LARGE GEARBOX VIBRATION MONITORING TECHNIQUES

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ABSTRACT

The vibration monitoring of planetary, multi-shaft gears and bearings system enclosed in a large casing gearbox (e.g. extruder and cement mill gearbox) is a challenging task for vibration analysts. Process industries use many condition monitoring techniques to diagnose gears and bearings faults. The primary program is vibration monitoring and diagnostics. Sometimes the standard vibration monitoring programs may not help to detect fault condition at right time due to the complexities involved in large gearboxes and results in unplanned outage of the plant. The vibration amplitudes and plots sometimes reveal machine is normal but suddenly failure may happen to gears, shafts or bearings. To have an effective vibration monitoring program, an advanced vibration analysis technique shall be incorporated in larger gearboxes. This paper discusses about the torsional vibration monitoring and analysis, time synchronous averaging, cepstrum analysis, high

frequency enveloping, trending, band analysis and alarm setting of large gearboxes.

INTRODUCTION

Generally, the standard vibration analysis features of gearboxes are

- Overall vibration amplitude measurements
- Overall vibration amplitude trending
- Vibration time waveform analysis
- Vibration spectrum analysis

Do these standard vibration analysis features predict the special purpose large gearbox fault? Most of the time the answer would be no, because before fault detection through standard vibration signals, the gearbox shall indicate its fault by generating abnormal noises or with other physical symptoms temperature, oil contamination, etc.

At this situation, the Vibration Analysts shall be under critics for the inability to detect the faults or take necessary corrective actions in time. The failure to diagnose in time could be due to one or several reasons of the following

- Improper selection of vibration sensors, mounting and locations
- Improper data collection settings and configuration
- Vibration signal attenuation due to casing to rotor weight ratio
- Long signal transmission path
- Low overall amplitude vibration
- Low dynamic range of instruments
- Improper alarm settings
- Infrequent vibration monitoring or no continuous monitoring system

To enhance a vibration monitoring program the following advanced techniques should be implemented in the condition monitoring programs of larger gearboxes.

- Torsional Vibration Monitoring
- Time Synchronous Averaging
- Cepstrum Analysis
- High Frequency Enveloping
- Order Tracking
- Band Trending, Band Analysis and Alarm Setting

Let us have a closer look of all these advanced techniques and its benefits.

TORSIONAL VIBRATION

Torsional vibration is an angular vibration exhibit in the system along its shaft axis of rotation. It is defined as a rotating mass of a system connected to shaft which allows the rotating mass to twist each other which creates a dynamic torque in the direction of shaft rotation. The torsional vibration can damage machinery components by generation of dynamic stresses in shaft, gears, coupling, etc.

The torsional vibration is complex because in nature many different frequency components may present in the system. Torsional vibration is one of the major concern in the gearboxes it can cause serious failures if it is not properly controlled. Mainly heavy duty machinery with a sum of different stages and speed variations for different production situations is critical to torsional resonances. Overloads resulting from frequent start-ups and unloading of large gearboxes causes transient torsional vibrations. Fixed speed drives may show torsional resonances during start-ups and coast-down operations. This happens more often by variable speed drives with multiple gears and shafts like extruder machines by change of the operational speed.

The information measured and recorded with special sensors can be used to understand the machine behaviour. The advanced monitoring system nowadays will also record and directly correlate the torsional signal patterns with the actual process parameters directly (e.g. material recirculation, throughput, temperatures, etc.).

TORSIONAL NATURAL FREQUENCY

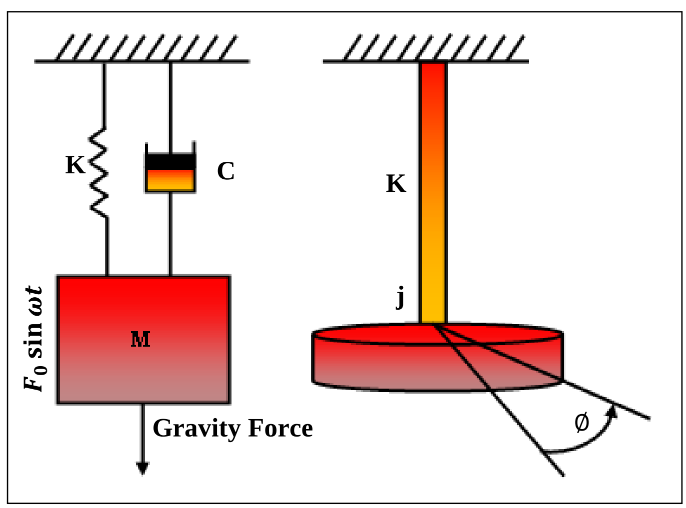

Figure 1 - Single Degree of Freedom - Lateral and Torsional

Torsional natural frequency is similar to lateral natural frequency exhibit in the rotating machinery. The comparison of lateral and torsional vibration is shown in figure 1. Like lateral analysis, the torsional analysis is also evaluated at early design stage. The gears are very susceptible to excite torsional natural frequency that depending on design, speed and operating loads.

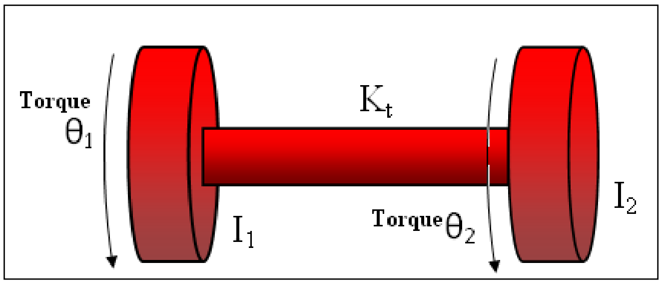

Figure 2 - Torsional Vibration – Two-Inertia System

The torsional natural frequency of the simple model of a system of two lumped masses at opposite ends of a massless shaft is shown in figure 2 is given by

$$f_T = \frac{1}{2\pi}\sqrt{\frac{K_t}{\frac{(I_1 \times I_2)}{I_1 + I_2}}}$$

Where

K_t - Shaft Torsional Stiffness, Nm/rad
I_1 - Polar Inertia 1, Kg-m-sec^2
I_2 - Polar Inertia 2, Kg-m-sec^2
θ_1 - Angular Displacement of Inertia 1, Deg
θ_2 - Angular Displacement of Inertia 2, Deg
f_T – Torsional Natural Frequency, Hz

The designer usually consider torsional natural frequency separation margin at least 10% below or above the operating speed to prevent torsional resonance. For some operational reasons, the torsional natural frequencies of gear systems are often being excited. The excitation of torsional natural frequency is not identified with standard vibration analysis or instrumentation because the vibration responses are rotational instead of lateral also may not show any symptoms until shaft or gear system fails. Thus, the torsional analysis is importance in diagnosis of the gearbox faults.

TORQUE MEASUREMENT AND SENSORS

The torque signals are generally measured either by demodulation method using magnetic pickup/proximity sensor or dynamic twisting angle measurement using strain gauges.

Figure 3 - Magnetic Pickup

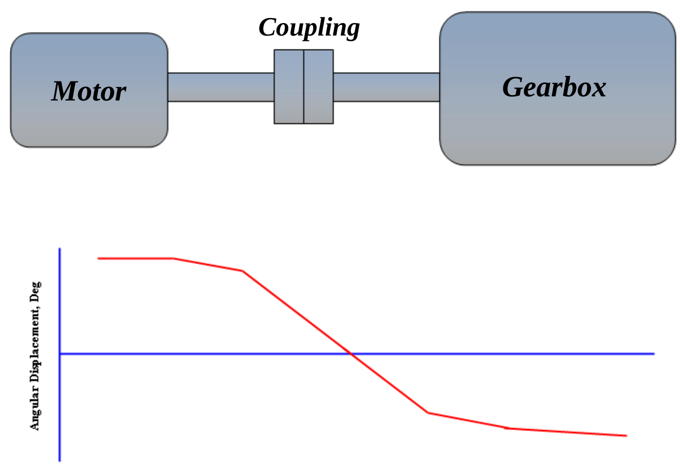

Figure 4 - Torsional Mode Shape

In demodulation method, magnetic pickup or gear teeth sensor figure 3 is used for measurement of steady pulse signal of gear teeth. The pulsing signal is a carrier wave, if there is no equidistant signal pulse phase means torsional vibration is exists in the systems that show up as frequency modulated signal in the plot. The generated steady pulse signals are passed through torsional demodulator and convert modulated signal to torsional vibration.

Another method is measurement of dynamic torque using strain gauges. These strain gauges are mounted at 45° angles to the shaft axis, so that bending of the shaft do not influence torque measurements. The dynamic torque can be measured anywhere on the shaft but the preferred location is a coupling because the stiffness is less than other shaft section also have large slope in the mode shape figure 4.

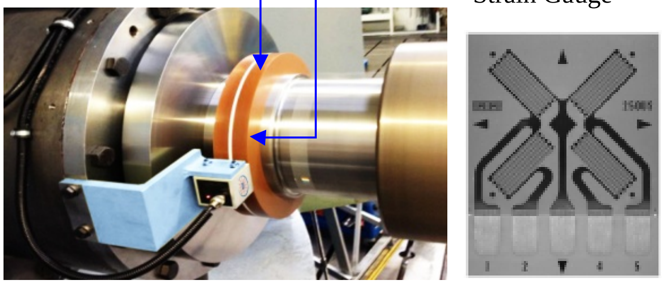

Figure 5 - Strain Gauge Assembly on Gearbox Shaft

The signals from strain gauges shown in figure 5 are connected to a telemetry system which transmits the signal via an antenna ring to a fixed antenna head. The outputs from the antenna signals are processed in the dynamic signal processor and converted to torsional vibration for trending and analysis. The monitoring of torque vibration is an important tool to understand the dynamics of the whole system, meaning how the gearbox/motor is reacting on their operation. Critical operations can be diagnosed as well as overloading of the machine can be avoided. High advanced monitoring systems will also record the main operation parameters of the machine at the same time (e.g. material recirculation/throughput, temperature, material transport conveyors, etc.) to correlate critical process situations indicated by the torque sensor with the actual process parameters directly. Operation errors can also analysed and machine operations streamlined for a longer lifecycle of the machine.

TORSIONAL VIBRATION MONITORING OF GEARBOX

Figure 6 - Extruder Gearbox

The extruder gearbox shown in figure 6 is operated by variable speed drive either a single or combination of fixed and variable speed drive as shown in figure 7 and 8. The opened view of gearbox with dual drive input and output side is shown in figure 9. These gearboxes have complex gear assembly with lot of gears, shafts and bearings.

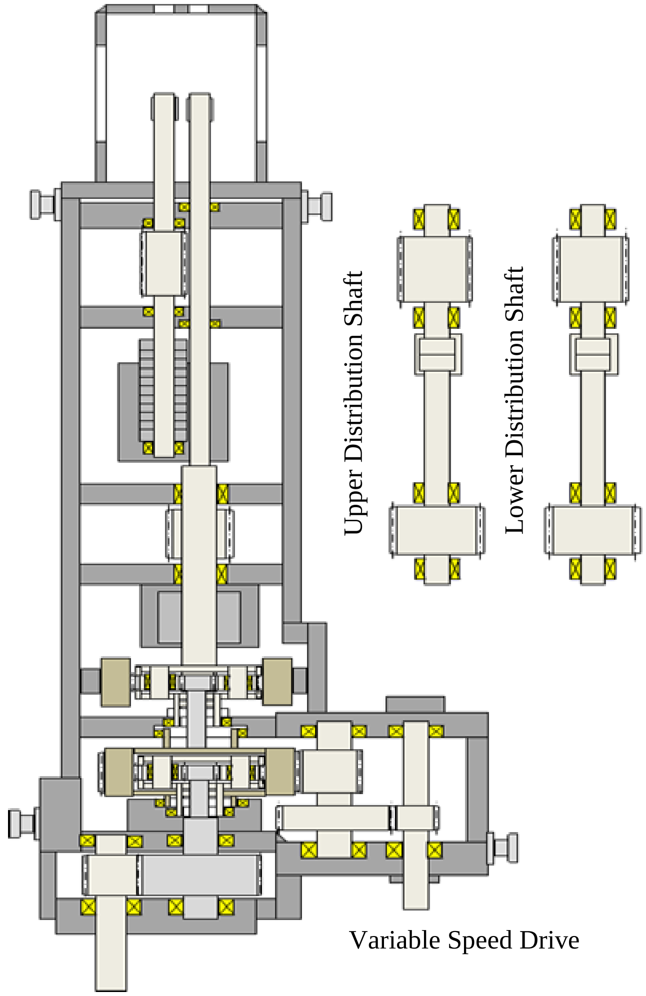

Figure 7 - Extruder Gearbox – Cross Section

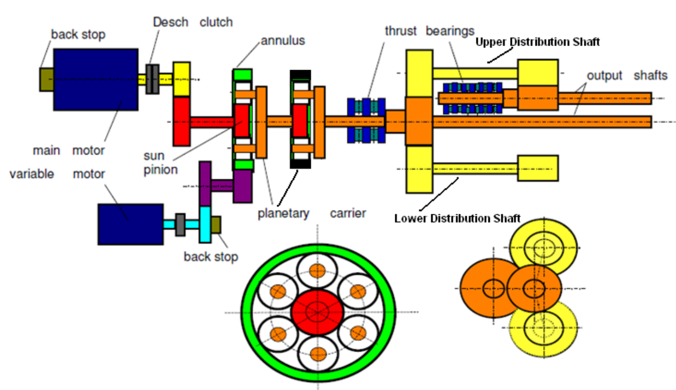

Figure 8 - Extruder's Motor - Gearbox Schematic Diagram

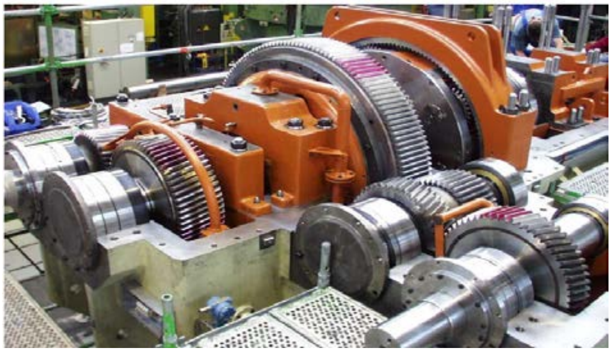

Figure 9a - Extruder Gearbox Input Shaft and Planetary Section

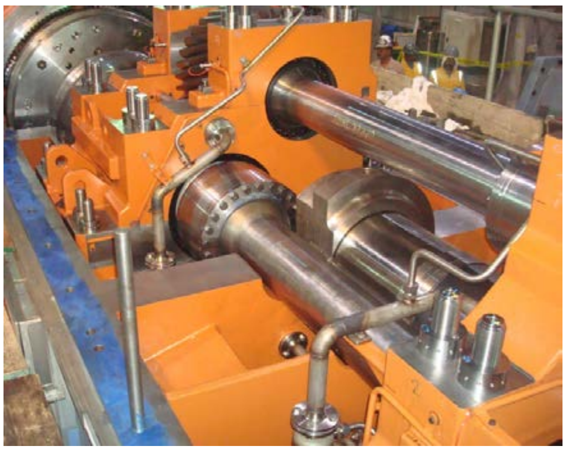

Figure 9b - Extruder Gearbox Torsion Distribution and Output Shaft Section

The figure 7 shows a 7.0 MW capacity extruder's gearbox which contains 30 gears, 10 shafts and 65 bearings including two big size tandem thrust rolling element bearings. Each output shaft is loaded with a maximum torque of 356.0 kNm that is used for operation of the extruder screws. This gearbox is equipped with 22 vibration sensors. Seven sensors are installed inside the remaining are installed outside the casing for continuous vibration monitoring. The gearbox is driving by dual motors one is fixed and another is a variable speed drive motor to change the operating load according to the production requirement. This type of gearbox is likely to pass or operate close to at least one torsional resonance during speed changes or some operational reasons, torsional natural frequency being excited. Apart from torsional resonance frequency, the gearbox can also exhibit gear and bearing fault frequencies that should be scrutinized for diagnosis to predict the exact fault on machine.

The torsional response spectrum shown in figure 10 was recorded from strain gauges mounted close to the drive coupling of output shaft of the extruder gearbox. The construction of plot is similar to standard vibration velocity spectrum, frequency in X-axis and torque in Y-axis. Torsional time base and trending plot is shown in figure 11 and 12 respectively.

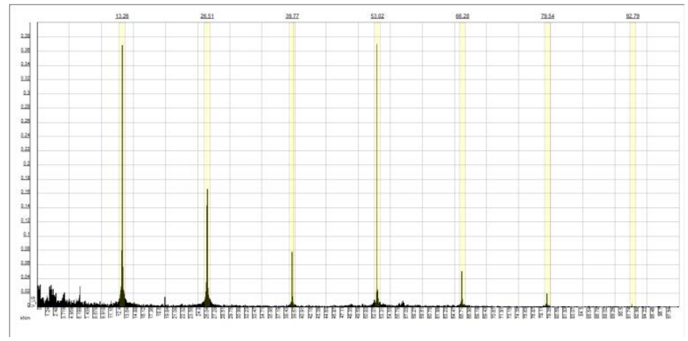

Figure 10 – Extruder Gearbox Torque Vibration Spectrum

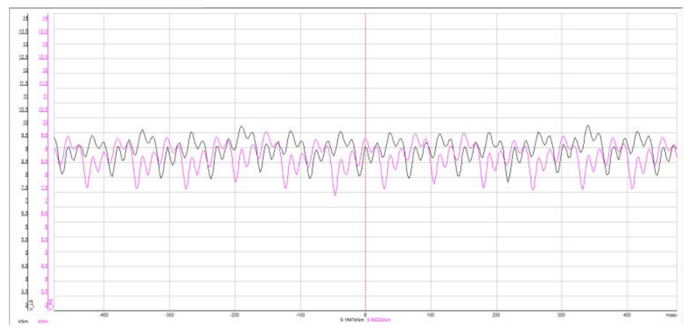

Figure 11 - Extruder Gearbox Torque Vibration Waveform

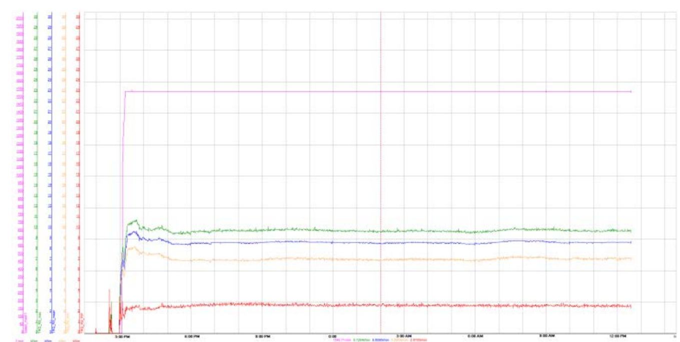

Figure 12 – Extruder Gearbox Torque Trending

The continuous monitoring of torque, trending, in depth analysis of torsional vibration plots and process data correlation shall help the vibration analysts to predict the fault precisely in advance and plan necessary corrective action and also eliminate unexpected plant shutdown and production losses.

Implementation of torsional vibration monitoring in addition to vibration monitoring will enhance the condition monitoring programs of larger size gearboxes. This technique shall help the analyst to predict fault at early stage and avoid the unexpected downtime of machines.

TIME SYNCHRONOUS AVERAGING

Time synchronous averaging is one of the vibration signal processing techniques that extracts periodic waveform signal from noisy data. This time domain averaging process is applied to a complex time waveform to eliminate random noise and unwanted signals. It is a powerful technique and should be used for analysis of multi gear and shaft system.

Time synchronous averaging is a different process from the RMS averaging process. In RMS averaging process, spectral peak shall be averaged out but do not eliminate all random noise and unwanted signal whereas in time synchronous averaging process, the unwanted signals and random noises are eliminated in the time wave form and spectrum.

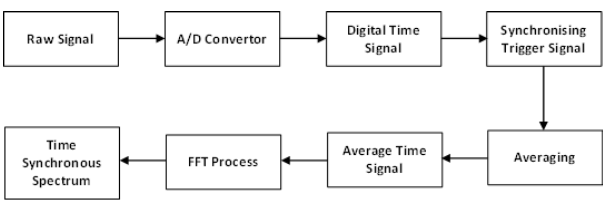

Figure 13 - Block Diagram Time Synchronous Averaging

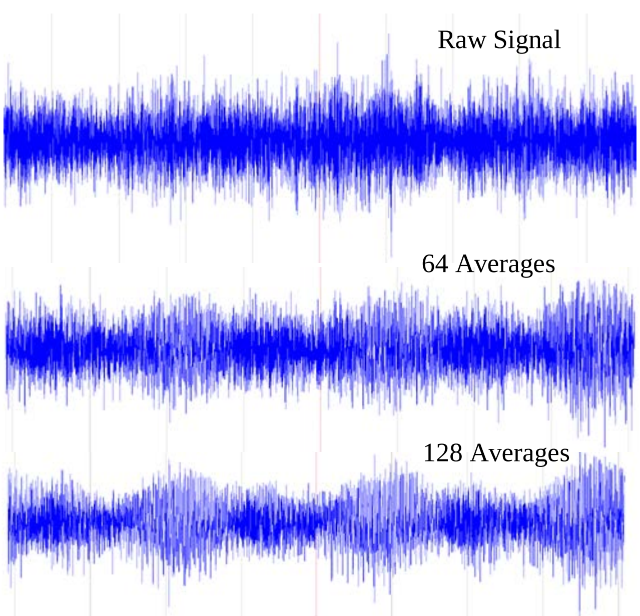

Figure 14 - Time Synchronous Averaging Illustration

The block diagram shown in figure 13 illustrates the TSA process. A signal from the sensor passed through A/D convertor which converts analog to digital signal. Then the digital time waveform signal is processed and averaged to desired numbers. Finally, FFT spectrum is computed from the time domain values. The figure 14 illustrates the averaging process and display synchronous waveform after 124 averages

Time synchronous averaging is well suitable for gearbox analysis which separates out discrete frequency in the presence of multiple gears and bearings fault frequencies in the waveform and spectrum display.

As stated in this paper, extruder gearbox contains many numbers of gears, bearings and shafts. Since the multiple gears and bearings assembled in single large casing, close to each other there will be a frequency transmission in between acceleration sensors. The vibration time domain and spectrum plots collected from these sensors may contain random noises, harmonics of various gears and bearings fault frequencies in the same plot.

Sometimes fault signals may be buried under the random noise floor itself. The analysis of this kind of time domain and FFT spectrum is very difficult also may not help to detect exact faults of the machine. Applying the time synchronous averaging process one can eliminate unwanted signals, random noises and separate out the required discrete signals for analysis.

The figure 15 is a raw time signal had taken from a 5000 kW VRM gearbox with double planetary gear stages figure 17. The figure 16 is after applied Time Synchronous Averaging process to the raw signal. The operating speed of the input shaft is 16.5 Hz and sun shaft speed is 1.23 Hz. The speed of the planet carrier is 0.24 Hz.

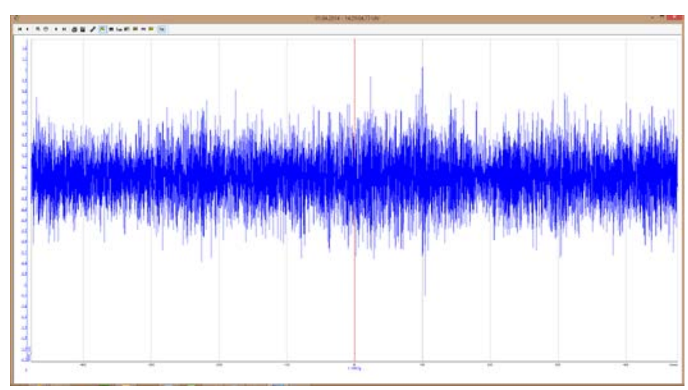

Figure 15 - Gearbox Input Shaft Raw Time Waveform

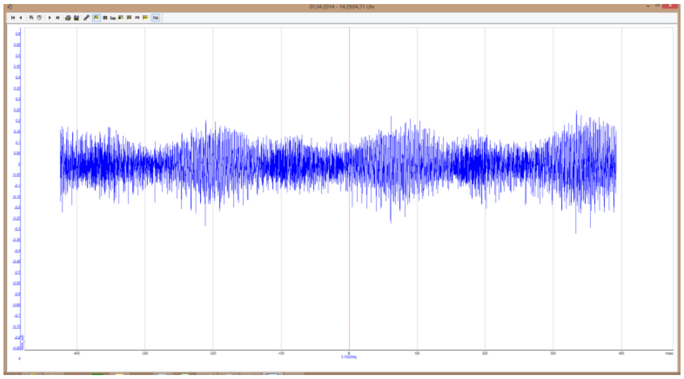

Figure 16 - Gearbox Input shaft Time Waveform after TSA

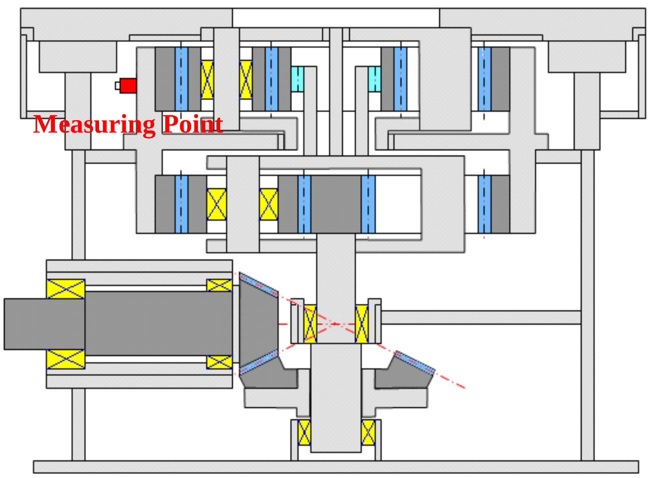

Figure 17 – Vertical Roller Mill Gearbox Schematic Diagram

The raw time signal shown in figure 15 is constructed with many discrete frequencies and random noises that are unable to distinguish the quested operating and gear mesh frequency for proper diagnosis.

The figure 16 is a time synchronous averaged plot after 124 times and synchronised for sun shaft speed 1.23 Hz using the trigger pulse. All non-synchronous frequencies and noises are averaged out from the time domain and only the synchronous frequency peak is displayed.

Figure 18 and 19 is a FFT spectrum plot taken on the VRM gearbox before and after time averaging with trigger pulse of the sun shaft speed 1.23 Hz. The vibration spectrum of plot display frequencies synchronised to shaft speed, other discrete frequencies and noises were averaged out.

Generally, the key phasor of either input or output shaft shall be used for trigger pulse to generate time synchronous averaging plots. But there are some hidden intermediate shafts in the mill gearbox that contain multiple gear mesh and bearing fault frequencies. Measuring the speed or trigger pulse from these shaft directly is difficult due to accessibility. The recording of time synchronous waveform and spectrum for intermediate shaft is not possible. In such condition, using the software program and logics, the trigger signal can be generated for the hidden shaft with respect to gear teeth, input and output speed in digital signal processor. If input or output speed changes, the software program will automatically calculates the intermediate shaft speed and times synchronous averaging of hidden gears and bearings frequencies are measured.

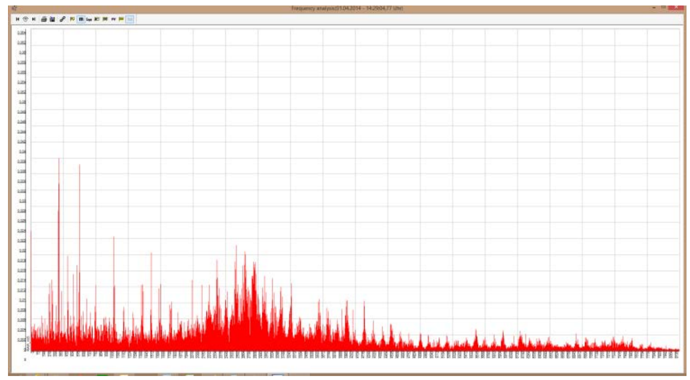

Figure 18 – VRM Gearbox Vibration Spectrum before TSA

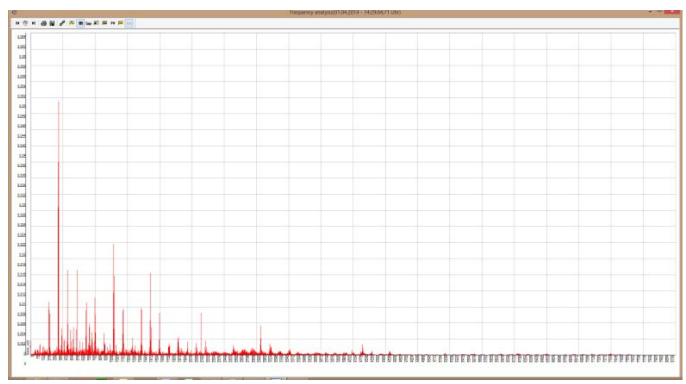

Figure 19 - VRM Gearbox Vibration Spectrum after TSA

Time synchronous averaging is one of the advanced monitoring and analysis techniques available in in latest analyzers and dynamic signal processors. The averaging of time domain and its FFT spectrum definitely improve the diagnosis and eases the analysis for vibration analyst to pinpoint the fault frequencies exactly.

CEPSTRUM ANALYSIS

A cepstrum is the result of taking the Inverse Fast Fourier transform (IFFT) of the logarithm of the vibration spectrum signal. The name is "cepstrum" derived by reversing the first four letters of "spectrum. This technique is applied to detect, analyse the sidebands growth of the gear mesh fault in the presence of multiple fault frequencies. The standard FFT spectrum of gearbox generally consists of many harmonics frequency components of operating speed, bearing and gears faults. In extruder and cement mill gearboxes sometimes it is likely to have a multiple gear mesh frequencies close to each other due to signal transmissions with modulation of operating speed sidebands in the same FFT spectrum. It may inconvenient to analyse the fault for Vibration Analysts. In this case the cepstrum process shall be very useful.

The FFT is a reversible process, constructed on real and imaginary values of time domain. If these real and imaginary values are the input to IFFT then it shall return to time domain values. Using this principle, cepstrum plot for diagnosis can be constructed.

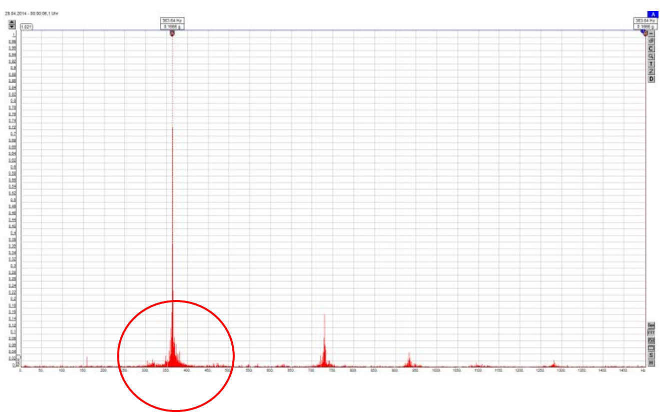

Figure 20 – VRM FFT Spectrum Plot No Fault

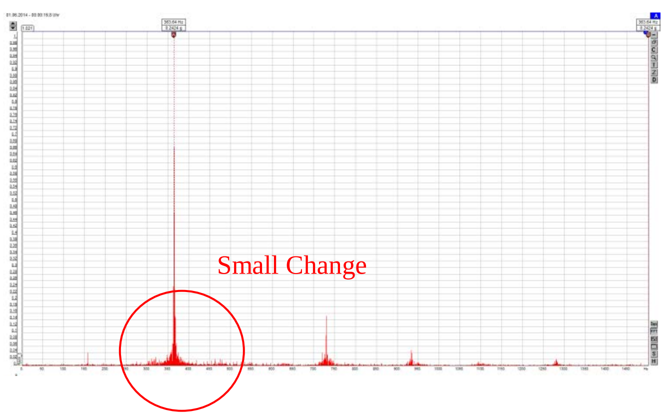

Figure 21 – VRM FFT Spectrum Plot Beginning Fault

The diagnosis of the growth of sidebands in the spectrum plots is important however with cluster of complex signals in the spectra may be difficult to recognize these sidebands severity. The cepstrum process has an ability to separate out the sidebands signal and frequency periodicity that may not immediately visible to the eye. By using cepstrum process one can easily see the growth of sidebands and fault severity also detect which gear is having fault.

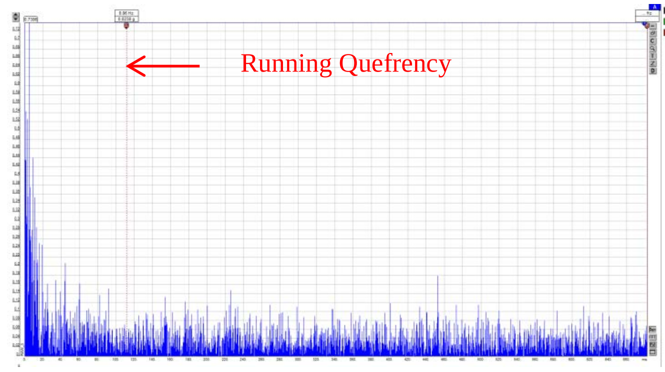

Figure 22 – VRM Cepstrum Plot No Fault

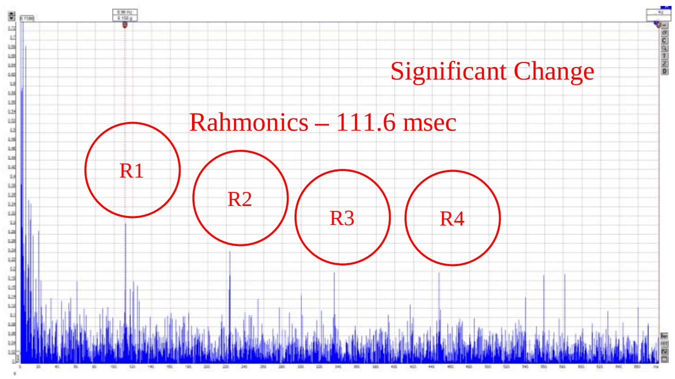

Figure 23 –VRM Cepstrum Plot Beginning Fault

The FFT spectrums figure 20 and figure 21 was obtained on a VRM (vertical roller mill) gearbox before no fault and with a beginning of single tooth crack on bevel gear wheel. The bevel gear running speed is 8.96 Hz and GMF is 363.64 Hz respectively. The spectral plots for the beginning fault did not shows significant changes of the frequency pattern and not apparent to the eye. Figure 22 and 23 is the corresponding cepstrum it shows a significant change in its pattern, operating speed periodicity quefrency 111.6 msec and it rahmonics (Reverse of harmonics) that are visible to the eyes.

The mathematical algorithm rules are programmed in the advanced signal analyzer, once machinery details are configured that will automatically perform the cepstrum plot. Employing this technique in the gearbox monitoring program, the vibration analyst can monitor fault frequencies in its early stage and detects which gear is having a significant defect in the presence of multiple fault frequencies.

HIGH FREQUENCY ACCELERATION ENVELOPING

High frequency acceleration enveloping (HFE) is a standard and most popular technique used to detect rolling element faults at early stage. The basic principle of the enveloping process is extraction of a bearing resonance signal results from repeatedly impact of rolling elements or structure. The excited resonance signals are usually a low energy and high frequency in nature at early failure stage. This may not be seeing in standard FFT spectrum because these peaks are buried in the noise floor. The process of extraction of signal using electronic circuits, HFD filters and software programs are said to be high frequency enveloping or demodulation. The figure 24 is a block diagram and illustrates the HFE process.

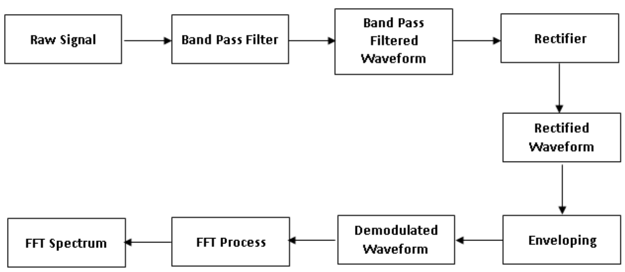

Figure 24 - High Frequency Enveloping Process

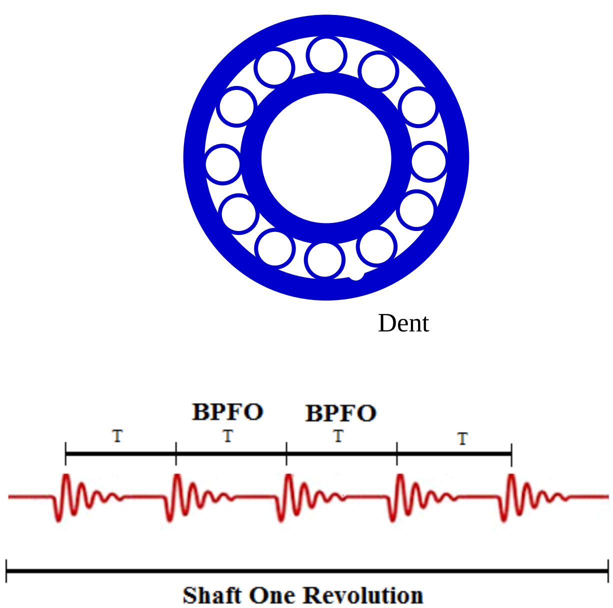

Figure 25 – Bearing Fault Frequency Illustration

For example, let consider a ball bearing contains 12 rolling elements and running at 3000 rpm. If the outer race is has a small crack or dent that causes 12 impacts in a single rotation of the shaft as shown in figure 25. Every impact excites the bearing natural frequency. This is similar to hitting the structure with a hammer. This resonance frequency is depends on bearing geometry, mass, stiffness and damping, usually range between 3.0 kHz to 50.0 kHz or more. This low energy impact and rolling elements in and out on the raceway in every rotation generates an amplitude modulation vibration signal at bearing fault frequencies intervals. The envelope process is applied over the modulated signal to generate demodulation signal, which is processed further to low frequency HFE spectrum.

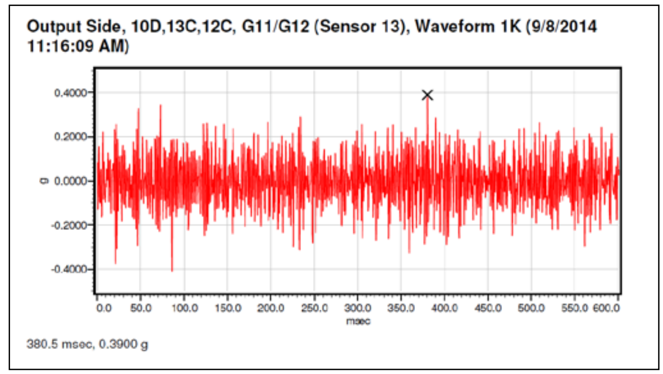

Figure 26 - Extruder Bearing Upper Torsion Distributor Shaft Bearing Raw Time Waveform

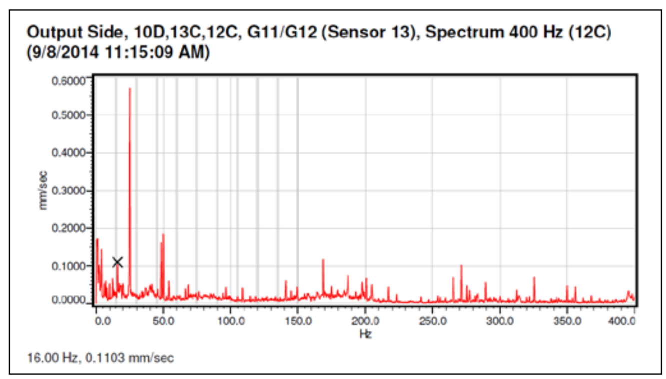

Figure 27 - Extruder Gearbox Upper Torsion Distributor Shaft Bearing FFT Spectrum

The figure 26 is a raw acceleration TWF signal and figure 27 is a standard FFT spectrum (400 Hz) of extruder gearbox upper torsion distributor shaft's bearing figure 7. At the time of recording the plots, shaft speed was 90.5 rpm and bearing fault frequencies are BPFI - 19.7 Hz, BPFO - 15.9 Hz and BSF - 5.5 Hz. Both of the plots did not shows clearly the bearing outer race fault frequency BPFO - 15.9 Hz because the fault is in early stage, has low energy and the signal is a mix of many other fault frequencies and random noise.

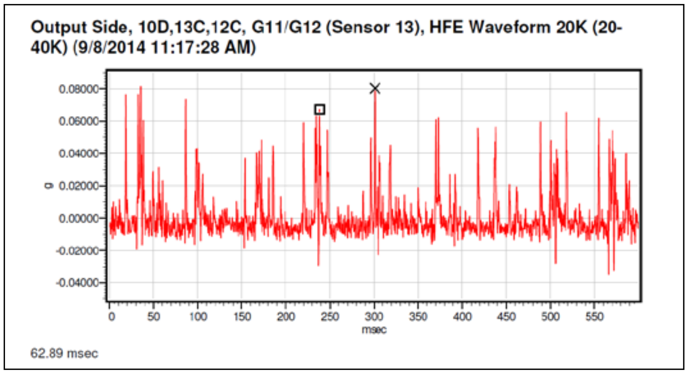

Figure 28 – Extruder Gearbox Upper Torsion Distributor Shaft Bearing Filtered Time Waveform (Filter 20 - 40K)

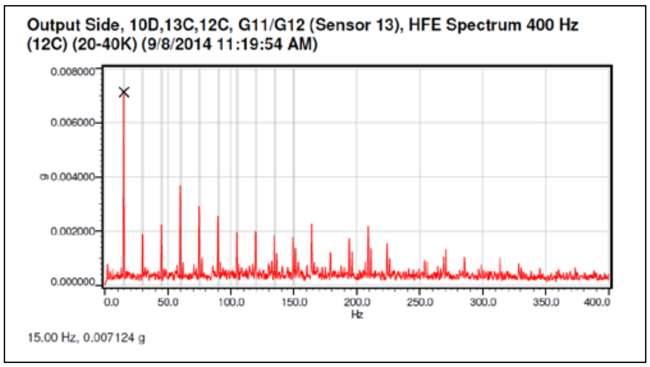

Figure 29 - Extruder Gearbox Upper Torsion Distributor Shaft Bearing HFE Acceleration Spectrum – (Filter 20 - 40K)

The same raw time waveform signal is passed through a 20.0 - 40.0 kHz band pass filter to process HFE spectrum. Note that the selection of filter is important in HFE process otherwise quality of data and spectrum may not show the defect frequencies or no good enough data to diagnose the bearing faults. This band pass cuts off the vibration signal below 20.0 kHz and above 40.0 kHz. The figure 28 shows a filtered waveform and figure 29 is displayed an enveloped FFT spectrum of an extruder gearbox upper torsion distributor shaft's bearing figure 7. Both these plots were clearly shows the bearing outer race fault frequency BPFI 15.9 Hz.

The other way to better understand, enveloping is a process of drawing a line over the band pass filtered time peak that makes an enveloping signal. To do envelope processing either analog or digital circuit is needed. The analog envelope detector circuit is larger in size. Presently software based enveloped detectors are used for this application and it is cheaper than analog filters. There are several algorithms used for digital enveloping process but widely used algorithm is the Hilbert transform method shown in figure 30.

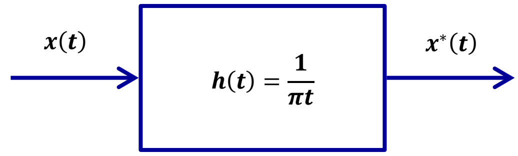

Figure 30 - Hilbert Transform Function

The Hilbert transform is a process of transformation of time to time signal where the output is exactly same as the input expect its phase at all frequencies that has shifted by 90 degrees. This mathematical calculation process are taken care in the signal processor itself, so this paper is not going to discus in details how the filters are functioning and signal processing; it is just to understand filter basics and how the envelope detectors are working.

Let look how to configure the analyzer properly and collect the enveloped spectrum because this is important for vibration analysts to diagnose the fault. Accelerometers are the best one to measure acceleration raw signal directly from the bearing locations. The preferred best method of mounting accelerometers is stud mount, so that high frequency response shall be obtained properly to have quality spectrum for analysis. Mount the accelerometer as close as possible to the bearings so that signal energy will not take long transmission path to reach sensors otherwise the energy pulse may low in magnitude or may not sufficient to compute spectrum for diagnose.

Next, the selection of band filter is also important in the envelope process. There are several pre-set band pass filters are available in the signal processor, generally they are 2.0 kHz - 5.0 kHz, 5.0 kHz – 10.0 kHz, 10.0 kHz – 20.0 kHz and 20.0 kHz – 40.0 kHz. Most of the bearings resonant above 3.0 kHz however, it is depend on bearing geometry, mass, stiffness and damping. The selection of number of lines and averaging is also plays a role in enveloping, analyst has to choose proper lines and averages to have good resolution spectrum.

Though the enveloping process is a very useful technique for diagnose the faults but there are some limitations. The energy pulse and magnitudes generation are very sensitive to the applied load and shall vary. There is no specific value to assess the severity of the bearing deterioration. The presence of the fault frequencies in the HFE spectrum is not means inspection and replacement of bearings should be planned immediately. It has suggest do close monitoring of the envelope spectrum once a defect frequency is seeing, increase the monitoring program and look for fault frequencies in standard FFT spectrum. Proper trending, alarm setting, evaluation of peak frequency, magnitude and sideband analysis are playing an important role in successful diagnosis of the fault condition using the enveloping. Based on severity condition, inspection of bearings shall be planned. However, it needs extensive experience to take decision for stoppage of the plant otherwise its end up with nothing. It is really a challenging task for vibration engineers.

ORDER TRACKING

Order analysis is a technique for analysis of vibration signals that contains rotational speed components and it harmonics. The order tracking processes will give good accuracy and resolution. To get accurate orders the signal should be always exactly in center of the bin. For that sample rate should be constant value and speed should not vary.

In a variable speed machine if the frequency of the signal is changes, location of the order peak also changes in FFT spectrum. To get a constant sample rate, the sampling rate should be function of the speed for a variable speed machine. Therefore, order tracking is important in analysis for a variable speed machine where frequently the speed changes.

As know, the sampling rate of the 400 lines spectrum is

$$F_s = 2.56 \times F_{max}$$

Or

$$F_{max} = \frac{F_s}{2.56}$$

Where

F_s – Sampling Rate, Hz
F_{max} – Maximum Frequency of Spectrum, Hz

Now if the sampling rate is some multiple of N speed, then the sampling rate is

$$F_s = N \times F_1$$

Then

$$F_{max} = \frac{N \times F_1}{2.56}$$

Where

N – Number, Power of 2
F_1 *– Speed, RPS*

If the running speed is located in the Bth bin

$$B = F_1 \times \frac{400}{F_{max}}$$

$$B = \frac{F_1 \times 400 \times 2.56}{N \times F_1}$$

$$B = \frac{1024}{N}$$

Now the bin B is independent of speed and equal to 1024/N. Using the above equation running speed signal bin location can be calculated.

Using this technique, one can compare the spectrum at different speeds with good accuracy and resolution. Note to have peak in center of the bin all the times then it should be a whole number, therefore N should be always power of two. This feature is programmed in advanced vibration monitoring software and order tracking will be performed automatically with a speed signal as reference whenever the gearbox speed changes, always keep the signal in center of the bin.

Another benefit is the filtered order analysis for evaluation of rotational speed harmonics. Applying band pass filter to the signal and filter out all other frequencies that are not of interested. Ordering tracking feature is an important application and must be used in analysis of large gearbox especially driving by a variable speed drives.

BAND TRENDING, FREQUENCY BAND ANALYSIS AND ALARM SETTING

In the above topics, so for discussed about some advanced vibration monitoring techniques for diagnosis of the large size gearbox that has multiple gears, bearings and shafts. Using these techniques faults can be diagnosed exactly where on the machine, but severity evaluation or decision to shut the machine for maintenance and do corrective actions are not possible. The machine condition evaluation and decision making for stoppage is the major tough task for analysts.

Though there is some monitoring systems are installed for analysis, an unexpected failure may be happened to the gearbox. If we asked a question, why the fault condition had not predicted before failure of the gearbox? The answers may be vibration amplitudes were not picked up by analyzers, low vibration amplitude or vibration not exceeded the alarm set level, no fault frequency seen in the plots, etc.

There are several reasons for not detection of fault condition of the equipment but the important reason is no proper trending and alarm settings in condition monitoring programs. Let discuss about overall vibration trending, band analysis and alarm setting in details.

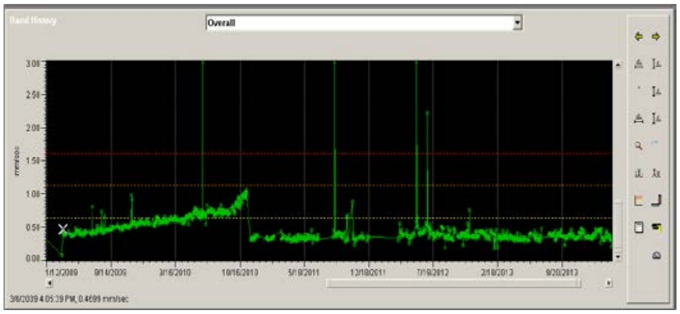

Figure 31 - Extruder Gearbox Tandem Bearing (T2AR) Velocity Trend

Overall vibration trending is a process where the overall vibration amplitude either displacement or velocity or acceleration of each location readings plotted against the time. It is simply a plot constructed by amplitude versus time that can be seconds, days, months or years. The trend is a good indicator about the deterioration of the machine in conjunction with proper alarm setting. The overall velocity amplitude of extruder gearbox output shaft's (figure 9b) tandem bearing (T2AR) is shown in figure 31.

Frequency band analysis is a trending and analysis of the fault frequency that is present in the desired frequency bands. The frequency span of vibration spectrum plot is divided into different sub frequency bands of mass unbalance, misalignment, bearing and gear meshing frequencies. The minimum and maximum frequencies of each band will be in units or orders of the running speed of the machine figure 32.

The frequency span of overall and band vibration analysis shall be atleast three times of the gear meshing frequencies or ten times of the shaft running speed whichever is greater. The frequency span should have enough resolution to resolve sidebands of the fault frequency.

Next, the understanding of the vibration acceptance limit and alarm setting for a continuous and periodic monitoring is an important especially for large size special purpose gearbox. There are ISO, API, AGMA, GM and manufacturer's standard guidelines available for vibration acceptance limit for new and rebuilt machines.

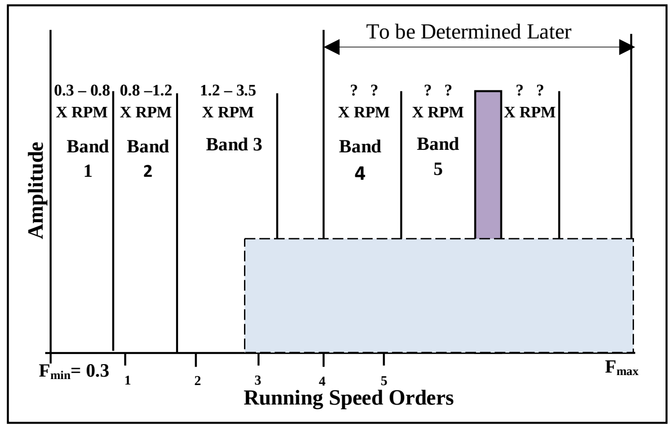

Figure 32 – Frequency Band Analysis (GM Standard)

Vibration Acceptance Limits				
Velocity mm/sec -RMS	Class I	Class II	Class III	Class IV
0.28	A	A	A	A
0.45	A	A	A	A
0.71	A	A	A	A
1.12	B	A	A	A
1.8	B	B	A	A
2.8	C	B	B	A
4.5	C	C	B	B
7.1	D	C	C	B
11.2	D	D	C	C
18.0	D	D	D	C
28.0	D	D	D	B
45.0	D	D	D	B

Equipment Classification	
Class I	Individual components of engines and machine integrally collected to the complete machine in its normal operating condition (< 15 kW)
Class II	Medium sized machines without special foundations (15 – 75 kW) and rigidly mounted machines on special foundations (up to 300 kW)
Class III	Large prime movers and machines with rotating mass mounted on rigid and heavy foundations where relatively stiff in the direction off vibration measurements
Class IV	Large prime movers and other large machine with rotating mass mounted on foundations which are relatively stiff in the direction of measurements (eg. Turbo-Generators and Gas Turbines > 10 MW)

Zone A - For newly commissioned machines
Zone B - For Unrestricted long-term operation
Zone C – For Unsatisfactory for long-term operations
Zone D – Have sufficient severity and cause potential damage

Table 1 - ISO 10816 - 1 Vibration Severity Chart

The ISO 10816 - 1 general guideline for evaluation of machines vibration standard and specified vibration acceptance limits are shown in the table 1. The large size gearbox (e.g. extruder, cement mill or marine engine gearbox) shall fall generally under class III category. It has suggested user should properly evaluate the machine constructions and design. Accordingly the vibration acceptance limits and classification shall be established either to new or rebuilt machines also for periodic condition monitoring programs.

Casing Vibration Levels		
	Velocity - RMS	**Acceleration True Peak**
Frequency Range	10 Hz – 2.5 kHz	2.5 kHz – 10 kHz
Overall	2.9 mm/sec (0.11 in/sec)	4.0 gs
Discrete Frequency	1.8 mm/sec (0.07 in/sec)	

Note: Discrete frequency shall be determined from FFT spectrum

Table 2 - API 613 Casing Vibration Limit

The vibration acceptance limit specified in API 613, a special purpose gearbox has given in the table 2 for newly purchasing machine. The limits specified in the table 2 is a guideline value only however the supplier and purchaser shall have mutually agreed values between them or follow OEM standards as limit.

MAXIMUM ALLOWABLE VIBRATION LEVELS FOR GEARBOXES WITH ≤ TWO (2) GEAR SETS			
VELOCITY LINE-AMPLITUDE BAND LIMITS			
Band	**Frequency Range Hz/CPM**	**Velocity**	
		mm/sec RMS	**In/sec -Pk**
1	(0.3 - 0.8) x RPM	0.718	0.04
2	[0.8 - 1.2] x RPM	1.35	0.075
3	(1.2 - 3.5]) x RPM	0.718	0.04
4	(3.5 - 8.5) x RPM	0.54	0.03
5	8.5 x RPM - 1,000 Hz (60,000 CPM)	0.54	0.03
6	1,000 - 2,000 Hz (60,000 - 120,000 CPM)	0.54	0.03
ACCELERATION BAND-LIMITED OVERALL AMPLITUDE LIMITS			
Band	**FREQUENCY RANGE Hz**	G's - RMS	G's Pk
1	0.3 x RPM - 3.5 x GMF Or 10 kHz Whichever is greater	0.707	1.0

Table 3 - Allowable Vibration Limit for Gearbox (GM Standard)

The General Motors (GM) vibration standard GM – 1761 acceptance limits for a new and rebuilt machine is specified in terms of "**LINE AMPLITUDE ACCEPTANCE LIMITS**" and "**BAND-LIMITED OVERALL AMPLITUDE LIMITS**". Refer Table 3.

The maximum allowable vibration limits specified in the table - 3 is for the gearbox contains equal or less than two gear sets. For gearboxes with more than two (2) gear sets, the acceptance limits shall be established between purchaser and supplier. It is the purchaser responsibility to ensure suitable acceptance value for the purchasing gearbox.

The GM standard defines **"LINE AMPLITUDE ACCEPTANCE LIMITS"** (Figure 33) is

- A line of resolution will have a band width ($\Delta f) = 5.0$ Hz (300 CPM) unless specified otherwise restriction would result in less than 400 lines of resolution over the frequency range specified for certification, in which case the resolution requirement will default to 400 lines. The greater resolution (i.e. $\Delta f < 5.0\ Hz$) may be required for low speed equipment, to resolve "Side Bands," or in Band 1 to resolve machine vibration between 0.3X and 0.8X Running Speed.
- The maximum amplitude of any line of resolution contained within a band shall not exceed the Line Amplitude Acceptance Limit for the band.
- If a line of resolution is coincidental with the F_{min}/F_{max} of two adjacent bands, that line of resolution will be included in "Line Amplitude Acceptance Limit" evaluation for each band.

The GM standard defines **"BAND-LIMITED OVERALL AMPLITUDE LIMITS"** (Figure 34) is

- The total vibration level "A" in a band, as calculated by the following equation that shall not exceed the Overall Amplitude Acceptance Limit specified for the Band.

$$A = \sqrt{\frac{\sum_{i=1}^{N} A_i^2}{1.5}}$$

Where

A - Overall vibration level in the Band
A_i - Amplitude in the ith line of resolution in the Band
(i = 1) - The first line of resolution in the Band
(i = N) - The last line of resolution in the Band
N - Number of lines of resolution in the Band
W - Window Factor (W = 1.5 for a Hanning Window)

- If the total energy in a peak is to be measured, a minimum of 5 lines of resolution must be used and the peak must be centred in the band.

- If a line of resolution is coincidental with the F_{min}/F_{max} of two adjacent bands, that line of resolution will be included in the "Band-Limited Overall Amplitude Acceptance Limit" evaluation for the band.

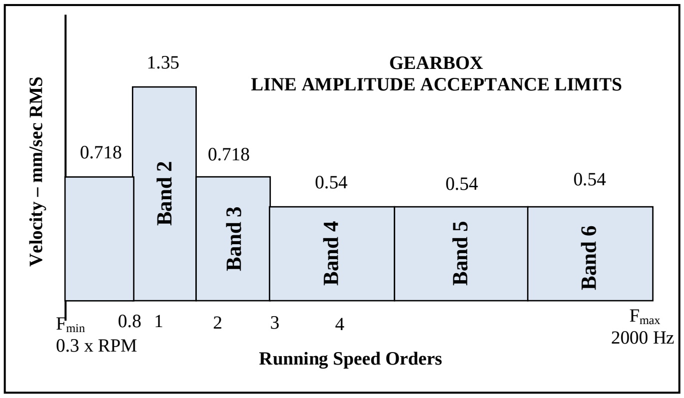

Figure 33 - Line Amplitude Acceptance Limits for Gearbox (GM Standard)

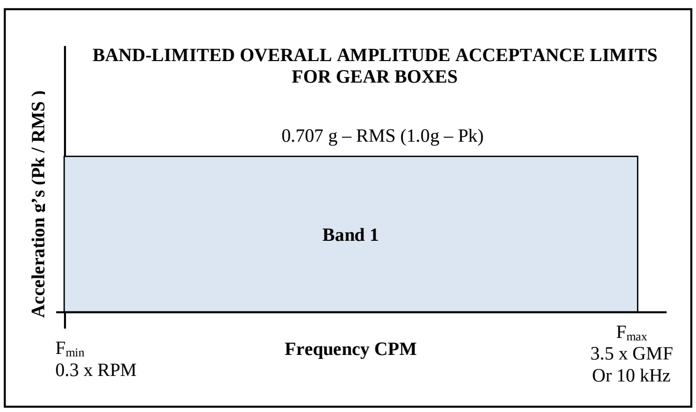

Figure 34 - Band Limited Overall Amplitude Limits (GM Standard)

The vibration acceptance limit values of various standards discussed above is a general guideline only. Though these standards are specified acceptable limits, all of them are also suggests trending and setting alarm with reference to baseline data would be the appropriate. Unless otherwise if new or repaired machine did not have specific baseline data, the vibration analyst shall use the above discussed acceptance limits till average baseline data for each locations of the gearbox is established. Once an average baseline value is established suitable alarms can be set in the monitoring programs.

The following are the three types of GM standard recommended alarm setting, shall be used for continuous vibration monitoring program for large size gearboxes

- The "**First Warning**" vibration levels shall set 1.5X the applicable new and rebuilt machine maximum acceptance levels (baseline data) for the machine under consideration. This First Warning Level would indicate a problem has developed and its' severity has reached a point where, although the machine can continue to be run, more frequent monitoring of the machine's "Health" is recommended.

- The "**First Alarm**" vibration levels shall set 2X the applicable new and rebuilt machine maximum acceptance levels (baseline data) for the machine under consideration. This First Alarm Level would indicate the severity of the problem has reached a stage where the developing cause of the vibration needs to be identified, necessary repair parts identified and ordered (if not in crib stock), date for repair established based on minimum production interruption, and skilled trades personnel identified and scheduled for the repair. Although the machine can continue to be run, it should be closely monitored, particularly if it is a "critical machine".

- The "**Second Alarm**" vibration levels shall set 2.5X failure pending or 3X - failure eminent. If the machine is a critical machine, it should be schedule for repairs ASAP.

The above said alarm is a general thumb role however after experience with on machine, the vibration levels for warning and alarm can be adjusted to fit the specific machine health conditions. The proper setting of alarm and trend monitoring plays an important role in monitoring program and helps to take right decision for maintenance on the machines timely. The advanced diagnostic system has the capability of band trending and frequency analysis. The signal processor and software is programmed to perform this task automatically. The user has to do machine configuration only according to their requirement.

The extruder gearbox is huge in size and acceptance limits may vary from location to location with respect to baseline vibration data. It is suggested that this type of big gearboxes should have permanent continuous monitoring system also facilities of advanced monitoring techniques. The alarm setting is to be established properly to alert operators and analyst for necessary corrective actions.

The figure 35 & 36 is Extruder gearbox figure 7 & 9 output shaft's two stages tandem bearing T2AR overall vibration amplitude and thrust plate fault frequency's band trend respectively. This gearbox is 7.0 MW capacity and normal operating speed range of each output shaft is from 100 rpm to 160 rpm shall vary based on process requirements. It has installed total 22 vibration sensors out of that 7 sensors are installed inside the gearbox and the remaining installed outside the casing for continuous vibration monitoring. The tandem bearing T2AR vibration monitoring sensor is mounted inside the gearbox in axial direction and near to the thick tandem bearing housing. The figure 37 & 38 is spectrum before and after T2AR thrust plate fault generation.

The average overall vibration amplitude and T2AR thrust plate fault frequency's band baseline energy of this tandem bearing had recorded as 0.32 mm/sec and 0.1 mm/sec respectively. Three types of alarm values had set in the monitoring programs with reference to the average baseline value of the band frequencies. If look the values recorded in the trend, total received peak energy was very low in magnitude and not easy to distinguish a fault if doing standard analysis. The low energy is because of low operating speed, sensor mounting location, signal transmission path, large and thick casing that attenuates signal transmission to sensors.

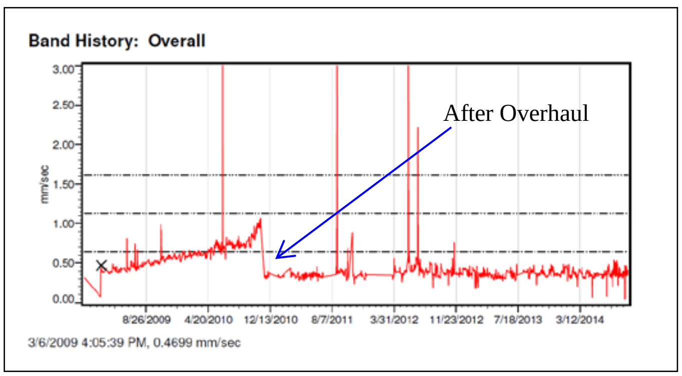

Figure 35 - Extruder Gearbox Tandem Bearing T2AR Overall Band Vibration Trend

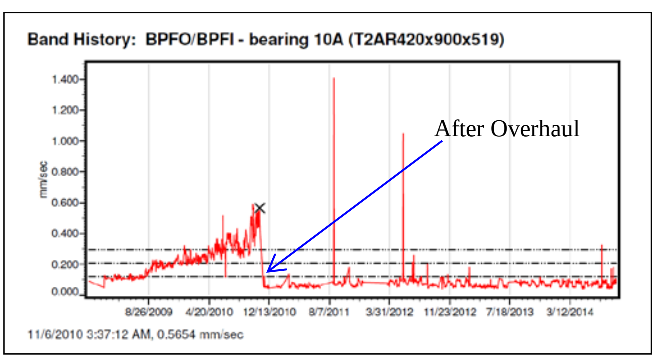

Figure 36 - Extruder Gearbox Tandem Bearing T2AR Thrust Plate Fault Band Vibration Trend

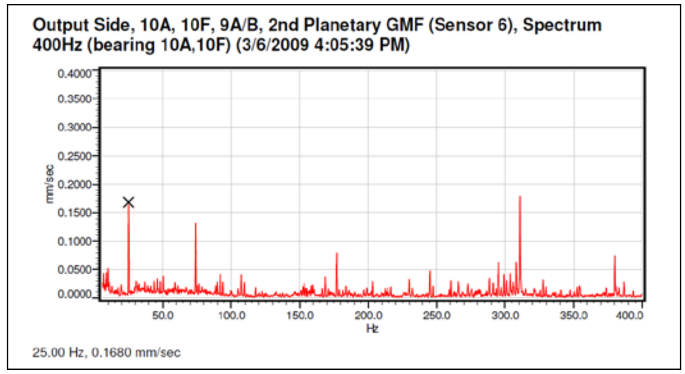

Figure 37 - Extruder Gearbox Tandem Bearing T2AR Vibration Spectrum - No Fault

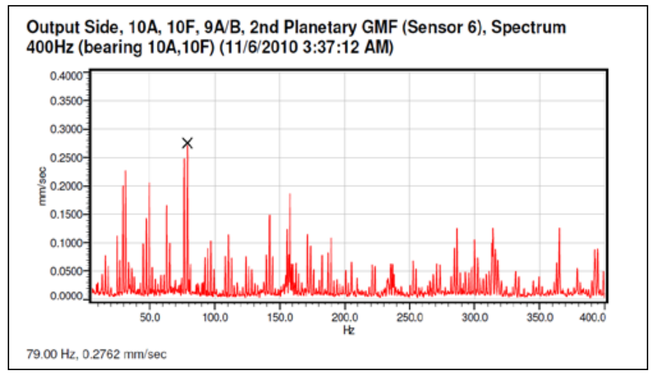

Figure 38 - Extruder Gearbox Tandem Bearing T2AR Vibration Spectrum - Thrust Plate Fault

The vibration amplitudes of the trend values were gradually increases and reached the alarm set values. The overall value increased to 1.1 mm/sec and reached above FIRST WARNING (0.65 mm/sec). T2AR thrust plate band trend value increased to 0.57 mm/sec and reached over the SECOND ALARM (0.3 mm/sec). Though the peak energy was low, the decision had made to stop the plant; do internal inspection and overhaul of the gearbox. The gearbox was opened for overhaul and found two stages tandem bearing (T2AR) completely damaged figure 39. Apart from this bearing another eight stages tandem thrust bearings (T8AR) and all output shaft bearings also found damaged and cracked. These bearings were given expected good lifetime and exceeded the OEM recommended basic bearing rating life.

This gearbox has 7 gear meshing fault frequencies, 76 bearings fault frequencies and 10 different stages of speed. If not properly evaluated machine construction, vibrations and operations parameters one can say there is no vibration in the gearbox and continued further. Though decision was made little delay due to operational constrains the major catastrophic failure was avoided by evaluating overall performance of the gearbox. After overhauling and replacement of all bearings, the vibration readings are came down to normal ranges. Since then the gearbox is in service with acceptable vibration levels and below the first warning limits. These are clearly seeing in the trend figure 35 & 36.

The frequency band analysis and trending is more important for big gearbox analysis where casing is heavy and have more thickness. The signal transmission to sensor may not have enough energy and most of the fault frequencies amplitudes will be in the noise floor level. The author personally have experience on these machines monitoring and suggests implementation of frequency band trending and advanced analysis on those machines are necessary rather standard overall amplitude monitoring and analysis.

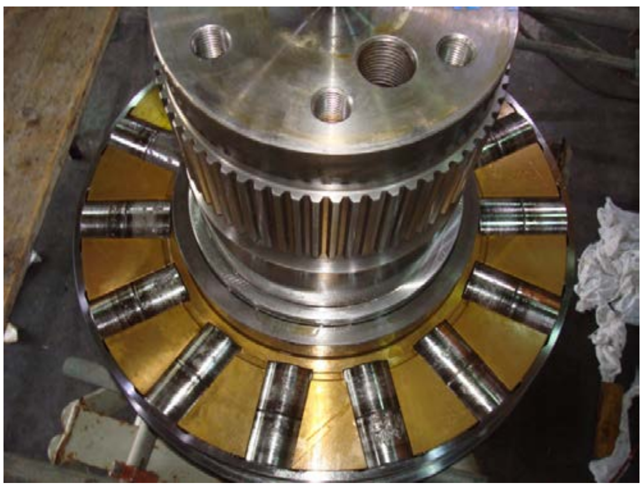

Figure 39a - Extruder Gearbox 2 Stages Tandem Bearing

Figure 39b - Extruder Gearbox 2 Stages Tandem Bearing Outer Race - Damaged Condition

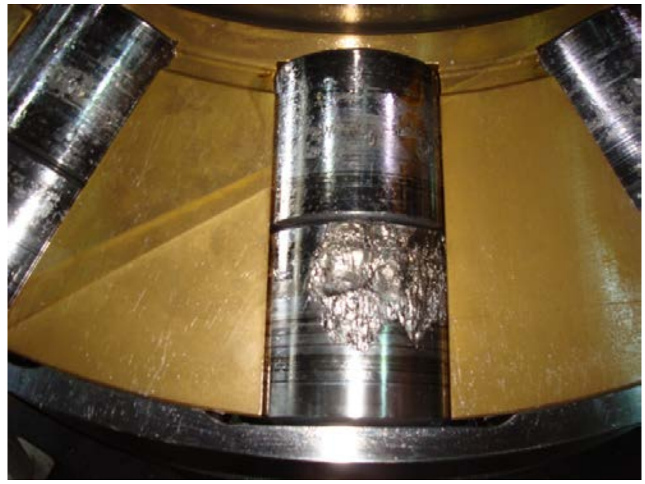

Figure 39c - Extruder Gearbox 2 Stages Tandem Bearing Roller - Damaged Condition

CONCLUSIONS

This paper briefed about currently available advanced analysis techniques for condition monitoring of large size gearboxes. The analysts must utilize the suitable techniques and implement in the appropriate condition monitoring programs of these gearbox or to any similar machines. The torque monitoring and analysis is an important technique in detection of large size gearbox fault that is to be considered in the monitoring program. Time synchronous averaging will be very useful method to separate out the discrete frequency from the complex waveform and spectrum.

For the large gearboxes or any machines there shall be multiple fault conditions may present in the same plot. The presence of energy level in the plots may be low and sometimes fault peaks buried in the noise floor itself. Distinguishing these fault condition is difficult task. The advanced monitoring techniques and signal processor, proper sensor selection and mounting, machine configurations, process parameter evaluations, correlation of vibration and process data shall improve the diagnostics capability of the machine. Since the condition monitoring techniques may vary from machine to machine, to have a success in diagnosis and effective analysis one must do proper condition evaluation and apply suitable condition monitoring techniques on the machines.

Every vibration analysts and engineers should properly understand evaluation of machine vibration severity and appearance of fault frequencies energy in the plots. This paper is trying to explain severity limit is not only based on the benchmark level and this limit shall vary machine to machine, sensor mounting locations, design and instrument configurations, etc. All standards also recommend the same, machine severity and condition evaluation is not only vibration limit specified in the standard it shall also from based on baseline data, machine characteristics, design and operating condition. So according to machine design and vibration severity; the condition monitoring and evaluation should be programmed.

NOMENCLATURE

K_t - Shaft Torsional Stiffness, Nm/rad
I_1 - Polar Inertia 1, Kg-m-sec^2
I_2 - Polar Inertia 2, Kg-m-sec^2
θ_1 - Angular Displacement of Inertia 1, Deg
θ_2 - Angular Displacement of Inertia 2, Deg
f_T - Torsional Frequency, Hz
N – Number, Power of 2
F_1 – Speed, RPS
A - Overall vibration level in the Band
Ai - Amplitude in the ith line of resolution in the Band
(i = 1) - The first line of resolution in the Band
(i = N) - The last line of resolution in the Band
N - Number of lines of resolution in the Band
W - Window Factor (W = 1.5 for a Hanning Window)
TSA – Time Synchronous Averaging
VRM – Vertical Roller Mill
GMF – Gear Meshing Frequency
OEM – Original Equipment Manufacturers

REFERENCES

Harris' Shock and Vibration Handbook – Author Cyril M. Harris and Allan G. Piersol, McGraw Hill - Firth Edition

Advanced Vibration Analysis Course Notes, Vibration Institute – USA

Fundamental of Signal Processing for Sound and Vibration Engineers, Author Kihong Shin – Andong National University, Republic of Korea and Josher K. Hammond – University of Southampton, UK, 2008, John Wiley & Sons Ltd, England

Practical Machinery Vibration Analysis & Predictive Maintenance – Author Cornelius Scheffer & Paresh Girdhar - First Edition 2004

Practical Implementation of Torsional Analysis and Field Measurement by Malcom E. Leader, P.E. – Applied Machinery Dynamics Co. and Ray D. Kelm, P.E. – Kelm Engineering, USA.

Analysis of Torsional Vibration in Rotating Machinery by J.C. Wachael & Fred R. Szenasi, Proceedings of 22nd Turbo Machinery Symposium

Torsional Vibration Analysis and Testing of Synchronous Motor Driven Turbo Machinery by Mark A. Corbo, Clifford P. Cook, Charles W. Yeiser, and Michael J. Costello, Proceedings of 31st Turbo Machinery Symposium, 2002

New Approach on To Torsional Vibration Monitoring by Lorenzo Naldi, Mateusz Golebiowski & Velerio Rossi, Proceedings of the 40th Turbo Machinery Symposium 2011, Houston, Texas

Time Domain Averaging with Order Tracking, Application Note by N. John Wismer, Bruel & Kjaaer, Denmark

Time Synchronous Averaging, Application Note – Feb 2009 by Crystal Instruments, Santa Clara, CA 95054

Cepstrum Analysis and gearbox fault diagnostics, Application Note – Second Edition by R.B. Randall, Bruel & Kjaer

Detecting Gear Tooth Cracks Using Cepstral Analysis In Gearbox Of Helicopters by Leila Nacib, Komi Midzodzi Pekpe and Saadi Sakhara, - International Journal of Advances in Engineering & Technology, Jan. 2013.

Vibration Monitoring: Envelope Signal Processing, Application Note Feb 2003 by Donald Howieson, Diagnostic Instruments, Inc, SKF Reliability Systems @ptitudeXchange San Diego, CA 92123, United States

Envelope Signal Processing, Application Note Feb 2006 by Tom Scott, Managing Director, Diagnostic Solutions Ltd, Rosset

Envelope Analysis for Diagnostics of Local Faults in Rolling Element Bearings - Application Note by Hans Konstantin-Hansen, Brüel & Kjær, Denmark

High Frequency Acceleration Enveloping by Using Labiew by Ilya Shulin, Purdue University, West Lafayette, IN 47907-2021

Signal Processing for Effective Vibration Analysis by Dennis H. Shreve IRD Mechanalysis, Inc – Columbus, Ohio, Nov 1995

Order Tracking with SKF @ptitude Observer, Application Note - Feb 2004, SKF Condition Monitoring Training Center, Lulea – Sweden

Rolling Element Bearing Analysis by Brian P. Graney & Ken Starry, Material Evaluation Technical Paper – Jan 2012, Vol 70, No.1 PP 78-85, ASME, INC

ISO 10816 - 1, Mechanical Evaluation of Machine Vibration by measurements on non-rotating parts – Guideline 1995

API - 613, Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Services, Feb 2003 Fifth Edition, Reaffirmed Aug 2007

General Motors/Delphi Vibration Standard for the Purchase of New and Rebuilt Machinery and Equipment - GM Specification V1.0a 1999 issued under the direction of the General Motors Corporation Vibration Standards Committee - GM-1761

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