100 kHz. Signals from selected gages were also recorded on a Datalab system with a frequency response of 250 kHz, but the reduction of this data is left for future work. This acquisition system has been used for many previous experiments so further detailed information on the system will not be repeated at this time. Of course in these experiments data is obtained from all sensors including strain gages, pressure sensors, heat flux gages, and encoders.

3.4 Bench Tests

To prove the feasibility of this unique method and guide the full-scale experiments, several bench studies were performed. These studies allowed for carefully monitored control of the technique and the opportunity to quantify the signal to noise ratio of the piezoelectric actuation method.
3.4.1 Experimental Setup for Bench Tests

The components used for the bench tests are exactly the same as in the rotating experiments (see Figure 3.8) except that there is no need for a slip ring. A photograph of the entire acquisition system used for the bench studies is shown in Figure 3.9. To isolate the measurement from surrounding vibrations, the test specimen is clamped on a granite table supported by air shocks.

![Figure 3.9 Photo of the Experimental Components in the Bench Tests](image)

After the blades have been instrumented with the piezoelectric actuators and strain gages, they are mounted in a broach block. The mounting block is simply a large mass of heavy metal that has a broach machined into the top, replicating the actual rotor firtree mount. This type of assembly is usually readily available since it can be used for a wide variety of static tests on blades. Set screws on the opposite surface of the block
provide a means of simulating the centrifugal load on the blade mount by inducing stress in this area.

To determine the appropriate torque applied to the set screws for simulating a particular rotational speed a calibration of the applied force was performed. The applied force is simply the torque on the set screw times the lever arm. This linear force can be approximated by Equation 3.1, which is the relation between the centrifugal force, $F_c$, and the rotational speed in RPM, $N$, based on the blade mass, $m_b$ and the disk radius at the hub, $r_{	ext{effective}}$.

$$F_c = m_b r_{	ext{effective}} \left( N \frac{\pi}{30} \right)^2$$  \hspace{1cm} 3.1

Calibrating this relation experimentally required the use of a load cell attached to the broach block, as seen in Figure 3.10. By tightening the set screw to a measured torque with a standard torque wrench, allowing the spacer to come in contact with the load sensor, the force from the set screw could be recorded. After converting these forces to the rotational speed using Equation 3.1 a calibration curve could be plotted, as seen in Figure 3.11. The curve shows the expected relation that the torque is proportional to the speed squared. This is only an approximation to how the broach block replicates the rotating blade mounted in a disk, due to the multitude of uncertainty in the procedure and validity of the centrifugal loading relation. Since it was not necessary to obtain an extremely accurate correlation of the simulated loading, this simple calibration provided a quick and easy estimation for the bench studies.
The torque applied to the set screws can be calibrated with the corresponding simulated wheel speed of the engine. Figure 3.12 shows resonant peaks for three torque settings on the blade. The increase in resonant frequency and decrease in damping (or steeper peaks) with increasing wheel speed can clearly be observed. The actual damping
values were also calculated showing a clear and measurable difference when the 
simulated centrifugal load was adjusted.

![Mode 2, 1st Torsion](image)

**Figure 3.12 Change in Blade Structural Response With Increasing Simulated Centrifugal Load by Adjusting Torque on Broach Block**

It should be noted that one of the major disadvantages with using a broach block is the inability to accurately simulate rotational speed. A broach block can only impose stress in the shank of the blade, not in the airfoil. It is very difficult to obtain a reliable and repeatable torque measurement due to all of the uncertainties associated with using set screws to apply a distributed force. Also, due to the large stresses it is often times hard to reach the actual stress levels in the blade mount without yielding the threads on the bolts. It is for these reasons that the only accurate way to perform vibrational
experiments on turbine blades is to use a spin test facility that places the blade in a realistic rotating environment.

3.4.2 Signal Noise Reduction

By far the largest problem in data acquisition is in the many forms of noise contaminating the strain gage signals. The broach block experiments allowed for a comparative study of methods for reducing the noise and obtaining the best possible signals. The two main sources of noise are system noise, which comes from the vibration of the surrounding medium, and capacitive cross-talk, which comes from the piezo actuation power lines. The first source is inherent to the rig assembly and cannot easily be reduced, especially in the rotating experiments. This component also includes the noise from the rotating contact in the slip-ring. The results from the broach block experiments do not include a noticeable amount of system noise, since the tests are performed in a stationary reference frame. Consequently, it is impossible to use the bench results for determining methods of reducing the system noise.

The second source of noise, capacitive cross-talk, can be minimized by shielding the wires as much as possible. In this application it is impossible to completely shield the wires, since very fine gage leads are used on the blade surface to the disk. Also, in the rotating experiments it is difficult to shield through a slip ring without expensive modifications or using two separate slip rings. There is always a portion of wiring that has the high power lines in close proximity to the strain gage signals. For this particular experiment, neither customization nor multiple slip rings were available, so the only way to minimize the cross-talk in pre-processing is to separate the conflicting leads as much
as possible. By using a combination of shielding and keeping the power lines on separate connectors the noise was minimized to a point that it could be removed with a unique post-processing technique. Chapter 4 explains the data reduction necessary for this routine.

3.4.3 Initial Comparisons with Finite Element Analysis

To determine if the piezos are actuating the blades in the proper manner it is important to compare the initial bench test results with the finite element analysis. Table 3.2 summarizes a comparison of the resonant frequencies, \( \omega_r \), at a simulated speed of 10,000 RPM (a torque on the broach block equal to 60 in-lb) with the predicted results at ambient temperature and 10,000 RPM.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Computed Finite Element ( \omega_r ) (Hz)</th>
<th>Measured Bench Test ( \omega_r ) (Hz)</th>
<th>Percent Error in Prediction</th>
</tr>
</thead>
<tbody>
<tr>
<td>1, 2\textsuperscript{nd} Bending</td>
<td>4402</td>
<td>3930</td>
<td>12</td>
</tr>
<tr>
<td>2, 1\textsuperscript{st} Torsion</td>
<td>7407</td>
<td>7151</td>
<td>3.6</td>
</tr>
<tr>
<td>3, Coupled</td>
<td>12646</td>
<td>11470</td>
<td>10</td>
</tr>
<tr>
<td>4, 2\textsuperscript{nd} Bending</td>
<td>14703</td>
<td>15000</td>
<td>2.0</td>
</tr>
<tr>
<td>5, 2\textsuperscript{nd} Torsion</td>
<td>18769</td>
<td>18770</td>
<td>5.3e-3</td>
</tr>
</tbody>
</table>

Table 3.2 Comparison of Predicted Resonant Frequencies from FEA with Measured Bench Test Results at a Simulated 10,000 RPM in Air

The results agree well within the scope of the bench experiments, since it has already been mentioned that the simulated speed is not very accurate. It appears as though the largest predicted error is for modes 1 and 3. Table 3.2 proves that it is feasible to use the piezos to measure the resonant frequency of the mode shapes of
interest in the rotational experiments. Fortunately, it should be noted that the error in the prediction of mode 5 just happens to be extremely low.

3.4.4 Intrusiveness of Measurement Components

With any data acquisition system it is necessary to recognize the extent to which a transducer is altering the actual measurement. In the case of monitoring the structural response of blades with embedded piezoelectric actuators there are three main sources of intrusiveness: the reduction in resonant frequency due to the added mass of the actuator, the additional damping contributed by both the actuator and the strain gage, and the reduction in stiffness due to the removal of blade material for the flush-mounted actuator. By knowing the relation between the resonant frequency, \( \omega_r \), and the blade mass, \( m_b \), and stiffness, \( k \), the explanation of these sources can be explained (shown in Equation 3.2).

\[
\omega_r = \sqrt{\frac{k}{m_b}}
\]  

(3.2)

A set of auxiliary experiments was performed to analyze these effects and their individual contributions to the total intrusiveness. The first set of data was taken with a typical arrangement of one actuator, mounted near the tip of the blade on the pressure surface, and two strain gages. The same blade was then used to attach an additional actuator in the same location, but on the suction surface. Data was taken with this second configuration by powering one of the two actuators. The results were used in conjunction with the original configuration to quantify the effects of a single mounted actuator.
3.4.4.1 Added Mass Effects

The first conclusion from these auxiliary experiments confirmed that the attachment of the actuator reduced the resonant frequencies, following the relationship of Equation 3.2. The average resonant frequency drop for this range of modes was 150 Hz or 2%. This effect is obviously dependent on the size of the actuator with respect to the blade and the amount of glue used for attachment. This case represents an extreme scenario with a very small blade and relatively large actuator. In most cases, the additional mass of the actuator is negligible with respect to the total blade mass, showing little or no reduction in resonant frequency.

3.4.4.2 Added Damping Effects

The second source of intrusiveness was observed by again comparing the two sets of data from the auxiliary experiments. In the current study the total system damping is defined by the critical damping ratio, \( \zeta \), which is equal to the ratio of the damping coefficient, \( c \), and the damping coefficient when the system is critically damped, \( c_0 \). A small, but measurable difference in \( \zeta \) was seen, as evidenced by the summary in Table 3.3. It is obvious that for this case, the addition of the piezo actuator results in a higher increase in damping for the bending mode, when compared to the torsion modes. This can be explained by the highly contoured spanwise nature of this family of mode shapes. The second bending mode shape from Figure 2.12 on page 26 shows a good example of this near the tip of the blade. Again, this effect depends on the size of the actuator, but for most practical cases the added damping is very small, yet noticeable.
Table 3.3 Additional Damping due to the Installation of a Piezoelectric Actuator

<table>
<thead>
<tr>
<th>Mode</th>
<th>$\zeta \times 10^4$</th>
<th>$\zeta \times 10^4$</th>
<th>$\zeta \times 10^4$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st Torsion</td>
<td>3.17$\pm$0.31</td>
<td>4.76$\pm$1.80</td>
<td>+1.59</td>
</tr>
<tr>
<td>Coupled</td>
<td>7.29$\pm$0.16</td>
<td>9.63$\pm$0.23</td>
<td>+2.34</td>
</tr>
<tr>
<td>2nd Bending</td>
<td>10.75$\pm$2.50</td>
<td>16.91$\pm$1.30</td>
<td>+6.16</td>
</tr>
<tr>
<td>2nd Torsion</td>
<td>13.74$\pm$2.50</td>
<td>21.56$\pm$1.50</td>
<td>+7.82</td>
</tr>
</tbody>
</table>

3.4.4.3 Removal of Material Effects

The third source of intrusiveness is due to the removal of material for the transducers and the corresponding change in stiffness. This becomes important in blades that have had significant alterations made for the flush-mounted instrumentation. Two investigations were performed to study this effect. The first used a set of finite element solutions replicating the removal of blade material and addition of piezoelectric material. This is in contrast to the removal of mass simulations which did not take into account the removal of material. The results showed a small change in the blade structural response (a reduction in the resonant frequency of less than 5%) even for an exaggerated actuator size. This limiting case was for a thin pocket and inserted actuator that traversed the entire span of the blade. Since the actuator is so diminutive with respect to the blade dimensions, the actual removal of material is negligible in most cases. The reduction in the blade stiffness appears to be a more dominant component of the intrusive effects, since the decrease in resonant frequency is larger for the removal of material source.
The second investigation compared the vibratory response of a nominal blade to that of the two heavily instrumented blades in the rig experiments. Table 3.4 summarizes the responses of a nominal blade and two of the blades that have the instrumentation shown in Figure 3.6 on page 37. The results of first bending (mode 1) is not shown because it was not excited well enough by the piezo to obtain a clear vibratory response. There is as much as a 10% decrease in resonant frequency due to the removal of a significant amount of blade material. The variation in the amount of reduction is probably due to the change in the blade stiffness as a result of material removal.

<table>
<thead>
<tr>
<th>Mode 2</th>
<th>Mode 3</th>
<th>Mode 4</th>
<th>Mode 5</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1st Torsion</td>
<td>Coupled</td>
<td>2nd Bending</td>
</tr>
<tr>
<td>ω₁ (kHz)</td>
<td>ω₁ (kHz)</td>
<td>ω₂ (kHz)</td>
<td>ω₂ (kHz)</td>
</tr>
<tr>
<td>Nominal Blade</td>
<td>7.55</td>
<td>11.41</td>
<td>14.77</td>
</tr>
<tr>
<td>Blade 54</td>
<td>6.63</td>
<td>11.43</td>
<td>13.53</td>
</tr>
<tr>
<td>Blade 66</td>
<td>6.97</td>
<td>11.02</td>
<td>13.23</td>
</tr>
</tbody>
</table>

Table 3.4 Reduction in Resonant Frequency for Heavily Instrumented Blades

This set of auxiliary experiments showed that the addition of a piezoelectric actuator can reduce the frequency response and add some damping, but in most cases these effects are so small that they don’t significantly alter the intended measurement. It is important to identify where the intrusive effects are severe so that they can be taken into account when creating the test plan for the rig experiments and in data reduction.

3.5 Full-Scale Turbine Rig

The main portion of the current study is a full-scale test of engine hardware that is exposed to representative flight conditions. A turbine rig containing the actual vane
nozzles and rotor blades of the AlliedSignal TFE 731-2 high pressure turbine is used to complete the full-scale study. Several valuable studies have already been completed using the same turbine rig including novel heat transfer and pressure measurement studies of Dunn and Hause (1982) and Dunn (1986). All components of the rig were made available for the current study and AlliedSignal Engines contributed a new turbine disk with a complete set of rotor blades.

Figure 3.13 Diagram of Turbine Rig Used in Current Experiments

Figure 3.13 shows a diagram of the rig, which consists of a forward transition section followed by a circular opening that faces the primary nozzle flow. The airflow then enters a complete 360° annular passage containing the vanes and blades. The flow exhausts through an exit tube that contains an interchangeable orifice plate, changing the mass flow through the rig. The rig is designed such that the axial Mach number of the flow exiting the vane is subsonic. A slip ring is located inside the forward bullet nose to
transfer electrical signals from the rotating to stationary reference frame. An air motor (or impulse turbine), located in the aft bullet nose drives the turbine disk and slip ring. All wires, coolant, air supply, and exhausts are fed through the support struts. A photograph of the instrumented disk with mounted blades and a photograph of the fully assembled rig can be seen in Figure 3.14 and Figure 3.16 respectively.

Figure 3.14 The Instrumented Turbine Wheel and Blades

Figure 3.15 Fully Assembled Rig Outside of Tunnel Facility
Figure 3.16 Fully Assembled Rig Mounted in Test Facility

The rotor blade tip to shroud clearance at 0 RPM was set to 0.025 inches, which is 2% of the blade height. At near engine speed, the centrifugal growth is expected to be about 0.008 inches (Dunn and Hause, 1982), so the actual operating clearance is closer to 0.020 inches. The axial gap between the rotor and stator at the midspan was set to 0.265 inches for the first set of tests and increased to 0.345 inches for a second set of experiments. This results in respective axial spacings which are 32% and 46% of the vane axial chord.

3.6 Turbine Test Facility Operation

The experiments are performed in the Turbine Test Facility (TTF) at The Ohio State University Gas Turbine Laboratory. A number of similar experimental studies of unsteady flow in turbines have been performed in this facility, including Dunn, et al. (1992) and Dunn (1986). The facility is a shock tunnel device that provides a clean, uniform, and well-known gas dynamic condition to the turbine inlet. Figure 3.17 shows
the static pressure upstream of the turbine stage during a typical run. For a short
duration (on the order of 10’s of milliseconds), the experimental technique duplicates the
design corrected flow, physical rotational speed, and design pressure ratios. Dunn et Al.
(1989), provides the most extensive description of the facility.

Figure 3.17 Pressure Trace from Static Wall Tap Just Upstream of the Vane

Since many papers have already discussed the operation of this facility, only a
brief summary is provided in this document. To obtain the short duration source of
heated and pressurized air, a long tube is pressurized with a mixture of helium and air
(the driver tube). Another tube (the driven tube) is filled with the test gas (in this case
air) and pressurized to a lower pressure than the driver tube. The 2 tubes are separated by a small cavity enclosed on either side by 2 copper diaphragms, which have been prescribed to burst at a certain pressure. To start the run, this cavity is rapidly vented causing the diaphragms to burst. The resulting shock wave heats and pressurizes the working gas, which travels down into a fast acting valve and into the test section. The facility can also be used as a blowdown type wind tunnel by simply pressurizing the driver and the driven tube with air (no diaphragms present) and opening the valve. Some of the runs in the current study utilized a blowdown mode when room temperature air is needed to match the design flow conditions. Blowdown runs are much cheaper and quicker to use, since costly helium and scored diaphragms are not used.

3.6.1 Details of the Test Facility

Figure 3.18 is a photograph of the facility with the large dump tank, expansion nozzle, and long shock tube. The facility consist of an 18.5 inch inner diameter helium/air driven tube with a 40 foot long driver tube, a 60 foot long driven tube, a primary nozzle, and a 9 foot diameter by 34 foot long dump tank. The length of the tube allows for a sufficient test time without the reflected wave system from the end of the driver tube interfering. The dump tank is initially evacuated (to about 5 torr) to reduce the air loading on the spin-up impulse turbine and improve the flow establishment characteristics.
Figure 3.18 Turbine Test Facility at The Ohio State University

The turbine rig is housed in a support structure inside of the dump tank as shown in Figure 3.19. This provides the structural support to the turbine during operation. The rig was designed to sit on two saddle supports that are attached to I-beams mounted inside the dump tank. The turbine rig inlet is located in the nozzle at a position along the expanding flow such that a desired total inlet pressure is matched.
3.6.2 *Application of the Facility to the Study of Forced Response*

The short duration facility operates on the principle that the time scale of a fluid particle travelling through the machine is much smaller than the run time scale of the tunnel. If the flow time is greater than an order of magnitude smaller than the run time, the experiment can be considered as operating under quasi-steady state conditions. It is also necessary to ensure that the test time accounts for the damping out of the blade response to the initial startup pressure from the transient. After the startup impulse is damped out, then the rest of the run time provides the forced response excitation of interest. By looking at the lowest damped mode and data points from previous experiments, it can be shown that the impact of the initial ramp decays by 99% after 5 milliseconds. Since the TTF has a run time of around 50 milliseconds, a sufficient time window of steady flow is available for the investigation of the blade forced response due to the upstream wake excitation.
Another physical parameter that is necessary to check is the ability of the rapidly increasing excitation to allow the blade to reach the peak amplitude of the resonant response. Since the turbine experiences a swift acceleration as the valve opens and the air enters the rig, the blade response may only reach a small percentage of the resonant response during a constant speed excitation. By investigating the transient excitation of a single degree of freedom system, this percentage of the peak resonant response, \( a/a_0 \), can be written as a function of the acceleration rate \( \Omega \), the damping \( \zeta \), the engine order \( EO \), and the resonant frequency \( \omega_r \), by the following equation, Kielb, R. (1999).

\[
\frac{a}{a_0} = 1 - e^{-\sqrt{\chi}}
\]

\[\chi = \frac{(EO)\Omega}{\omega_r^2 \zeta^2} \tag{3.3}\]

By keeping \( \chi \) below 2, the transient response is more than 75% of the steady state resonant response. For the current study the expected values for \( \chi \) are between 1 and 1.5, which allows the excitation to reach 80% to 85% of the peak resonant response. This should be a sufficient amount for the purpose of the study, but the acceleration rates will be taken into account when the data is reduced.

### 3.6.3 Defining the Test Matrix

In order to properly study the forced response, it is necessary to understand the conditions that exist during excitation and how to replicate them in the rig studies. The major source of excitation in this study is the wake loading on the rotor blades. Since there are 41 vanes, in the reference frame of the rotor there is an unsteady pressure loading occurring 41 times per revolution (or the 41 EO excitation). This excitation is
plotted, along with the modal frequencies, as a function of speed on the Campbell Diagram in Figure 3.20. The modal frequencies represent the predicted values that are extrapolated to speed from the bench data (taken using the broach block) on the heavily instrumented blades. These lines correspondingly include the slight reduction of modal frequency due the installation of the transducers. Also plotted is the modal frequency line expected during operation of the engine. Since the blades experience an elevated material temperature in operation, a softening of the modal frequency is seen. In the current study there will not be a temperature softening effect as the material temperature rise in the short duration experiments is no more than a few degrees.

Figure 3.20 Revised Campbell Diagram for Turbine Rig
The Campbell Diagram can be used to find the crossing of a blade resonance with the wake excitation. In the range of interest there are 4 crossings where it might be possible to study rotor blade forced response. The mode 5 crossing is particularly interesting because this is the point at which the engine experienced blade fatigue failure. The goal of the current experiments is to match the conditions at these modal crossings. To replicate these conditions in the rig the following conditions must be matched: rotational speed, corrected speed, pressure ratio, and inlet temperature.

Using the Campbell Diagram as a starting point, the physical speed of the turbine can be found for each crossing. The piezoelectric actuation experiments in vacuum are then performed to get an even clearer estimate of the resonant frequencies with aerodynamic excitation. To determine the other conditions for the aerodynamic excitation experiments, the non-dimensional parameters of corrected speed, corrected flow, and pressure ratio are needed at the aerodynamic design point of operation.

The total inlet temperature, $T_{14}$, can then be calculated for each modal crossing using Equation 3.5, based on the turbine speed, $N$.

$$T_{14} = T_{14,design} \left( \frac{N}{N_{design}} \right)^2$$

3.5

Also, to match the aerodynamic design point, the ratio of inlet to exit pressure, $1/\pi$, across the stage remains constant. The magnitude of the inlet pressure is primarily arbitrary, but some caution is taken to maintain balance between values that are either too high or too low. If the pressure is very high the turbine accelerates too quickly and if the pressure is very low the transducers cannot resolve the electrical signal. Another consideration for setting the inlet pressure is where the rig can physically be installed in...
the expansion nozzle. A simple isentropic expansion calculation of choked flow allows for a plot of the axial pressure distribution through the nozzle based on the ratio of choke area to inlet area and the starting pressures that are feasible for the facility. Inlet pressures between 20 and 80 psia were chosen to provide a reasonable acceleration rate and clear pressure signals.

Once the inlet pressure is determined, the exit pressure, or essentially the mass flow, needs to be set so that the pressure ratio is matched. A standard ASME sharp orifice choke plate is used for this purpose. A similar isentropic expansion calculation, as was used in the inlet pressure calculation can be used to find the appropriate choke diameter. By relying on past data and knowledge of a discharge coefficient through the sharp orifice, the pressure ratio associated with various choke plate diameters is estimated. The run time pressure ratio is measured using the rig static pressure transducers (PF5 and PF10). Using a larger diameter orifice allows for more mass flow, yielding a higher exit pressure and lower pressure ratio. Slight modifications to the choke plates in between runs allowed for the best replication of the pressure ratio across the stage.

3.7 Summary of Experimental Runs

As mentioned in the previous section, the goal of the experiments is to match the modal crossings from the Campbell Diagram. The conditions and data acquisition details for the piezoelectric actuation experiments and the aerodynamic excitation experiments are summarized in Table 3.5 and Table 3.6, respectively. To excite mode 1, the temperature of the tunnel needs to be cryogenic, which was not possible to obtain.
The other natural modes of excitation were attempted using three separate classes of experimental conditions: shock tunnel runs at the matching high temperatures, blowdown runs at the matching ambient temperature, and blowdown runs with either positive or negative incidence on the blades. The crossing of modes 4 and 5 fall into the first class where the shock tunnel is needed to supply the high temperature air. The mode 3 crossing is in the second class since the matching inlet temperature is very close to room temperature. The mode 2 crossing fits into the final class since it was not possible to obtain the required inlet temperature of 226°F. For the mode 2 excitation the turbine stage operated with 10° of positive incidence on the airfoils. Several mode 4 crossings were also analyzed in blowdown, with 50° of negative incidence.

Seventeen total runs were completed with runs 1-9 completed for the nominal axial rotor/stator spacing and runs 10-17 being at the 10% increase in spacing. Unfortunately, the experiment for the mode 5 crossing was at such a high speed that the centrifugal loading caused the instrumentation leads to fail prior to the run. The different pressure ratios seen for the high incidence blowdown runs were used to somewhat compensate for the unmatched inlet temperature condition. By reducing the mass flow (i.e. using a smaller choke plate) the velocity diagrams could be altered to lower the flow incidence.
Table 3.5 Summary of Conditions for Piezoelectric Actuation Runs

<table>
<thead>
<tr>
<th>Speed (RPM)</th>
<th>Pressure (psia)</th>
<th>Freq. Range (kHz)</th>
<th>Resolution (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Atmosphere</td>
<td>0-20</td>
<td>1.25</td>
</tr>
<tr>
<td>0</td>
<td>Vacuum</td>
<td>0-20</td>
<td>1.25</td>
</tr>
<tr>
<td>5000</td>
<td>Vacuum</td>
<td>0-20</td>
<td>1.25</td>
</tr>
<tr>
<td>9300</td>
<td>Vacuum</td>
<td>0-20</td>
<td>1.25</td>
</tr>
<tr>
<td>12000</td>
<td>Vacuum</td>
<td>0-20</td>
<td>1.25</td>
</tr>
<tr>
<td>15000</td>
<td>Vacuum</td>
<td>0-20</td>
<td>1.25</td>
</tr>
<tr>
<td>19500</td>
<td>Vacuum</td>
<td>0-20</td>
<td>1.25</td>
</tr>
</tbody>
</table>

Table 3.6 Summary of Operating Conditions for Aerodynamic Excitation Runs

<table>
<thead>
<tr>
<th>Run</th>
<th>Type</th>
<th>Modal Crossing</th>
<th>Speed Range (RPM)</th>
<th>Approx. Incidence (°)</th>
<th>Inlet Pressure (psia)</th>
<th>Pressure Ratio</th>
<th>Axial Spacing</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Blowdown</td>
<td>M3</td>
<td>15500-17700</td>
<td>0</td>
<td>42</td>
<td>2.1</td>
<td>32%</td>
</tr>
<tr>
<td>2</td>
<td>Blowdown</td>
<td>M3</td>
<td>15900-17250</td>
<td>0</td>
<td>42</td>
<td>1.8</td>
<td>32%</td>
</tr>
<tr>
<td>3</td>
<td>Blowdown</td>
<td>M4</td>
<td>19500-19900</td>
<td>-50</td>
<td>42</td>
<td>1.4</td>
<td>32%</td>
</tr>
<tr>
<td>4</td>
<td>Shock</td>
<td>M4</td>
<td>19800-20700</td>
<td>0</td>
<td>81</td>
<td>2.1</td>
<td>32%</td>
</tr>
<tr>
<td>5</td>
<td>Shock</td>
<td>M4</td>
<td>19500-20200</td>
<td>0</td>
<td>46</td>
<td>2.1</td>
<td>32%</td>
</tr>
<tr>
<td>6</td>
<td>Blowdown</td>
<td>M3</td>
<td>15900-17500</td>
<td>0</td>
<td>42</td>
<td>2.0</td>
<td>32%</td>
</tr>
<tr>
<td>7</td>
<td>Blowdown</td>
<td>M2</td>
<td>9900-12100</td>
<td>+10</td>
<td>42</td>
<td>1.6</td>
<td>32%</td>
</tr>
<tr>
<td>8</td>
<td>Blowdown</td>
<td>M2</td>
<td>9000-9700</td>
<td>+10</td>
<td>20</td>
<td>1.6</td>
<td>32%</td>
</tr>
<tr>
<td>9</td>
<td>Shock</td>
<td>M4</td>
<td>17700-18600</td>
<td>0</td>
<td>45</td>
<td>2.2</td>
<td>46%</td>
</tr>
<tr>
<td>10</td>
<td>Shock</td>
<td>M4</td>
<td>17000-17100</td>
<td>0</td>
<td>12</td>
<td>2.0</td>
<td>46%</td>
</tr>
<tr>
<td>11</td>
<td>Blowdown</td>
<td>M3</td>
<td>15900-17800</td>
<td>0</td>
<td>42</td>
<td>2.0</td>
<td>46%</td>
</tr>
<tr>
<td>12</td>
<td>Blowdown</td>
<td>M4</td>
<td>20200-2100</td>
<td>-50</td>
<td>42</td>
<td>1.8</td>
<td>46%</td>
</tr>
<tr>
<td>13</td>
<td>Blowdown</td>
<td>M2</td>
<td>11900-12350</td>
<td>+10</td>
<td>42</td>
<td>1.6</td>
<td>46%</td>
</tr>
<tr>
<td>14</td>
<td>Blowdown</td>
<td>M2</td>
<td>9700-11800</td>
<td>+10</td>
<td>42</td>
<td>1.6</td>
<td>46%</td>
</tr>
<tr>
<td>15</td>
<td>Blowdown</td>
<td>M2</td>
<td>10000-11100</td>
<td>+10</td>
<td>21</td>
<td>1.6</td>
<td>46%</td>
</tr>
<tr>
<td>16</td>
<td>Shock</td>
<td>M4</td>
<td>19500-20250</td>
<td>0</td>
<td>82</td>
<td>2.1</td>
<td>46%</td>
</tr>
<tr>
<td>17</td>
<td>Shock</td>
<td>M4</td>
<td>19850-20200</td>
<td>0</td>
<td>46</td>
<td>2.2</td>
<td>46%</td>
</tr>
<tr>
<td>18</td>
<td>Shock</td>
<td>M5</td>
<td>26000-?</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Lost Instrumentation

66
CHAPTER 4

DATA REDUCTION

Most of the data reduction methods used in this study follow traditional techniques, like time averaging and using the Fast Fourier Transform (FFT). Some of the data requires unique reduction methods and a careful evaluation of system errors is needed. In particular, the data obtained from the strain gages during the piezoelectric actuation experiments requires several steps of reduction to resolve the needed computation of the blade damping. This includes removing the capacitive cross-talk and performing a fit of the data to a single degree of freedom model.

4.1 Post-Processing of Piezoelectric Actuation Strain Gage Data

For many of the bench tests and all of the spin tests in vacuum it was necessary to post-process the strain gage signals. Capacitive cross-talk between the high voltage signal carried on the actuator leads and the low voltage signal carried on the strain gage leads must be isolated to allow further post processing.

4.1.1 Removal of Capacitive Cross-Talk

A typical output from a strain gage during piezoelectric excitation is shown in Figure 4.1a. When cross-talk occurred it was necessary to adjust the dynamic range of
the data acquisition system to a 5 Volt range. At this range and with the 18 bit data acquisition system it was possible to resolve voltages as low as ±10 μV. This enabled the swept sine routine to detect the lower level strain gage signal from amongst the cross-talk. To separate the two signals a tenth order polynomial least squares fit was employed. The signal that results after this filter is applied to remove the cross-talk is displayed in Figure 4.1b, showing distinctive resonant peaks.

![Graph](image)

**Figure 4.1** Strain gage data obtained from the swept sine measurement technique showing a) the raw signal before removal of the capacitive cross-talk signal, and b) the same signal after use of the polynomial curve fit routine.
4.1.2 Damping Calculation Techniques

One of the main goals of this technique was to determine the amount of damping in a blade-disk assembly. The term used to quantify this effect is the critical damping ratio $\zeta_c$ or the ratio of the system damping $c$, to the critical system damping $c_c$. The model used to replicate the blade response is the single degree of freedom (SDOF) equation of motion given by:

$$m \ddot{x} + c \dot{x} + kx = F \sin(\omega t)$$

4.1

with the solution for the displacement $x(t)$ being:

$$x(t) = A \sin(\omega t + \phi)$$

4.2

The important variables are the amplitude of the response, $A$, the amplitude of the forcing function, $F$, and the phase shift of the response, $\phi$. By fitting a range of the experimental data at a particular mode of interest to this SDOF model, the identification of the system parameters can be obtained. Since the data collected from the strain gages is based in the frequency domain, the reduction into identifying $\zeta$ can be found in many different ways. A complete listing and description of the indirect frequency domain methods for system identification can be found in Maia and Silva (1997). Methods such as the peak-amplitude (or half power) and the quadrature response (or slope of phase) yielded sufficient initial guesses for $\zeta$, but a more precise measurement was needed. The Nyquist circle fit method was investigated, but the low frequency resolution prevented the ability to plot a well defined circle. The most reliable method was to fit single degree of freedom (SDOF) curves to the model equations of both amplitude and phase.
4.1.2.1 Amplitude Method

The majority of the amplitude method was developed by Jeffers and is detailed in Jeffers (1999). To calculate the damping ratio for a particular mode, a least squares fit routine was used. It was based on the arbitrary curve fit routine described in Bevington (1969). The arbitrary function used in this case was the single degree of freedom (SDOF) forced vibration magnitude given in the following equation:

\[
\frac{A}{F} = \left( \frac{1}{\sqrt{1 - \left( \frac{\omega_r}{\omega} \right)^2 \left( 2\zeta \frac{\omega_r}{\omega} \right)^2}} \right)
\]

4.3

The excitation frequency of the response is \(\omega\) and the resonant frequency is \(\omega_r\). The values for \(A\), \(\omega_r\), and \(\zeta\) are modified by the least squares routine to minimize the variance between the fit and the data.

By using the SDOF amplitude fit, it was assumed that the experimental resonant response approximated a single degree of freedom over the frequency range to which the curve fit was applied. This was not always the case, especially when a blade is mounted in a disk assembly where blade to blade vibration coupling through the disk made the SDOF amplitude fit less effective. To remedy this, the curve fit routine was modified by applying a weighting function designed to emphasize the values near the maximum resonant peak. Figure 4.2 shows an example of this curve fit applied to a resonant peak.
Figure 4.2 Sample of the Strain Gage Data with a Curve Fit Applied to a Resonant Peak

4.1.2.2 Phase Method

In observing the SDOF model, a mode occurs when the response is delayed in time relative to the forcing function, or in mathematical terms when a rapid phase shift of 180° occurs. Resonance occurs at the frequency when the blade and actuator are exactly 90° out of phase from one another (ϕ = -90°). The solution for the phase from Equation 4.2 becomes:

\[
\tan \phi = \frac{2\zeta}{\omega / \omega_r - \omega_r / \omega}
\]

Performing a least squares fit to the experimental data around a resonant frequency using Equation 4.4, allows for the computation of \(\zeta\) and \(\omega_r\). Figure 4.3 shows one example of experimental data along with the SDOF fit of Equation 4.4. The only
frequency range of interest occurs within a few data points around resonance. The reason for this is explained by taking the derivative of Equation 4.4 with respect to the frequency \( \omega \), showing that the damping is inversely proportional to both the slope of the phase at resonance and the natural frequency. For very low damping ratios (on the order of \( 10^{-4} \)) the slope is so steep that it is difficult to obtain a sufficient number of data points with which to fit the data. The amount of points with which to fit a curve then becomes strongly dependent on the resolution or sampling rate specified during the experimental setup. This trend points out one of the disadvantages with using the phase method. On the other hand, this method provides the best estimation of the natural frequency and is useful for higher damping or if the experiment allows for a variable frequency resolution for data acquisition.

![Graph showing phase response and area of interest](image)

Figure 4.3 Sample of the Strain Gage Data with a Curve Fit Applied to a Resonant Mode
4.2 Post-Processing of Aerodynamic Excitation Strain Gage and Piezoelectric Sensor Data

The same data reduction techniques to calculate the damping and resonant frequencies mentioned above were used in the aerodynamic actuation experiments. The only difference is that the frequency resolution of this set of data was much less because of the need to perform the FFT's over small time windows. As a result there is much more error in the damping calculation and the weighting function of the SDOF curve fit is less effective.

The data from the piezoelectric sensors seemed to provide signals with greater sensitivity than the strain gages. The lack of an intense uncertainty analysis for these sensors prevented the ability to show the results for the current study. Using a piezoelectric ceramic as a strain sensor shows a lot of promise for use in future studies in rotating environments.

4.3 Post-Processing of Pressure Sensor Data

The majority of the pressure data was reduced with familiar methods to resolve the aerodynamics of the flow. The time-averaged pressure results set the flow conditions for each run, while the FFT's of the pressure sensor data provide details about the unsteadiness of the flow. Care is taken to make sure that the averages and FFT's are performed over small enough time windows to keep the speed change at a minimum, while still allowing for a clear averaging without excessive noise corruption. Also, the averages and FFT’s were taken at off resonance, to keep the blade vibration from interfering with the pressure signal. For each run, the average was taken between 5 and 10 ms based on the turbine disk acceleration during that run. This time window is less
than a 1% change in speed, but includes over 2 revolutions of the disk (at least 100 wake passing events). For the FFT's specifically, it was found that if a larger time window was used the peak amplitudes would get washed out due to the large change in speed and hence the change in excitation frequency. In other words, a single resonance of interest would show up as multiple peaks spread out over a frequency range.

Another determining factor of the time window selection, is to allow for the best frequency resolution when performing the FFT's. Since the resolution is equal to the inverse of the time window size, using a 10 ms window only provides frequency data at 100 Hz intervals. Using this resolution it was still possible to distinguish resonant peaks among the noise. Since the main concern was to find the integer multiples of the wake excitation (41 EO), an approximate location of the harmonics was known before performing the FFT. To accurately find the peak amplitude a power spectrum fit of the data around resonance was used to extract the value and frequency of the resonance in between data points. Similarly, a linear interpolation between phase data points was used to find the phase at resonance. To avoid the possibility of interpolating between large jumps, the process is performed on the unwrapped phase or the continuous phase measurements that have not been forced in between $-180^\circ$ and $180^\circ$. 
CHAPTER 5

ERROR ANALYSIS

The unique piezoelectric actuation and reduction technique requires a careful estimation of the precision and accuracy of the damping calculation. The total uncertainty is divided into errors from the experimental components during data acquisition and errors due to post processing of data. By analyzing each portion of error separately, the largest sources of uncertainty can be easily identified. A large portion of this chapter focuses on the uncertainty during the piezoelectric actuation experiments, while the final section discusses the uncertainty in the pressure measurements.

5.1 Experimental Uncertainty

The error that occurs in the piezoelectric data acquisition studies is attributed to storage of both voltage and time. The voltage error is limited by the accuracy of the 18 bit system and losses in signal transmission, while the time error is simply a function of the 50 kHz sampling rate. Using Figure 3.8 on page 4 as a reference, the voltage error can be reduced to a lumped system including strain gages, amplifiers, DAS, and wires. By recording the strain gage response with no excitation to the actuator, the error due to the lumped system was measured and found to approximately be a constant ±10μV. The time error, or more importantly phase error, is directly proportional to the frequency.
Combining all the measurement errors and propagating them through the model equations using a root square technique provides an estimate of the error in $\zeta$ due to the acquisition system. This error is a function of each data set and consequently is different for each response.

5.2 Error from Data Reduction Techniques

Not only is there uncertainty from the measurement system, but a significant source comes from the various data reduction routines. This begins with the initial cross-talk filter, then propagates to both of the SDOF models for calculating the damping ratio.

5.2.1 Capacitive Cross-Talk Polynomial Filter

To help quantify the error due to the polynomial fit, which is used to remove the capacitive cross-talk signal, the order of the fit was changed. The effect of changing the order of the fit was then determined by running the damping calculation to obtain the influence of the fit’s order on the damping calculation. The results of this study show that as the order of the curve fit was varied from 8 to 12, the damping ratio varied less than one percent when compared to the standard 10th order fit.

Since the cross-talk filter essentially removes any information about the baseline vibrational response, the magnitude SDOF curve fit routine was tested to determine the effect an offset would have on the damping. The results show that for a 1% increase in offset there is a corresponding 1.65% increase in the damping ratio. As a result these effects were considered very minimal compared to other error and consequently ignored.
5.2.2 Effect of Noise on the SDOF Fits

To determine the deviations in the damping reduction methods, artificial responses with known damping ratios and randomly generated error were repeatedly put through both SDOF routines. A statistical analysis then provided a prediction of the contribution of bias and precision errors to the total data reduction error.

Figure 5.1 Data Reduction Uncertainty in the Estimation of the Critical Damping Ratio in the Presence of Noise for both a) the Amplitude Method and b) the Phase Method.
The results of this study for both the magnitude and phase method are shown respectively in Figure 5.1. For lower damping ratios (around $5 \times 10^{-4}$ or 0.05%) the magnitude method has little bias and precision errors, while at higher values (around 0.01 or 1%) both errors are of significant value. The curve fit routine has more difficulty in fitting the amplitude SDOF equation to highly damped (wider) peaks, causing the higher uncertainty. On the other hand, the phase method has lower errors at the higher damping values because of the larger supply of accurate data points around resonance.

As a result, the chosen method depends on $\zeta$, $\omega_r$, and the resolution. Using both methods and comparing the results provides a confident level of the system parameters.

### 5.3 Overall Damping Uncertainty

Most of the data presented had a noise level between 5% and 10%. Table 5.1 provides a summary of the approximate total uncertainty as a function of $\zeta$, $\omega_r$, and the resolution for both methods. As expected, in most cases the largest portion of error is attributed to the data reduction technique. The need for exploring other reduction schemes mentioned in Maia and Silva (1997) for the particular application to turbomachinery blading is evident.

<table>
<thead>
<tr>
<th>Resolution (Hz)</th>
<th>$\zeta \times 10^{-4}$</th>
<th>$\omega_r$ (kHz)</th>
<th>Amplitude Method</th>
<th>Phase Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>5</td>
<td>7.5</td>
<td>±1.09</td>
<td>±3.50</td>
</tr>
<tr>
<td>10</td>
<td>20</td>
<td>18</td>
<td>±2.56</td>
<td>±1.32</td>
</tr>
<tr>
<td>1</td>
<td>5</td>
<td>7.5</td>
<td>±0.66</td>
<td>±1.80</td>
</tr>
<tr>
<td>1</td>
<td>20</td>
<td>18</td>
<td>±0.50</td>
<td>±1.50</td>
</tr>
</tbody>
</table>

Table 5.1 Overall Uncertainty Due to both Experimental and Data Reduction Errors for Various Ranges of $\zeta$ and Resolution
5.4 Repeatability of the Damping Calculation

To determine the repeatability of the damping calculations, experiments were performed by taking data on a blade with a series of remountings to the same setting in the broach block. Table 5.2 presents a summary of the results from the repeatability experiments for both damping reduction schemes. The first conclusion from the table is that the standard deviation is very low when looking at each method individually, proving that both modes yield repeatable values for $\zeta$. Secondly, the median and standard deviation are very comparable between methods, showing that both methods yield very similar results.

<table>
<thead>
<tr>
<th></th>
<th>1 Hz Resolution</th>
<th>10 Hz Resolution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amplitude</td>
<td>Phase</td>
<td>Amplitude</td>
</tr>
<tr>
<td>Method $\zeta * 10^{-4}$</td>
<td>$\zeta * 10^{-4}$</td>
<td>$\zeta * 10^{-4}$</td>
</tr>
<tr>
<td>median</td>
<td>2.52</td>
<td>6.21</td>
</tr>
<tr>
<td>std. dev.</td>
<td>$\pm 0.15$</td>
<td>$\pm 0.58$</td>
</tr>
</tbody>
</table>

*All Data Taken At Same Conditions

Table 5.2 Experimental Uncertainty with a Comparison of Amplitude and Phase Methods, for Estimating the Damping, Showing a Set of $\zeta$ Values for Repeated Experimental Runs

5.5 Entire Uncertainty in the Pressure Measurements

To determine the uncertainty in the pressure measurements, the sensors are calibrated numerous times throughout the course of the experiments. By pressurizing the dump tank through the range of expected pressures a calibration curve can be created. By comparing the values to a very accurate Heise pressure sensor located in the
dump tank, the standard deviation of the pressure readings are obtained. Most of the sensors had a standard deviation of less than 1 kPa (0.15 psi) for all of the runs.

During data reduction, the pressure error from the calibration carries through to the average and FFT calculations. For the average pressure distributions, one standard deviation of the calculation is used as the error band (±1 standard deviation). The measurement error is so much lower in this case that the data is treated as only having data reduction error. For the unsteady peak amplitudes obtained by the FFT, the measurement error becomes important since the peak amplitudes are so low with respect to the actual pressure magnitude. For lower inlet pressures the error is even higher since it is more difficult to pick the peak amplitude from amongst the noise in the FFT calculation.

There is also error in the measurement of the sensor location on the blade, since the sensor has a finite width. The pressure signal is treated as being an average over the width of the gage. Since the blade chord is so small, the width of the sensor is a significant percentage (around 5% of the chord).
CHAPTER 6

RESULTS: VIBRATORY RESPONSE AND STRUCTURAL DAMPING IN VACUUM

The results of the piezoelectric excitation spin experiments in vacuum provide crucial information on the vibratory response and structural damping in the absence of any aerodynamic forces. They also provide a controlled structural response data set to guide the more realistic experiments with the aerodynamic excitation. The results from selected runs of all the strain gage signals can be seen in Appendix C. This chapter has the following major objectives: 1) explain how the mistuning in the system contributes to the strain gage signals and how this effects the data reduction process, 2) define the change in structural damping with increasing rotational speed, and 3) compare the rotating data to the finite element predictions and the bench data.

6.1 Mistuning

A word of caution is in order before analyzing the results from the spin tests in vacuum. The strain gages not only sense the vibration of the particular blade to which they are attached, but they also measure the vibration of neighboring blades during the spin tests. In other words, the vibration of a neighboring blade affects the vibration of the instrumented blade through coupling within the disk or the contact between
platforms. Since each blade inherits small geometric differences from the manufacturing process, the resonant frequencies are slightly mistuned from each other. In addition, the resonant frequencies of the blades containing instrumentation were shown to have noticeably lower values than the other nominal blades. This allows for enough separation in frequencies such that the vibratory response and damping can be quantified.

Examples of vibratory response measurements on 2 different blades at the same speed are shown in Figure 6.1 and Figure 6.2. These figures show the strain signals obtained at both the midspan and base of two neighboring blades. The graphs labeled as (a) and (b) show the experimental runs where the blade is excited over a wide frequency range (0-20 kHz) with a coarse frequency resolution of 6.25 Hz. Mode 2 and mode 4 exhibit the largest response throughout most of the study.

The wide frequency range runs are mostly used for viewing the characteristics of the response. In order to obtain an accurate damping and resonant frequency measurement, a narrower frequency range is selected during data acquisition to allow for data to be taken with a smaller frequency resolution. The graphs labeled as (c) and (d) show the experimental runs taken over a smaller frequency range of interest (12-14 kHz) and a correspondingly smaller resolution of 1.25 Hz. It should be emphasized that the high resolution runs are taken separately from the lower resolution runs, causing a slight difference in the comparison of the peak responses of both cases.

It is obvious that the vibration of several blades affects the instrumented blade. The question then becomes which peak resonance belongs to each blade. In this case, and generally in all of the other cases, the response of the blade that the sensor is
attached to is determined to be the highest peak resonance of the many peaks shown in the plots. This conclusion is made by comparing the results from different sensors during the same runs and by using the bench data which can be seen in Appendix C. Figure 6.1 clearly shows a large resonance of blade 66 around 12.6 kHz on both sensors and some smaller resonances. The peak around 12.8 kHz is determined to be the response of blade 54 by comparing this plot with that in Figure 6.2. In addition, the bench data from blade 54 isolated in the broach block showed a similar resonant response, which allows for a confident identification of the peak resonance.

Since the response of 6 different blades are sensed by 9 total strain gages, the experiments provide a significant database of the vibratory response characteristics of a mistuned blade-disk assembly. The main purpose for this database in the current analysis is to differentiate the resonant peaks by the methods mentioned in the previous paragraph. This data could also be used to study the mistuning characteristics of this particular turbine, but this task is left for future studies.
Figure 6.1 Response of Blade 66 at 9300 RPM for 2 Frequency Resolutions

Figure 6.2 Response of Blade 54 at 9300 RPM for 2 Frequency Resolutions
6.2 Change in Blade Response with Increasing Rotational Speed

As the centrifugal load on the blade increases due to the change in rotational speed, the normal force between the blade-disk interfaces also increases, causing the frictional force to decrease. In the turbine used for the current study the only form of structural damping comes from this frictional force, so it is necessary to quantify the decrease in structural damping with increasing rotational speed. This set of experiments in vacuum also helps to define the run conditions for the aerodynamic excitation experiments.

The mode 4 (2nd bending) and mode 2 (1st torsion) vibratory responses of blade 54 (SG31) at different speeds are plotted in Figure 6.3 and Figure 6.4. The increases in both the amplitude of the peak response and the resonant frequency are seen from the curves. The smaller damping at higher speeds is also evident, since the width of the resonant peak at the half power points is narrower. The computed resonant frequency and damping for each response is listed in the same figure.
Figure 6.3 2nd Bending Vibratory Response of Blade 54 at 6 Different Rotational Speeds

Figure 6.4 1st Torsion Vibratory Response of Blade 54 at 4 Different Rotational Speeds
Figure 6.3 and Figure 6.4 provide a visual display of the change in response with speed, but to analyze the details of the change in structural damping, the critical damping ratio is plotted for 2 different modes on 2 blades in Figure 6.5. These plots suggest that the damping is proportional to 1 over the square of the speed, especially for speeds above 5000 RPM. Since the damping is proportional to the inverse of the centrifugal load, which is proportional to the square of the speed, the data indicates that the measured damping is solely from this increased loading. The only exception occurs for lower speeds where nonlinear effects from the more complex friction interfaces could exist.

![Diagram of damping ratio vs speed](image)

Figure 6.5 Change in Structural Damping with Rotational Speed (2 Blades, 2 Modes)

The damping of modes 2 and 4 are presented here because the responses from these modes allowed for the best identification of peaks for the data reduction programs.
Modes 1 (1\textsuperscript{st} bending), 3 (coupled), and 5 (2\textsuperscript{nd} torsion) were harder to distinguish in the data. Mode 3 seemed to be excited well, but there were so many closely spaced peaks that it was not easily possible to associate a given resonance with a given blade. A sample of the average damping ratios and a range of resonant frequencies for mode 3 are summarized in Table 6.1. There were similar problems with trying to distinguish the mode 5 resonance. It was not excited well by the piezoelectric and consequently it was difficult to find the resonant peaks for calculating the damping. A few of the experiments provided some deducible information about the resonance, which is also summarized in Table 6.1. Both modes appear to show the same type of trends found with modes 2 and 4.

<table>
<thead>
<tr>
<th>Speed (RPM)</th>
<th>( \zeta ) (e-4)</th>
<th>( \omega_r ) (Hz)</th>
<th>( \zeta ) (e-4)</th>
<th>( \omega_r ) (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15000</td>
<td>3.0</td>
<td>9.5-11.0</td>
<td>5.0</td>
<td>16.2-18.0</td>
</tr>
<tr>
<td>12000</td>
<td>5.0</td>
<td>9.2-11.5</td>
<td>6.0</td>
<td>16.0-14.0</td>
</tr>
<tr>
<td>9300</td>
<td>6.0</td>
<td>9-10.2</td>
<td>16.0</td>
<td>15.8-17.2</td>
</tr>
<tr>
<td>5000</td>
<td>10.0</td>
<td>8.5-9.5</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 6.1 Summary of Damping Ratio and Resonant Frequency of Mode 3 (Coupled) and Mode 5 (2\textsuperscript{nd} Torsion)

The change in the resonant frequencies with speed can be seen in the experimental Campbell Diagram of Figure 6.6. The resonant frequencies plotted here represent an average of the response for the instrumented blades. Consequently, the frequencies are lower than those from the predicted Campbell Diagram (by as much as
The results of Figure 6.6 provide a detailed estimate of the modal crossings for determining the speeds to match during the aerodynamic excitation experiments.

Figure 6.6 Experimental Campbell Diagram for Instrumented Airfoils Obtained from Vacuum Spin Tests

6.3 Comparison of Results with Finite Element Predictions

To show a comparison of the resonant frequency predictions and measured values at different speeds during the vacuum spin tests, the average resonant frequencies of the blades without instrumentation are tabulated in Table 6.2 with the predicted values. The predictions agree to within 20% of the nominal blade measured response.
Also, the finite element results over predict the resonant frequencies in almost every case, which is to be expected due to the damping in the measurements.

<table>
<thead>
<tr>
<th>Speed (RPM)</th>
<th>Mode 2 $\omega_r$ (Hz)</th>
<th>Mode 3 $\omega_r$ (Hz)</th>
<th>Mode 4 $\omega_r$ (Hz)</th>
<th>Mode 5 $\omega_r$ (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FEA</td>
<td>Data*</td>
<td>FEA</td>
<td>Data*</td>
</tr>
<tr>
<td>10000</td>
<td>7407</td>
<td>6900</td>
<td>12646</td>
<td>10200</td>
</tr>
<tr>
<td>15000</td>
<td>7425</td>
<td>7100</td>
<td>12656</td>
<td>10700</td>
</tr>
<tr>
<td>20000</td>
<td>7450</td>
<td>7200</td>
<td>12670</td>
<td>11300</td>
</tr>
</tbody>
</table>

*Taken from average of nominal blades (non-instrumented)

Table 6.2 Comparison of Free Vibration Finite Element Analysis Predictions of Resonant Frequencies with Measured Values

6.4 Comparison of Results with Broach Block Experiments

It is important to compare the static experiments in the broach block and the rotating experiments in the disk to see how well the bench simulations represent the realistic rotating response. The comparisons also can help in clarifying the mistuning characteristics. Figure 6.7 shows the vibratory responses for the simulated speed of 12000 RPM in the broach block and the corresponding responses at speeds of 12000 and 15000 RPM in the vacuum spin tests. The relative peak of each mode is similar in both the broach block and the rotor. It is clear from the plots in b that the mistuned responses of the other blades are sensed by the strain gages. The data from the broach block (a, b) provides a “clean” response for determining which response belongs to each blade.
Figure 6.7 Comparison of Blade 54 Vibratory Response in the Broach Block (a, b) and the Spinning Rotor in Vacuum (c, d)

It is also important to look at how the resonant frequency and damping changes between the static and rotational experiments. These comparisons are summarized in the plots in Figure 6.8 and the results of Table 6.3. It is difficult to accurately determine which speed the bench experiments actually simulate. Since the bench tests were not performed in vacuum the comparisons are not straightforward. One should expect the resonant frequency and peak amplitude to drop and the damping to increase for the bench results. With the exception of mode 3 for blade 66, the damping levels are almost the same in both the rotor and the bench experiments.
Figure 6.8 Comparison of Blade 54 Vibratory Response in the Broach Block and the Spinning Rotor in Vacuum for Mode 4 and Mode 2

<table>
<thead>
<tr>
<th></th>
<th>Mode 2</th>
<th>Mode 3</th>
<th>Mode 4</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\omega_n$ (Hz)</td>
<td>$\zeta$ (e-4)</td>
<td>$\omega_n$ (Hz)</td>
</tr>
<tr>
<td>Blade 54 Rotor (12000 RPM)</td>
<td>6519</td>
<td>7.6</td>
<td>10376</td>
</tr>
<tr>
<td>Blade 54 Bench (~12000 RPM*)</td>
<td>6628</td>
<td>7.2</td>
<td>11432</td>
</tr>
<tr>
<td>Blade 54 Rotor (15000 RPM)</td>
<td>6690</td>
<td>6.9</td>
<td>10650</td>
</tr>
<tr>
<td>Blade 66 Rotor (12000 RPM)</td>
<td>6658</td>
<td>7.2</td>
<td>10271</td>
</tr>
<tr>
<td>Blade 66 Bench (~12000 RPM*)</td>
<td>6966</td>
<td>6.8</td>
<td>11025</td>
</tr>
<tr>
<td>Blade 66 Rotor (15000 RPM)</td>
<td>6816</td>
<td>6.5</td>
<td>10555</td>
</tr>
</tbody>
</table>

*Simulated rotational speed in the broach block

Table 6.3 Comparison of Broach Block and Rotor Measured Resonant Frequency and Damping for Blades 54 and Blade 66
CHAPTER 7

RESULTS: AERODYNAMIC FORCING FUNCTION AND VIBRATORY RESPONSE WITH AIR LOADING

The results in this chapter document for the first time the measurement of both the aerodynamic and structural response of a high pressure turbine rotor to the high pressure turbine vane wake excitation. To begin the investigation of the aerodynamic component the steady state pressures around the airfoil are presented. The unsteady loading is then analyzed by investigating the amplitude and phase of the pressure at the wake excitation frequencies. The pressure harmonics are compared at the nominal axial rotor/stator spacing and the increased spacing to determine the effect of the increased gap. Both the steady and unsteady pressures are compared against the CFD predictions for validating the code at different conditions. To understand the change in the unsteady pressure loading as the blade goes through vibration, the amplitude of the harmonic is analyzed both at resonance and off resonance. Finally the comparison of the vibratory response with aerodynamic excitation and in vacuum is made to determine the change in structural response and damping with the aerodynamic forces present.

This chapter is organized so that the first two sections (7.1 and 7.2) present data that establishes the overall flow characteristics and repeatability of the measurements.
The last three sections (7.3, 7.4, and 7.5) present the comparisons with predictions and other new findings obtained during the course of this study.

7.1 **Time-Averaged Pressure Distributions**

To establish the overall flow characteristics for every run, the time-averaged pressure distributions over the blade are presented. A summary of these results is presented in this section, but all of the time-averaged distributions can be found in Appendix A.

The steady state distributions are classified into three separate categories: runs with no incidence, runs with negative incidence, and runs with positive incidence. The majority of the runs are at zero incidence. One example of this type is plotted in Figure 7.1, showing both measurements at the midspan and near the tip. The results at midspan come as expected, with a relatively constant distribution across both surfaces and a higher loading on the pressure surface. These characteristics are seen throughout almost all of the runs having no incidence. The data taken near the tip shows a higher variation across each surface. This is not surprising, due to the complex 3D flow structures at the endwall, between the tip and casing. Figure 7.1 also shows that the average loading is increasing with increasing span location. In this case, there is approximately a 10% average increase in pressure as the span location increases from midspan (50%) to near the tip (85%).
Figure 7.1 Time-Averaged Pressure Distribution at 2 Spans with No Incidence (Run 2)

The runs with a large negative incidence occurred when trying to match the mode 4 crossing at room temperature with a blowdown run. The pressure distributions from Figure 7.2 show a varying signal across both surfaces. When air flows along the surface of the blade, areas of re-circulating flow can develop. These separation "bubbles" are especially noticeable on the pressure surface when there is large negative incidence because the inlet flow to the blade is not aligned with the blade in the relative frame of reference. Figure 2.6b on page 18 shows this visually in the results of the CFD simulations with negative incidence. The measurements show a large pressure near the leading edge at 85% span on the suction surface (8% chord). This corresponds to a loss in pressure differential (i.e. lift) across the airfoil.
Figure 7.2 Time-Averaged Pressure Distribution with Negative Incidence (Run 3)

The runs with positive incidence were experienced when obtaining the mode 2 crossing at room temperature in a blowdown run. With positive incidence, the inlet flow hits the pressure surface and the flow is then forced to follow the shape of the airfoil as seen in Figure 2.6 on page 18. There will consequently be an area of flow re-circulation near the leading edge of the suction surface. Figure 7.3 shows the relatively constant average pressure distributions for this type of run. The results are very similar to a run with no incidence since the incidence angle is very small. The measurements do not show the high loss in pressure differential seen in the large negative incidence runs.
The preceding results set the overall flow characteristics for each run and show the spanwise pressure variations. Another comparison of interest is the difference in the average distribution between the nominal axial rotor/stator spacing and the increased spacing. Figure 7.4 and Figure 7.5 show the measured pressures at both axial spacings at the midspan and near the tip, respectively. The midspan shows almost exact agreement and the tip measurements are very similar except for the gage near the 40% axial chord location. This means that the flow field for both the spacings is very similar, which allows for the comparisons of the pressure harmonics in the next section.
Figure 7.4 Comparison of Time-Averaged Pressure Distribution from Two Different Axial Rotor/Stator Spacings at 50% Span (Run 4 and Run 16)

Figure 7.5 Comparison of Time-Averaged Pressure Distribution from Two Different Axial Rotor/Stator Spacings at 85% Span (Run 4 and Run 16)
7.2 Harmonics of Aerodynamic Forcing Function

The time-averaged distributions of the previous section only indicate the overall nature of the flow during the run. To find the forcing function content of the pressure data, the amplitude and phase of the unsteady pressures are analyzed in the frequency domain. The data presented in this section was taken at operating points off of blade resonance. Therefore, the blade vibration amplitudes are small and the effect of blade motion on the measured unsteady pressures should be minimal.

After performing the FFT’s, the data exhibited measurable lower order harmonic content, but the main concern for the current study is the vane wake excitation at higher frequencies. The lower order harmonics were found to mostly be attributed to realistic engine order excitations (i.e. 1st EO, 3rd EO strut excitation, and 13th EO vane sector excitation). This was found by determining that the harmonics scale with speed and are not measured on the static pressure transducers upstream of the rotor. The data shows that the amplitude of the first and second harmonic (or the 41st and 82nd EO) are distinguishable from the noise on most of the gages for all of the runs. An example of one particular FFT showing 3 harmonics of interest and the lower order harmonics is seen in Figure 7.6.
Figure 7.6 Harmonic Content of Pressure Signal (Run 4, Gage PR10, 57-67 ms)

The surface distributions of the 1st and 2nd pressure harmonic amplitude and phase were plotted for every run and are presented in Appendix B. The cases presented here (only the 1st harmonic amplitude and phase) are meant to summarize the trends at each modal crossing and analyze the repeatability of the measurements and FFT reduction. Lines connecting the data points are only used to make easier comparisons and see the overall trends, not to necessarily reflect the actual continuous surface distributions. The progression of plots in this section begin with the pressure harmonic amplitude distributions at 3 modal crossings: mode 4 (2nd bending), mode 3 (coupled), and mode 2 (1st torsion). The amplitude distributions are followed by the phase distributions in the same order.
7.2.1 First Harmonic Amplitude

The peak amplitudes (non-dimensionalized by the inlet static pressure) of the 41 EO excitation at the midspan and near the tip for each modal crossing are shown in Figure 7.7 through Figure 7.12. Each plot shows data from 2 similar run conditions for the 3 modal crossings to compare the repeatability of the experiments for slightly different operating conditions. The data sets on each graph only differ by either the amount of acceleration (inlet pressure) or the pressure ratio across the stage during the run.

The first trend to notice is that the amplitude distribution for repeated conditions is within the error of the measurement for most of the runs. This means that the experiment is mostly repeatable and the FFT is doing a reasonable job in determining the peak resonance. One exception occurs in the comparisons for the mode 2 crossing. This is probably due to the lower speed change that the turbine experiences when run 7 is compared to run 8 (Figure 7.11 and Figure 7.12). Another exception to the generally reasonable comparisons occurs frequently for the gage located near 90% axial chord. This sensor (PR9) consistently shows large discrepancies throughout the test matrix, and eventually failed during one of the runs. The other trend to notice is that the amplitude of the data varies quite dramatically across the surface of the blade. Consequently the continuous distribution can look different than the distribution obtained from connecting the data points. With a limited amount of sensors on a small blade it is difficult to state the distribution trends clearly, but the data can still provide a good comparison for the prediction tools, which is shown in Section 7.3.
Figure 7.7 Surface Distribution of Pressure Harmonic Amplitude for Mode 4 Crossing (50% Span, 32% Axial Spacing)

Figure 7.8 Surface Distribution of Pressure Harmonic Amplitude for Mode 4 Crossing (85% Span, 32% Axial Spacing)
Figure 7.9 Surface Distribution of Pressure Harmonic Amplitude for Mode 3 Crossing (50% Span, 32% Axial Spacing)

Figure 7.10 Surface Distribution of Pressure Harmonic Amplitude for Mode 3 Crossing (85% Span, 32% Axial Spacing)
Figure 7.11 Surface Distribution of Pressure Harmonic Amplitude for Mode 2 Crossing (50% Span, 32% Axial Spacing)

Figure 7.12 Surface Distribution of Pressure Harmonic Amplitude for Mode 2 Crossing (85% Span, 32% Axial Spacing)
7.2.2  *First Harmonic Phase*

The phase angle at the peak resonant frequency for the 41 EO excitation is shown in Figure 7.13 through Figure 7.18, following the same progression as the previous section. Since the phase angle is a measurement relative to a reference that is not identified by the data reduction system, it is more difficult to make comparisons. In order to make a comparison, one of the data sets is offset so that the trend in the phase lines up for both runs. This offset is determined by aligning the measurements of a gage that shows the lowest amount of uncertainty for a particular run. Another problem is that the phase angle is a circular measurement that is not represented well on a rectangular axis, so values that measure close to 180° look very different than values near 180°. A prime example of this is seen in Figure 7.13 for the –85% chord location. In reality values for the nominal high acceleration and nominal low acceleration cases are very close. Keeping the above cautionary notes in mind, it is not surprising that there is less repeatable agreement in the phase measurements than there is in the amplitude data. Still, reasonable comparisons can be made for repeated runs in most cases, specifically for the mode 4 crossing. As was the case with the amplitude, it is difficult to state a clear trend in the data, with the varying axial distribution, but the comparisons with prediction tools (shown in the proceeding sections) are still useful.
Figure 7.13 Surface Distribution of Pressure Harmonic Phase for Mode 4 Crossing (50% Span, 32% Axial Spacing)

Figure 7.14 Surface Distribution of Pressure Harmonic Phase for Mode 4 Crossing (85% Span, 32% Axial Spacing)
Figure 7.15 Surface Distribution of Pressure Harmonic Phase for Mode 3 Crossing (50% Span, 32% Axial Spacing)

Figure 7.16 Surface Distribution of Pressure Harmonic Phase for Mode 3 Crossing (85% Span, 32% Axial Spacing)
Figure 7.17 Surface Distribution of Pressure Harmonic Phase for Mode 2 Crossing
(50% Span, 32% Axial Spacing)

Figure 7.18 Surface Distribution of Pressure Harmonic Phase for Mode 2 Crossing
(85% Span, 32% Axial Spacing)
7.2.3 Variation of Harmonic Amplitudes with Axial Spacing

An important question to be answered in this study is how the amplitude of the unsteady pressure loading varies with the axial rotor/stator spacing. Engine designers like to have a small gap between the two components to save length and weight. However, if the gap is too small the vane wake loading may be too high on the rotor blades.

The first harmonic amplitude pressure distributions for the mode 4 and mode 2 crossings are shown at the midspan and tip, respectively, in Figure 7.19 through Figure 7.22. As expected, the amplitudes are higher in general at the closer spacing for both crossings. This appears to be a much larger increase near the tip than at the midspan and a higher increase on the suction surface. Since the steady loading was seen to be higher near the tip, it is not surprising that the unsteady wake excitation is larger at 85% span.

These results show that by using the 32% axial spacing configuration the amplitude of the forcing function can be up to 4 times higher than if a configuration with a 10% larger axial spacing was used. This could significantly increase the unsteady stresses that the blade encounters during operation and cause an HCF problem.
Figure 7.19 Comparison of 1st Harmonic Amplitude Pressure Distribution from Two Different Axial Rotor/Stator Spacings at 50% Span for Mode 4 (Run 4 and Run 16)

Figure 7.20 Comparison of 1st Harmonic Amplitude Pressure Distribution from Two Different Axial Rotor/Stator Spacings at 85% Span for Mode 4 (Run 4 and Run 16)
Figure 7.21 Comparison of 1st Harmonic Amplitude Pressure Distribution from Two Different Axial Rotor/Stator Spacings at 50% Span for Mode 2 (Run 7 and Run 13)

Figure 7.22 Comparison of 1st Harmonic Amplitude Pressure Distribution from Two Different Axial Rotor/Stator Spacings at 85% Span for Mode 2 (Run 7 and Run 13)
7.3 Comparison of Pressure Distributions with CFD

One of the most important aspects of the current study is to provide CFD code validation. Comparisons of the data and results from a quasi-three dimensional code (UNSFLO) are presented here as one example of the ability to validate all of the other codes. UNSFLO has been validated in several other similar studies, Giles and Haimes (1993), but this is the first time that it has been validated for a subsonic turbine. The time-averaged distributions are compared first, followed by the amplitudes and phases of the harmonics.

7.3.1 Time-Averaged Pressure

As mentioned earlier, the time-averaged pressure distributions are calculated to obtain the overall flow characteristics for each run. It is difficult to move on to comparisons of the harmonics if the code cannot predict the steady pressures. Following the same progression of section 7.1, the comparisons are shown for runs with no incidence, negative incidence, and positive incidence.

Figure 7.23 and Figure 7.24 show the measured and predicted distributions of the runs with no incidence at midspan and near the tip, respectively. There is reasonable agreement at the midspan in the trend of the data and the level of the pressure. The relative pressure differential between suction and pressure surface is matched well by the code. One exception occurs on the gage near 60% axial chord (PR9) which was shown earlier to have large discrepancies in the measurement throughout the experiments. There is less agreement near the tip, but this is not surprising since UNSFLO is not a true three-dimensional code. Still, the level of the data is similar to the prediction.
Figure 7.23 Comparison of Predicted and Measured Time-Averaged Pressure Distributions for No Incidence (50% Span, 32% Spacing, Run 4)

Figure 7.24 Comparison of Predicted and Measured Time-Averaged Pressure Distributions for No Incidence (85% Span, 32% Spacing, Run 4)
The midspan and tip distributions for a run with negative incidence are shown in Figure 7.25 and Figure 7.26, respectively. There is less agreement for this case since the code has a difficult time in accurately determining the location of the re-circulating flow on the pressure surface. Sources of entropy outside of the thin viscous grid around the surface of the blade are not handled by a code of this type. A better comparison can be made on the pressure surface since there are no separation bubbles. The level and trend of the pressure surface data are still in reasonable agreement.

The final steady results to compare are the positive incidence runs, shown in Figure 7.27 and Figure 7.28. There is close agreement, especially at the midspan between the data and the prediction. Near the tip there is less agreement on the suction surface.

These comparisons of the time-averaged pressures show that UNSFLO can predict the overall steady-state flow characteristics with some accuracy, especially at the midspan. This provides the basis for making the same comparisons with the unsteady pressures.
Figure 7.25 Comparison of Predicted and Measured Time-Averaged Pressure Distributions for Negative Incidence (50% Span, 32% Spacing, Run 3)

Figure 7.26 Comparison of Predicted and Measured Time-Averaged Pressure Distributions for Negative incidence (85% Span, 32% Spacing, Run 3)
Figure 7.27 Comparison of Predicted and Measured Time-Averaged Pressure Distributions for Positive Incidence (50% Span, 32% Spacing, Run 7)

Figure 7.28 Comparison of Predicted and Measured Time-Averaged Pressure Distributions for Positive Incidence (85% Span, 32% Spacing, Run 7)
7.3.2  Amplitude and Phase of Pressure Harmonics

The most important comparison for the purpose of a forced response analysis is to determine how well the code can predict the unsteady forcing functions. The comparisons presented here are the data at the modal crossings of importance and the corresponding unsteady solution from UNSFLO. The UNSFLO unsteady pressures are only those due to vane wake passing with no blade motion. The main comparison of interest begins with the first harmonic (41 EO) amplitude and phase at the midspan for both the mode 4 and mode 3 crossings. Also, the comparison of the mode 4 crossing near the tip is shown to observe the overall trend in all of the 85% span data. Finally, one example of a comparison for the second harmonic amplitude at midspan is shown.

The first harmonic amplitude and phase at the mode 4 crossing can be seen in Figure 7.29 and Figure 7.30, respectively, which is followed by the same plots at the mode 3 crossing in Figure 7.31 and Figure 7.32. The overall levels of the amplitudes agree well with the data. There is better agreement in the trends of the distribution on the suction surface, although this is difficult to tell with the oscillatory nature of the distribution. The phase has less agreement at both crossings. This is a finding that has been observed in other similar studies (Manwaring et al., 1996; Abhari and Giles, 1995) showing that improvement is still needed in the ability of current CFD codes to predict phase.
Figure 7.29 Comparison of Predicted and Measured 1st Pressure Harmonic Amplitude Distributions at Mode 4 Crossing (50% Span, 32% Spacing, Run 4)

Figure 7.30 Comparison of Predicted and Measured 1st Pressure Harmonic Phase Distributions at Mode 4 Crossing (50% Span, 32% Spacing, Run 4)
Figure 7.31 Comparison of Predicted and Measured 1st Pressure Harmonic Amplitude Distributions at Mode 3 Crossing (50% Span, 32% Spacing, Run 2)

Figure 7.32 Comparison of Predicted and Measured 1st Pressure Harmonic Phase Distributions at Mode 3 Crossing (50% Span, 32% Spacing, Run 2)
Only one example of the code validation for flow near the tip is presented because similar conclusions can be made for the other cases. Since UNSFLO is a quasi-three dimensional code, good agreement is not expected. Figure 7.33 shows the surface distribution of the first harmonic amplitude for the 85% span location and the corresponding prediction. The pressure level and trend of the first harmonic is off by a significant amount in the prediction, but the trend on the suction surface seems to be matched. When comparing the results at the midspan and the tip, the code shows a slight decrease in the level of the harmonic distribution with increasing span, while the data seems to show a slight increase.

![Graph showing comparison of predicted and measured 1st pressure harmonic amplitude distributions at mode 3 crossing (85% span, 32% spacing, Run 2)](image)

Figure 7.33 Comparison of Predicted and Measured 1st Pressure Harmonic Amplitude Distributions at Mode 3 Crossing (85% Span, 32% Spacing, Run 2)

The final CFD comparison is made for the amplitude of the second harmonic (82 EO). Only one data set, the 50% span location for the mode 4 crossing, is presented here.
in Figure 7.34. The level of the response is in general agreement with the prediction, but the trend does not appear to follow the data very closely. In particular, there is better agreement on the pressure surface. It is more difficult to reduce the 2\textsuperscript{nd} harmonic amplitudes at each sensor location since the peak pressures are so small when compared to the noise in the total FFT distribution.

Figure 7.34 Comparison of Predicted and Measured 2\textsuperscript{nd} Pressure Harmonic Amplitude Distributions at Mode 4 Crossing (50\% Span, 32\% Spacing, Run 4)

7.4 Unsteady Pressure Loading During Blade Structural Resonance

All of the pressure data so far has been analyzed when the blade is not vibrating. It is important to see the effect of the blade vibration on the unsteady loading in order to understand the interaction between the structural and fluid systems. Figure 7.35 shows the strain gage signals from 2 different blades, in the time domain, from an experimental
run at the mode 3 crossing. All of the time domain strain gage data from this run can be seen in Appendix D. It is obvious that blade 39 is going through a structural resonance beginning at around 75 ms, due to the higher peak to peak signal. Blade 54 does not show as strong of an indication of going through resonance, although there is some vibratory response near the end of the time period.

![Graph showing strain gage signals for Blade 39 and Blade 54](image)

Figure 7.35 Strain Gage Signals of Two Neighboring Blades During Resonance (Run 6)

To obtain a clearer picture of both the vibratory response and corresponding pressure loading for each blade, the signals need to be analyzed in the frequency domain. The problem with using the frequency domain is that the turbine experiences a large acceleration during the run, so the forcing frequency of the vane excitation is increasing rapidly. If the FFT is taken over a large time period (above 20 ms) the peak resonances appear to “wash” out due to the changing excitation. Consequently, it is necessary to analyze the vibratory response over a very small portion of the run time, making the
acceleration relatively constant. This creates another problem because when a smaller time period is used for performing the FFT, the frequency resolution becomes very low. Several methods for increasing this resolution, including repeating the small data set window to artificially aid the FFT, failed to produce better results. The best results with the data were found if a 10ms time window was used, which results in a 100 Hz frequency resolution.

One method for overcoming the acceleration problem is to perform the FFT over small “sliding” time windows as a particular blade goes through resonance. By using time windows of 10 ms the balance between a small speed change and high frequency resolution was optimized. The “sliding” windows were taken several times to cover the entire time period of the signals shown in Figure 7.35. Both the pressure and strain gage harmonic for blade 39 and blade 54, deduced from the sliding FFT technique, can be seen Figure 7.36. The trend in both cases is that the pressure amplitude goes to a minimum as the structural response goes to a maximum. The response of blade 39 sees 2 “dips” over the time (or frequency) period. The first valley, occurring near 85 ms, is from the vibration of the blade that the sensor is attached to (blade 39). The second valley, occurring near 110 ms, seems to appear when the neighboring blade (blade 54) enters resonance. Blade 39 senses the vibration of blade 54 as evidenced by the smaller resonant peak around 110 ms, but it does not appear as though blade 54 senses blade 39’s resonance. Consequently there is not a dip in the pressure harmonic of blade 54. It is quite evident that this occurrence is not a creation of the data reduction programs, since the trend occurs on both gages at distinct times. It should be noted that the large resonance during this run is from the 1st bending mode (mode 1), not the targeted mode 3.
crossing that was intended to be excited by the vane wake during this run. Both resonant responses are plotted in the figure, showing that the mode 3 response is much lower than the 1st bending response. This still does not change the conclusions made about how the pressure field responds to the blade going through vibration.

![Graphs showing pressure and strain data](image)

**Figure 7.36 Peak Amplitudes of Pressure and Strain Gage Data as Two Neighboring Blades Go Through Structural Resonance**

The phenomena of Figure 7.36 may be explained by considering the unsteady pressure due to blade motion as a destructive interference with the unsteady pressure
field due to wake passing. In other words, the unsteady pressures are at the same frequency, but have a different phase angle.

7.5 Comparison of Damping with Aerodynamic Excitation vs. Piezoelectric Excitation in Vacuum

The goal in using two different excitation methods was to quantify the effect of the aerodynamic excitation on the vibratory response of the blade. It is necessary to first compare the vibratory responses from the 2 forms of blade actuation. Of course the aerodynamic excitation methods are the most realistic for classifying the structural response of a turbine blade in operation. The vibratory responses over a wide frequency range for both methods can be seen in the plots of Figure 7.37. The strain gage signals from the two different gages on blade 54 show expected response levels for the 5 expected modes of interest. It is clear that the piezoelectric excitation method does indeed provide a feasible excitation for classifying the vibratory response in vacuum, when compared to the responses from the aerodynamic excitation experiments. At first glance there are two indications of the aerodynamic damping from the two strain gage signals (Figure 7.37 c and d). The first is the larger peak amplitude for the response in vacuum. There is also an indication that the resonant frequency decreases and the width of the peak decreases during the vacuum experiments.
Figure 7.37 Vibratory Response Over a Large Frequency Range in Vacuum (Run 72) and with Aerodynamic Excitation (Run 4)

To actually determine the damping level it is important to focus on the mode 4 response, since this is the modal crossing for which the aerodynamic excitation experiments were performed. Figure 7.38 shows the mode 4 vibratory responses for both excitation experiments. These plots provide a more detailed view of the damping increase with aerodynamic excitation. The resonant responses from both gages show a reduction in resonant frequency and wider peak resonance. This conclusion is confirmed
with the results of the SDOF damping reduction technique shown in Table 7.1. The damping levels in this comparison indicate that a large component of the total damping is due to the aerodynamic loading on the blade.

![Graph showing vibratory response in vacuum and air](image)

Figure 7.38 Vibratory Response of Mode 4 in Vacuum and with Air Loading (Run 4)

<table>
<thead>
<tr>
<th>Gage</th>
<th>Blade</th>
<th>Location</th>
<th>Vacuum (kHz)</th>
<th>Air (kHz)</th>
<th>Vacuum (c-4)</th>
<th>Air (c-4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SG 31</td>
<td>54</td>
<td>Midspan</td>
<td>13288</td>
<td>13250</td>
<td>9.0</td>
<td>55.0</td>
</tr>
<tr>
<td>SG 32</td>
<td>54</td>
<td>Base</td>
<td>13288</td>
<td>13250</td>
<td>9.0</td>
<td>65.0</td>
</tr>
<tr>
<td>SG 33</td>
<td>66</td>
<td>Midspan</td>
<td>13038</td>
<td>13090</td>
<td>9.0</td>
<td>65.0</td>
</tr>
<tr>
<td>SG 34</td>
<td>66</td>
<td>Base</td>
<td>13038</td>
<td>13090</td>
<td>9.0</td>
<td>70.0</td>
</tr>
</tbody>
</table>

Table 7.1 Comparison of Resonant Frequency and Damping of Two Blades during the Mode 4 Excitation in both Vacuum and Air

It should be noted that the frequency resolution of the FFT results from the aerodynamic excitation experiments is much lower than the resolution from the vacuum
tests (100 Hz compared with 1.25 Hz). This creates difficulty when computing the
damping with the SDOF methods, since there are few points in the aerodynamic
excitation response with which to fit a curve around resonance. It is also necessary to
make sure that there is a small change in speed during the FFT window to avoid a
resonant response that shows a smoothed out peak response over a rapidly changing
excitation frequency. By using time windows of 10 ms, the speed change is assured to
be less than 5% over the time frame. There is also the effect of not being able to reach
the steady state resonant peak with an excitation source of rapidly increasing frequency,
as discussed in Chapter 3. Both of the previous results actually make the damping seem
about 20% higher than it truly is for the aerodynamic loading experiments. While this is
a high value, it still does not account for the high increase (over 500%) when compared
to the measured structural damping value.

The caveats mentioned in the previous paragraph encapsulate the problems with
using a shock tunnel for forced response studies. For future studies it is necessary to
avoid large accelerations during the run so that the wake excitation frequency remains
constant. This is possible by investigating a turbine with a higher inertia or through the
use of a flywheel assembly.
CHAPTER 8

CONCLUSIONS

The results presented in this study have provided a coupled aerodynamic and structural data set and supporting computational analysis for the investigation of the forced response of a rotating turbine at engine conditions. The experimental surface pressure and corresponding strain gage data have provided for the first time an investigation of the HPT vane wake excitation of a rotating HPT blade at engine conditions. Both the aerodynamic and structural data can be used by the gas turbine industry to verify a wide range of computational prediction tools.

Before obtaining the experimental data, a series of predictions were obtained using typical computational tools. UNSFLO was used to predict the unsteady full-stage surface pressures at the midspan and near the tip of the blade. The results showed that the average pressure distributions have a very flat distribution while the surface harmonics have an oscillatory nature. It was also seen that a significant increase in the peak amplitude of the harmonic is found when the rotor is 20% closer to the vane axially, due to the magnitude of the wake excitation. ANSYS was used for the finite element structural response predictions. A single blade was modeled with the firtree attachment, platform, and airfoil to provide the vibratory response characteristics. The
results showed that there are 5 widely spaced modes between the frequency range of interest (1-20kHz). The finite element analysis also showed that the change in vibratory response due to the installation of instrumentation was minimal.

This analysis provided a background for the experimental program including the optimal use of the transducers, and setting of the run conditions. The combination of the piezoelectric blade excitation technique and aerodynamic excitation through the use of a shock tunnel facility allowed for the acquisition of a coupled fluid and structural interaction data set at engine conditions. The unique piezoelectric actuation method provided continuous blade excitation at high frequency while monitoring the structural response in a spinning rotor. The historical problem with this method of cross-talk interfering with the measurement signals was resolved through experimental techniques and post-processing. A complete uncertainty analysis showed that the critical damping ratio of the blade could be calculated to within ±1e-4. It was necessary to prove that a shock tunnel facility could be used to study forced response, but a note of caution was needed to show that there is significant acceleration during the run. As a result, the vane wake excitation has a rapidly changing frequency nature. This excitation was shown to allow the blade to reach over 75% of the peak amplitude of a steady excitation. By looking at small time windows of data the turbine acceleration could be kept under control.

The first series of experiments in the evacuated spin facility showed the existence of mistuning in the blade vibratory response, but the resonant peaks were far enough apart to classify the nature of the response and damping characteristics. The damping
was shown to be inversely proportional to the square of the speed. Comparisons with finite element analysis and bench data provided reasonable agreement.

The results from the aerodynamic experiment provided the pressure and structural response data at 3 modal crossings of interest. The time-averaged pressures were resolved to prove that the experimental conditions provided the proper flow. Comparisons of the time-averaged results were in good agreement with the UNSFLO predictions. Reasonable comparisons were also seen between the pressure harmonics and the predictions. Both the level and trend of the data were picked up well by the predictions at midspan for the first amplitude harmonics. Similarly, the second harmonic amplitude matched reasonably well with the predictions. There was considerably less agreement for the first and second phase of the harmonics. In some cases, the trend of the response was comparable, but in others there was not much agreement.

Another major conclusion from the shock tunnel experiments came with the comparison of the pressure distributions between the two different axial rotor/stator spacings. The steady pressure comparisons showed the expected close agreement between both spacings. The first harmonic amplitude variation with spacing showed a large increase for the closer spacing, especially near the tip. The harmonic loading also seemed to show a larger level on the suction surface for this same comparison.

The structural response of the blade was investigated with the representative aerodynamic excitation during this set of experiments. The harmonic of the wake excitation was shown to “dip” as the blade goes through resonance. This means that the vibratory response acts as a destructive interference to the unsteady pressure field. In
addition, the vibratory response in air was compared with the response in vacuum to quantify the additional aerodynamic damping to the total damping level. The results indicate up to a tenfold increase in the total damping with the air loading.

The combination of computational and experimental methods in this study has provided insight into the forced response of a turbine blade at actual engine conditions. The experimental results provide a much needed data set for unsteady pressure loading and vibratory response for turbines. The data and accompanying analysis presented here provide the basis for obtaining a clear understanding of the complex fluid and structural interaction that has caused engine failures in the past through HCF. By determining the physical processes that make up an HCF problem of this manner many of the safety and cost issues in gas turbine engine design can be solved.

Although the results presented in this document summarize the most important conclusions, there is the potential for additional knowledge that can be ascertained from the experiments with future work. A full forced response prediction, by applying the CFD results as a forcing function to the finite element model, would allow for better comparison of the strain gage measurements during the aerodynamic excitation experiments. In addition, because of the amount of strain gage data obtained from numerous sensors on several blades, the experiments provide information on the mistuning characteristics of a blade-disk assembly. This data could be used to perform a more extensive mistuning analysis. Only the surface of this analysis was touched for the current study. Also, the coupled pressure and strain gage harmonic data was thoroughly explained for only one particular run. A careful investigation of this coupling for the other runs could provide interesting information about the variations with different
modes and different run conditions. In order to analyze these effects, more work on the
damping reduction techniques for low frequency resolutions would be helpful. The
potential for using piezoelectric ceramics as strain sensors was also demonstrated.

As computational prediction tools continue to improve the ability of predicting
the physical processes in gas turbines, the experimental data and analysis obtained in the
current study could help the designer develop improved method for making a better
engine. It is hopeful that this data set is the first of many future experimental programs
aimed at investigated the forced response of turbines.
BIBLIOGRAPHY


AlliedSignal Contributions to the OSU/GUIde Aerelastic Measurement Program, 21-9771, August 1997.


APPENDIX A

TIME-AVERAGED PRESSURE DISTRIBUTIONS

The following plots represent the average pressure distribution (non-dimensionalized by the inlet pressure) over the blade as a function of the percent axial chord during the established flow of every run. The error bars on the pressure axis represent 1 standard deviation in the non-dimensional average while the error bars on the position axis represent the width of the sensor. The notation follows as 0 being designated the leading edge with the positive chord locations being the suction surface.

Pressure Gage Location Guide (PR's)

<table>
<thead>
<tr>
<th>% Span</th>
<th>18</th>
<th>17</th>
<th>16</th>
<th>15</th>
<th>14</th>
<th>13</th>
<th>12</th>
<th>11</th>
<th>10</th>
<th>9</th>
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</thead>
<tbody>
<tr>
<td>100%</td>
<td>1</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
<td>7</td>
<td>8</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

-100% -75% -50% -25% 0% 25% 50% 75% 100%

% Axial Chord
Run 4 Average Surface Pressure Distributions (P/P_{in})

% Axial Chord

% Span

% Axial Chord
Run 11 Average Surface Pressure Distributions (P/Pin)
Run 12 Average Surface Pressure Distributions (P/P∞)

% Axial Chord

% Span

-100  -80  -60  -40  -20    0    20    40    60    80    100

0.1  0.2  0.3  0.4  0.5  0.6  0.7  0.8  0.9  1.0

-100  -80  -60  -40  -20    0    20    40    60    80    100

0.1  0.2  0.3  0.4  0.5  0.6  0.7  0.8  0.9  1.0

149
APPENDIX B

HARMONIC PRESSURE DISTRIBUTIONS

The following plots represent the first and second harmonic pressure distributions (non-dimensionalized by the inlet pressure) over the blade as a function of the percent axial chord during the established flow of every run. The error bars on the pressure axis represent 1 standard deviation in the non-dimensional average while the error bars on the position axis represent the width of the sensor. The notation follows as 0 being designated the leading edge with the positive chord locations being the suction surface.

Pressure Gage Location Guide (PR's)

% Span

18 17 16 15 14 13 12 11 10 9

% Axial Chord

-100% -75% -50% -25% 0% 25% 50% 75% 100%

155
Run 10 1st Harmonic Pressure Amplitude (P/Inlet)

% Axial Chord

85% Span

0.025

0.02

0.015

0.01

0.005

0

-100 -80 -60 -40 -20 0 20 40 60 80 100

% Axial Chord

50% Span

0.08

0.07

0.06

0.05

0.04

0.03

0.02

0.01

-100 -80 -60 -40 -20 0 20 40 60 80 100

% Axial Chord
APPENDIX C

STRUCTURAL RESPONSE DATA FROM VACUUM SPIN EXPERIMENTS

The following plots represent the structural response measurements from selected piezoelectric vacuum spin experiments. The responses from all 8 strain gages are included (4 on each page). The plots are titled by the corresponding filename for that particular run. A description of the experimental conditions for these runs is summarized in the table below. A diagram of the strain gage locations on the numbering scheme can be seen on the following page.

Run Conditions for Piezoelectric Actuation Experiments

<table>
<thead>
<tr>
<th>Run Number</th>
<th>Modal Crossing</th>
<th>Speed (RPM)</th>
<th>Pressure</th>
<th>Freq. Range (KHz)</th>
<th>Excitation Method</th>
<th>Resolution (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>41</td>
<td>All</td>
<td>0</td>
<td>atmosphere</td>
<td>0-20</td>
<td>chirp</td>
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<tr>
<td>42</td>
<td>All</td>
<td>0</td>
<td>vacuum</td>
<td>0-20</td>
<td>chirp</td>
<td>6.25</td>
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<tr>
<td>52</td>
<td>All</td>
<td>5000</td>
<td>vacuum</td>
<td>0-20</td>
<td>chirp</td>
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<td>54</td>
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<tr>
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<td>6.25</td>
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<tr>
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<td>All</td>
<td>19500</td>
<td>vacuum</td>
<td>0-20</td>
<td>chirp</td>
<td>6.25</td>
</tr>
</tbody>
</table>
Strain Gage Location Guide

(Measured as a Percent of the Entire Span)
APPENDIX D

STRUCTURAL RESPONSE DATA FROM AERODYNAMIC EXCITATION EXPERIMENTS

The following plots represent the structural response measurements from a selected aerodynamic excitation experiment (Run6, Mode 3 Blowdown, ~16000 RPM, \( P_{\text{inlet}} = 42.0 \) psia, \( P_{\text{ratio}} = 1.96 \)). The plots are converted into microstrain and in the time domain. Responses from 8 strain gages are included (SG32 is missing since it failed before the run). A diagram of the strain gage locations on the numbering scheme can be seen on the following page.
Strain Gage Location Guide

(Measured as a Percent of the Entire Span)