GEARBOX FAULT DIAGNOSIS

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ABSTRACT

This paper presents cases of gearbox fault diagnosis in the field, and the methodology used in their diagnosis. The techniques used in the diagnosis include spectral analysis, time waveform analysis, envelope analysis, phase analysis, ODS analysis and resonance testing. The cases presented have a wide range including cases of misaligned gearboxes, casing distortion, backlash, wear, teeth overload and teeth resonance. The methods used in diagnosis and the corrective action are presented. These cases are from a wide range of industries including: cement, steel, and petrochemical industries.

INTRODUCTION

Diagnosis of gearbox fault problems in the field tend to be the most difficult problems for vibration analysts. There are a variety of reasons for this difficulty. In general, there is significant activity in a gearbox spectrum, that requires particular attention in monitoring, and in many cases heavy gearbox casings preclude the use of seismic sensors, which when used, can mislead the analyst. Moreover, a variety of faults occur in a gearbox, from installation faults, to wear, to overload, which require considerable knowledge both in the gearbox fundamentals and vibration analysis tools.

The techniques used for diagnosis range from spectral analysis, where particular frequencies are monitored (shaft speed frequencies, gear meshing frequencies, sidebands,… etc.), to time waveform analysis, where the waveform is evaluated for shape and form [1-5]. Special analysis tools such as envelope analysis, operational deflection shape (ODS) analysis and resonance testing may have to be used in special cases [6, 7].

This paper presents several case histories from a variety of industries, including: cement, steel, and petrochemical industries. The cases presented are all unique and required elaborate analysis. The techniques varied in the diagnosis of each of the cases. The above mentioned techniques were all employed in one or the other of the cases presented. It is quite instructive to follow the application of each technique in the relevant cases presented.

VIBRATION TECHNIQUES FOR GEARBOX ANALYSIS AND DIAGNOSIS

Spectral analysis is the core of the diagnosis of gearboxes. Spectral data are usually taken as velocity data, but also as acceleration data for high speed gearboxes and as displacement data for low speed gearboxes.

These spectral data should be measured on all bearings on the gearbox, in all three directions, horizontal, vertical and axial. Complete knowledge of the gearbox should be available to identify characteristic frequencies. The purpose of the spectral analysis is to identify the frequencies causing the gearbox to vibrate. If all vibration amplitude levels are within acceptable limits [8], then the gearbox would be accepted as normal. However, if any of the spectral components has a high amplitude, then spectral analysis is used to correlate the frequency of the high amplitude vibration to a gearbox frequency.

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The result of the spectral analysis of the high amplitude vibration is one of three cases: a) at a known frequency, b) at an unknown frequency, or c) at the running speed. By a known (unknown) frequency it is meant that the reason for the presence of this frequency in the spectrum is known (unknown). The first case is the easiest to analyze. If the high amplitude vibration is at a known frequency, then the problem is correlated to that known frequency. For example, if 2x or 2xGM (Gear Mesh) vibration is identified then this is usually correlated to misalignment. If decreasing harmonics of the running speed are present in the spectrum, then this spectrum shape is usually correlated with looseness in the bearings or the skid. If however, there were unknown frequencies in the spectrum, then additional testing would be required to determine the source of the unknown frequencies. Amongst the additional testing that may be required are: resonance testing [6] (including impact (bump) test and critical speed test), and modal testing. The purpose of the resonance testing is to correlate the unknown frequency to natural frequencies (stationary components) or critical speeds (rotating components) of the machine. Modal testing is a more advanced form of resonance testing, where all the modal characteristics of the machine are determined, including natural frequencies, damping ratios, and mode shapes. Modal testing is rarely used in the field, as it is an elaborate testing method, and is usually time consuming and costly. However, when justified, it can be a very powerful tool to obtain the machine characteristics and both identify clearly the unknown frequency in the spectrum, and suggest a solution to the problem.

The most difficult case occurs when the spectral analysis reveals high 1x or high GM vibration. There are many faults, related to installation problems, that lead to high 1x vibration. Amongst these faults are unbalance, misalignment, casing distortion, tilted foundation, skid leveling, piping strain and excessive bearing clearance. In this case, special vibration measurements have to be conducted on the machine to describe the nature of this 1x vibration, and to distinguish between the different 1x faults. These measurements include: time waveform measurement, phase measurement, and measurement of the ODS.

The time-waveform measurement can be used to distinguish between misalignment, piping strain and excessive bearing clearance. For piping strain, it will be quite clear that the forcing on the machine is directional, usually in the horizontal direction, and this directional force would be acting on the whole machine. Inappropriate bearing clearance also results in directional forces, however this would be localized at the bearing with the inappropriate clearance. This is particularly true for special geometry bearings, such as lemon bore or multi-lobe bearings.

The phase analysis is quite important to diagnose unbalance, misalignment, bent shaft and casing distortion. In many cases, misalignment (the main installation anomaly) manifests itself as vibration at 1x only. One of the best ways to distinguish between 1x vibration due to unbalance and 1x vibration due to misalignment is to measure phase across the coupling. If there is a 180° phase shift across the coupling, then the problem is misalignment [1]. If no phase shift occurs across the coupling, then the problem is unbalance. A bent shaft would produce a 180° phase shift in the axial direction across the machine (end-to-end), corrected for transducer orientation. Casing distortion can be easily identified by 180° phase shift across the machine (side –to-side or end-to-end) in the horizontal, vertical and/or axial directions. A cocked bearing can be identified by measuring phase around the bearing housing and noticing the phase shift due to the wobbly action of the cocked bearing [6]. In many cases a coupled time-waveform-phase analysis is quite useful in visualizing the vibration pattern and identifying the problem.

If the 1x vibration problem is still not solved after the time waveform and phase analysis, then an Operational Deflection Shape (ODS) should be measured. The ODS is useful in identifying problems of tilted foundation, skid leveling, skid looseness, and shaft parallelism. In the ODS measurement, phase-referenced 1x vibration is measured at grid points on the machine structure or skid. This reveals the actual deflection shape of the machine under the operating load, and at the operating speed. Note that the ODS is not a mode shape of the machine or structure, unless the machine is in resonance, but it can be considered as a summation of the contribution of all of the modes of vibration at the operating speed. ODS analysis can be quite useful in identifying installation problems, as it provides a visualization of the actual vibration pattern of the machine and /or skid. In particular, if a machine skid exhibits a node in its ODS, then this is a clear indication of a tilted foundation or a leveling problem in the skid. Accurate measurement of skid and/or foundation levels would then be required to confirm the results of the ODS analysis.

The presence of a high GM frequency in the spectrum does not in itself indicate a problem in the gearbox. However, a significant change of the GM, or the appearance of multiples of GM or sidebands, then this indicates a gear meshing problem, probably wear. However, many similar problems have these

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general symptoms. Narrow band analysis will be required to distinguish between various meshing problems. For example, backlash results in a family of sidebands (characteristic of frequency modulation) around the gear mesh frequency. An eccentric gear would result in sidebands to the GM at the frequency of the shaft speed of the eccentric gear. Excessive wear would, in general, result in the presence of both multiples of the GM frequency as well as sidebands. Common factor problems result in subharmonic multiples of the GM frequency, and can be high even for a new gearbox. However, this gearbox will have accelerated wear.

Recently, it had become evident that envelope analysis can be quite useful in detecting gearbox faults early on. Envelope analysis has been quite useful in detecting rolling element bearing problems. The advantage of envelope analysis is that it accentuates moderate frequency components that excite high frequency vibration. This results in clear identification of the gear mesh frequencies and sidebands in the envelope spectrum.

ENVELOPE ANALYSIS

The envelope analysis is based on filtering high frequency vibration, usually around the accelerometer resonant frequency, thus amplified. The premise is that this high frequency vibration is due to machine resonances caused by multiples of the moderate frequency vibration on the machine.

The envelope of the filtered high frequency vibration is processed [6]. The spectrum of the envelope is simply an accentuation of the moderate frequency vibration that would be otherwise obscured in a normal spectrum. This is particularly true of sidebands. The sidebands to a gearmesh frequency in an overloaded gearbox will appear clearly in an envelope spectrum well before they appear in a normal spectrum.

It should be noted that different instrument manufacturers have different ways to apply the enveloping technique. It is thus not immediately possible to compare envelope data from different instruments.

RESONANCE TESTING

In a resonance test, it is required to find any natural frequencies or critical speeds that may be excited by the machine. Usually an impact test is conducted on the machine to determine the natural frequencies of stationary parts, while a critical speed test is required to determine the natural frequencies of rotors.

To conduct an impact test [6], a mallet is used to strike the machine structure while stationary. This impact excites many modes on the structure. The vibration is measured during and immediately after the impact. A special set-up with a uniform window, small pre-trigger, and no averages is to be used. The measured spectrum would contain only the excited natural frequencies. In many cases a log or dB scale would be necessary to view the spectral peaks. It is recommended to repeat the impact test at several points to make sure that the measurement is not conducted at a node of one of the modes, and that there is enough modal activity at the measurement point.

OPERATIONAL DEFLECTION SHAPE (ODS) ANALYSIS

The operational deflection shape (ODS) measurement is an actual visualization of the machine behavior, at any frequency (but usually at the running speed), under its normal operating conditions. It is important to measure not only the amplitude of vibration, but also the phase at all points on the machine. This allows the visualization of the actual relative deflection of the machine at its operating condition.

To perform an ODS measurement, it is customary to divide the machine into a grid that characterizes the actual machine shape. Measurements are conducted at all points on the grid. It is best to use a computer program where this grid is plotted.

To conduct measurements on a machine grid using a single channel analyzer, a phase reference has to be used. In this case, it is best to use a photoelectric probe viewing a reflective tape mounted on the machine. A roving accelerometer is then used to measure all points on the grid. The measurement should be set-up to measure amplitude and phase at the running speed. The collected data are then transferred to the computer to be visualized. The relative deflection shape of the structure can be plotted and perceived problems identified.

To use a dual channel analyzer for measurement, it is not necessary to attach a phase reference. Rather by using two accelerometers on the grid, one as a reference sensor and the other as a roving sensor, and using the transfer function capabilities of a dual channel analyzer, it is possible to find the relative amplitude and phase for any point on the grid, relative to the reference point. The advantage of this method, is that the ODS need not be at the running speed, but at any frequency, measured on the transfer function. The data are then also transferred to the computer to be visualized and plotted. Perceived problems due to the actual relative deflection of the machine can be clearly identified.

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TOOTH RESONANCE PROBLEM: BUMP TEST

This case is from the cement industry. A cement mill had its Girth gear (which is a huge gear surrounding the mill body, and used to rotate the mill at a low speed (about 15 rpm)) set changed, and replaced by a new one from a different manufacturer. However, the machine exhibited high vibration (up to 14 mm/s rms), particularly at the pinion bearings. The cause of the vibration was unknown. Moreover, pitting on the Girth gear teeth started to occur (Fig. 1). Upon inspection of the spectrum (Fig. 2), it could be clearly seen that the gear mesh frequency at 67 Hz was beating with another unknown frequency at 65 Hz. This was confirmed from the beating pattern on the time waveform (Fig. 3). The cause of this unknown frequency that is beating with the gear meshing frequency and causing the pitting was unknown. Moreover, Fig. 2 shows also some high frequency activity at about 132 Hz, whose source is also unknown. Phase measurements at the pinion bearings tracking the gear mesh frequency indicated abnormal twisting of the pinion bearings. This was probably caused by the beating phenomenon.

Thus the diagnosis of the cause of the high vibration, on this cement mill, was due to beating between the gear mesh frequency and an unknown frequency. It was decided to investigate the cause of this unknown frequency by performing a resonance test. A bump test was performed on the pinion bearings and skid, on the mill body, and on the pinion and girth gear teeth. All showed resonant frequencies that were quite far from the high gear meshing frequency, except for the gear teeth resonant frequencies. Figure 4 shows the results of the bump test for the girth gear teeth, where resonant frequencies of 62 Hz and 129 Hz were evident. The resonant frequency of 62 Hz of the girth gear teeth was deemed to be the beating frequency with the gear mesh frequency, thus amplifying the vibration and causing the gear pitting. The resonant frequency at 129 Hz also appears to explain the vibration activity at 132 Hz. The difference of 3 Hz, between the sought after frequencies and the measured tooth resonance frequencies, can be explained by the fact that the resonance test was conducted at stand still and at no load. It is conceivable that the teeth natural frequencies at load and speed, for such a huge gear, should have some considerable stiffening both due to loading and due to the centrifugal force, even though it is operating at a low speed.

Thus the final diagnosis was that the gear-pitting problem is caused by resonances in the gear teeth at the gear meshing frequency. This causes a profound beating between the two frequencies, resulting in multiple impacts of the teeth during their meshing cycles, causing the pitting in the gear teeth. Based on the results of this analysis, the owner decided to replace the new gear set with the old worn gear set from the original equipment manufacturer (OEM), since they do not exhibit these teeth resonance problems, pending the arrival of a new gear set from the OEM.

Fig.1 Pitting of the Girth Gear

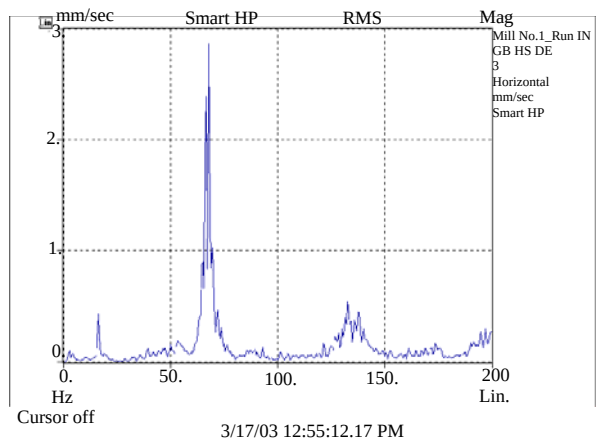

Fig. 2 Spectrum Showing beating with Gear Mesh Frequency

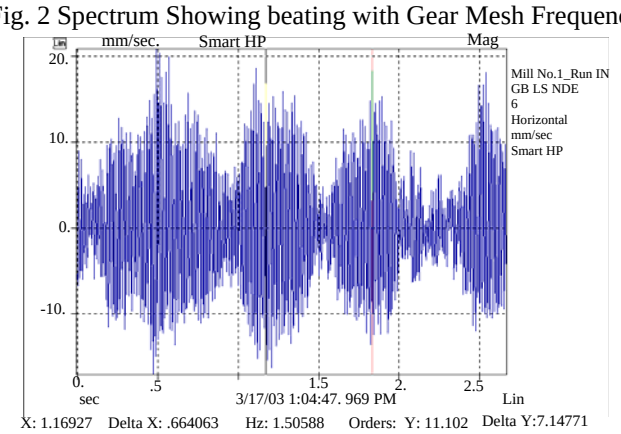

Fig. 3 Beating in Time Waveform

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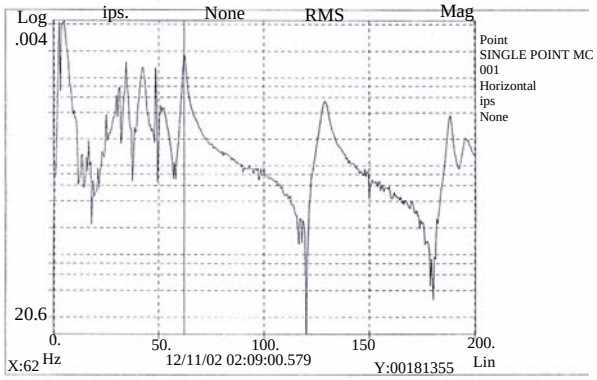

Fig. 4 Bump test showing resonant frequency of 62 Hz

GEARBOX VIBRATION DUE TO TOOTH LOADING: ENVELOPE ANALYSIS

This case is also from the cement industry. A new cement plant in Egypt has compactly designed gearboxes driving the cement mills. These mills are each driven by a 3900 KW electric motor through the gearboxes. The gearboxes are mounted on fluid film bearings. These gearboxes have several classes, 4 MW class, and 8 MW class, …etc. The particular gearboxes described here were 4 MW class. It was decided to monitor these critically loaded gearboxes early on, as part of a condition monitoring program. Due to the complex nature of this particular gearbox design, the monitoring program involved measuring several spectra on all gearbox bearings. One of the gear mesh frequencies monitored was at about 348 Hz, and was monitored monthly along with its multiples in a 1000 Hz band velocity spectrum. A typical spectrum is shown in Fig. 5, where the gear mesh (GM) frequency component is evident at about 3 mm/s rms. Over a period of three years, the GM remained constant in amplitude, and other than the appearance of seemingly benign sidebands of amplitudes of 0.2 mm/s (indicating some wear), the spectrum remained fairly normal.

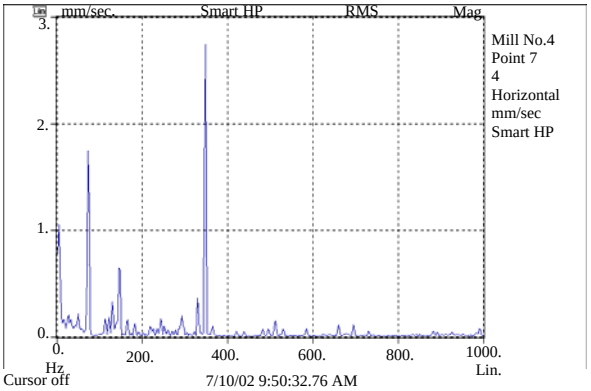

Fig. 5 Normal Spectrum on the Gearbox showing GM frequency

On a routine inspection of one of these gearboxes, an excessive wear pattern was visually identified. However, no indication of that excessive wear was evident neither in the monitored spectra, nor the time waveforms. This confused the monitoring program. If a condition monitoring program cannot detect such excessive wear patterns, it must be revised. To improve on the condition monitoring, it was decided to measure envelope spectra, which are routinely measured on machines with rolling element bearings, but not on machines mounted on fluid film bearings such as this gearbox.

A sample of the measured envelope spectra is shown in Fig. 6. It clearly shows the GM and its multiples, plus huge sidebands that are of the same order of magnitude as the GM. This was clearly abnormal. Machines on rolling element bearings would fail well before the sidebands reach the value of the fault frequency, even in an envelope spectrum.

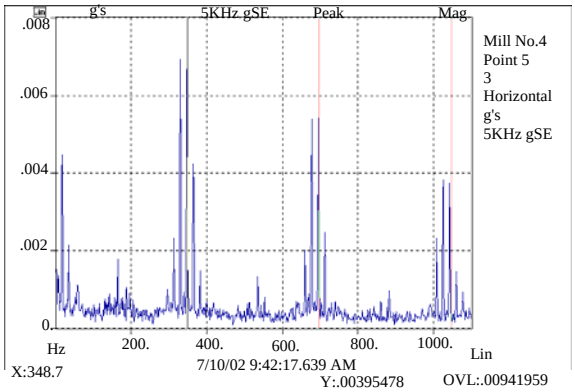

Fig. 6 Envelope Spectra showing huge sidebands to the GM frequency

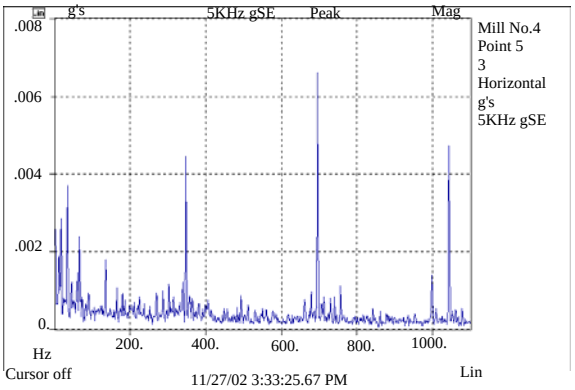

Fig. 7 Envelope Spectra after replacing the gear set showing no sidebands

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It was decided that the gear set was excessively worn and was replaced with a new set. The envelope spectrum of the new set is shown in Fig. 7, where the GM and its multiples appear, but no sidebands. Since then, envelope spectra are routinely measured on these gearboxes, as they are clearly more sensitive to the gears' condition than normal spectra. Any increase of the sidebands on the envelope spectrum is considered an early warning of wearing gears.

It is hypothesized that the reason the envelope spectrum is such an effective monitoring tool for these gearboxes, is due to the fact that since it is a high frequency measurement at the accelerometer's natural frequency, then it is possible to detect stress waves by the envelope spectrum. Since these gearboxes are heavily loaded (3900 KW for a 4 MW class gearbox), then it is expected that the heavy tooth loading generates stress waves that act as a carrier to the gear meshing frequency, which in turn is modulated by the gear running speed. The envelope spectrum detects these relatively low frequencies, and gives early warning of the modulation effects (thus the sidebands) well before a normal spectrum detects any change in condition. It is recommended that envelope spectra become an integral part of any gearbox monitoring program because of this capability.

MISALIGNMENT PROBLEM: TIME WAVEFORM ANALYSIS

This case is from general utility industry, and concerns a gearbox for a major chiller used for air conditioning in a major Saudi building. A group of chillers were newly installed, and had a proximity probe based monitoring system installed for these heavy machines that consisted of a motor-gearbox-compressor combination. All machines were mounted on fluid film bearings. The gearbox had a single stage with a gear meshing frequency of 4590 Hz, with motor speed at 30 Hz and compressor speed at 68.5 Hz.

All chiller systems performed very well, except one in which the gearbox consistently operated at its alarm level, as indicated from the proximity probe monitoring system. The manufacturer suggested increasing the alarm level. However, the owner decided to investigate the root cause of this problem.

A complete survey was performed on this particular unit. Time waveforms as well as spectra were measured on the casing at the bearing locations, by an accelerometer based system. Also, time waveforms and spectra were collected from the panel for the permanently mounted proximity probe monitoring system.

Fig. 8 shows a sample of the high frequency acceleration spectra, which has very low amplitudes, with no significant pattern. Moreover, the time waveform of the casing data also had no significant pattern. The spectra of the displacement panel data showed many multiples of the running speed, all at low amplitude. The key to diagnosing this problem was the time waveform of the panel displacement data, shown in Fig. 9.

Investigating the waveform in Fig. 9, it is clear that the shaft travels about 5 µm in the negative direction and about 25 µm in the positive direction. Moreover, the time waveform is spiked in the positive direction. It thus becomes clear that the gear is providing directional forces pushing the shaft to travel into the upper part of the bearing. This is a clear pattern of misalignment in gearboxes [5].

The misalignment was corrected at the next shut down. The machine has since then operated well below the alarm level. The key to diagnosing this problem was from the analysis of the time waveform shape, which revealed the directional forces caused by misalignment. Analysis of the time waveform is often overlooked in condition monitoring programs in favor of spectral monitoring. This case study clearly indicates the need to analyze time waveforms to diagnose gearbox problems.

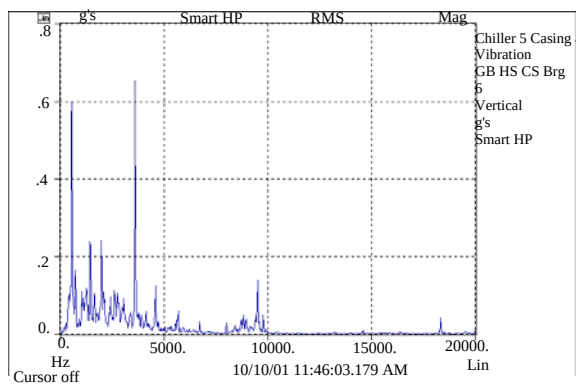

Fig. 8 Casing High Frequency Spectra

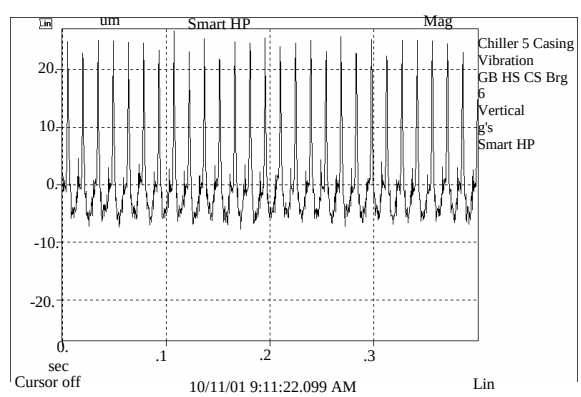

Fig. 9 Panel Displacement Time Waveform Indicating Misalignment Problem

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CRACKED TOOTH PROBLEM: CREST FACTOR

This case is from the steel industry. Baseline data were being collected for a brand new state-of-the-art steel plant in Egypt. Complete vibration data were being collected, including spectra, time waveform, envelope spectra and overall values. The vibration levels were generally well within limits established in international standards [8].

One of the intermediate rolling stands, which was driven by a 7 MW DC motor, had an output speed of 75 rpm (1.25 Hz), and was driven by a double reduction gearbox with gear mesh frequencies of 46.25 Hz and 30 Hz. Spectral measurement on the gearbox casing indicated very low amplitude levels at less than 0.3 mm/s, as shown in Fig. 10. However, on the inspection of the time waveform for the same point, which is on the high speed shaft motor side, the shape of the waveform indicated pulsations at the running speed of 1.25 Hz, as shown in Fig. 11. This is quite worrisome for such a brand new gearbox under warranty.

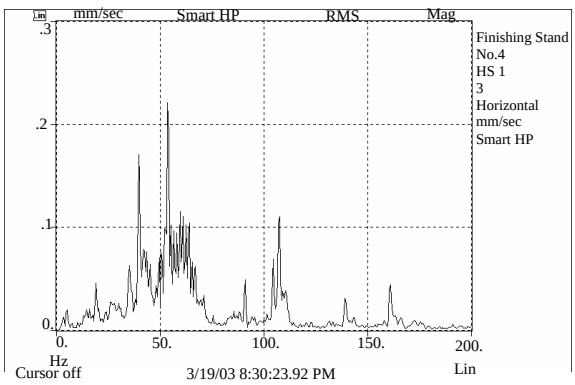

Fig. 10 Spectrum of Gearbox Vibration

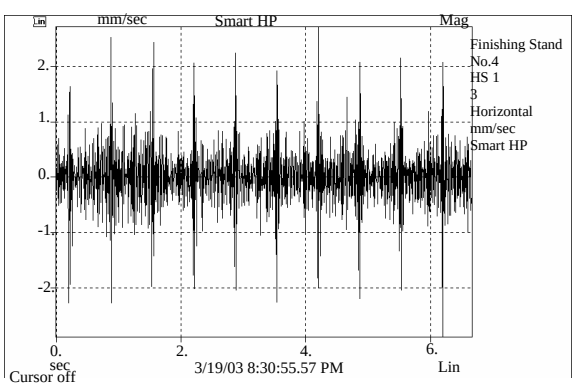

Fig. 11 Pulses in Time Waveform

The gearbox manufacturer first rejected the idea that there is a problem on this new gearbox just by investigating the time waveform shape, particularly that the ISO 10816-3 standard [8], would clearly indicate that this gearbox is in good condition. However, when presented with crest factor data (ratio of peak value to rms value), shown in Table 1 for points on the high speed and low speed shafts of the first stage of the gearbox, the manufacturer accepted that there is a problem with the gearbox.

The crest factor for a square wave is 1 and for a harmonic signal is 1.414. As any signal exhibits sharpness or spikiness, the crest factor increases. This would indicate that the peak value is much larger than the rms value. Having crest factors in the range of 2 to 4 is not uncommon for many sharp non-harmonic signals. Usually, crest factors above 5 would indicate a problem [9]. However, for this gearbox the crest factor approached 10 on some points, as shown in Table 1. This is a sure sign for a problem.

The reason for the pulsations is yet to be determined. However, the pulses at 1x in gearboxes are usually associated with gear tooth cracks. As the cracked tooth enters a loading cycle once per revolution, it produces a sharp peak, while the vibration energy, represented by the rms value, does not change much. The crest factor is thus a valuable tool for diagnosing tooth crack problems.

Table 1 Crest Factor Data for the Points on the Gearbox First Stage

Point	Point No.	Direction	Units	rms Value	Peak Value	Crest Factor (Peak/rms)
HS 1	3	Axial	mm/sec	0.245	1.36	5.6
HS 1	3	Horizontal	mm/sec	0.309	2.9	9.4
HS 1	3	Vertical	mm/sec	0.14	0.6	4.3
LS 1	5	Axial	mm/sec	0.166	1.64	9.9
LS 1	5	Horizontal	mm/sec	0.368	1.7	4.6
LS 1	5	Vertical	mm/sec	0.341	1.32	3.9

VIBRATION OF GEARBOX SUPPORTING STRUCTURE: ODS ANALYSIS

This case study is from the petrochemical industry, and is for the supporting frame of a vertical air cooling fan supported on a horizontal steel frame with a horizontal motor and a bevel gearbox (see figure 12). The unit is one of 16 units supported on an elevated steel frame, used to cool air for 3 Gas Turbines 9 MW each. The units were experiencing frequent shear failures of the shaft connecting the motor and the gearbox at the input shaft of the gearbox. The manufacturer suggested the use of a

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larger diameter shaft. However, this resulted in accelerated wear in the gearbox.

Upon investigation of the problem, a spectral analysis indicated that the main exciting frequency is at 6x (blade passing frequency). A bump test was conducted, but was not conclusive, because of transmitted vibration and it was not possible to shut down all coolers at the same time. An ODS was conducted at 6x, the exciting frequency, and the results are shown in Figures 13 and 14. It is seen quite clearly that the main deflection occurs in the middle of the frame, right where the repeated failure occurs, both in the horizontal and vertical directions. The reason of the failures is quite clear: the flexibility of the supporting structure at the gearbox input shaft position results in shaft failure. Increasing the shaft diameter only transmitted the problem to the gearbox, resulting in the accelerated wear. The root cause of the problem is the flexibility of the supporting structure. This supporting structure should be stiffened.

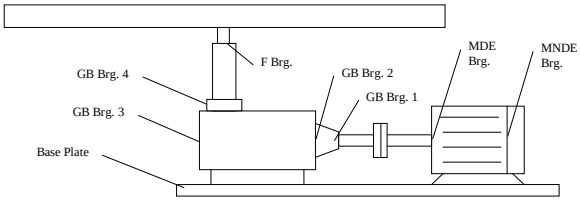

Figure 12 schematic layout of the machine and measurement points definition

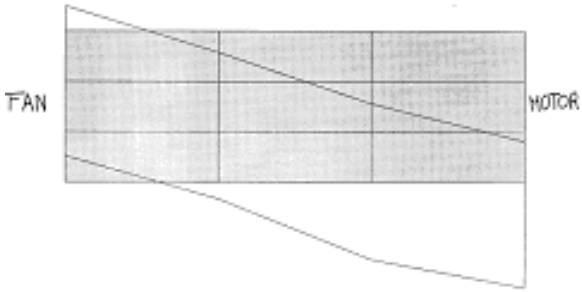

Figure 13 Base plate ods-horizontal direction

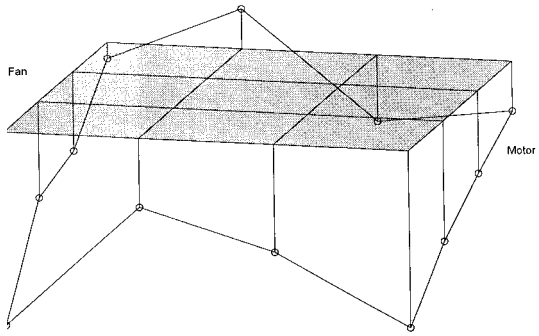

Figure 14 Base plate ods-vertical direction

CONCLUSION

This paper presented a variety of gearbox vibration problems from various industries. The problems represent a set of perhaps unusual cases, where a variety of vibration analysis tools had to be used. A bump test had to be used to diagnose a tooth resonance problem. Envelope analysis had to be used to diagnose a tooth loading problem. A time waveform analysis along with spectral analysis had to be used to diagnose a misalignment problem. A crest factor analysis was used along with the time waveform analysis to diagnose a cracked tooth problem. An ODS analysis revealed the root-cause of a recurrent gear box input shaft / gear wear problem. All these tools have to be used for effective gearbox analysis and diagnosis. It is hoped that this paper will provide some guidance and insight to practicing vibration engineers who have to diagnose gearbox problems.

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