Turbine engine rotor blade fault diagnostics through casing pressure and vibration sensors

J Cox and P Anusonti-Inthra
University of Tennessee Space Institute, 411 B H Goethert Pkwy, Tullahoma, TN 37388, USA
Email: ry_co@hotmail.com

Abstract. In this study, an exact solution is provided for a previously indeterminate equation used for rotor blade fault diagnostics. The method estimates rotor blade natural frequency through turbine engine casing pressure and vibration sensors. The equation requires accurate measurements of low-amplitude sideband signals in the frequency domain. With this in mind, statistical evaluation was also completed with the goal of determining the effect of sampling time and frequency on sideband resolution in the frequency domain.

1. Introduction
Rotor blade failures cause catastrophic damage to turbine engines, and therefore blade faults must be found before they are allowed to propagate. A common method of fault detection involves measuring shifts in the rotor blade natural frequencies, which can indicate blade damage. Current methods of rotor blade vibratory frequency identification are costly and overly intrusive, including drilling holes in the casing surface, or placing sensors directly on rotor blades, in the harsh environment of the engine internal flow path. A new, less intrusive method in development uses the engine casing vibratory response to determine rotor blade vibratory frequency.

Forbes and Randall determined that the turbine engine casing acts as a filter to the internal dynamic pressure, and so the frequencies contained in both signals are the same [1]. In a turbine engine fan, the casing internal pressure peak travels with the rotor blade tips. The resulting internal dynamic pressure signal contains the blade pass frequency (BPF), the number of blades multiplied by the rotor rotational frequency. Blade vibration causes modulation in the dynamic pressure signal, leading to sidebands in the frequency domain. The casing internal pressure fluctuation is not sinusoidal and therefore includes a number of harmonics. The possible harmonics include all integral multiples of the engine rotational frequency, not just the BPF.

Forbes and Randall showed that each harmonic of the rotor speed carrier frequency produces its own set of sidebands, as shown in Figure 1 [1]. To determine rotor blade vibratory frequency from the casing internal pressure response in the frequency domain, Forbes and Randall developed the following equation

$$\omega = \frac{\Delta f + q\Omega}{2},$$  \hspace{1cm} (1)

where $\omega$ is the blade vibratory frequency, $\Delta f$ is the sideband spacing, $\Omega$ is the rotor rotational frequency, and $q$ is an integer based on the possible sources of the sideband frequencies [1]. Equation
1 is linear in two variables (ω and q), and thus indeterminate. Additionally, the amplitude of the sidebands in the frequency response is on the order of the blade deflection amplitude, which may fall well below the noise floor.

![Figure 1](image.png)

**Figure 1.** Blade vibration causes sidebands equally spaced around multiples of rotor speed, adapted from [1].

![Figure 2](image.png)

**Figure 2.** Assuming a constant modulation frequency, the sidebands shift with the carrier frequency.

2. Method

To overcome the indetermination problem of Equation 1, a rotor speed change was employed, such that a system of equations was formed. The system of equations was then solved to determine blade vibratory frequency.

An initial study of the amplitude problem was undertaken to determine the effect of a number of variables on the detectability of the sidebands in the frequency domain. Design of experiments (DOE) and analysis of variance (ANOVA) were used to determine influence of the blade vibratory amplitude, the signal-to-noise ratio, the sampling time, and the sampling frequency on the sideband and noise amplitudes in the frequency domain.

3. Results

The indetermination problem of Equation 1 is overcome by employing a rotor speed change (rotor frequency $\Omega_1$ to $\Omega_2$), assuming a constant rotor blade natural frequency, $\omega$, as shown in Figure 2. The rotor speed change produces a set of linear equations,

\[ m\Omega_1 \pm \omega = f_{sb1}, \]
\[ m\Omega_2 \pm \omega = f_{sb2}, \]

where $m$ is a multiple of rotor speed, and $f_{sb1}$ and $f_{sb2}$ are the sideband frequencies at rotor frequencies $\Omega_1$ and $\Omega_2$ respectively. Equations 2 and 3 are solved through substitution to obtain

\[ \omega = \frac{\Omega_1 f_{sb2} - \Omega_2 f_{sb1}}{\Omega_2 - \Omega_1}. \]

The question of sideband amplitude was then studied using simulated data. An equation was derived to represent the pressure at a point on the internal surface of the fan casing, in the direct path of the fan blades. A simple modulated sine wave was used to represent the casing internal surface dynamic pressure, as shown

\[ P_n = rand(-1,1) + \frac{(2SNR)^{1/2}}{6.6} \cos(2\pi b t + \epsilon \cos(2\pi t)). \]
where $P_n$ is the noise-normalized pressure, $SNR$ is the signal-to-noise ratio, $b$ is the number of rotor blades, $t$ is time, and $\epsilon$ is the rotor blade vibratory deflection amplitude. The objective of Equation 5 is to provide a means to determine the sampling time ($t_f$) and frequency ($f_s$) required to resolve the sideband amplitude with a given $SNR$ and $\epsilon$.

Principles from design of experiments (DOE) and analysis of variance (ANOVA) were used to determine the influence of each of the four main effects ($t_f$, $f_s$, $SNR$, and $\epsilon$) and any interactions. The sideband amplitude was compared to the highest noise peak amplitude, as shown in Figure 3, and the percent difference was recorded as the output variable.

Through ANOVA, it was found that the greatest factors affecting sideband amplitude were, in order from highest to lowest: 1) $\epsilon$, 2) $f_s$, 3) $SNR$, and 4) $t_f$, with two and three factor interactions contributing significantly less. The controllable factors, $f_s$ and $t_f$, contributed approximately 75% and 61% (respectively) as much as the highest contributing factor, $\epsilon$. As expected, noise in the FFT was reduced by using higher sampling frequencies and times.

4. Conclusions

In this study, an exact solution was determined for Forbes’ and Randall’s equation, such that the rotor blade frequency is determined by employing a shift in rotor speed, and then measuring the sideband frequency change as detected in the casing vibratory frequency response (or internal dynamic pressure frequency response). This equation works when the sideband amplitudes rise above the noise level.

Using simulated data, it was found that the rotor blade deflection amplitude has the largest influence on sideband detectability, followed by the sampling frequency, the signal to noise ratio, the sampling time, and various interactions.

Proposed future work includes 1) using a more realistic representation of the engine internal dynamic pressure, employing a saw tooth wave form and random rotor speed fluctuations for simulated data, and 2) collecting data from a fan in development at UTSI to analyze and verify the techniques proposed herein.

References

[1] Forbes G and Randall R 2013 Estimation of turbine blade natural frequencies from casing pressure and vibration measurements Mech. Syst. and Signal Process, http://dx.doi.org/10.1016/j.ymssp.2012.11.006.