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GEARBOX VIBRATION DIAGNOSTICS TEST

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Abstract: The dynamic loads that appear in these conditions can be considerable, in comparison with the static forces, and their being token into consideration at the gearing planning is compulsory. The interior sources are represented by the deviations from the tooth-processing precision, especially the error of the measured step on the basis circle

Keywords: diagnostics, vibration, frequency

I VIBRATION PROPAGATION

For the purposes of condition monitoring, we will consider gearboxes (gears, shafts, bearings, and casings) to comprise a linear mechanical system and the gear motion errors to be the sources of vibration. If the gear motion errors are the input signals, then the gearbox can be modeled as a multiple input, single output system. The measured vibration signal on the gearbox casing can be represented by the following summation over N gears and M number of transmission paths for the k^{th} gear.

$$\tilde{s}(t) = \sum_{k=1}^{N} \sum_{n=1}^{M} h_{kn}(t) * s_{ek}(t) + w(t) \qquad (1)$$

Where $h_{kn}(t)$ is the impulse response of the k^{th} input signal via the n^{th} path, $s_{ek}(t)$ is the k^{th} gear motion error signal and w(t) is any external noise. Taking the Fourier transform of $\tilde{s}(t)$ results in the following summation over the same indices where convolution has been replaced by multiplication.

$$\tilde{S}(t) = \sum_{k=1}^{N} \sum_{n=1}^{M_k} H_{kn}(f) S_{ek}(f) + W(f) \qquad (2)$$

The transfer functions $H_{kn}(f)$ can be very complicated structural frequency response functions consisting of the gears, shafts, bearings and casing. Local resonances and time delays due to the propagation time will cause phase changes in the signal. Structural resonances will act as mechanical amplifiers to boost the vibration signal in certain frequency bands. After the vibration propagates through the structure, it is reasonable to expect phase changes, but the underlying frequency components will remain unchanged. It should come as no surprise that good accelerometer placement and mounting are important considerations in gearbox diagnostics. Also, it is important that any automated signal processing techniques used for diagnostics be wideband, since different gearboxes have drastically different structural resonances. Where one narrowband technique may work well centered on a specific frequency, it may fail to give good results on a different class of gearboxes.

II.TIME INVARIABLE PROPAGATION

When dealing with accelerometers mounted on relatively small, rigid gearboxes it is reasonable to assume the vibration propagation is time invariant. The wave speed is high enough and the propagation path short enough to say, to a very good approximation, that there is no time delay between different propagation paths. If neglect the time delay between the gear and the measurement transducer, then $h_{kn}(t) = c_{kn} * \delta(t)$. If then we apply an ideal comb filter spaced at multiples of the shaft rotation rate, equation 3 simplifies to the following simple expression for the vibration due to the k^{th} gear.

$$\tilde{s}_k(t) = \left(\sum_{n=1}^{M_k} c_{kn} \right) s_{ek}(t) = c_n s_{ek}(t) \qquad (3)$$

The frequency content of the measured signal is proportional to the gear motion error signal. This illustrates that the comb filtering technique is able to isolate the vibration from a single gear in the presence of non-harmonically related noise.

III. GEAR TOOTH FATIGUE FAILURE

The most common mode of failure for overloaded gearboxes is fatigue failure of the gear teeth due to excessive bending stresses. The overloading may be caused by loose bearings, misalignment, torsional vibration or operator error. Of these failures, it is most common for the gear to fracture before the pinion, due to the larger gear tooth spacing on the gear. A crack normally begins to propagate at the root of the tooth where the tensile stresses are greatest.

For diagnostic purposes, it is important to identify cracked gear teeth before they break. If we can identify a cracked gear tooth, it will give us the longest amount of lead time to schedule machinery down time and maintenance. The condition of the gear can quickly deteriorate once a gear tooth breaks free. Tooth debris can become caught in the mesh, causing further damage. The missing gear teeth carry no load, often causing adjacent teeth to fracture from excessive bending stresses. Figure 1 shows two broken teeth on the output gear for MDTB..

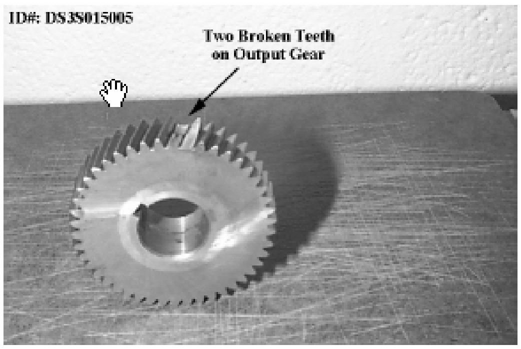


Figure 1. Two Adjacent Gear Teeth Fractures

IV.MECHANICAL DIAGNOSTICS TEST BED

The interdisciplinary field of engineering called machinery condition monitoring is charged with developing new technologies to diagnose and predict machinery problems.

One of the principle tools for diagnosing rotating machinery problems has been vibration analysis. Through the use of different signal processing techniques, it has been possible to give vital diagnostic information to equipment operators before equipment catastrophically fails. A problem with this approach is that it requires constant human interpretation of the results. The logical progression of these condition monitoring technologies is the automation of the diagnostic process. Automated diagnostics coupled with an accurate prognostic capability can save lives as well as millions of dollars in costs per year. To automate the diagnostic process, the Penn State Applied Research Lab is currently involved in a project to develop a wireless intelligent sensor system, where each gearbox in a complex system is equipped with individual sensors, data acquisition and microprocessor modules.

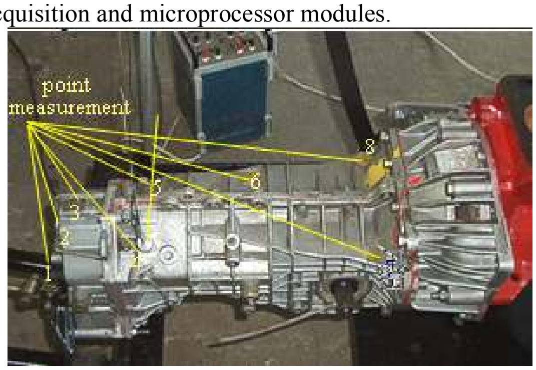


Figure 2 Gearbox Sensor Configurations

The processing modules compute signal features and transmit them back to a central processor for predicting the remaining useful life of the component.

The sensor data is recorded by a PC data acquisition system. The gearbox is mounted on the test stand and driven by a 30 Hp electric motor while the load is supplied by a 75 Hp electric motor/generator. In all, 52 sensors monitor the gearbox, of which, eight measure vibrations.

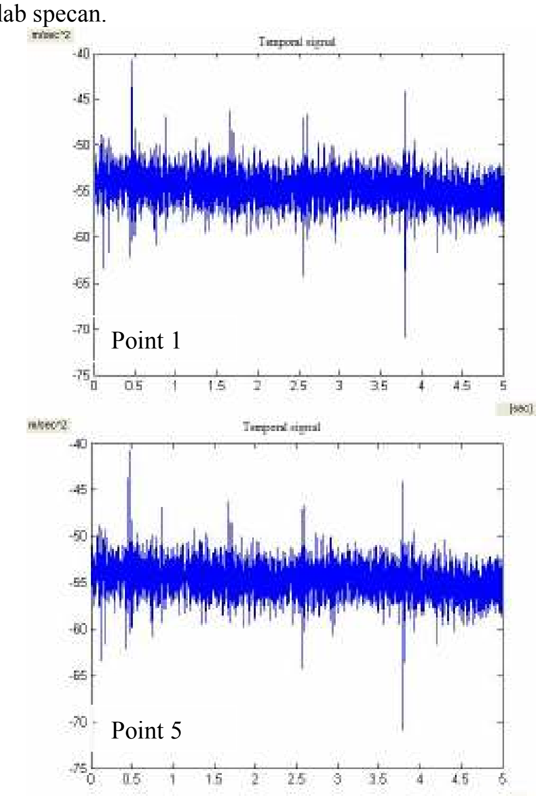
There are seven single axis accelerometers and one triaxial accelerometer. All experimental vibration data were recorded in a 16 bit binary data format with a 20 kHz sampling rate. Ten second data snapshots were taken in half hour intervals and saved to disk for further analysis. The gearboxes are driven at 200-300% of their maximum rating until two accelerometers exceed 150% of their nominal RMS vibration levels. The test is then shutdown and a gearbox autopsy is performed to determine the mode of failure.

V. EXPERIMENTAL DETERMINATIONS

The Mechanical Diagnostics Test Bed at the University of Oradea Lab was created to record transitional sensor data from gearboxes undergoing failure

Used transducers type were piezoelectric KD42 has a sensitivity Bqa = 140 [pC / g]. Experimental measurements were performed in the 8 points down the gearbox.

The acquisition signals were produced with a PCI-1200 acquisition boards, manufactured by National Instruments connected to a PC using a program in Math lab specan.



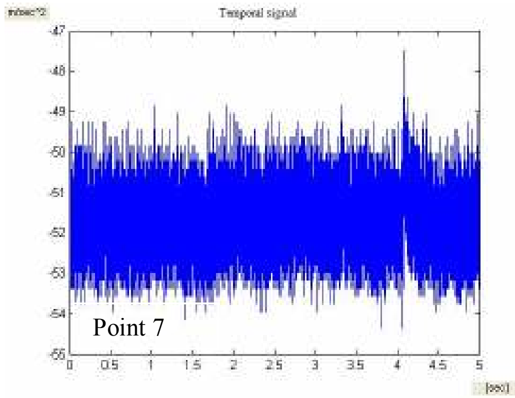


Fig. 3 Temporal signal

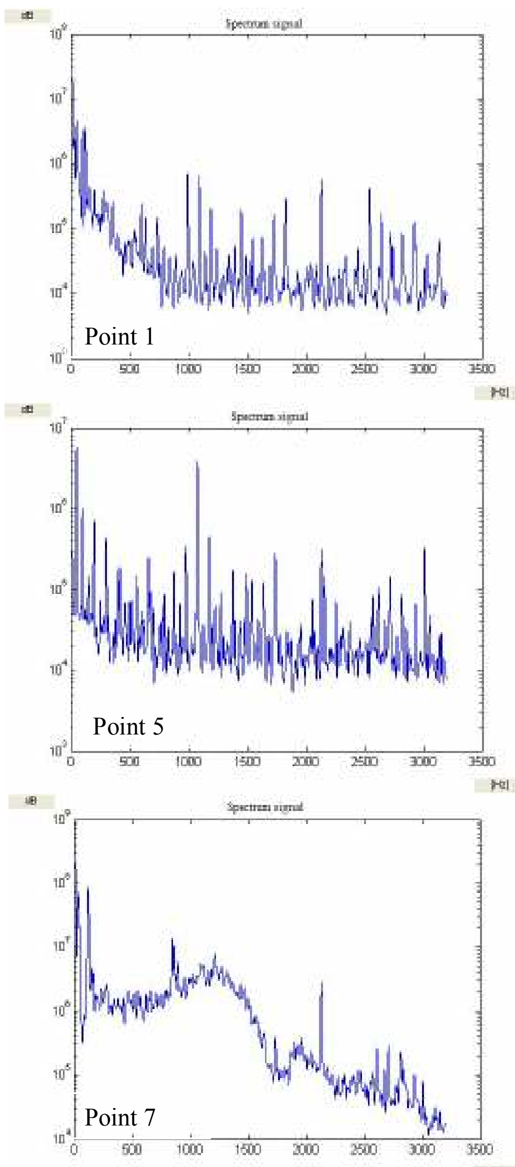


Fig. 4 Spectrum signal

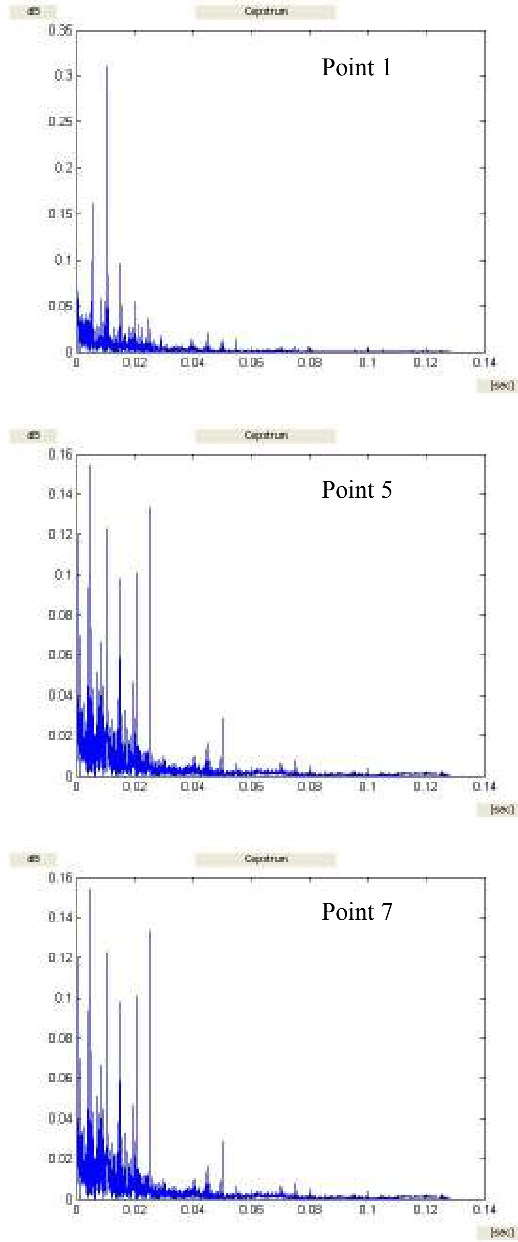


Fig. 5 Cepstrum signal

The acquiring input from a piezoelectric transducer every 5 seconds. The program is written in Visual C using standard libraries and library functions to its purchase card Nat. Instr. PCI 1200. Acquired data are stored in a data file. At the end of data acquisition, the program checks if it was done correctly and displays a confirmation screen.

Analysis of the results led to the following conclusions:

- Temporal signal for measuring points 1-5 are cast in a narrow band just noticed it dispersion amplitudes and for points 6-8 wider band amplitudes
- Amplitude spectrum signals have approximately 10 dB at all points of measurement

- Cepstrum analysis shows double amplitude of 0.3 dB measurement point 1 compared to the other 7 measuring points where the amplitude is 0.14 dB

To obtain the diagrams we used a computer program, *Cepstrum analysis*, in MathLab.

The *cepstru* subprogram builds the cepstrum analysis of the purchased signal.

After defining constants is associated with each type of transducer own constant. Analyzed signal is filtered through filter Hanning after running cepstrum analysis and displays the result.

```
% Calculation and display cepstru
% Constants Program
global x1
fre_esant = 20000;
nr_esant = 2 ^ 16;% must be a power of 2
time = 5;
pct_med = 64;% must be a power of 2
domeniu_plot_frecv = 3200;
domeniu_plot_timp = 0.128;
% Transformation in volts
 x1 = x1. * (10/4096);
% Subtract the offset value
 x1 = x1 - mean (x1);
% Constant KD42 transducer
kd42 = 0.0415;
kd35 = 0.0439;
% Calibration
x1 = x1./kd35;
% Transformation in m / s ^ 2
x1 = x1 * 9.81;
% Generate filter
H = Hanna (pct_med) / pct_med;
% spectrum
amp = real (fft (x1, nr_esant)). ^ 2;
amp = amp (1: nr_esant / 2);
amp = abs (amp);
% Filter (mediation Hanning filter)
amp = conv (H, amp);
amp = abs (amp (pct_med / 2: length (amp) - (pct_med /
2)));
freq = (0: length (amps) -1) * fre_esant / nr_esant;
freq = freq (:);
dom_spec = floor (domeniu_plot_frecv * nr_esant /
fre_esant);
% logarithm
amp = 20 * log10 (amp / (10 ^ (-5)));
% IFFT
amp_ceps = abs (real (ifft (amp)));
% Chart cepstru
timp_ceps = (0: length (amp_ceps) -1) / fre_esant;
dom_ceps = floor (domeniu_plot_timp * fre_esant);
figure, plot (timp_ceps (10: dom_ceps) amp_ceps (10:
dom_ceps), 'k');
title (strcat ('Cepsrum'));
xlabel ('Time [s] '');
ylabel (''dB (m / s ^ 2) ');
```

The final conclusion is that transmission errors caused by defects gears in a gearbox is high in measuring points 1and 2.

REFERENCES

[1] Andrews, S.A., **Noise and Vibrations of Engines and. Transmissions**, MIMechE. University of Waster Australia, Conference Publications, 1979, p (47-57).
[2] Boyes, J.D. **Analysis Technique for Gearbox-Diagnosis Using the High Resolution FFT Analyses**, Bruel & Kjaer, Application Notes, nr.106, 1981.
[3] Brown, D.N. & Jorgensen, J.C., **Machine Condition Monitoring Using Vibration Analysis**, Bruel & Kjaer, Application Note, 1987.
[4] Bruce, B., **Precise control of vibratory stress relief**, Poolinght & Production, nov. 1983, p.(64-66).
[5] Dempsey, P.J., and Zakrajsek, J.J., **Minimizing Loan Effects on NA4 Gear Vibration Diagnostic Parameter**, NASA/TM-2001-210671, 2001.