I, Christopher G Larsen, hereby submit this original work as part of the requirements for the degree of Master of Science in Mechanical Engineering.

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Structural FRF Measurements up to 50 kHz to Assist Frequency Band Selection for Machinery Health Monitoring

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Structural FRF Measurements up to 50 kHz to Assist Frequency Band Selection for Machinery Health Monitoring

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1 Abstract

To set up condition indicators for machinery health monitoring it is preferable to run multiple machines with seeded defects and with no defects and measure the resulting vibration signatures; however, this can be a prohibitively expensive undertaking, especially when the machinery has many monitored components. Instead, measurement of the defect-to-monitoring-sensor transfer path dynamics provides a picture of the frequency bands which will amplify the desired vibration signal and those bands which will attenuate it. Since measurement of the true transfer path (from gear tooth or bearing race to the vibration sensor) can also be a costly task, this work seeks to demonstrate that, in cases where the monitored bearing is near the machine surface, a transfer path measurement from the surface near the bearing load zone reasonably approximates the true transfer path, sufficiently for selecting frequency bands of vibration signal amplification and avoidance of bands of signal attenuation.

To demonstrate this in the lab, piezoelectric bearing excitation devices were constructed and placed in a US Army Apache helicopter’s intermediate gearbox and the true-transfer-path measurements were compared to surface-mounted excitation FRFs. As these compared favorably, the surface measurement method was used on all the gearboxes for each of the following rotorcraft: Apache, Kiowa, Chinook, and Blackhawk. Since the Chinook and the Blackhawk are built in a special-ops version which is equipped with a rotor brake, those components were measured on both standard and special-ops versions. In the interest of space, only the complete results from the Blackhawk are shown here; however, the US Army has made all of the data and reports from the project publicly available.
2 Acknowledgments

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3  Acronyms Used

HUMS – Health and Usage Monitoring System
RMS – Root Mean Square
PSD – Power Spectral Density
DFT – Discrete Fourier Transform
FFT – Fast Fourier Transform
FRF – Frequency Response Function
IGB – Intermediate Gearbox
TGB – Tail Gearbox
TTO – Tail Takeoff
MXMSN – Main Transmission
4 Introduction

The goal of this work is to demonstrate that high-frequency transfer path measurements can be made on rotating machinery to assist in the selection of frequency bands for enveloping/demodulation techniques for condition-based health monitoring. The motivation was to improve the false alarm and missed detection rates of the US Army’s helicopter monitoring systems. These transfer-path measurements were made from 0 – 50 kHz, requiring the design and selection of a measurement system and the demonstration that these high-frequency measurements are repeatable and meaningful.

Demonstration that surface-excitation FRFs near a bearing’s load zone are similar to “true transfer path” FRFs was also done, since surface-excitation FRFs was the proposed measurement method to be used. Additionally, it was shown that gearbox preload did not significantly alter the measurements, giving further confidence that the measurements would be meaningful and useful.

Finally, these measurements were made on six Army rotorcraft: two versions of the Blackhawk, two versions of the Chinook, the Kiowa, and the Apache, though in the interest of space, only the data from the Blackhawk test is presented here. The data is currently being used by the Army to adjust the frequency bands for the enveloping/demodulation techniques in the installed HUMS monitoring systems. All of the data and reports from this project have been made public by the Army (the public release notice can be found in the Appendix, Notice of Public Release).
5 Background

Helicopters are complex systems with little in the way of redundancy. While fixed-wing aircraft can generally glide some distance after a complete engine failure, any major failure in a rotary-wing aircraft’s drivetrain will typically result in a crash. While maintenance is conservatively scheduled to inspect and replace wear items like gears and bearings before they pose a threat to a helicopter’s crew, components can fail prematurely, due to debris, improper installation or maintenance, or manufacturing defects. Unfortunately, it was a crash of a BV 234 (a commercial version of the US Army Chinook) in the North Sea in November, 1986 that claimed 45 lives which lead to the requirement that all helicopters operating in that area have a health-monitoring system [1].

The accident investigation concluded that failure of the ring gear in the forward transmission caused the forward and aft rotors to collide. The aft pylon (which attaches the engines, the combining transmission, and the rear transmission, see Figure 1) was torn from the craft, which then immediately plummeted to the sea. Miraculously, there were two survivors.
The airframe was only five years old and had been inspected and serviced in accordance with all requirements. The forward transmission had been overhauled the year prior and fitted with the new standard -6 spiral bevel ring gear. Further inspections (even at 120 service hours before the crash) all showed the transmission to be in good working order. The gearboxes were also equipped with chip-detection devices which monitor for metal particles in the lubrication oil, and these showed no signs of impending failure.

The accident investigation concluded that the primary failure was circumferential cracking of the attachment flange of the forward transmission spiral bevel ring gear. A small groove formed by wear and corrosion in the maritime environment lead to fatigue cracks which ultimately caused the gear to
fail. Slight design changes between the -5 and the -6 revisions of the gear caused the wear of the groove in the new gear to be a critical failure point, and the process that certified the new revision was also flawed, as it was demonstrated in an aft transmission, which did not wear in the same way. Examination of other in-service -6 ring gears showed the initiation of fatigue cracks in many cases, and an immediate redesign was called for.

While the redesigned ring gear and the certification process were clearly to blame for the accident, the report also plainly showed that health monitoring would have saved these lives. Other than chip detection (maintainers look for magnetic particles on the oil drain plugs), vibration health monitoring was generally not done at this time; however, there was an audio flight recorder. Spectral analysis of the audio showed that for the entire thirty minutes of recorded time, the forward spiral ring gear mesh frequencies were definitely perceptible (and quite loud minutes before the crash) and probably could have been detected many hours beforehand [2]. In 1990 the UK CAA (Civil Aviation Authority) mandated that all rotorcraft operating in the North Sea area would have to be equipped with health monitoring devices beginning in December 1991.

While the first monitoring systems installed were certainly an aid to reduced accidents due to mechanical failure, these systems were plagued with false alarms and some missed detections. A high false-alarm rate is a significant problem, as the “boy who cries wolf” is quickly ignored, particularly after a maintenance depot spends tens of thousands of dollars replacing costly components which were in perfectly good health. Adjusting the monitoring-systems’ algorithms to reduce the false alarm rate can also increase the missed-detection rate, and failing mechanical components may be missed, leading to a crash.

Both commercial interests and the US Army have funded considerable efforts to develop better monitoring systems over the years; in 2005 the Army approved current systems for widespread use on the fleet [3]. IAC (Intelligent Automation Corporation) and Goodrich are both major manufacturers of
these monitoring systems, now known as HUMS (Health and Usage Monitoring Systems), and both
companies are suppliers to the US Army. While the primary benefit has always been safety,
maintenance savings have also been realized; Goodrich claims that in one year of use, HUMS saved the
4th Battalion’s 101st Aviation Regiment $44.9 million in parts and maintenance man-hours [3].

While the Army has had many success stories with HUMS, there are still some bearings on some
craft which demonstrate a high false-alarm rate, and there have been some documented missed
detections. It is the goal of this work to improve these rates through better understanding of the
vibration transfer paths, from bearing defect to monitoring sensors.

5.1 Bearing Fault Detection

When rolling-element bearings begin to fail, generally a small spall begins to develop on one of
the bearing races or rolling elements. As the rolling elements spin over the spall, this creates periodic
impulses. The period of these impulses is dependent on the bearing geometry and the rotating speed.
The outer race defect frequency (ballpass frequency over the outer race) is:

\[ BPFO = \frac{n_f r}{2} \left( 1 - \frac{d}{D} \cos \phi \right) \]  \hspace{1cm} (4.1)

The inner race defect frequency (ballpass frequency over the inner race) is:

\[ BPFI = \frac{n_f r}{2} \left( 1 + \frac{d}{D} \cos \phi \right) \]  \hspace{1cm} (4.2)

The fundamental train frequency (cage speed) is:

\[ FTF = \frac{f_r}{2} \left( 1 - \frac{d}{D} \cos \phi \right) \]  \hspace{1cm} (4.3)

The ball (or roller) spin frequency is:

\[ BSF = \frac{D}{2d} \left[ 1 - \left( \frac{d}{D} \cos \phi \right)^2 \right] \]  \hspace{1cm} (4.4)
Where \( f_r \) is the shaft speed, \( n \) is the number of rolling elements, \( d \) is the individual rolling element diameter, \( D \) is half the diameter between the inner and outer races, and \( \phi \) is the angle of the load from the radial plane [4, pp. 47 - 47].

The Fourier transform of these repeated impulses (also known as a “Dirac comb”) is also an impulse train in the frequency domain. This means that the base defect frequency and its harmonics are found in the measured response, and this can be advantageous for defect detection. If the harmonics are amplified by the transfer path, they can be easily detected. Looking for harmonics of defect frequencies is generally known as “Enveloping” or “Demodulation.” This is often done with a zoom-FFT, but can also be done using the Hilbert transform [4, p. 96].

Even in the idealized model where the bearing defect generates a perfect impulse train, the measured response at the monitoring accelerometer is highly dependent on the transfer path. The measured acceleration due to the bearing defect is essentially the defect force convolved with the transfer path dynamics; this means that one cannot blindly look for just any harmonic of the base defect frequency, as a zero in the transfer path at that frequency will yield very poor energy transmission, and the defect frequency will be difficult to detect.

This can easily be demonstrated with a numerical model. Consider a bearing defect with a fundamental frequency of 1 kHz; the power spectrum of this defect with its harmonics (and some added noise) is shown in the top of Figure 2. The defect transfer path is shown in the bottom of Figure 2. The measured response (i.e. the input force convolved with the transfer path) is shown in Figure 3. Because of the FRF peak at 20 kHz, there is a large measured response signal, and the green band shown in Figure 3 would be an ideal region for enveloping. However, due to the zero in the transfer path at 30 kHz, there is very little response there, and trying to envelope in the red region shown in Figure 3 would not provide any information. The Matlab code for this model can be found in the Appendix (transferPathHole.m).
Note that there are dozens of other signal processing techniques for detecting failing bearings, such as:

- **Minimum Entropy Deconvolution** – an adaptive filter is used to maximize kurtosis in the bearing vibration, with the assumption that the forcing function is made up of impulses.
- **Acoustic Emission** – essentially monitoring for elevated broadband noise due to rolling elements impacting a defect. Usually between 20kHz and 1MHz.
- **Shock-Pulse-Method (SPM)** – a form of enveloping/demodulation where the chosen frequency band is simply at the accelerometer’s resonance
- **Kurtosis** – 4th statistical moment – a measure of the "peakedness" of the probability distribution of the signal (a normal distribution has zero kurtosis)
- **Bispectrum** – looking for a family of harmonics of a defect frequency. Should be correlated by phase, large values at frequency pairs \((2f,f), (3f,f), (2f,4f),\) etc.
- **RMS (Root Mean Square)** – tracking the overall energy content of the signal
- **Cepstrum Analysis** – the cepstrum is the inverse Fourier transform of the log of the magnitude of the Fourier transform. Peaks in the ‘quefrequency’ (the so-called time domain of this inverse Fourier transform) indicate periodicity in the frequency domain, which identifies families of harmonics
- Variance – measures the statistical dispersion of a signal

- Crest Factor – peak amplitude of a waveform divided by the RMS value, gives a measure of how "peaky" a waveform is (or measure of impacting). A perfect sine wave of amplitude 1 (RMS = 0.707) has a crest factor of 1.41; crest factors higher than this have some degree of impacting.

- M6 – sixth statistical moment

- Envelope RMS – generally the RMS energy content in a given frequency band

- Short Time Fourier Transform – used to determine the sinusoidal frequency and phase content of local sections of a signal as it changes over time. Helpful with non-constant shaft speeds.

- Short Time Energy – a demodulation technique that employs a sliding window to calculate the variation of vibration energy over time.

- Autocorrelation – cross-correlation of a signal with itself (a measure of similarity of between observations as a function of time separation between them). Finds "hidden" repeated events in a signal (musical beats, bearing impacts, etc.).

These techniques are not currently used in the HUMS systems and will not be discussed further here.

### 5.2 Condition Indicators

Even healthy bearings may exhibit some low-level amplitude of their defect frequencies; setting up an “alarm,” or condition indicators requires some knowledge of how the amplitude of the defect frequencies changes with defect severity. For a simple, single-speed machine, it is usually possible to make some baseline measurements when it is new or the bearings are known to be in good condition and then simply monitor for increased vibration amplitude, particularly at the known defect frequencies. However, any change in operating conditions such as speed, load, temperature, etc. can also cause a change in the measured acceleration amplitudes – it is very difficult to distinguish these changes from actual changes in the bearing condition. For this reason, setting up condition indicators is generally a
non-trivial task – ideally, one should have a large matrix of data: known good bearing data from multiple machines as well as known faulted bearing data and all of this across all relevant operating conditions [5].

Obviously, this testing, particularly acquiring the known faulted bearing data, can be very costly. For example, the ideal test plan for gathering this data for the Blackhawk helicopter would require running each of its seven gearboxes on a test rig with seeded defects of each type (outer race, inner race, and rolling element) in each bearing individually, and doing this over a range of operating conditions. Clearly, this would be a monumental task for just one model of helicopter, and the US Army uses several; this testing would have to be done for each.

5.3 Transfer Path Measurements

An alternative is to measure the vibration transfer paths, from the bearing defect to the monitoring accelerometer. This provides clear information about frequency bands which will readily transmit energy and bands which do not. This does not provide a complete picture; however, as the frequency content of the defect force is unknown. That is, one might choose an enveloping band in a high-frequency region of strong structural response, but a typical defect in the bearing of interest may not excite the structure at sufficiently high frequency. Still, knowledge of the transfer path dynamics is definitely superior to simply guessing at the frequency bands chosen for demodulation.

Measuring the transfer path is only somewhat easier than running seeded-defect tests, however. Exciting at the defect locations is extremely difficult, and this must be done with the gearbox in its assembled condition, because if it is disassembled for access, the system dynamics will be considerably different. The excitation device must be installed at each bearing defect location, and this means disassembling each gearbox at least once. Additionally, since monitoring is to be done up to 50 kHz, the measurements have to be made up to 50 kHz.
However, many of these bearings are near the surface of the gearbox, and it was found that these FRFs are similar enough to the true transfer path FRFs for the purpose of selecting enveloping bands (this will be demonstrated later). These measurements are simple and easy to make, as they can be done directly on a helicopter in the hangar without any major disassembly.
6 Piezoelectric Bearing Force Transducers

To make the “true” transfer path measurements, the bearings of interest were modified to make them into excitation devices. One rolling element was removed from the bearing and replaced with a piezoelectric reaction-mass shaker coupled with a piezoelectric force transducer [6].

6.1 Piezoelectric Chip Selection – Resonance Modeling

For a force sensor to provide flat response up to the desired measurement frequency, its first resonance must be sufficiently above the measurement range such that the response is strictly along the “stiffness line.” The piezoelectric elements selected for this testing were from Physikinstrumente: 5x5x2 mm miniature multilayer piezo stack actuators, model PL055. The resonant frequency of these actuators is listed as “> 300 kHz;” however, it was decided that some simple modeling to estimate the free-free response of the chips would be in order. This form of the piezoelectric constitutive equations relates stress, strain, voltage, and the piezoelectric coupling constants [7]:

$$\mathbf{S} = \mathbf{s}_e \mathbf{\sigma} + \mathbf{d}^T \mathbf{V}$$  \hspace{1cm} (4.1)

where \(\mathbf{S}\) is a 6x1 vector of the three normal strains and three shear strains, \(\mathbf{s}_e\) is a 6x6 matrix of coupled compliance terms (which account for Poisson’s ratio), \(\mathbf{\sigma}\) is a 6x1 vector of the three normal and three shear stresses, \(\mathbf{d}\) is a 3x6 matrix of the piezoelectric coupling constants, and \(\mathbf{V}\) is a 3x1 vector of the applied voltages per length of piezoelectric material (generally only one of these is non-zero). Displacement is simply the normal strain multiplied by the original length. Note that unconstrained (free-free), the stress \(\mathbf{\sigma}\) is zero (also assuming a static application), and the displacement is directly proportional to the applied voltage. To determine the resonant response, however, it is assumed that the chip will be strained by its own inertia as it moves due to an applied voltage. For a simple approximation, it will be assumed that \(\frac{1}{2}\) the mass contributes to this inertia. The stress is then the inertial force divided by the cross-sectional area. Figure 4 shows the piezo with the coordinate system.
Assume voltage is only applied in the z-direction and ignore shear stresses; the constitutive equation can be expanded:

\[
\begin{pmatrix}
\frac{1}{h} & 0 & 0 \\
0 & \frac{1}{a} & 0 \\
0 & 0 & \frac{1}{a}
\end{pmatrix}
\begin{pmatrix}
x \\
y \\
z
\end{pmatrix}
= s_E \left( \frac{m}{2} \right)
\begin{pmatrix}
\frac{1}{A_x} & 0 & 0 \\
0 & \frac{1}{A_y} & 0 \\
0 & 0 & \frac{1}{A_z}
\end{pmatrix}
\begin{pmatrix}
\ddot{x} \\
\ddot{y} \\
\ddot{z}
\end{pmatrix}
+ \begin{pmatrix}
0 \\
0 \\
\frac{v_{13}}{a}
\end{pmatrix}
\] (4.2)

Now the equation is Fourier transformed, and the appropriate areas are substituted in:

\[
\begin{pmatrix}
\frac{1}{h} & 0 & 0 \\
0 & \frac{1}{a} & 0 \\
0 & 0 & \frac{1}{a}
\end{pmatrix}
\begin{pmatrix}
X \\
Y \\
Z
\end{pmatrix}
+ s_E \left( \frac{m(j\omega)^2}{2} \right)
\begin{pmatrix}
\frac{1}{a^2} & 0 & 0 \\
0 & \frac{1}{ah} & 0 \\
0 & 0 & \frac{1}{ah}
\end{pmatrix}
\begin{pmatrix}
X \\
Y \\
Z
\end{pmatrix}
= \begin{pmatrix}
0 \\
0 \\
\frac{V_{33}}{a}
\end{pmatrix}
\] (4.3)

Rearranging to produce a displacement/voltage FRF yields:
\[
\begin{bmatrix}
\frac{1}{h} & 0 & 0 \\
0 & \frac{1}{a} & 0 \\
0 & 0 & \frac{1}{a}
\end{bmatrix} + S_E \left( \frac{m(j\omega)^2}{2} \right) \begin{bmatrix}
\frac{1}{a^2} & 0 & 0 \\
0 & \frac{1}{ah} & 0 \\
0 & 0 & \frac{1}{ah}
\end{bmatrix} \begin{bmatrix}
X \\
Y \\
Z
\end{bmatrix} = d^T \begin{bmatrix}
0 \\
0 \\
\frac{V_{33}}{a}
\end{bmatrix}
\]

(4.4)

\[
[H] = \left[ \begin{bmatrix}
\frac{1}{h} & 0 & 0 \\
0 & \frac{1}{a} & 0 \\
0 & 0 & \frac{1}{a}
\end{bmatrix} - S_E \left( \frac{m\omega^2}{2} \right) \begin{bmatrix}
\frac{1}{a^2} & 0 & 0 \\
0 & \frac{1}{ah} & 0 \\
0 & 0 & \frac{1}{ah}
\end{bmatrix} \right]^{-1} d^T
\]

(4.5)

This was modeled in Matlab (code in Appendix: piezoModel.m), and the FRFs are shown in Figure 5. It can be seen that the 1st resonance is at about 100 kHz, and the response is flat to 20 kHz and very close to flat to the desired 50 kHz.
This simple model does not account for shear strain, and it simply assumes that the inertia in each direction is due to half the mass. Additionally, the chip itself is a multi-layered piezo stack; that is, it is created like a “sandwich” of even smaller chips – this produces more displacement or force at lower voltage. However, based on this simple model and the manufacturer’s assertion that this chip’s resonance is over 300 kHz, it was assumed that these piezo elements would provide sufficiently accurate force measurements.

6.2 Excitation Signal and FRF Computation

The choice of excitation signal was very important for making good quality measurements. Piezo actuators are driven with a command signal from a function generator or acquisition system source card, and this command signal must be amplified, generally by an amplifier designed specifically for piezoelectric elements. Driving noise into a piezo element for excitation spreads the available power from the amplifier over the entire spectrum, and the power driven at any given frequency is very low. This can be illustrated as follows. Assume that the amplifier has available power $P$. A signal’s power in a given frequency band can be calculated by integrating the PSD over that frequency band, as shown in Equation (4.6) [8].

$$P = \int_{f_1}^{f_2} S(f) df$$  \hspace{1cm} (4.6)

Therefore, if the amplifier’s power is spread evenly over 50 kHz, the available power-per-Hz will be $1/50,000$ the total available power. This would provide inadequate force input into the system and the measurements would be very poor.

Driving a fixed frequency can provide all the available power from the amplifier at that frequency, and this provides the best signal-to-noise ratio possible; additionally, the selected frequency can be driven for as long as desired, permitting significant averaging. This is called stepped-sine
excitation. The FRF amplitude and phase can be estimated using only a 2-channel oscilloscope, and if amplitude is the only interest it could even be done with calibrated RMS values from a volt meter. The obvious downside for this method is that this must be repeated for each desired spectral line, and this can be very time consuming.

An excellent alternative to stepped-sine excitation is a chirp. A five-second chirp swept from 0 – 50 kHz was chosen to provide the best balance between testing speed and accuracy. The actual source used was a VXI E1445a card – this is essentially a typical function generator in a VXI ‘C-size’ card [9]. This card does not generate a true continuously-swept sine; it discretely steps through the frequencies – the desired frequency band is divided into 800 discrete frequencies, so for a 50 kHz sweep, the frequency interval is 62.5 Hz, which is more than fine enough for the larger general FRF trends that will dictate frequency-band selection. This effect is easily viewed when looking at the recorded force, as the amplitude steps slightly with frequency. While this does technically create a (very low level) step response as the source steps through the frequencies, both input and response are measured, so the FRF computation is unaffected. A typical force measurement example is shown in Figure 6.
One area of concern for using chirp excitation was the fact that very little time is spent at any given frequency – with a true continuous chirp, this is only an infinitesimal amount of time. It was thought that this could be an issue with heavily damped systems, as some time is required for the system to reach steady-state, and this might mean that the estimated FRF amplitude could be low. However, this is not the case: as long as both the force and the system response are accurately measured, the FRF will be correct. Of course, if the response amplitude is in the noise floor, the FRF estimate will be poor, but so will the coherence. This transient effect would only be an issue with stepped-sine excitation when estimating the FRF spectral line by spectral line and making a simple amplitude/phase estimate, as the response amplitude would be low until steady-state is reached.

Sixty seconds of data were collected for each test – this provides twelve complete chirps and twelve averages. The FRFs were processed such that the block size matches the chirp length, using a uniform window (i.e. no window), and no overlap processing. Because the blocks repeat with the chirps,
this satisfies the Discrete Fourier Transform assumption that the signal is repeated infinitely before and after the block being transformed [10], eliminating the possibility of leakage and the need for a window. In fact, the chirp does not have to be triggered with the acquisition system, because the block can begin somewhere in the middle of a chirp and still repeat itself. Periodicity of the DFT can be shown from the definition of the DFT itself:

\[
X_{k+N} = \sum_{n=0}^{N-1} x_n e^{-\frac{2\pi i}{N}(k+N)n} = \sum_{n=0}^{N-1} x_n e^{-\frac{2\pi i}{N}kn} e^{-2\pi i n} = \sum_{n=0}^{N-1} x_n e^{-\frac{2\pi i}{N}kn} = X_k
\] (4.7)

6.3 Reaction-Mass Excitation

The same type of piezoelectric element also served as the force transducer. A 2.0 gram \( \frac{1}{4}'' \) diameter tungsten-carbide sphere was used as the reaction mass. Tungsten carbide was chosen because its density is approximately twice that of steel, providing more mass for the limited space available. The force from a reaction-mass exciter is proportional to acceleration, and the displacement of the driving piezo element (as a 1st order approximation) is proportional to voltage. Consider equation (4.1), repeated here for convenience: \( S = s_e \sigma + d^T V \). If the piezoelectric element is blocked, the strain \( S \) is zero, and the stress is proportional to voltage. This is the maximum force available from a piezoelectric element. The force from the reaction-mass exciter is therefore (approximately) proportional to the square of frequency. This means that very little force will be available at low frequency; typically measurements below approximately 3 kHz were poor, so impact testing was done to fill in the low frequency range. Splicing impact FRFs with shaker FRFs is discussed below.

6.4 Crosstalk

One area of concern with this type of transducer is crosstalk between the excitation piezo element and the force-measurement piezo element, as this would seriously corrupt the FRF estimate.
To test the level of electrical crosstalk, two piezos were mounted in close proximity to one another using an alligator-clip fixture, as shown in Figure 7. The two piezos were separated by approximately 0.1 mm, which is the thickness of the mica used to separate and insulate the piezos from each other on assembly.

A 0 – 50 kHz chirp was used for excitation to one of the piezos, generated by the VXI E1445a card, and this was amplified by the ISI amplifier. The voltage of the un-driven piezo was measured, and the power spectrum is shown in Figure 8. The red trace shows the crosstalk of the current setup, the green trace shows the improved system, (discussed below), and the black trace shows the noise floor (no commanded voltage to the driving piezo element). The blue trace shows a typical measurement for comparison. Note that the force measurement power is more than four orders of magnitude higher than the crosstalk amplitude throughout the frequency range.
As this crosstalk rises with frequency, it was assumed that it is electrical interference between the long unshielded test leads, shown in Figure 9.

Figure 8: Crosstalk Power Spectra
Figure 9: Unshielded Test Leads

The test leads were shortened as much as possible, to a few millimeters, and soldered to shielded coaxial cables. For further testing and easier handling, one of the piezos was potted onto a ¾” stainless-steel hexagon. This is shown in Figure 10.

Figure 10: Pizeo Crosstalk Test Setup
There are four possible polarity orientations of the two piezos (assuming the sides of the two square piezos are to be parallel), and all four were tested to determine if one method is superior in reducing crosstalk. In short, the piezo orientation did not matter as far as crosstalk was concerned.

6.5 FRF Comparison with Commercial Shaker

FRFs made with the small reaction-mass piezoelectric exciter were compared with FRFs made with a commercial piezoelectric shaker. The force measurements for the commercial shaker were made with a high-frequency PCB load ring (model number 201B02). To perform this test, an Apache IGB was instrumented with six PCB high-frequency accelerometers (model number 352A60). The accelerometer mounting locations are shown in Figure 11 and Figure 12. The lab-built shaker is shown mounted on the gearbox in Figure 13 and Figure 14 shows the Piezomechanik shaker mounted in the same location on the same mounting pad.
The same chirp excitation was driven to each shaker; sixty seconds of data was recorded on both the force and response channels at 196,608 samples/second. FRFs were made for each case. The FRFs were normalized and plotted together; these are shown for three locations in Figure 15, Figure 16, and Figure 17. The FRFs made with the commercial piezoshaker are shown with the blue trace, and the FRFs made with the Etegent-built piezoshaker are shown with the red trace. It can be seen that they compare reasonably well; for the purposes of selecting good energy-transfer frequency bands, the results are perfectly adequate.

Figure 15: FRF to High Frequency Accel #2
Figure 16: FRF to High Frequency Accel #4

Figure 17: FRF to High Frequency Accel #5
7 Surface FRFs – Testing Methodology Confirmation

Because these measurements were made on in-service helicopters (meaning that the aircraft were scheduled for five days of testing each, but modifications which inhibit an immediate return to service were not permissible), there were severe limitations as to how sensors and the shaker could be mounted. Obviously, drilling and tapping holes was not an option. Therefore, the methodology used for the testing procedure had to be demonstrated to produce meaningful, repeatable results.

7.1 Threaded vs. Glued Mount

It is generally assumed that a threaded mount is the only way to make high-frequency structural measurements, and for good reason. Threaded mounts would seem to provide the most rigid, repeatable means of attaching transducers and sensors to a structure. However, most desired test locations on these gearboxes have no tapped hole, and it was not feasible to machine a flat surface and tap a hole in those locations for this testing. Therefore, it was necessary to test the effects of mounting the sensors with an adhesive.

For this test, a scrap aluminum block was drilled through and tapped 10-32 for mounting both the piezoshaker (Piezomechanik) with load cell (PCB 201B02) and the high-frequency accelerometer (PCB 352A60). These were screwed to the block, a 5 second 0 – 5 Volt, 0 – 50 kHz chirp was commanded to the exciter, and 30 seconds of time data was recorded. FRFs were produced in post-processing. Figure 18 shows the piezoshaker-side of the test setup, and Figure 19 shows the high-frequency accelerometer-side of the test setup.
Data was recorded three times, with the test setup disassembled between tests to gauge repeatability. Note that the transducer and sensor were tightened without the use of a torque wrench and that may improve repeatability. Next, the threads for mounting the accelerometer were clearance-drilled, the accelerometer was glued in place (super glue), and the test was repeated three times. Between tests, the accelerometer was removed, cleaned, and re-glued; however, the piezoshaker (with load-cell) was left in place to test just the repeatability of gluing the sensor to the surface.

**7.2 Results**

The key takeaway is that the glued-accelerometer FRFs compare very well with the threaded accelerometers, so using super-glue to mount the sensors and actuator will not appreciably color the results. Figure 20 shows the average of the three threaded FRFs (blue trace) plotted with the average of the three glued FRFs (red trace). This clearly shows that the two methods are quite similar, particularly below about 40 kHz. The maximum difference between the two methods is about 8 dB (which occurs at about 47 kHz); however, for the purpose of selecting frequency bands of maximum energy transmission, this difference is unimportant. Furthermore, it is shown below that this difference is less than that seen between repeated measurements of the threaded condition.
Figure 21 shows the three FRFs taken for the threaded conditions, and while repeatability is not perfect, it is acceptable for the purpose of selecting frequency bands where energy transmission is at a maximum. Note that the maximum difference in peaks is about 15 dB, which occurs at about 22 kHz. Figure 22 shows the FRFs taken for the glued conditions, and it can be seen that repeatability here is somewhat better than that for the threaded conditions. It was thought that this could be either due to not removing and reinstalling the shaker, or due to better consistency with mounting the accelerometer with glue, or some combination of both. Three cases of mounting and dismounting the shaker to the gearbox and comparing the resulting FRFs are shown for two accelerometer mounting locations in Figure 23 and Figure 24 (the accelerometer mounting locations are shown in Figure 11 and Figure 12). As can be seen, two of the cases match quite well, while the third shows differences in amplitude by as much as 15 dB. The reason for this difference is unclear; however, for the purpose of selecting frequency bands that readily transmit bearing energy, it appears that nominal care in mounting the piezoshaker and accelerometers produces repeatable results. Note that the poor low-frequency
response is due to the low force output with reaction-mass excitation at low frequency (discussed above).

7.3 Cable Noise and Linearity
As the measured acceleration could be quite low for large gearboxes and certain frequencies, noise was always a concern. Since PCB makes both solid-core and stranded-core coaxial cables, these were both tested to determine if one provided consistently cleaner measurements. Generally, the solid-core cables are sold for charge-based (no internal IEPE amplifier) sensors which can be very low signal.
intensity levels. Additionally, the system was tested for linearity by driving at two distinct amplitudes and comparing the FRFs.

7.3.1. Test
The IGB was set up with the lab-based excitation system, with the PCB high-frequency accelerometer attached to a mounting pad which was super-glued to the case. The test setup is shown in Figure 25.

![Figure 25: IGB Test Setup](image)

FRFs were made between exciter and high-frequency accelerometer, and the coherence between 5 kHz and 50 kHz was used as the metric.

The gearbox was laid on foam on a table in the lab to provide some level of isolation. The piezo actuators used were the 5x2x2 mm chips from Physick Instrument; they were separated from each
other, the gearbox, and the reaction-mass with microscope-slide covers. The reaction mass for this test was the head of a 3/8" bolt. The E1445A card was used as the source, and this drove the ISI-SYS piezo amplifier. A 0 – 50 kHz chirp was driven with a chirp time of 5 seconds. The ‘high’ level chirp was commanded with a 1V signal with a 2V offset. The ‘low’ level chirp was set at 324 mV, which was the lowest level permitted by the soft front panel. The measured force of the ‘high’ level was about 10 dB higher than that measured by the ‘low’ level. The force and response spectra for both ‘high’ and ‘low’ excitation levels are shown in Figure 26 and Figure 27.

![Figure 26: Force Spectrum](image1)

![Figure 27: Response Spectrum](image2)

The actuator was installed in the location shown in Figure 25, next to the Chadwick-Helmuth accelerometer. The CH accel was powered with a 9V battery, and a custom cable was made to facilitate this. The Dytran accel was grounded to the gearbox (as this is a 3-pin connector, it is possible to float this ground). The Dytran accel was also used with a custom cable.

Both Dytran and PCB accels received IEPE power from the IEPE breakout boxes (this was enabled in DAC Express). Sample rate was 196,608 Sa/s (maximum available from VXI E1433A card). Capture time was set for 30 seconds, which would give 6 full 5-second chirps; however, the large amount of data streamed to disk created a FIFO overflow error (even with the maximum possible block
size of 16,384), and the recorded data was typically about 27 seconds. It was later discovered that displaying the large blocks of data while recording was the reason for the overflow; not displaying the time data while recording permitted logging complete sixty-second data sets of six channels.

7.3.2. Results
Not surprisingly, the difference between the cables was negligible. To evaluate the quality of the measurement, the FRF (accel/piezo) coherence was integrated from 5 kHz to 50 kHz. This was normalized such that a value of 100 means perfect coherence throughout that frequency range and a value of 0 means no coherence in that range. Figure 28 shows the FRFs and coherence between the 7 tests. The difference between the cables is negligible, as shown by the normalized coherence values (which range from 73.5 – 73.9). These minor differences do not follow any trend, and these differences (which only amount to a 0.5% spread) are likely due to experimental variability.

Figure 28 also shows that the FRFs are essentially identical between ‘low’ and ‘high’ force levels, which indicates that the system is linear (at least this gearbox in this particular test configuration). Fortunately, this indicates that our force measurement is linear between these two levels of excitation, further indicating that the force measurement made by the lab-made excitation system is reasonable.

One area of potential concern is the FRF between about 48 – 50 kHz. There is some variability in the amplitude in this region (as much as 15 dB), though all the coherence values in this region indicate a ‘good’ measurement (coherence is above 0.99 in this region for all measurements). Interestingly, the coherence dips to about 0.8 between about 21 – 25 kHz, though the FRF amplitudes remain consistent across the tests.

Another thing to note is the low-frequency cutoff. For the ‘high’ force cases, this is about 4 kHz, whereas for the ‘low’ force cases, this is at about 6 kHz.
As piezoelectric shakers were purchased from two different manufacturers (Piezomechanic and ISI-SYS), these were tested to determine if similar FRFs could be made with each.

### 7.4 Piezoshaker Comparison Test Setup

For the comparison test, a steel block was used for mounting the piezoshakers, load cell (PCB 201B02) and the high-frequency accelerometer (PCB 352A60). Mounting pads were super glued to the block, one on each side, with the accelerometer threaded into one of the mounting pads and the piezoshaker attached to the mounting pad on the other side with the PCB load cell. The Piezomechanik shaker/load cell combination was screwed in directly to the mounting pad, with a washer placed between the shaker and load cell to increase contact between the two. The ISI-SYS shaker had to be
attached using a second mounting pad super-glued to the first mounting pad, as it was designed to use custom pads. For both cases, a 5 second 0 – 5 Volt, 0 – 50 kHz chirp was commanded to the exciter, and 60 seconds of time data was recorded. FRFs were produced in post-processing. Figure 29 shows the Piezo Mechanik setup, and Figure 30 shows the ISI-SYS setup.

![Figure 29: Piezoshaker Comparison Test Setup, Piezomechanik Shaker](image1)

![Figure 30: Piezoshaker Comparison Test Setup, ISI-SYS Shaker](image2)

Data was recorded twice using NI LabVIEW SignalExpress software and the NI PXI-4462 data-acquisition card. The accelerometer was kept in the same location for the shaker comparison tests.

7.4.1. Results
The FRFs from the two shakers are quite similar. Figure 31 shows the Piezomechanik FRF (blue trace) plotted with the ISI-SYS FRF (red trace). This shows that the two shakers are quite similar below 35 kHz. The two FRFs differ slightly about 35 kHz, though this may be attributable to the differences in mounting pads. This result is encouraging, as it further indicates that these high-frequency measurements are meaningful and repeatable.
Figure 31: Comparison Between FRFs Made with Piezomechanik and ISI-SYS Shakers

7.5 Gluing Mounting Pads Directly to Epoxy Coating

For efficient test setup, it is more convenient to glue the mounting pads directly to the gearbox epoxy coating, rather than carefully remove the coating first. To test whether this affects the measured FRF results, a mounting pad was glued to the side of the gearbox, FRFs were recorded, the location was carefully marked, the pad and epoxy were removed, the pad was re-glued to the gearbox, and the test was repeated. The accelerometer mounting location is shown in Figure 32, with a close-up shown in Figure 33. The piezoshaker mounting location is shown in Figure 34, and the accelerometer mounting location with epoxy removed is shown in Figure 35.
7.5.1. Results
The FRFs between these two test conditions are practically identical – we can confidently glue the mounting pads directly to the epoxy coating so long as it is sound. These FRFs are shown in Figure 36, with the blue trace showing the FRF with the pad glued directly to the coating and the red trace showing the FRF with the epoxy scraped off and the pad glued directly to the gearbox casting. Though it has not been tested, gluing the pad to a coating which is crazed or flaking is expected to affect the results.
7.6 Splicing Impact FRFs with Shaker FRFs for 0 – 50 kHz FRFs

Because reaction-mass piezoshakers are not good for getting energy into a system at low frequency (in this case below about 3 kHz), a complete 0 – 50 kHz FRF will require another excitation system for the low-frequency region. Impact testing only produces good FRFs up to about 6 kHz, so this provides a 3 kHz overlap region between the two methods to evaluate the quality and comparability of the two measurements. These spliced FRFs are shown below in Figure 37, Figure 38, Figure 39, and Figure 40, and it can be seen that these match very well. The FRF from the modal hammer is shown by the red trace, and the FRF from the piezoshaker is shown by the blue trace. This also lends further confidence that these measurements are repeatable and meaningful.
7.7 Comparison of Surface FRFs to True Transfer Path FRFs

To determine whether surface-measured FRFs would be similar enough to the “true transfer path,” an Apache IGB was selected for testing. To make a true transfer path measurement, it is important to know the bearing load zones, because this provides a likely area where the bearing will begin to spall. Certainly, several helicopter bearings have no fixed load zone, such as swashplate bearings, where the load direction is dependent on the flight direction as well as wind; however, many bearings do, and to best estimate where the actual transfer path would be, it was important to have good estimates about these loading directions. This information was provided by AMRDEC for each helicopter under consideration. The bearing exciters were placed such that excitation would be in the load zone; also, when making surface FRFs, the shaker was placed as near the location and direction of the load zone as possible.

7.7.1. Reaction-Mass Bearing Exciter and Gearbox Instrumentation

A duplex ball bearing in the Apache IGB was selected for conversion to a bearing excitation device to demonstrate similarity between true defect transfer paths and close-proximity surface-measured FRFs. The duplex ball bearing was disassembled, and one rolling element was removed. Part of the cage was notched to provide clearance, as shown in Figure 41. The surface of the outer race was ground slightly to permit the piezo element to sit flat on the surface and the outer race was notched slightly to provide clearance for the wires (Figure 42).
5x5x2 mm piezos (PhysikInstrumente PL-055) were wired to shielded coaxial cable and glued to the bearing outer race, with a tungsten-carbide sphere (ground flat on the bottom) for the reaction mass. Tungsten-carbide was chosen, as it is roughly twice as dense as steel, and that permitted a 2.0 gram ¼” diameter sphere which fit neatly inside the bearing. Figure 43 shows an as-purchased piezo on the right, and an assembled piezo/cable unit on the left. Figure 44 shows the assembled reaction-mass exciter glued to the outer race.
Next, the cage was placed into the outer race with one of the split inner-race pieces with some epoxy to prevent the bearing from accidentally turning and damaging the exciter (Figure 45). The other inner-race piece was epoxied in place (Figure 46), and the instrumented bearing was installed in the gearbox.

A notch was ground into the bearing housing to provide clearance for the wires (Figure 47) – this notch was not directly under the bearing load zone so as not to interfere with energy transfer from reaction-mass exciter and the remainder of the gearbox. Additional wire clearance was necessary, so a shallow notch was also ground into the outer race of the upper duplex ball bearing outer race, as shown in Figure 48. Again, this notch is not in the bearing’s load zone to minimize effect on the transfer path.
The instrumented lower duplex bearing was then pressed into the housing (Figure 49), and then the upper bearing was pressed in place over the lower bearing (Figure 50).

The bearing housing was then reassembled into the gearbox. Instead of creating an internal bearing exciter for the upper duplex ball bearing, a reaction-mass exciter was made and placed externally on the bearing housing in the load zone. Because of the small ledge on the housing, 3x3x2 mm piezos were used (PhysikInstrumente PL-033). The bearing housing is not especially thick at this point, and the difference in transfer path is expected to be minimal. The smaller piezos are more difficult to handle, but they provide sufficient force and measurement sensitivity, as shown later. Figure 51 shows the exciter, and Figure 53 gives a view of its location.
As the roller bearing does not have its own inner race (the gear shaft directly contacts the rollers), it was not possible to create a bearing exciter for it. As with the upper duplex bearing, the reaction-mass exciter was placed on the casting outside the bearing, as shown in Figure 52. As there were no space limitations here, a machine nut was used as the reaction mass. The cables were routed out of the gearbox through the oil sight plug.

The gearbox was instrumented with six high-frequency accelerometers (PCB 352A60), and the two HUMS accelerometers (Dytran 3062A1 and Chadwick-Helmuth 4177B) complete the response instrumentation for this test. Figure 53 shows the locations of the Chadwick-Helmuth accelerometer, the external exciter, the upper duplex bearing exciter, and the locations of high-frequency accelerometers numbers 1, 2, and 5. Note that locations 1, 2, and 5 were chosen because they are closest to the load zones for the output bearings, and locations 3, 4, and 6 were chosen because they are closest to the load zones for the input bearings (the load zone locations and directions were provided by the Army).
Figure 53: Accelerometer and External Exciter Locations

Figure 54: Accelerometer Locations
7.8 Test Setup for Pre-loading Gearbox

To verify that the measured FRFs on a parked helicopter will not differ significantly from those on a running craft with hundreds of foot-lbs of torque transmitted through the gearboxes, a fixture was made to permit locking one side of the gearbox and applying a measured torque to the other side. Figure 55 shows a side view of this fixture. The input side of the gearbox is locked with an aluminum plate which is bolted to the input flange. As shown in Figure 56, applying torque to the gearbox is accomplished by tightening the tensioning nuts which pull on the aluminum plate bolted to the output flange. The two DC load cells equally spaced from the centerline guarantee that a true torque is applied to the shaft by ensuring the force measurements are the same.

7.9 Testing Excitation, Acquisition, and Processing

The commanded excitation for these tests was a 0 – 5 Volt, 5-second chirp from 0 – 50 kHz. Time-data was acquired with a VXI system with a high-frequency E1433a card at 196,608 samples per second. 50 seconds of data was captured for each test, triggered such that there were 10 complete chirps per data set. The FRFs were post-processed using a 983,040 block size (5 * sample rate) so that each block is one complete chirp, and no window was used, since leakage would not be an issue with each completely observed block. The FRFs therefore have 10 averages, and the coherence shows that these measurements are generally quite good through the frequency range. Exceptions are low frequency
(below about 3 kHz for the commercial exciter system, and below 5 – 8 kHz for the miniature exciters), since reaction-mass exciters are not generally good at getting low-frequency energy into a system. Also, the coherence is generally not great above about 40 – 45 kHz for the HUMS accels, as this is above their resonance and their response is very low (and dominated by noise) in this range. Additionally, the coherence is not good at the deep anti-resonances, as the response amplitude at these frequencies is also very low.

Also, no impact testing was done for these tests. For one thing, it isn’t possible to do impact testing for the internal bearing exciter locations; the only place would have been at the external exciter location. Another reason was that these tests were not conducted to provide complete 0 – 50 kHz FRFs but to determine if external gearbox testing would provide similar information to internal bearing “true transfer path” FRFs. Finally, it appears that the FRF amplitude at low frequency could be consistently lower than that at higher frequencies, so it ultimately may not be important to have this information for selecting amplification regions of the transfer path.

7.10 True Transfer Path vs. Surface Transfer Path Results

The primary goal of these tests was to determine if making FRFs from the surface of the gearbox near the bearing load zone to the response measurement locations would compare well to FRFs made from the actual bearing load zone to the same response locations. In short, the answer is that 5 of the 6 cases tested here (2 HUMS accelerometers and 3 bearing exciters) match fairly well between surface and “true transfer path” measurements. As can be seen from the following plots, the “internal” and “external” FRFs are generally similar enough to merit testing gearboxes on-aircraft.

7.10.1. Roller-Bearing Transfer Path

Roller-bearing exciter FRFs with the external-exciters FRFs are shown in Figure 57 through Figure 66. Note that Figure 57 shows the internal/external FRFs for the HUMS Chadwick-Helmuth accelerometer, and Figure 59 shows the internal/external FRFs for a high-frequency accelerometer.
installed at the same location. Likewise, Figure 58 shows the FRFs for the HUMS Dytran accelerometer, and Figure 60 shows the FRFs for a high-frequency accelerometer in the same location. Figure 61 through Figure 66 show the FRFs for the 6 high-frequency locations.

First, it is obvious that the HUMS accelerometers do not have a flat response up to 50 kHz; both FRFs drop substantially above 35 – 40 kHz when compared to FRFs made with high-frequency accelerometers. Also note that the Dytran FRF has a large dip at about 17 kHz. It is very encouraging to note that the internal and external FRFs for these locations do not differ substantially, whether made with the HUMS accelerometers or with high-frequency accelerometers. The high-frequency accelerometer locations 1, 2, and 5, however, differ by as much as 20 dB at some frequencies. Recall that these locations correspond to those for the load zones for the bearings under test (output side). The significance of this is yet unknown.

For the HUMS accelerometers installed in this gearbox, it would appear that a frequency band of about 11 – 17 kHz would be appropriate for the output roller bearing for the Chadwick-Helmuth accelerometer, whether selecting bands from the Chadwick FRF or the high-frequency accelerometer FRF. In selecting bands for the Dytran accelerometer, one might choose 32 – 35 kHz; however, when looking at the high-frequency accelerometer FRF, 27 – 32 kHz might look like a better region.
Figure 57: Internal/External FRF Comparison, Roller Bearing, Output Side, HUMS Accelerometer (Chadwick Helmuth)

Figure 58: Internal/External FRF Comparison, Roller Bearing, Output Side, HUMS Accelerometer (Dytran)

Figure 59: Internal/External FRF Comparison, Roller Bearing, Output Side, High-Frequency Accel HUMS Chadwick Location

Figure 60: Internal/External FRF Comparison, Roller Bearing, Output Side, High-Frequency Accel at HUMS Dytran Location

Figure 61: Internal/External FRF Comparison, Roller Bearing, Output Side, High-Frequency Accelerometer, Location #1

Figure 62: Internal/External FRF Comparison, Roller Bearing, Output Side, High-Frequency Accelerometer, Location #2
7.10.2. Lower Duplex Ball-Bearing Transfer Path

For this bearing, the transfer path differs substantially between the internal and external FRFs for the Chadwick HUMS accelerometer (Figure 67), as well as the high-frequency accelerometer in the same location (Figure 69). For the internal case, a band of about 20 – 25 kHz looks ideal; however, when measured externally with the HUMS accelerometer, one might choose about 10 – 15 kHz instead. Note that with the high-frequency accel at this location, 20 – 25 kHz would probably still be considered the best frequency band.

The Dytran HUMS accel differs less for this bearing between inside/outside FRFs (Figure 68), and a band of 32 – 37 kHz might be considered good for this bearing as well (this is likely the accelerometer’s
resonance). However, when considering the high-frequency accel response at this location (Figure 70), one might choose substantially different bands: 40 – 45 kHz from the internal measurement and 20 – 25 kHz for the external measurement.

Figure 67: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, HUMS Accel (Chadwick Helmuth)

Figure 68: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, HUMS Accel (Dytran)

Figure 69: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, HF Accel at HUMS CH Location

Figure 70: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, HF Accel at HUMS Dytran Location
Figure 71: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #1

Figure 72: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #2

Figure 73: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #3

Figure 74: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #4

Figure 75: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #5

Figure 76: Internal/External FRF Comparison, Lower Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #6
7.10.3. Upper Duplex Ball-Bearing Transfer Path

These FRFs do not differ as much between internal and external excitation. Note that the internal excitation is not truly inside the gearbox for this bearing, but instead it is on the outside of the bearing housing directly at the load zone; see Figure 53 to view the excitation locations. Interestingly, the best frequency band for the Chadwick accel might be the range where the two measurements differ the most – 7 – 10 kHz (Figure 77). For the Dytran accel (Figure 78), the accel’s resonance of 32 – 35 kHz might be the best region (probably their mounted resonances).
7.10.4. HUMS Accelerometer Dynamics

Figure 81: Internal/External FRF Comparison, Upper Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #1

Figure 82: Internal/External FRF Comparison, Upper Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #2

Figure 83: Internal/External FRF Comparison, Upper Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #3

Figure 84: Internal/External FRF Comparison, Upper Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #4

Figure 85: Internal/External FRF Comparison, Upper Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #5

Figure 86: Internal/External FRF Comparison, Upper Duplex Ball Bearing, Output Side, High-Frequency Accel, Location #6
For a clear demonstration of the way the HUMS accelerometers color the bearing response, the FRFs with the HUMS accels are plotted with the FRFS with the high-frequency accels in the HUMS locations. These are shown below in Figure 87 through Figure 94. From these plots, it is easy to see that both accelerometers’ response drops off substantially past about 35 kHz. The Dytran accel’s resonance at about 33 kHz is also clearly visible.
7.10.5. Pre-loaded Gearbox Tests

These tests were made to determine if pre-loading the gearbox significantly changes the FRFs.

This is because all of the tests will be made with the rotor-craft sitting idle (not running) with no gear preload, and it is best to be sure the FRFs measured in this manner will not differ significantly from those on a running craft with gearboxes transmitting hundreds of ft-lbs of torque. Fortunately, the FRFs do not differ significantly between zero, 100, or 165 ft-lbs of torque. The maximum torque for this gearbox is about 360 ft-lbs\(^1\); however, testing beyond 165 with this fixture would have required some

\(^1\) Max horsepower of 330 at 4815 RPM yields 360 foot-lbs torque
minor modifications (ran out of travel for the load cells), and since there was no difference at this level, this was not pursued further. Figure 95 through Figure 110 show these FRFs.
Figure 99: FRF Comparison With Preload, Lower Duplex Bearing Exciter, Chadwick Helmuth

Figure 100: FRF Comparison With Preload, Lower Duplex Bearing Exciter, Dytran

Figure 101: FRF Comparison With Preload, Lower Duplex Bearing Exciter, High-Frequency Accel Location #3

Figure 102: FRF Comparison With Preload, Lower Duplex Bearing Exciter, High-Frequency Accel Location #4

Figure 103: FRF Comparison With Preload, Upper Duplex Bearing Exciter, Chadwick Helmuth

Figure 104: FRF Comparison With Preload, Upper Duplex Bearing Exciter, Dytran
Figure 105: FRF Comparison With Preload, Upper Duplex Bearing Exciter, High-Frequency Accel, Location #5

Figure 106: FRF Comparison With Preload, Upper Duplex Bearing Exciter, High-Frequency Accel, Location #6

Figure 107: FRF Comparison With Preload, External Excitation, Chadwick Helmuth HUMS accelerometer

Figure 108: FRF Comparison With Preload, External Excitation, Dytran HUMS accelerometer

Figure 109: FRF Comparison With Preload, External Excitation, High-Frequency Accel, Location #1

Figure 110: FRF Comparison With Preload, External Excitation, High-Frequency Accel, Location #2

7.11 Conclusions
For the HUMS accelerometers for the three bearings tested, the “true transfer path” FRFs (i.e. from bearing to accel) are very similar to the easily-measured FRFs with the exciter mounted on the external surface of the gearbox near the load zone. This is in some part due to the nature of the accelerometer locations, and in some part due to the large influence of the accelerometer dynamics on the measurements. This is not to say that the FRFs are identical; they differ by more than 20 dB at many frequencies; however, the general form of the FRFs and therefore the locations of the peaks are quite similar, and therefore the external measurement would seem to be perfectly satisfactory for selecting frequency bands for any of the three bearings tested. As repeated before, the real proof of selecting an ideal frequency band will be use of the band with known faulted bearing data.
8 Repeatability – On Aircraft Testing

An important concern for this series of tests is the repeatability of these high-frequency measurements both on the same aircraft, and across multiple aircraft. As reported above, repeatability on an individual gearbox in the lab was quite good, through removing and remounting sensors, mounting pads, and the shaker. Some of this repeatability testing was also done on the Blackhawk. Validation across multiple serial numbers was another issue, and this was tested by taking measurements on the Blackhawk IGB on three different aircraft. Four excitation locations were chosen to be at or as close as possible to the load zones of each of the four bearings in the gearbox.

These tests were made on an MH-60M (tail No. 20007) at the 160th at Fort Campbell, KY during the week of March 7 – 11, and on two UH-60s (tail Nos. 20764 and 20145) at the Redstone Arsenal March 23 – 25, 2011. Tail No. 20007 was a full test (e.g. all gearboxes), 20764 was a partial test (i.e. main module, oil cooler, and IGB), and 20145 was only the IGB (tested for repeatability).

8.1 Repeated Test w/o Changes

Repeatability when leaving all sensors in place and recording multiple data sets was tested. This means that the shaker and load cell were screwed to the mounting pad at location #1 and left in place between tests, and the accelerometers were not disturbed between tests. The test setup is shown in Figure 111.
Ten sixty-second data sets were recorded, and the results for the two accelerometer locations are shown in Figure 112 and Figure 113. While it is clear that this level of repeatability is more than adequate for the purpose of frequency-band selection, it should be noted that the results are the most similar from test to test below about 22 kHz, and 20 Hz - 20 kHz is listed as the limit (flat response region) for these accelerometers, though it is not known if repeatability is usually questionable outside this range. Further evidence that the accelerometer's repeatability beyond its normal measurement range is suspect can be seen by examining the FRF input (load cell) and output (accelerometer) power spectra, and noting trends in repeatability. Figure 114 shows the accelerometer power spectrum for the FRF shown in Figure 112, and Figure 115 shows the force power spectrum. While the force spectrum shows some variability between about 25 and 35 kHz, this could be true input variation due to increasing shaker temperature or some other effect, and therefore true measured input (although this is unknown). Note that the ripple on the force spectrum is due to the fact that the commanded force
input is not a true, continuous swept sine, but instead a stepped sine (i.e. discrete frequencies incremented in time). The accelerometer spectrum varies across a much greater frequency region, indicating that the variability noted in the FRF is more likely due to the variation in the accelerometer’s response. Note that while the repeatability seen here is more than sufficient for the purpose of enveloping band selection, this effect should be kept in mind when using a sensor outside of its normal frequency range for monitoring purposes.

Figure 112: IGB Input Accelerometer Location FRFs, 10 tests Without Disturbing the Test Setup

Figure 113: IGB Output Accelerometer Location FRFs, 10 tests Without Disturbing the Test Setup

Figure 114: IGB Input Accelerometer Power Spectra, 10 tests Without Disturbing the Test Setup

Figure 115: IGB Load Cell Power Spectra, 10 tests Without Disturbing the Test Setup
8.2 Dismounting and Remounting Shaker

Repeatability was also tested by leaving mounting pads in place, but then removing and reinstalling the shaker between tests. This was done three times at each of the four locations on the Blackhawk IGB, and these are shown here for the Input accelerometer location. Figure 116, Figure 117, Figure 118, and Figure 119 show the results of these tests, and the repeatability is quite good, particularly below 20 kHz. The blue, red, and black traces show the results of each test.

8.3 Removing and Reattaching Mounting Pads

Small differences in excitation and response locations can dramatically change measured FRFs, particularly at high frequency, so the shaker mounting pads were removed and reinstalled between two

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tests at two different locations to examine repeatability. Figure 120, Figure 121, Figure 122, and Figure 123 show the results of this test. As can be seen, repeatability is generally very good below about 20 kHz, and above this region, though the amplitudes differ by as much as 20 dB in one case (Figure 121, around 36 kHz), the amplitude trends are generally preserved and are suitable for the purpose of energy-transmission frequency band selection.

8.4 Rotating the Blades or Drivetrain
As these FRFs were all measured on parked, static aircraft, there was some question as to whether the FRFs would change if the blades (and drivetrain) were rotated between tests. Figure 124, Figure
125, Figure 126, and Figure 127 show the FRFs taken with the blades in two different positions. It can be seen that the FRFs do not differ significantly.

8.5 Torque Applied to Drivetrain

Again, as these tests are being performed on parked aircraft, there was some question as to whether the FRFs would change with torque applied to the drivetrain. This loads the gears and in most cases applies some radial load to the bearings, potentially altering clearances and affecting transfer path dynamics. This effect was tested in the lab on the Apache IGB; however, to be thorough, it was also tested on a complete Blackhawk. To test this on the Blackhawk, the torque was applied to both the
main rotor and the tail rotor in opposite directions. This was done in both the normal direction of rotation and in the opposite direction. Figure 128 and Figure 129 show how this was done. Figure 130 and Figure 131 show the results for this test – it can be seen that there is little difference between these FRFs.

8.6 Repeatability Between Standard HUMS and High-Frequency Accelerometers

During the full test on the MH-60M, the HUMS accelerometers were removed and replaced with high-frequency accelerometers (PCB 352A60), and these results can be compared. Note that these comparisons must be made carefully, as none of the HUMS accelerometers has a flat calibration above 20 kHz and many do not above 5 kHz. For example, the 3062A1 Dytran (called ‘sparkplug accels’) have a
resonance typically around 35 kHz, and the response of these accelerometers falls off rapidly above the resonance. These tend to generally match the high-frequency accelerometers up to 20 kHz or so, respond with larger amplitude up to slightly past their resonance, and then respond with lower amplitude beyond their resonance. Also, there is the issue of mass-loading the structure, which means that the mass of the accelerometer affects the response of the system, and this is particularly more of an issue at higher frequency. Effectively, the high-frequency accelerometer measurements help to demonstrate the effects the HUMS accelerometers have on the measurements. Figure 132 shows the difference when comparing the high-frequency accelerometer to the Dytran ‘sparkplug’ accelerometer – notice the 10 dB higher response at 30 kHz and the nearly 30 dB lower response at 45 kHz. Figure 133 shows the comparison for an Endevco 6259. The Endevco is has a claimed resonance at about 40 kHz, and this matches the more than 10 dB difference seen at that frequency. Note that the recommended frequency bands will be discussed later.

8.7 Repeatability Across Serial Numbers

A critical assumption for successful CBM on a fleet of aircraft is that the same enveloping frequencies can be used for a given bearing across the fleet. For example, to assert that a 5 – 10 kHz frequency range is ideal for monitoring the Goodrich-equipped Blackhawk IGB assumes that this
frequency band will readily transmit bearing-defect energy to the HUMS accelerometers for any Blackhawk IGB, whether it was manufactured in 1979 or 2009 and no matter how many times it has been rebuilt. One feature which will lend itself to the consistency in these transfer paths is the fact that these gearboxes are built to very close tolerances – meaning that, for the most part, they should be quite similar from S/N to S/N. There is, however, the fact that the manufacturers may have seen fit on occasion during 30 years of production to add or subtract ribs, filets, housing thicknesses, etc. as trouble areas arise or unnecessary material is removed. Depending on the magnitude of the design change, this could slightly or dramatically affect the transfer path of interest. Additionally, despite the fact that these gearboxes are made to close tolerances, the housings are almost certainly green-sand cast, and this is not an exact process. Due to slight differences in patterns and the inexact process of cope and drag\(^2\) alignment, the housings can differ from one another slightly. It is suspected that transfer-path differences due to the casting process would be most noticeable at high frequency, though this remains to be tested. Ultimately, for any CBM method to work on a fleet, defect-frequency transfer paths must be ‘consistent enough.’

**8.7.1. IGB**

To investigate consistency across S/Ns, FRFs were measured on a total of three Blackhawk IGBs, and these are compared below. The good news is that, while there is at least 10 years difference in age of the three gearboxes (tail number 20764’s IGB was wearing an overhaul tag which was hard to read, but it looked like the year 2000, and tail number 20007 was practically brand new), there were no obvious external physical differences. One complicating matter in comparing these FRFs is the fact that the load cell used to make the measurements on each gearbox was damaged somehow after testing the first. The load cell’s sensitivity dropped by approximately a factor of 10. Unfortunately, this means that the FRFs measured on two of the gearboxes are suspect; however, they are consistent between each

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\(^2\) The cope and drag are the top and bottom halves of a green-sand mold, which are separated to remove the pattern, and then reassembled to pour the casting.
other, and this is good news for finding repeatability across serial numbers. Figure 134 through Figure 141 show these FRFs (note that the frequency band recommendations are discussed later). The blue trace is the FRF measured on tail number 20764, the red trace is the FRF for tail number 20145, and the black trace is for tail number 20007 (which was multiplied by 10 to facilitate comparison between the three FRFs). As can be seen in the figures, 20764 and 20145 match fairly well, particularly below around 20 kHz. The FRFs from 20007 do show some marked similarity; however, these differ more appreciably. For example, in Figure 134 and Figure 135, the general character of the FRFs match reasonably well, except between 10 – 15 kHz and 30 – 35 kHz. Some laboratory calibration measurements of the damaged load cell were made, yet applying these to the recorded data did not significantly improve the comparison with the 20007 data. It is quite possible that the calibration of the damaged sensor is changing over time. Despite the differences seen in these FRFs, it would appear that if one were to choose 7 – 12 kHz as an ideal region, this would be a good band of energy transmission for all S/Ns and test locations, avoiding some of the unpredictable differences seen at higher frequency. Only one aircraft (tail No. 92-26415) shows a condition indicator for the IGB on the IAC IMDS website, and it is a ‘yellow’ indicator. The spectrum of this indicator shows content at approximately 1900 Hz and 5100 Hz (Figure 142); however, these bands seem to show up in most ‘green’ indicators as well at lower amplitude (Figure 143), so it may be due to another component and not indicative of bearing condition.
8.7.2. Main Module

In an effort to determine if adding a rotor brake to a Blackhawk main module alters the ideal frequency bands for enveloping/demodulation, the main module was tested on one with (tail number 20007) and one without (tail number 20764) a rotor brake. This permits another opportunity to compare FRFs across S/Ns. Here is one example of a minor external difference between gearboxes: Figure 144 shows the location #3 (near the top, front of the gearbox) on tail number 20764 and Figure 145 shows the same location for 20007. One can see that the integrated boss on 20007 is not present on 20764. While these particular differences are not likely to significantly affect the vibration transfer path for the present HUMS accelerometer locations, it significantly affects the input force – while the force direction for 20007 is horizontal, the input force direction for 20764 is close to a $45^\circ$ angle,
inputting force in both the horizontal and vertical directions. The differences between the FRFs can be easily seen in Figure 146, Figure 147, and Figure 148 (note that the frequency band recommendations are discussed later).

Figure 144: Tail No. 20764, Location #3 on Main Module, no Built-in Boss

Figure 145: Tail No. 20007, Location #3 on Main Module, with Built-in Boss

Figure 146: Main Module, Location #3, Right Input Accelerometer, Goodrich System

Figure 147: Main Module, Location #3, Left Input Accelerometer, Goodrich System

Figure 148: Main Module, Location #3, Recommended Frequency Band
Figure 148: Main Module, Location #3, Tail Takeoff Accelerometer, Goodrich System

Location #5 was apparently identical between the two gearboxes, and the FRFs are shown below in Figure 149, Figure 150, and Figure 151. The FRFs match to a reasonable extent below approximately 20 kHz and differ rather significantly thereafter at the Right Input location. The Left Input location and the TTO are at least similar above 20 kHz.

Figure 149: Main Module, Location #5, Left Input Accelerometer, Goodrich System

Figure 150: Main Module, Location #5, Right Input Accelerometer, Goodrich System
Plotted in Figure 152, Figure 153, and Figure 154 are the FRFs taken closest to the TTO and the rotor brake. As this part of the transmission is definitely different between the two tail numbers, one would expect the FRFs to be different as well, and they are, particularly at the TTO accelerometer location.
As the issue of S/N repeatability is of prime importance for CMB monitoring in a fleet, it would be advisable to further test this, perhaps by selecting one gearbox (such as the Blackhawk IGB) and testing 5 – 10 different examples. Also, if a gearbox design change (e.g. ribs, bosses, thickness, etc.) can be identified, then testing and comparison of pre/post design-change gearboxes would be very helpful for examining consistency across the fleet.

Figure 154: Main Module, Location #6, Tail Takeoff Accelerometer, Goodrich System
9 Complete Test – Blackhawk

As stated above, four complete and two partial helicopter tests were made, on an MH-60M (complete), a UH-60L (partial), an MH-47G (complete), a CH-47D (partial), an OH-58D (complete), and an AH-64 (complete). However, in the interest in space, only the complete results for the Blackhawk (MH-60M and UH-60L) will be shared here. Full reports for the other helicopters and all the data have been made publicly available by the Army.

9.1 Goodrich System

Tail number 20007 was delivered to the 160th with a Goodrich HUMS system; this was replaced with an IAC system, but most of the mounting pads for the Goodrich system were left in place. This permitted effectively testing two systems simultaneously. The results for the Goodrich system are discussed here. Note that in some cases, the Goodrich pads were removed and these locations could not be tested. Also note that a few gearboxes (e.g. accessory modules) use the same location between the Goodrich and IAC systems for the accelerometers, and only the IAC accelerometers were used for the test. Ideally, the FRFs recorded would be identical; however, as has been shown with the high-frequency accelerometers, this is not necessarily a good assumption (particularly at high frequency), since the accelerometer dynamics and inertial properties strongly affect the measurements.

9.1.1. Input Modules

The input modules on the Goodrich system are monitored by an Endevco 6259 accelerometer at both the input and output locations. The right (#2) input module on tail no. 20007 was missing the mount for the input accelerometer, as clearance was needed for a wire loom. Figure 155 through Figure 159 show the input location FRFs for the left input module, and Figure 160 through Figure 169 show the output location FRFs for both left and right modules. These are shown so that identical locations left to right can be easily compared. The frequency band selected for these modules was 5 – 12 kHz; looking
through the figures will illustrate that this is the best compromise for maximum energy transfer at these locations.

Figure 155: Left Input Module, Location #1, Input Accel

Figure 156: Left Input Module, Location #2, Input Accel

Figure 157: Left Input Module, Location #3, Input Accel

Figure 158: Left Input Module, Location #4, Input Accel

Figure 159: Left Input Module, Location #5, Input Accel
9.1.2. IGB
This transmission was discussed in detail in the Repeatability section, and the results can be found there.

9.1.3. Main Module
As mentioned above, the two main modules tested differed somewhat in the fact that one had a rotor brake and the other did not. Locations 3, 5, and 6 are shown above in Repeatability, Main Module (Figure 146 through Figure 154), so they will not be repeated here. As this gearbox is very large, and the transfer paths are generally very long, the selection of frequency bands was much more of a compromise. The 5 – 10 kHz band was chosen to provide adequate energy transmission for the largest number of bearings. The FRFs are shown in Figure 170 through Figure 193.
Figure 170: Main Module, Goodrich System, Left Input Accelerometer, Location #1

Figure 171: Main Module, Goodrich System, Right Input Accelerometer, Location #1

Figure 172: Main Module, Goodrich System, Tail Takeoff Accelerometer, Location #1

Figure 173: Main Module, Goodrich System, Left Input Accelerometer, Location #2

Figure 174: Main Module, Goodrich System, Right Input Accelerometer, Location #2

Figure 175: Main Module, Goodrich System, Tail Takeoff Accelerometer, Location #2
Figure 176: Main Module, Goodrich System, Left Input Accelerometer, Location #4

Figure 177: Main Module, Goodrich System, Right Input Accelerometer, Location #4

Figure 178: Main Module, Goodrich System, Tail Takeoff Accelerometer, Location #4

Figure 179: Main Module, Goodrich System, Left Input Accelerometer, Location #7

Figure 180: Main Module, Goodrich System, Right Input Accelerometer, Location #7

Figure 181: Main Module, Goodrich System, Tail Takeoff Accelerometer, Location #7
Figure 182: Main Module, Goodrich System, Left Input Accelerometer, Location #8

Figure 183: Main Module, Goodrich System, Right Input Accelerometer, Location #8

Figure 184: Main Module, Goodrich System, Tail Takeoff Accelerometer, Location #8

Figure 185: Main Module, Goodrich System, Left Input Accelerometer, Location #9

Figure 186: Main Module, Goodrich System, Right Input Accelerometer, Location #9

Figure 187: Main Module, Goodrich System, Tail Takeoff Accelerometer, Location #9
Figure 188: Main Module, Goodrich System, Left Input Accelerometer, Location #10

Figure 189: Main Module, Goodrich System, Right Input Accelerometer, Location #10

Figure 190: Main Module, Goodrich System, Tail Takeoff Accelerometer, Location #10

Figure 191: Main Module, Goodrich System, Left Input Accelerometer, Location #11

Figure 192: Main Module, Goodrich System, Right Input Accelerometer, Location #11

Figure 193: Main Module, Goodrich System, Tail Takeoff Accelerometer, Location #11
9.1.4. Oil Cooler

The oil cooler bearing was tested at two excitation locations – next to the sensor (pseudo-driving point), and mounted on the driveshaft just forward of the fan. These are noisy measurements, and this should be considered along with the following recommendations. Figure 194 through Figure 197 show the FRFs along with the recommended enveloping band of 18 – 25 kHz – this would appear to be the mounted sensor’s resonance.
9.2 IAC System

The IAC system was thoroughly tested on tail number 20007, and the results are sorted by gearbox below.

9.2.1. Accessory Gearboxes

Both left and right accessory gearboxes were tested at four primary bearing locations, and as these gearboxes are fairly small, the measurements are very clean. Figure 198 through Figure 205 show both left and right gearbox FRFs and these are displayed so that identical locations can be compared between the left and right. For these gearboxes, 15 – 20 kHz might have worked quite well, were it not for location 4, where response drops off dramatically right at 15 kHz. Fortunately, 10 -15 kHz is still a band of good energy transmission for all locations, so this was the band selected for recommendation. Looking at the IAC VMEP website shows only one indicator for a UH-60, tail No. 83-23853, accessory gearbox #2. This is a yellow indicator, and the spectrum shows content at 2 – 3 kHz and 6 – 8 kHz and to a lesser degree at about 12 kHz (Figure 206). However, inspection of spectra of green indicators shows similar content in this area, so this may not be significant.

![Figure 198: Left Accessory Gearbox, Location #1](image1)

![Figure 199: Right Accessory Gearbox, Location #1](image2)
Figure 200: Left Accessory Gearbox, Location #2

Figure 201: Right Accessory Gearbox, Location #2

Figure 202: Left Accessory Gearbox, Location #3

Figure 203: Right Accessory Gearbox, Location #3

Figure 204: Left Accessory Gearbox, Location #4

Figure 205: Right Accessory Gearbox, Location #4
9.2.2. Hanger Bearings

The hanger bearings were the noisiest measurements made on the Blackhawk, next to the oil cooler bearings. As such, the recommendations made here must be treated as tentative. Figure 208, Figure 209, and Figure 210 show the forward, mid, and aft hanger bearing FRFs, respectively. These measurements were made with the shaker mounted to the driveshaft – this transfer path is through a viscous bearing, and this is inherently not a path which readily transmits vibration energy. The recommended band is 15 – 22 kHz.
9.2.3. Input Modules

Both left (#1) and right (#2) input modules were tested on tail number 20007. Five locations were selected to measure transfer paths which ideally mimic a bearing fault’s true transfer path. The FRFs are displayed in Figure 211 through Figure 220. These are displayed so that left and right locations can be easily compared. The 15 – 22 kHz band selected seems to look quite good for each of the five locations on both gearboxes.
9.2.4. IGB

The recommended frequency band for the IAC IGB is 14 – 20 kHz, and this is shown in Figure 221, Figure 222, Figure 223, and Figure 224. This band seems to maximize response amplitude for the four bearings being monitored.
9.2.5. Main Module

Again, the main module is quite large, and there is no obvious frequency-band which transmits energy best for all locations. However, as can be seen in Figure 225 through Figure 246, the chosen band of 5 – 10 kHz would appear to be the best compromise, as none of the eleven excitation-location FRFs on either the main accelerometer or the ring-gear accelerometer show this region as an area of poor energy transmission.
Figure 227: Main Module Accelerometer, Excitation Location #2

Figure 228: Main Module Ring Gear Accelerometer, Excitation Location #2

Figure 229: Main Module Accelerometer, Excitation Location #3

Figure 230: Main Module Ring Gear Accelerometer, Excitation Location #3

Figure 231: Main Module Accelerometer, Excitation Location #4

Figure 232: Main Module Ring Gear Accelerometer, Excitation Location #4
Figure 233: Main Module Accelerometer, Excitation Location #5

Figure 234: Main Module Ring Gear Accelerometer, Excitation Location #5

Figure 235: Main Module Accelerometer, Excitation Location #6

Figure 236: Main Module Ring Gear Accelerometer, Excitation Location #6

Figure 237: Main Module Accelerometer, Excitation Location #7

Figure 238: Main Module Ring Gear Accelerometer, Excitation Location #7
Figure 239: Main Module Accelerometer, Excitation Location #8

Figure 240: Main Module Ring Gear Accelerometer, Excitation Location #8

Figure 241: Main Module Accelerometer, Excitation Location #9

Figure 242: Main Module Ring Gear Accelerometer, Excitation Location #9

Figure 243: Main Module Accelerometer, Excitation Location #10

Figure 244: Main Module Ring Gear Accelerometer, Excitation Location #10
9.2.6. Oil Cooler

As mentioned elsewhere, the oil cooler measurements are very noisy, as these transfer paths do not easily transmit vibration energy. Figure 247 through Figure 252 show the measured FRFs. Excitation was performed on the driveshaft just forward of the oil cooler fan, as a pseudo-driving point measurement (i.e. the shaker was mounted next to the accelerometer on the shroud), and at the forward hanger bearing. For the shroud-mounted accelerometer, the band of 25 – 32 kHz (likely the sensor’s mounted resonance) is the best band of energy transmission. For oil-cooler monitoring with the forward hanger bearing accelerometer, 5 – 12 kHz was deemed the best frequency band.
9.2.7. Swashplate

As the swashplate bearing is not load-zone specific, six excitation locations were chosen, two on the fixed portion (#2 and #3, both near the sensor, #2 vertical and #3 horizontal), and 4 on the rotating portion (all 90° apart). As Figure 253 through Figure 258 show, these are very clean measurements, and the selection of the 32 – 37 kHz band was very easy.
Figure 253: Swashplate Test Location #1

Figure 254: Swashplate Test Location #2

Figure 255: Swashplate Test Location #3

Figure 256: Swashplate Test Location #4

Figure 257: Swashplate Test Location #5

Figure 258: Swashplate Test Location #6
9.2.8. TGB

The tail gearbox was tested for each of its four bearings, and the FRFs are shown below in Figure 259 through Figure 262. While these FRFs do not share a band quite as conveniently as the swashplate, the 10 – 15 kHz region appears to be a good compromise.
10 Conclusions and Recommendations

10.1 Conclusions
The work presented here demonstrates that meaningful, repeatable transfer path measurements can be made up to 50 kHz. A chirp is an excellent excitation source for providing maximum energy from the piezoelectric shaker amplifier at any given frequency, and provides a good compromise between measurement speed and quality. Electrical crosstalk is always a concern when making high-frequency measurements, but careful shielding can be used to mitigate this risk. Sensors can be carefully mounted on pads with super glue without significant degradation in measurement fidelity.

Measurements can be made on a gearbox surface close to the bearing load zone for an effective approximation of the true defect-location-to-monitoring-sensor FRF measurement. The transfer path measurements were not different between loaded and unloaded conditions. Because of this, it was deemed worthwhile to make these measurements on six rotorcraft for the US Army: two versions of the Blackhawk, two versions of the Chinook, the Kiowa, and the Apache. The data from this work is currently being used by the Army to adjust the frequency bands for the enveloping/demodulation techniques used by the installed HUMS boxes.

Knowledge of the transfer path dynamics is valuable for setting up condition indicators for machinery health monitoring, as it reveals spectral regions which do and do not readily transmit energy and therefore the information needed to diagnose machinery health. While testing multiple machines with seeded gear or bearing defects is the most definitive way to establish condition indicator levels, this technique is considerably faster and cheaper.

10.2 Recommendations
Fleet monitoring presents some challenges in regards to setting up condition indicators, as otherwise small differences between machines can, in some cases, drastically alter the transfer path dynamics. Major design changes in a machine will obviously affect the transfer path, but minor
differences between components due to normal manufacturing techniques may affect the transfer path as well. Further investigation into the transfer path repeatability across gearbox serial numbers is recommended.

Some evidence for sensor variability when used above its advertised “flat response” region has been presented here. That is, a given accelerometer produced somewhat inconsistent results when used above its recommended frequency range. Further investigation into this phenomenon and its implications for using sensors outside of their recommended range for monitoring is recommended.

While it was shown that the transfer path measurements do not change with preload, it would be valuable to compare measurements made on a running gearbox with those made on the same gearbox when static. A significant change in transfer path dynamics would be unfortunate, as making these measurements on running machinery would be significantly more challenging. Good measurements would be more difficult to make, as the operational vibration would generally dwarf the vibration provided by the piezoelectric shaker; stepped-sine excitation may be the best approach for acceptable measurement quality.

As stated elsewhere, this work identifies transfer path frequency regions which readily transmit or amplify the energy generated near the bearing load zone. However, the energy generated by a defective bearing may not produce much content in that region; therefore, it is preferable that frequency bands identified in this way not be used alone to set up condition indicators. Ideally, comparison to known faulted-component data would be used to select the indicators, but in the absence of data, conservatively-set condition indicators may help avoid a missed positive (though it will increase the likelihood of false alarms), and for all cases, storing all data for cases of known faulty components will provide valuable information for the refinement of the monitoring techniques in use.
11 References


12 Bibliography

D. D. Howieson, "A Practical Introduction to Condition Monitoring of Rolling Element Bearings Using Envelope Signal Processing (ESP)," Diagnostic Instruments, Ltd..


13 Appendix

13.1 Notice of Public Release

RDMR-TM-SD

15 June 2011

MEMORANDUM FOR PERSONNEL CONCERNED

SUBJECT: Request for Public Release Approval: “Bearing Resonance Test Results” FN5315

1. The Information Product “Bearing Resonance Test Results” was reviewed by AMRDEC personnel to ensure it met Technical, OPSEC, Military Critical Technologies List Determination, Foreign Disclosure, and Public Affairs requirements.

2. IAW AR 360-1 the Information Product was approved for Public Release.

3. The Information Product was assigned the Control Number: FN5315 and a copy of it and the review paperwork will be maintained in the Public Affairs office.

4. POC for this action is the undersigned.

[Signature]

MERVIN E. BROKKE
AMRDEC PAO
13.2 transferPathHole.m

% show defect force convolved with good/poor transmission path in FRF

Sa = 1000; % sample rate
dt = 1/Sa;
blkSz = 10*1000; % block size
time = 0:dt:blkSz*dt - dt; % time vector
df = 1/time(end);
F = 0:df:Sa/2; % frequency vector

freqs = [5 11 12 20 27 33 33 37 43]; % frequency, Hz
damp = [1 .8 .7 .5 1 1.1 .7 1.3 .4]; % damping, rad/sec

lam = -damp + j * 2 * pi * freqs;
A = .001*[7 3 4 .8 11 7 9 -2 4 3]; % Amplitude, inches

h = zeros(1,blkSz);
for idx = 1:length(freqs)
    temp = A(idx) * exp(lam(idx) * time);
    h = h + imag(temp);
end

H = fft(h); % system FRF

% generate impulsive force with harmonics and noise
force = zeros(1,10e3);
force(1:1000:end) = .1;
force(1:1000:end) = .1;
force = force + .01*(rand(size(force))-.5);
FORCE = fft(force);

% deconvolve the response from the transfer path and force
X = FORCE .* H;
x = ifft(X);

figure,subplot(211),plot(F,db(FORCE(1:length(F)))),xlim([0 50])
ylabel('dB Amplitude')
grid
title('Force Spectrum')
ylim([-40 10])

subplot(212),plot(F,db(H(1:length(F)))),xlim([0 50])
xlabel('Frequency (kHz)')
ylabel('dB Amplitude')
grid
title('Defect Transfer Path')
ylim([-40 30])

figure,plot(F,db(X(1:length(F)))),xlim([0 50])
hold on
yAxis = get(gca, 'YLim');
plot([18*[1,1], yAxis,'k--','linewidth',2)
plot(22*[1,1], yAxis,'k--','linewidth',2)
plot(28*[1,1], yAxis,'k--','linewidth',2)
plot(32*[1,1], yAxis,'k--','linewidth',2)
patch([18,18,22,22],[yAxis(1) yAxis(2) yAxis(2) yAxis(1)], rgb('green'), 'FaceAlpha', .4)
patch([28,28,32,32],[yAxis(1) yAxis(2) yAxis(2) yAxis(1)], rgb('red'), 'FaceAlpha', .4)
xlabel('Frequency (kHz)')
ylabel('dB Amplitude')
grid
title('Response Spectrum')

13.3 piezoModel.m

% compute the strain/voltage FRF for an unladen PZT cube

% Piezoelectric Constitutive Equation:
% http://www.efunda.com/materials/piezo/piezo_math/math_index.cfm

% constants are from PI Material Coefficients PIC255

rho = 7800; % density, kg/m^3
sE = 1e-12 *[15.9  -5.699  -7.376  0  0  0;
             -5.699  15.9  -7.376  0  0  0;
             -7.376  -7.376  20.97  0  0  0;
              0   0   0   44.92  0  0;
              0   0   0   0   44.92  0;
              0   0   0   0   0   43.19]; % compliance, m^2/N

% piezoelectric coupling, m/V, note that d33 is in the direction of the
% voltage and d31 is perpendicular
d = 1e-12 *[0  0  0  0  535 0;
            0  0  0  535  0  0;
           -174 -174  394  0  0  0]; % m/V

a = 5e-3; % side of block, m
h = 2e-3; % height of block, m

m = a^2 * h * rho; % mass of cube, kg
nu = .36; % Poisson's Ratio

freq = linspace(0,1e6,1e4); % frequency vector, Hz
w = 2 * pi * freq; % frequency vector, Rad/s

% mass factor - assume the stress due to inerita is due to 1/mf
mf = 2;

% displacement over voltage FRF
z_V_FRF = 1 ./ (-rho * a^2 * w.^2 / (mf * d(3,3) * 1/sE(3,3)) + 1/d(3,3));
x_V_FRF = 1 ./ (-rho * a * h * w.^2 / (mf * d(3,1) * 1/sE(1,1)) + 1/d(3,1));
figure, plot(freq/1e3, db(z_V_FRF))
hold on
plot(freq/1e3, db(x_V_FRF), 'r')
title('1D FRFs')
ylabel('dB Amplitude')
legend('Voltage Direction','Wire Direction')

length/strain factor
LSF = diag([1/h 1/a 1/a 1 1 1]);

area factor
AF = diag([1/a^2 1/(a*h) 1/(a*h) 1 1 1]);

loop through for the FRF
nestedWaitbar('New','Computing FRFs');
for idx = 1:length(w)
    stuff = LSF - sE * m / mf * w(idx)^2 * AF;
    FRF(:,:,idx) = a * stuff \ d. ;
end
nestedWaitbar('Close')

figure, loglog(freq, 1e6*abs(squeeze(FRF(1,3,:))), 'linewidth', 2)
hold on
semilogy(freq, 1e6*abs(squeeze(FRF(2,3,:))), 'r', 'linewidth', 2)
semilogy(freq, 1e6*abs(squeeze(FRF(3,3,:))), 'k', 'linewidth', 2)
grid
legend('X Direction','Y Direction','Z Direction','location','southwest')
xlabel('Frequency (Hz)')
ylabel('Amplitude (\mum/V)')