CHAPTER 5

EXPERIMENTAL TESTING

Experimental data is extensive at PWC. For certification purposes, all HP or CT blades have a strain gage test (SGT) performed to determine the resonances in the running range and the vibratory stress associated with them. A strain gage test is done in test cell using a real engine as a test vehicle. Strain gage are attached to any components as required by the engineer. When the engine is running, a data acquisition system records the strain gage signal as well as the engine rotating speed. A Fast Fourier Transform is performed by the data acquisition system on the strain gage recorded time signal. This step transforms the time signal into the frequency domain and using the rotating speed, a waterfall plot is generated. For a turbine blade certification, these tests require the application of strain gages on different blades, the gages being located on high strain areas based on the FEM model. Therefore, the accuracy of the mode shape is of prime importance in order to assess the HCF life of the component. Due to highly complex and expensive method of performing these strain gage tests, a static test at normal temperature is developed to further study the contact elements as boundary condition in the blade FEM model. The main goal of the experimental testing is to determine the friction coefficient for the model in order to reproduce the mode shapes at the correct natural frequency values. Furthermore, the need for a specific friction coefficient for every mode shape might arise. In addition, contact testing using chalk between the blade and disc contact faces will be applied and the results will be correlated with the FEM modelling.

5.1 Experimental Test Model

To perform the experimental testing, a blade and disc will be used. To correlate the results of the experimental testing on the FEM results, the boundary conditions have to be the same. For the experimental test, the blade will be assembled on the disc. The disc will be held in a specially designed fixture to avoid any resonance in the frequency range of interest. To simulate centrifugal force (CF) loading, two screws are forced inside the chamfer of the rivet hole, which will create an upward force due to its conical shape (Figure 11). Refer to Figure 12 for illustration of the conical shape of the rivet hole.

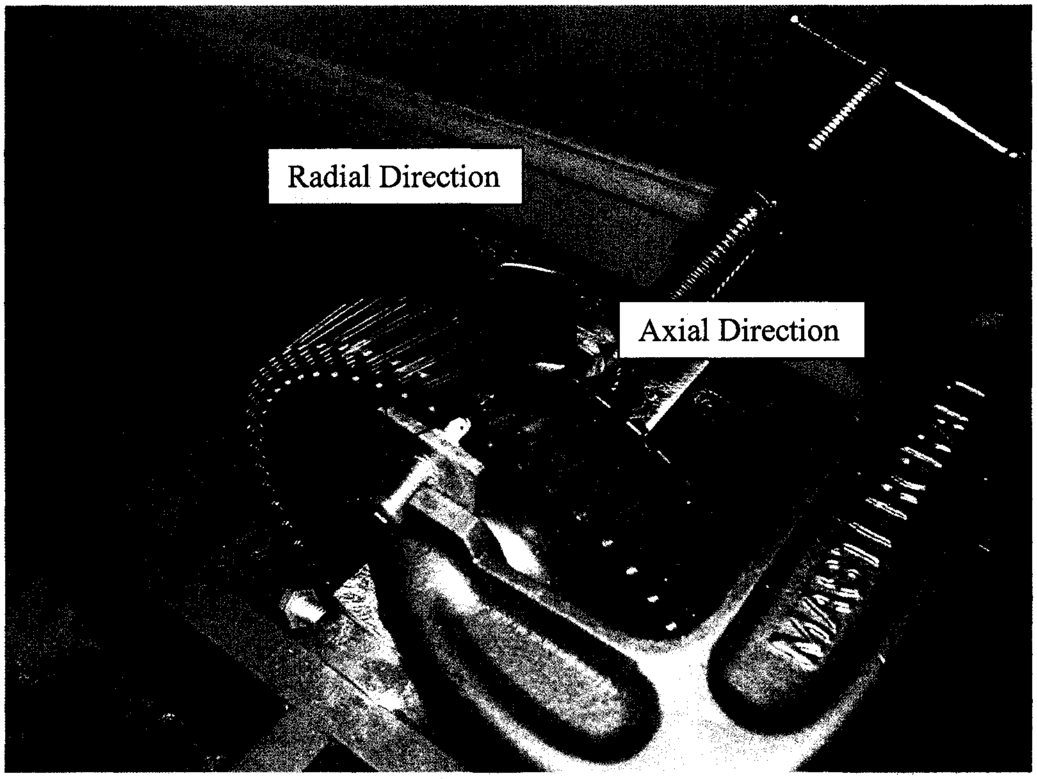

Figure 11 Experimental test mount simulation of centrifugal force

To recreate the same boundary conditions in the FEM model, the centrifugal force was removed and replaced by a displacement of 0.1 inch (approximate value) in the axial and radial directions based on the conical shape at which the screws are inserted (Figure 12).

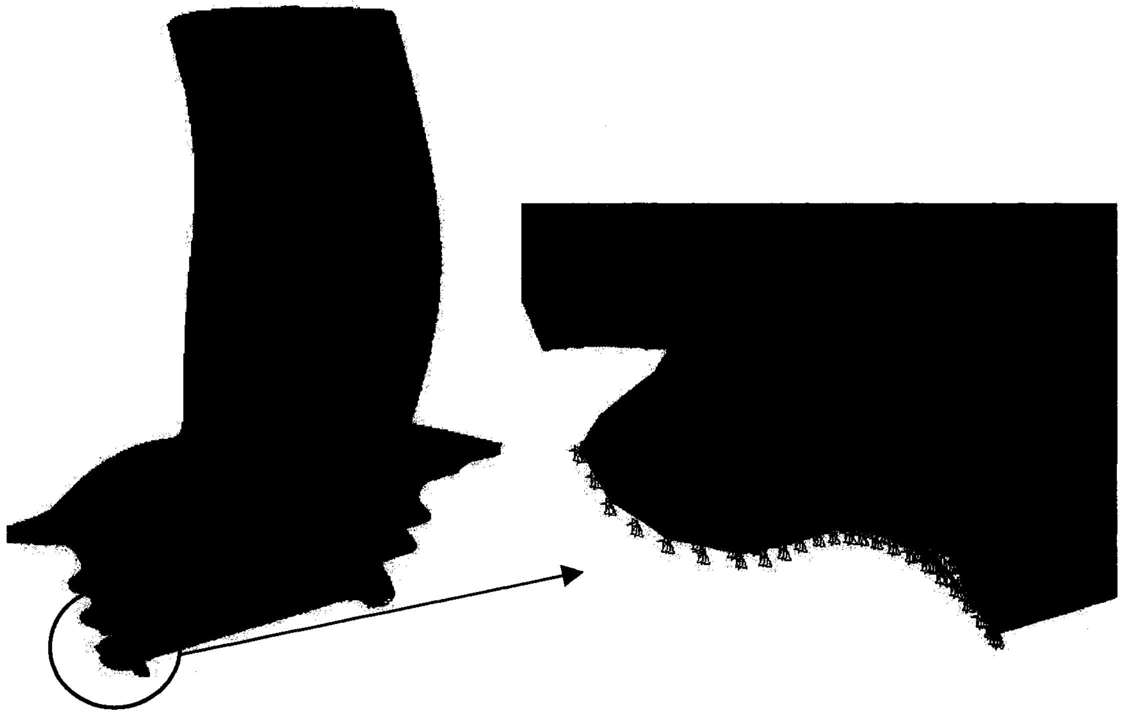

Figure 12 FEM model experimental boundary conditions

5.2 Response Signature Recording

When performing modal testing, usually a hammer is used to excite the component while an accelerometer is used to register the response signal of the component. This is not a concern when the component weighs significantly more than the accelerometer. In this case, the weight on the CT blade is less than ten (10) times the weight of the smallest accelerometer. Therefore, to avoid the shift in frequency due to weight of the accelerometer, a PolyTec laser vibrometer will be used instead (Figure 13).

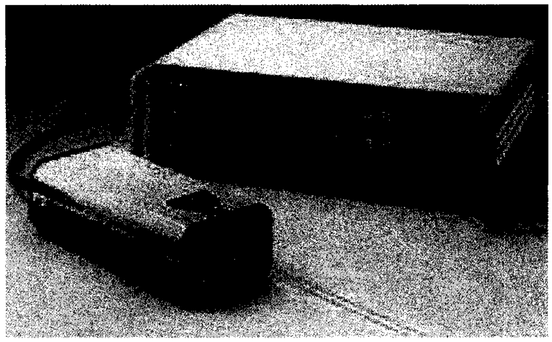

Figure 13 PolyTec Laser Vibrometer

The laser vibrometer will record nine (9) different points of the blade's airfoil so that a mode shape can be created using all the signals (Figure 14).

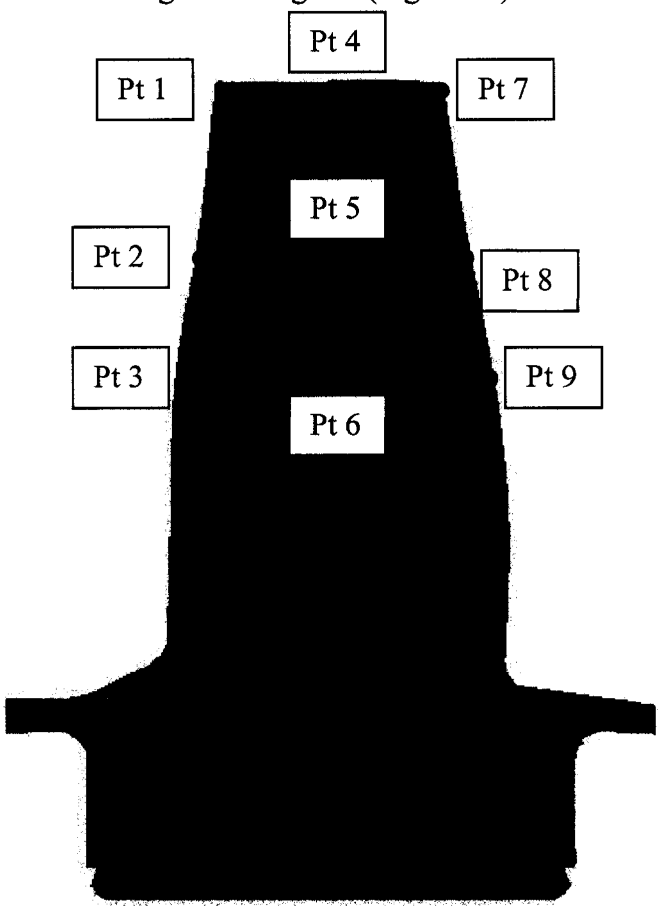

Figure 14 Blade signal recording locations

5.3 Excitation

To excite the CT blade, instead of a typical hammer, a high frequency speaker, JBL Professional Series Model No. 2425 coupled to a Model No. 2306 horn will be used (Figure 15).

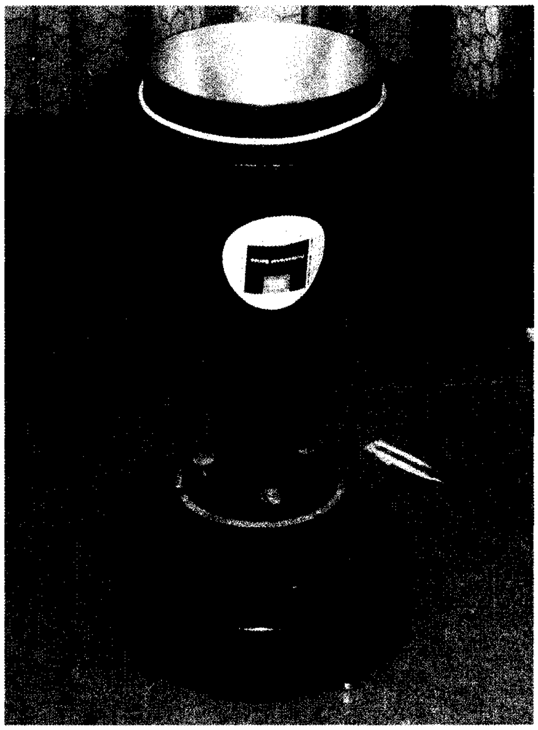

Figure 15 JBL Professional Series Model No. 2425 High Frequency Speaker coupled to Model No. 2306 Horn

A frequency generator is used to create the sine sweep from 2000 to 20000 Hz. The frequency generator is connected through a mixer Mackie Micro Series 1202-VLZ and then to an amplifier from TOA Corporation Dual Power Amplifier Model: IP-300D from which its output is connected to the JBL high frequency speaker.

5.4 Data Acquisition

The data acquisition system used is a Zonic Medallion, 8 channels 0 to 20 kHz. The velocity output of the vibrometer is directly connected to one of the channels. The output of the speaker is recorded through a sensitive microphone placed next to the blade and connected to another channel. The data acquisition parameters used during the experimental testing will give the best frequency resolution throughout the 2000 to 20000 Hz frequency range. The data acquisition program generates a Frequency Response Function (F.R.F.) by dividing the laser vibrometer signal by the microphone signal. The real and imaginary parts of the F.R.F will be used to determine the natural frequencies of the blade and the associated mode shape. The results are presented in section 7.3.

CHAPTER 6

VIBRATORY STRESS ANALYTICAL PREDICTION

Turbine blades are subjected to vibratory stresses due to unsteady flow in the gas path. The unsteadiness of the flow creates different load paths on the blade airfoil and coupled with the natural mode shape of the blade at that exact frequency, resonance occurs at which, high vibratory stresses are associated. This problematic is also known as "aero elasticity". Many sources of unsteady flow exist in turbomachines, such as:

- Blade / Vane wakes
- Blade / Vane potential fields
- Tip vortices
- End wall vortices

Most unsteady flows are circumferentially periodic and an integer multiples of rotor speed.

To predict turbine blade vibratory stresses analytically, both FEM modal solution and CFD solution at the resonance speed have to be coupled [2]. The modal solution was presented in section 4.2. The CFD solution is not presented or studied in this research but in a condensed form; an Euler CFD solution is performed to determine the steady part of the flow and to calculate the aerodynamic damping. The unsteady part of the flow is determined using a Navier-Stokes CFD solution where a turbulence model is introduced. Both steady and unsteady (vs. time) solutions are required to predict the aerodynamic load on the turbine blade.

6.1 FLARES Analytical Tool

To couple the aerodynamic solution to the mechanical model, an analytical tool called "FLutter And REsonance Stress Prediction System" (FLARES) has been developed by Pratt & Whitney East Hartford [2]. The following description of the code has been derived from the FLARES technical manual.

For FLARES to determine the vibratory stresses, it has to solve the following turbomachinery aeromechanics equation:

$$[M]\left\{\ddot{u}\right\}+[C(\ddot{u},\dot{u},u,\Omega)]\left\{\dot{u}\right\}+[K(u,\Omega)]\{u\}=\left\{P(\ddot{u},\dot{u},u,t)\right\}+\{F(u,\Omega)\} \tag{6.1}$$

[M]	Structural mass matrix
[C]	Structural damping matrix
[K]	Geometrically nonlinear stiffness matrix including centrifugal stiffness and softening
{F}	Nonlinear centrifugal force vector
{u}	Structural position vector
Ω	Engine Rotational Speed
t	Time (sec)

For the aerodynamic part, equation 6.1 can be solved with the steady state equation at the blade running position and steady stress. The perturbation assumption plus the separation of motion and gust loads are solved with the following equations:

$$\{u\}=\{\overline{u}\}+\{\widetilde{u}(t)\} \tag{6.2}$$

$$\left\{P(\ddot{u},\dot{u},u,t)\right\}=\{\overline{P}(\overline{u})\}+\{\widetilde{P}_M(\widetilde{u},\overline{u})\}+\{\widetilde{P}_G(\overline{u},t)\} \tag{6.3}$$

$\{\overline{u}\}$	Time-averaged position
$\{\widetilde{u}(t)\}$	Time dependent displacement
$\{\overline{P}(\overline{u})\}$	Time average aerodynamic forces

$\{\widetilde{P}_M(\widetilde{u},\overline{u})\}$ Airfoil vibratory motion dependent forces

$\{\widetilde{P}_G(\overline{u},t)\}$ Unsteady aerodynamic forces caused by "gust".

The aeroelastic equation is separated into the independent part of the airfoil vibratory motion:

$$[K(\overline{u},\Omega)]\{\overline{u}\}=\{P(\overline{u})\}+\{F(\overline{u},\Omega)\} \tag{6.4}$$

$[K(\overline{u},\Omega)]$ Geometrically nonlinear stiffness matrix

$\{F(\overline{u},\Omega)\}$ Nonlinear centrifugal force

Equation 6.4 is iteratively solved for $\{\overline{u}\}$.

The turbine blade natural frequency and mode shape for the specific resonance speed is solved while assuming an airfoil simple harmonic motion and in a vacuum structural dynamics:

$$\{\widetilde{u}(t)\}=\{\phi\}e^{iwt} \tag{6.5}$$

$$[[K(\overline{u},\Omega)]-\omega^2[M]]\{\phi\}=0 \tag{6.6}$$

$\{\varphi\}$ Mode shape, eigenvector

ω Natural frequency, eigenvalue

The assumption of the airfoil simple harmonic motion is based on PWC's experience where the HPT or CT blade modes are not coupled. The mode shape and natural frequency are solved using ANSYS®.

The turbine blade vibratory motion is a linear combination of orthogonal mode shapes:

$$\{\widetilde{u}\}=[\Phi]\{q\} \tag{6.7}$$

The motion dependent loads are a sum of loads from orthogonal mode shapes:

$$\{P_M\}=[P(\Phi)]\{q\} \tag{6.8}$$

$[\Phi]$	Normal modes
$\{q\}$	Normal or modal coordinates
$[P(\Phi)]$	Aerodynamic forces from the normal modes.

If substituted in the aeroelastic equation and premultiplied by $[\Phi]^T$, the orthogonality of $[\Phi]^T$ is creating an advantage and the assumed structural damping is represented by ζ:

$$[I]\left\{\ddot{q}\right\}+\left[2\zeta_j\omega_j\right]\left\{\dot{q}\right\}+\left[\omega_j^2\right]\{q\}-[Q]\{q\}=\{L(\omega)\} \tag{6.9}$$

While simple harmonic motion is assumed: $\{q\}=\{q_0\}e^{iwt}$

$$\left[\omega^2\left[\frac{\omega_j^{\ 2}}{\omega^2}-1+2i\zeta_j\frac{\omega_j}{\omega}\right]-[Q(\omega)]\right]\{q_0\}=\{L(\omega)\} \tag{6.10}$$

$[Q(\omega)]=[\Phi]^T[P(\Phi)]$: the generalized airfoil motion dependent forces;

$[L(\omega)]=[\Phi]^T[P_G(\omega)]$: the generalized gust dependent model force; $[P_G(\omega)]$ is the Fourier Transform of $[P_G(t)]$.

Therefore, for a single mode:

$$q_0=\frac{L(\omega)}{\mathrm{Im}\left[2i\zeta\omega^2-Q(\omega)\right]} \tag{6.11}$$

Knowing the modal coordinates {q}, the physical vibratory displacements and stresses $\{\tilde{u}\}$, can be determined by:

$$\{\tilde{u}\}=[\Phi]\{q\} \tag{6.12}$$

Practically, the single mode equation can be written this following way:

$$\sigma=F\cdot\sigma_{\mathrm{mod}\,al} \tag{6.13}$$

where:

$$F = \frac{\pi L}{\omega^2 (\delta_{aero} + \delta_{mech})} \tag{6.14}$$

σ	Vibratory stress for a single mode
$\sigma_{\mathrm{mod}\,al}$	Modal vibratory stress for a single mode (given by ANSYS)
F	Modal amplification factor
L	Modal force
δ_{aero}	Aerodynamic damping (logarithmic decrement)
δ_{mech}	Mechanical damping (logarithmic decrement)

Using the CFD analyses for particular resonance speeds, the modal amplification factor will be determined by FLARES. This factor will then multiply the modal vibratory stress vector in ANSYS® to obtain the analytical predicted turbine blade resonance vibratory stresses. The results are presented in section 7.4.