LOW SPEED BEARING ANALYSIS

Bruce Weathersby
KOSA
P.O. Box 4, Mail Drop 1.6
Highway 70 W
Salisbury, NC 281144

ABSTRACT: Accurately measuring and interpreting vibration data on bearing problems in low speed applications (below 200 rpm) can be difficult. A wealth of information is published on diagnosing bearing problems in higher speed applications. Far less data exists with shaft speeds below 200 rpm. A case history of repetitive bearing failures with shaft turning speeds between 3 and 12 rpm is presented. A discussion of the pitfalls associated with collecting low speed vibration data, analyzing it properly, and using it to predict bearing condition is given. The technique used is conventional vibration analysis with the frequency spectrum taken from accelerometers and FFT analyzers/data collectors.

To demonstrate the problems with instrumentation in this low frequency area different FFT analyzers/data collectors were used to analyze low frequency sine wave signals produced from a signal generator. While adequate agreement exists when analyzing 300 cpm signals, the agreement was very poor at 30 cpm and deteriorated as signals approached 0 cpm. A discussion of the reasons for the problem is presented. Ways to avoid the problems associated with instrumentation are presented.

INTRODUCTION:

In 1998 several of our large slow speed reactors were experiencing bearing problems that resulted in unplanned outages. From historical data we were averaging 4 unplanned shutdowns per year because of it. We knew that as many as 4 of these reactors were in some phase of bearing degradation at any given time. Each unplanned outage caused between 5 and 7 days of lost production.

Although we routinely analyzed and predicted bearing distress in higher speed equipment, we had not been able to do so at the 3 to 12 rpm turning speeds of these reactor shafts. In 1998 we began an investigation into techniques that would help us predict bearing condition in these reactors. A technique called PeakVue® was examined and did predict some of the early failures. Ultimately we settled on conventional techniques as being the most informative for our particular application. This was true only after we identified several pitfalls (associated with instrumentation) that exist in this slow speed area and learned to work our way around them.

By 2000 we were able to more accurately assess the condition of each bearing and plan repairs to avoid production losses. In 1999 we also identified the design flaw causing these bearing failures. We have subsequently changed the equipment design. There have been no unplanned outages (due to bearing failure) since we learned to predict the beginnings of bearing degradation.

The large physical size of the reactor and bearings involved may be of importance so a thorough description is given for comparative purposes.

In today's world the data collector far outnumbers any other type of FFT instrument used to collect vibration data. It is most commonly used with an accelerometer. These devices have become so portable and powerful they have largely displaced the dedicated "real time analyzer" of just a few years ago. Actually the terms "data collector" and "real time analyzer" are rather vague and misleading. Later in the paper they are more accurately described. The important point here is to recognize that some data collectors/FFT analyzers have moderate to severe limitations in this low speed area, especially the closer the speed gets to 0 rpm. A discussion of analyzer performance is given while analyzing signals down to 3 cpm.

The nature of diminishing amplitude levels that describe bearing faults (in inches per second) as you approach 0 rpm is very important. Bearing defects, at low speeds, produce smaller amplitudes (in inches per second) than would be normal at higher speeds. A discussion with accompanying graphs is presented.

Accelerometer performance is also very important in this area. Their performance and positive solutions to the problems they present are covered.

REACTOR—BEARING DESCRIPTION:

The reactors are large horizontal heated shells with a rotating agitator shaft as shown in Figure 1. The reactor shaft is supported by bearings contained in the end frame of the reactor shell. The reactor mechanical shaft seals are also mounted in the end frames between the reactor support bearings and the reactor end walls. The bearings that were failing were NOT the reactor support bearings, but rather the bearings installed inside the mechanical seal. The reactor shaft is driven through a variable speed drive connected to a shaft supported gearbox.

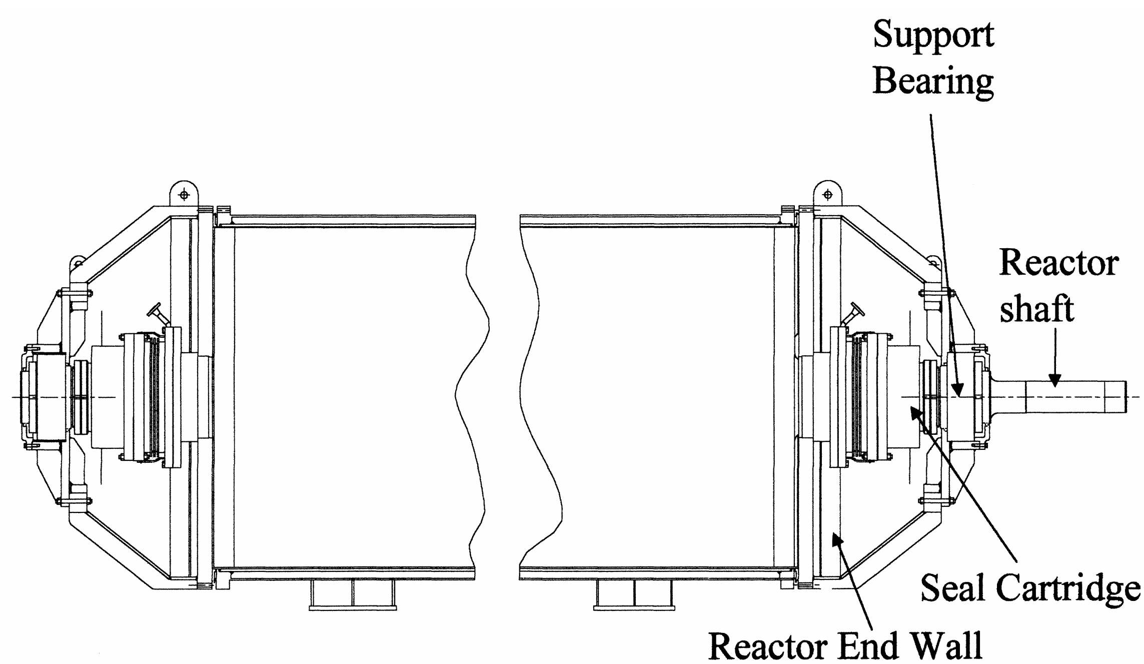


Figure 1. Reactor Outline Drawing Showing Seal Cartridge

When mechanical seals get to these large sizes (the shaft in the seal sleeve area is approximately 20”) they frequently are designed with their own bearings to insure that runout of the seal sleeve is kept to a minimum. Figure 2 shows the parts of each mechanical seal, the bearing that consistently failed, and the location of the permanently mounted accelerometers installed to get vibration data. As an example of size, each seal cartridge weighs approximately 3500 pounds.

Shaft agitator speed can be controlled between 3 and 12 rpm

IINSTRUMENTATION   CONCERNS:

There are many difficulties to overcome to be able to assess bearing condition accurately at these low speeds. I have listed them and will cover each in its own section. Only after identifying these can they either be understood or worked around. Many of these difficulties have to do with the actual transducers and instruments you use to gather data. You may be able to use your existing equipment to make good measurements OR you might need to acquire new instruments to obtain accurate data. Even after obtaining the data there are some difficulties present in interpreting it. Each is listed below:

1. Accelerometers
2. Units of measure
3. Integration
4. Analyzers
   A. AC or DC Coupled
   B. Band width
   C. Sine Wave Generator Test

To demonstrate these topics a signal generator was used to generate a low level amplitude, low frequency sine wave that was then analyzed by each of the data collectors/FFT analyzers. Graphs are presented to show how accurately different analyzers analyzed the known signal that was generated.

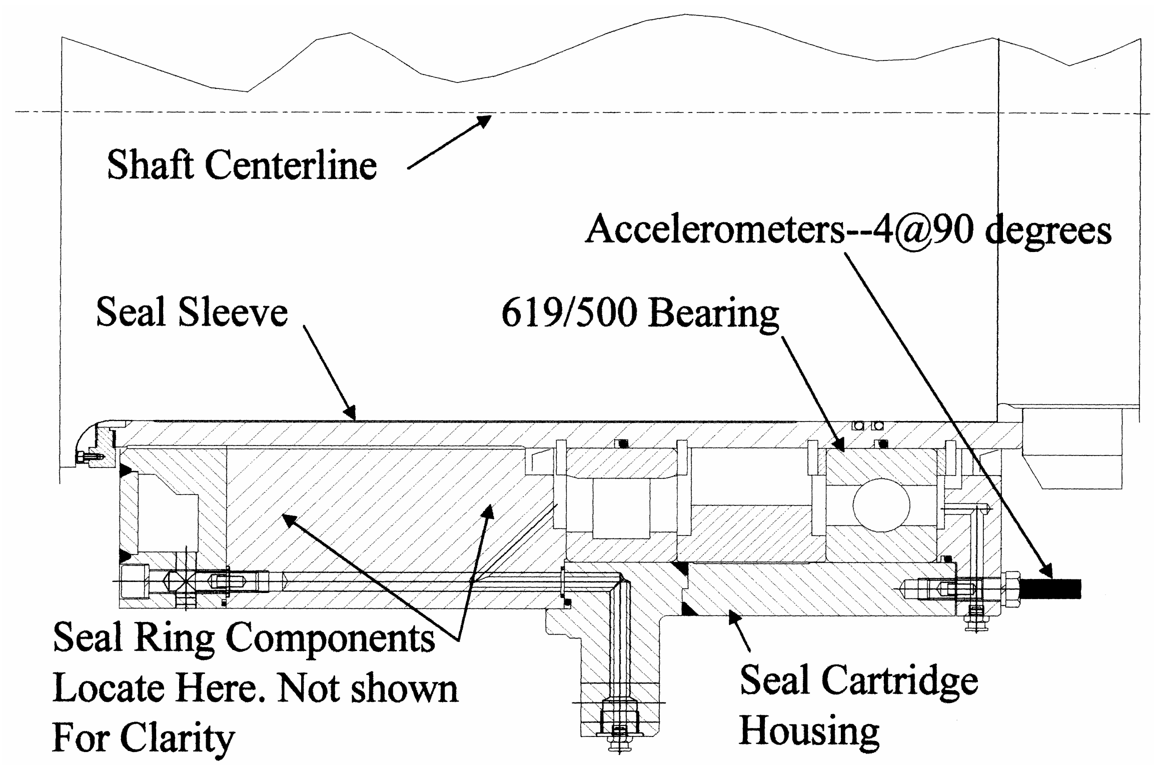


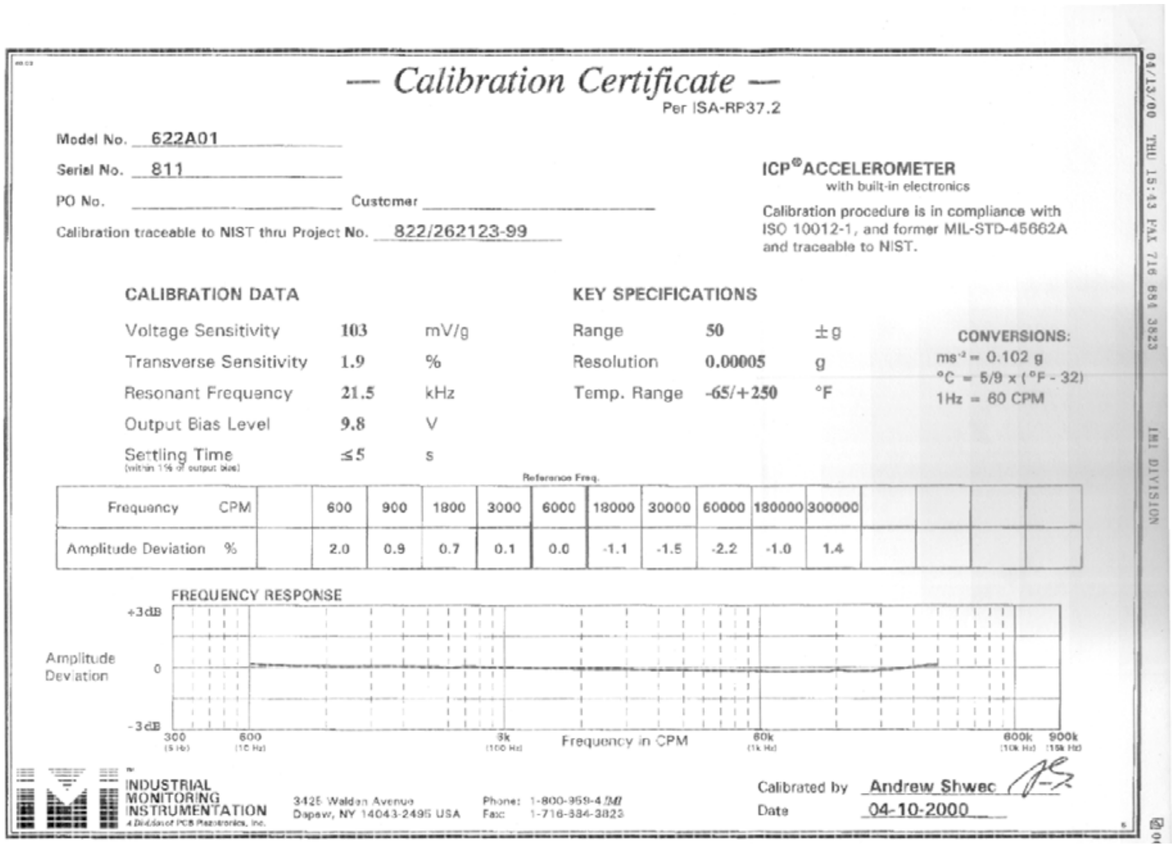
Figure 2. Seal Cartridge Section Drawing

ACCELEROMETERS:

At slow speed it can be very advantageous to consider the use of displacement measuring devices such as proximity probes or LVDT's. They measure displacement directly with no integration needed. Unfortunately, several characteristics of our application made their use not possible. Permanently mounting accelerometers on the seal housing proved to be the most advantageous way of collecting data. Accelerometers were mounted in in the axial direction (in line with the agitator shaft) as shown in Figure 2. Bolts that held the housing cover in place were removed, drilled, tapped, and faced. When reinstalled they provided a permanent way to mount the accelerometers close to the bearings. Cables were run outside the reactor end bell and readings were taken on a once per month basis.

Most data collectors are supplied with AC coupled general purpose accelerometers. This type has a "roll off" at low frequencies. Another way to say it is they under report the true amplitudes as the frequencies get closer to 0 cpm. By paying particular attention to their low frequency performance it is possible to select one that provides less than 5% error at 30 cpm. Manufacturers provide this information, however in many cases no information is provided for frequencies less than 600 cpm. Special requests have to be made to see their performance to as low as 30 cpm. Manufacturers may not provide data below 30 cpm because these accelerometers were never intended to work in this range.

A word of caution! The accelerometer that comes with your data collector may not give good performance at these low frequencies. Check it out! By careful review of specifications we selected a general purpose accelerometer that provided less than 5% error at frequencies of 30 cpm. ALL reactors were outfitted with this same accelerometer in order to keep differences in measurement to a minimum. Figure 3 shows the calibration certificate normally supplied with this accelerometer. Notice that no information is provided below 10 hz (600 rpm). By special request we obtained the data shown in Figure 4, which is an extended range calibration certificate. Notice that there is only 3 % error down to .5 hz (30 cpm). The manufacturer makes no claims about performance below 30 cpm.



— Calibration Certificate —
Per ISA-RP37.2

Model No. 622A01
Serial No. 811
PO No. Customer
Calibration traceable to NIST thru Project No. 822/262123-99

ICP® ACCELEROMETER
with built-in electronics
Calibration procedure is in compliance with ISO 10012-1, and former MIL-STD-45662A and traceable to NIST.

CALIBRATION DATA

Voltage Sensitivity	103	mV/g
Transverse Sensitivity	1.9	%
Resonant Frequency	21.5	kHz
Output Bias Level	9.8	V
Settling Time (within 1% of output bias)	≤5	s

KEY SPECIFICATIONS

Range	50	±g
Resolution	0.00005	g
Temp. Range	-65/+250	°F

CONVERSIONS:
ms⁻² = 0.102 g
°C = 5/9 x (°F - 32)
1Hz = 60 CPM

Reference Freq.

Frequency CPM		600	900	1800	3000	6000	18000	30000	60000	180000	300000				
Amplitude Deviation %		2.0	0.9	0.7	0.1	0.0	-1.1	-1.5	-2.2	-1.0	1.4				

FREQUENCY RESPONSE
Amplitude Deviation
+3dB
0
-3dB
300 (5 Hz)
600 (10 Hz)
6k (100 Hz)
Frequency in CPM
60k (1k Hz)
600k (10k Hz)
900k (15k Hz)

INDUSTRIAL MONITORING INSTRUMENTATION
A Division of PCB Piezotronics, Inc.
3425 Walden Avenue
Depew, NY 14043-2495 USA
Phone: 1-800-959-4IMI
Fax: 1-716-684-3823

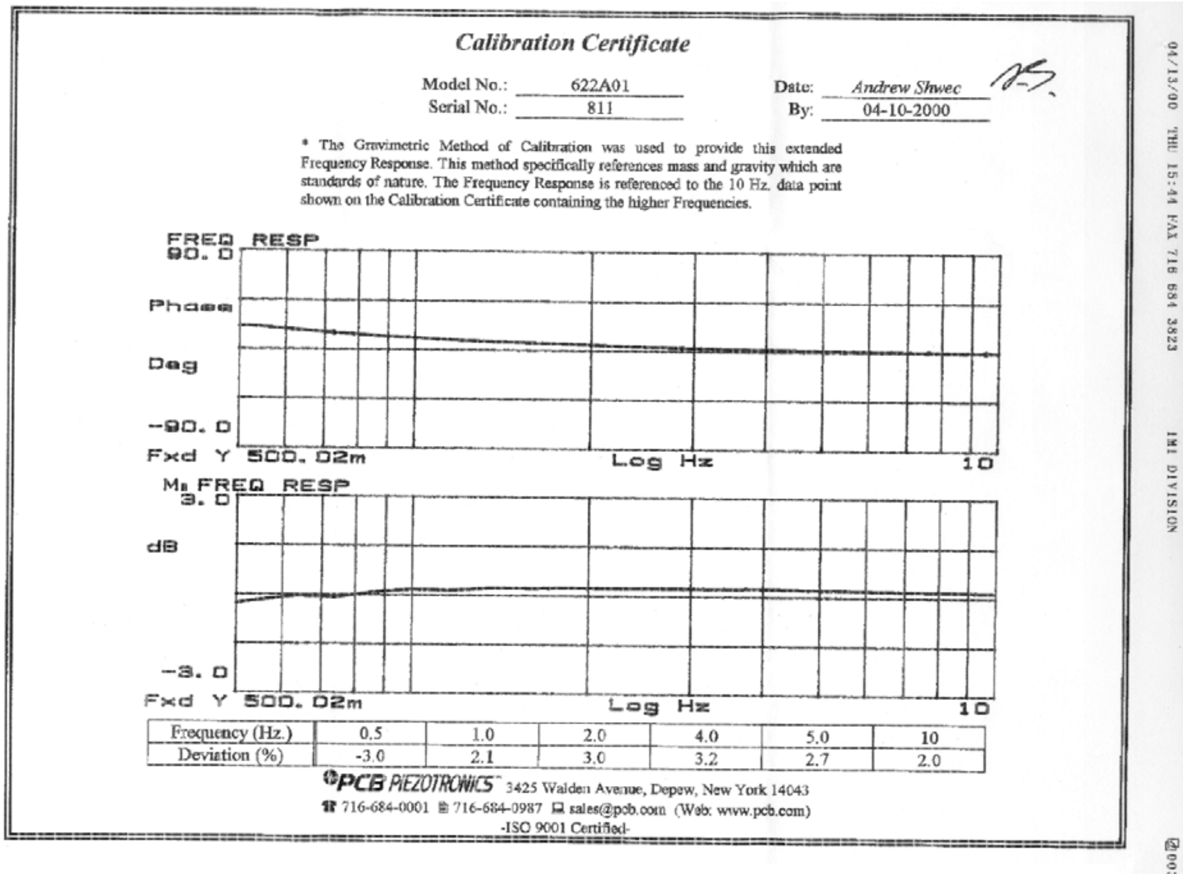
Calibrated by Andrew Shwec
Date 04-10-2000

04/13/00 THU 15:43 FAX 716 684 3823 IMI DIVISION

Figure 3. Calibration Certificate Normally Supplied

There is a wealth of information published on this subject and most accelerometer manufacturers can provide you with help as well as data on their product's performance. Recently several manufacturers have offered industrial style DC coupled accelerometers that perform almost perfectly down to 0 rpm. Be aware that DC coupled accelerometers do not do a good job in the higher frequency ranges.

Either of the ways described above should help you get good data at low frequencies and both are relatively inexpensive ways around the problems associated with low frequency accelerometer usage.



Calibration Certificate

Model No.: 622A01 Date: Andrew Shwec
Serial No.: 811 By: 04-10-2000

* The Gravimetric Method of Calibration was used to provide this extended Frequency Response. This method specifically references mass and gravity which are standards of nature. The Frequency Response is referenced to the 10 Hz. data point shown on the Calibration Certificate containing the higher Frequencies.

Frequency (Hz.)	0.5	1.0	2.0	4.0	5.0	10
Deviation (%)	-3.0	2.1	3.0	3.2	2.7	2.0

PCB PIEZOTRONICS 3425 Walden Avenue, Depew, New York 14043
716-684-0001 716-684-0987 sales@pcb.com (Web: www.pcb.com)
-ISO 9001 Certified-

04/13/00 THU 15:44 FAX 716 684 3823 IMI DIVISION 003

Figure 4. Extended Range Calibration Certificate

The 100 mv/g accelerometer selected provided a good signal to noise ratio and the actual frequencies generated in the bearings were reproduced nicely . If you have a problem with this you can also investigate accelerometers with 1000mv/g and even higher sensitivities. They can help with the signal to noise ratio problem. I can make no general statement that covers their performance at these low frequencies. You will have to check that for yourself.

Accelerometers measure acceleration (G's). You are probably much more familiar with using velocity (Inches per Second) or displacement (mils) at these low frequencies. To get to Velocity the signal from an accelerometer must be integrated once and to get to displacement it must be integrated twice. Although this integration can be done in the accelerometer package itself, it is much more common to do it in the vibration analyzer or data collector. Integration in all analyzers is not done the same way and can lead to a significant source of error, in (you guessed it) the low frequency range. Once again, the closer to 0 cpm the more severe the problem.

UNITS OF MEASURE:

As mentioned earlier, it may be better to look at displacement (mils) at slow speed. Later we will describe reasons why this may not be good if you are using an accelerometer to obtain your data. If you decide to use velocity (usually Inches per second –peak) or acceleration (G peak or rms) readings you must be very careful in the amplitudes that you consider as significant. Defective bearings at low shaft turning speeds give lower amplitudes of bearing defect frequencies than they do at higher shaft turning speeds.

There are many published graphs of displacement, velocity and acceleration compared to frequency. Most of them share one thing in common. They are VERY BUSY. A lot of data is contained in them. Sometimes there is so much you begin to loose the importance of the relationship between amplitude and frequency. For a simple explanation I have used the table in Figure 5. I picked a common bearing defect frequency of a BPFO (Ball pass frequency outer race) at 100hz (6000 cpm) at an amplitude of .06 Inches per Second Peak. Note that it would be .2 mils displacement. Most analysts would be very concerned with this amount or would already have removed the bearing from service. For purposes in the table I took the .2 mils displacement created and looked to see what value of velocity and acceleration it would produce if it occurred at continually lower frequencies.

CPM	**DISPLACEMENT (Mils Pk-Pk)**	**VELOCITY (IPS Peak)**	**G's (RMS)**
6000	.2	.062	.072
3000	.2	.031	.018
600	.2	.0063	.00073
300	.2	.0031	.00018
30	.2	.0003	.0000018

Figure 5. Velocity and Acceleration Levels of a .2 Mil Displacement at Differing Frequencies

From Figure 5 you see that at 30 cpm a BPFO signal of .2 mils amplitude only creates .0003 IPS. I am NOT advocating removing bearings at .0003 IPS at these low speeds, but the data in Figure 5 does show you are going to have to start thinking differently about what amplitudes you can dismiss as insignificant, and what you should consider important at these low turning speeds

INTEGRATION:

Integration can be done by analog or digital methods. The actual hardware to do it and how each manufacturer does it is outside the scope of this paper. It is not done the same way by each manufacturer and in some cases can introduce a SIGNIFICANT error in the data displayed.

Whether it is done by analog or digital as you get very close to 0 cpm EVERYBODY has a problem with it. The last few filter bins close to 0 rpm give the ski slope effect covered in other articles under integration. In fact, some manufacturers do not even report the findings of the lowest filter bins when integrating, they just artificially set them to 0 amplitude. Other analyzer manufacturers allow you to set them to 0 yourself and allow you to pick how many you want to change. This obviously is not a good tactic if your predominant frequency of interest is in one of these filter bins. At any rate if your signal of interest is in the first 4 or 5 filter bins of the spectrum you had better find out just how your manufacturer handles it. Either that, or stay away from integration.

An in-depth discussion of signal integration is covered in references 1 and 2.

ANALYZERS:

Earlier we referred to “data collector” and “real time analyzers”. Now its time to set the record straight. The basic problem here is not directly tied to these terms. Rather it has to do with HOW the analyzer processes the data. IF the analyzer uses AC coupling then it will have to deal with the low frequency amplitude reporting problems described earlier. Since most, if not all, data collectors are presently AC coupled, you can see the problem. There is a category of analyzers that are changeable between AC coupled and DC coupled that perform well down to DC or 0 rpm. It takes a few more pieces of hardware to be DC coupled, which require room and WEIGHT (the dreaded enemies of data collectors, which are famous for their portability and conveniently small size). From here on we will refer to analyzers as AC coupled or DC coupled. Remember that
some of the “real time analyzers” we referred to earlier are switch able between AC or DC coupled.

ANALYZER BAND WIDTH:

Another important aspect of taking good measurements in this low speed range is the bandwidth of the analyzer. The maximum frequency selected for analysis (Fmax) should be fairly low since we are really only interested in what happens at low frequencies. The Fmax selected, divided by the # of lines of resolution, will give the LRF (lowest resolvable frequency). The bandwidth is the LRF time the Window factor. The window factor for the Hanning window I used is 1.5. In general, selecting an Fmax of 10 hz (600 cpm) is adequate. If you use 800 lines of resolution then the LRF is 600/800 or .75cpm. The only drawback here is the time it takes to acquire the data. For only one average the acquisition time is 800lines / 10hz which gives 80 seconds to acquire the data. This is the amount of acquisition time for each average taken. You should probably take at least 2 or 3 averages. Overlap processing can shorten the time required to acquire the subsequent averages after the first one is taken.

If you are planning on getting the exact multiplier of the BPFO (Ball Pass frequency Outer race) for instance, you had better have very accurate turning speed information. Instead of relying on the analyzer to find this speed it is a good idea to confirm it by separate means to two or three decimal places. If the shaft is visible you can easily use a good stopwatch to get this turning speed to the second or 3rd decimal place accuracy you will need. Of course this assumes that your shaft speed is constant. In some cases it will not be. Since my shaft speeds were very constant this was not a problem. If your speeds vary you may try to look at orders of turning speed instead of cpm or rpm.

Each analyzer also does overall vibration values differently. I found them to be so varying as to be almost useless. Trending the overall value is normally a very useful tool, but I would be VARY WARY of using the overall as a guide at these low values. In the analyzers that I used the value of the first filter bin was sometimes abnormally high and was not a REAL indicator of vibration taking place. Most overall values would count it even though you know it is wrong. For that reason I would not use Overall level as a tool to trend (with the analyzer set up mentioned above) unless I was ABSOLUTELY sure it was reporting correctly

Finally, you need to be very wary of accepting the values in the first few filter bins as accurate. This is precisely why the graphs and data shown in Figures 6 through 11 stop at 3 cpm. With my analyzer setups, 3 cpm was in at least the 3rd filter bin in each measurement. Integration and other problems often lead to wrong values in these first few filter bins of the spectrum. As mentioned elsewhere in this paper, some manufacturers choose to purposely null the values in the first few filter bins. Others even let you select how many to null. IF you are interested in these values you need to find out how your analyzer manufacturer handles it.

SINE WAVE GENERATOR TEST:

In order to see just how several data collector/FFT analyzers performed in the low speed area a known signal at these low frequencies was fed into each one and analyzed. The signal was a sine wave from a Wavetek Model 195 Waveform Generator. The voltage from the generator was 50mv peak to peak. The signal was attenuated so that the actual voltage into each analyzer was .001Volt RMS. Each analyzer was calibrated for an accelerometer at 1000mv/g. This amplitude value was kept constant and the frequency of generation was varied from 5 hz down to .05 hz in increments. If analyzed correctly each data collector/FFT analyzer should show a .001G RMS level at each frequency of testing.

The results in Figures 6 and 7 are for a G's RMS reading from each analyzer. The results in Figures 8 and 9 are for the same .001G RMS level but are integrated into Velocity IPS Peak levels. Figures 10 and 11 show the results of the .001G RMS signal when double integrated to MILS Peak to Peak. Some analyzers have "corrected" values as well as different AC filters and DC coupled readings.

Clearly at the 5 hz (300 cpm) level most analyzers do a reasonable job of reporting correctly, even when integrated. The results change drastically the closer the signal of interest gets to 0 cpm. These results show conclusively that different analyzers report different results, many of them with large margins of error as the signals get closer to 0 cpm.

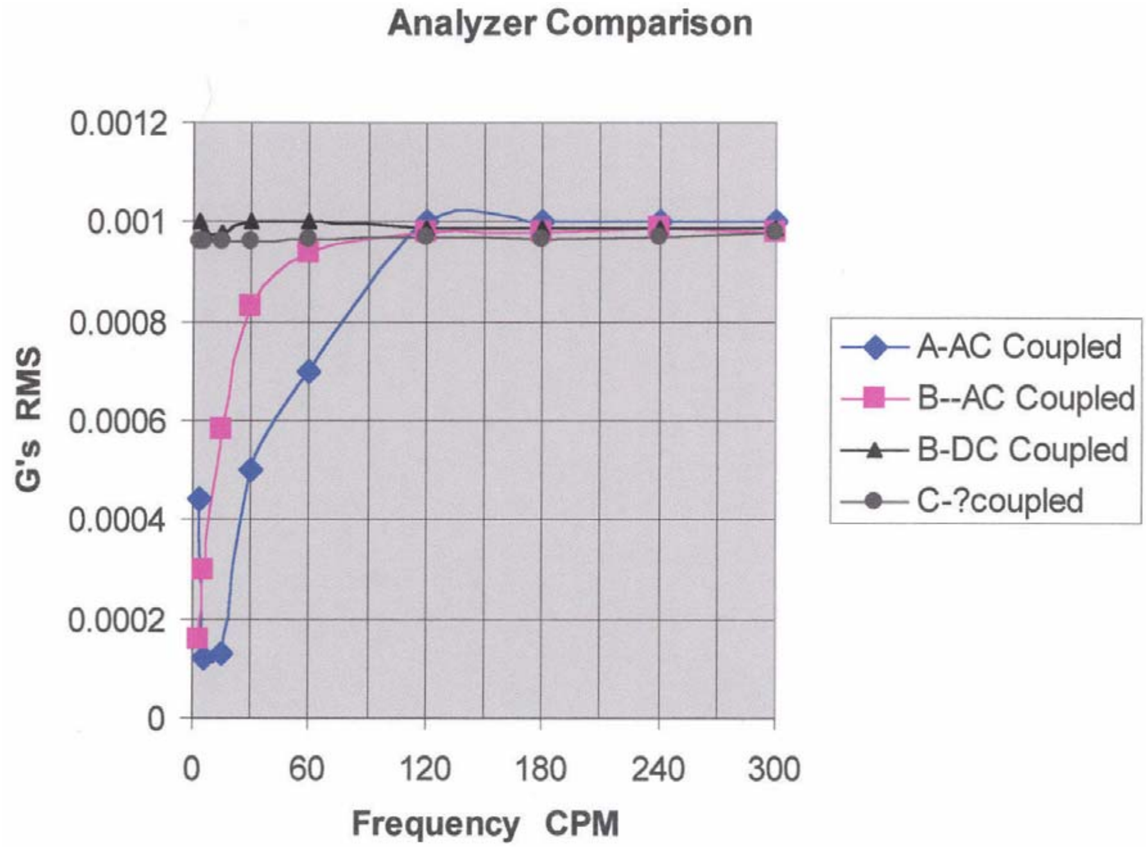


Figure 6. Results from Analyzer Brands A, B and C with .001 G RMS Input Signal at Different Frequencies

CPM	G's RMS Input	Brand A AC Coupled G RMS	Brand B AC Coupled G RMS	Brand B DC Coupled G RMS	Brand C G RMS
300	.001	.001	.00098	.00099	.00098
240	.001	.001	.00099	.00099	.00097
180	.001	.001	.00098	.00099	.00097
120	.001	.001	.00098	.00099	.00097
60	.001	.0007	.00094	.001	.00097
30	.001	.0005	.00083	.001	.00096
15	.001	.00013	.00058	.00098	.00096
6	.001	.00012	.00030	.00098	.00096
3	.001	.00044	.00016	.001	.00096

Figure 7. Data for Graph in Figure 6

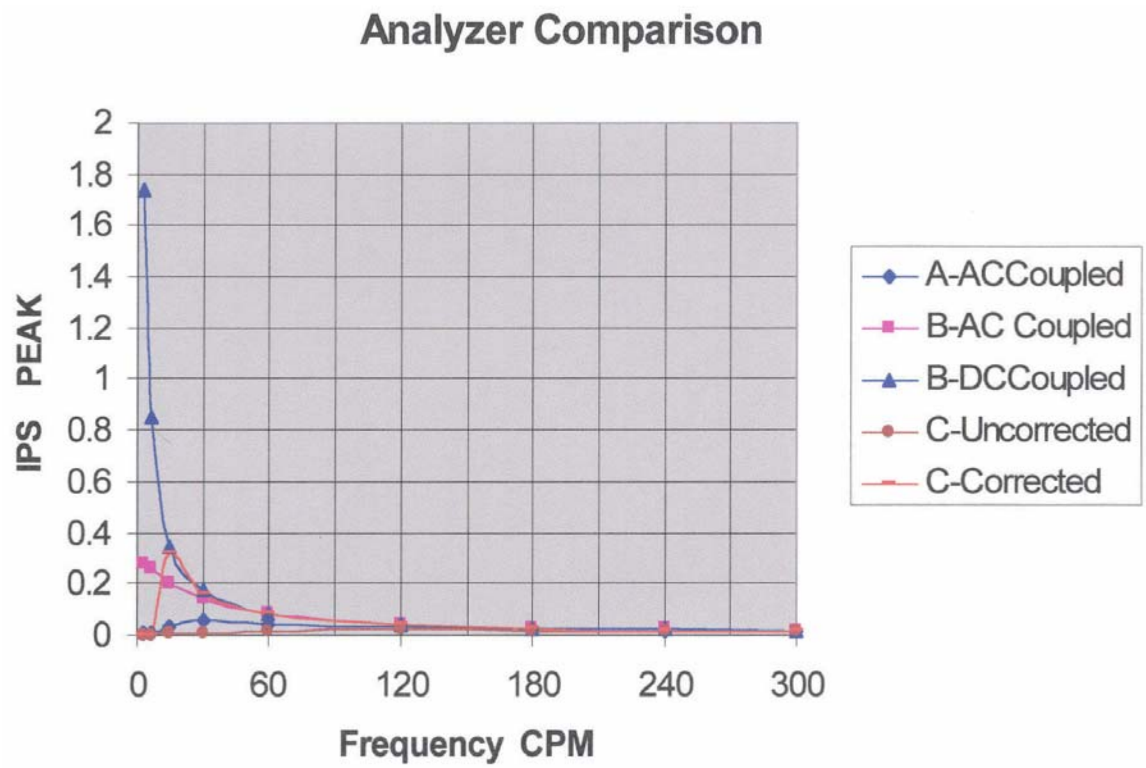


Figure 8. Results, Integrated to Velocity, from Analyzer Brands A, B and C with .001 G RMS Input Signal

CPM	G's RMS Input	Brand A AC Coupled IPS Peak	Brand B AC Coupled IPS Peak	Brand B DC Coupled IPS Peak	Brand C Uncorrected IPS Peak	Brand C Corrected IPS Peak
300	.001	.016	.0171	.0172	.015	.017
240	.001	.019	.0215	.0215	.0173	.021
180	.001	.027	.0285	.0286	.0199	.0279
120	.001	.036	.0424	.0429	.02227	.0416
60	.001	.046	.0814	.087	.01827	.08156
30	.001	.061	.144	.1742	.011	.1631
15	.001	.035	.2015	.342	.00584	.3247
6	.001	.0084	.2601	.8506	.00236	.00236
3	.001	.0073	.2774	1.738	.00119	.00119

Figure 9. Data for Graph in Figure 8

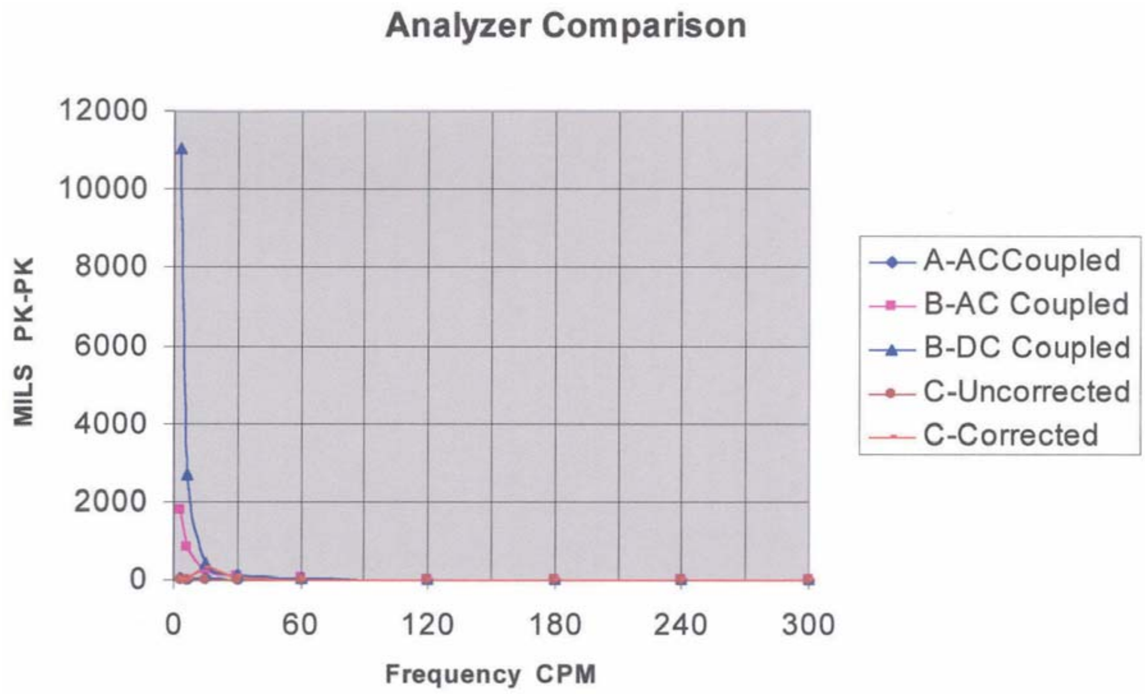


Figure 10. Results, Double Integrated to Displacement, from Analyzer Brands A, B and C With .001 G RMS Input Signal at Different Frequencies

CPM	G's RMS Input	Brand A AC Coupled Mils Pk-Pk	Brand B AC Coupled Mils Pk-Pk	Brand B DC Coupled Mils Pk-Pk	Brand C Uncorrected Mils Pk-Pk	Brand C Corrected Mils Pk-Pk
300	.001	.51	1.089	1.094	.6789	.9622
240	.001	.783	1.711	1.714	.8455	1.425
180	.001	1.45	3.019	3.033	1.019	2.391
120	.001	2.89	6.744	6.824	1.115	5.05
60	.001	7.345	25.92	27.69	.6789	19.24
30	.001	19.432	91.66	110.9	.2333	75.26
15	.001	22.903	256.6	435.4	.06419	301
6	.001	14.3	828	2708	.0107	.0107
3	.001	31.1	1766	110675	.0027	.0027

Figure 11. Data for Graph in Figure 10

These results do not even account for any errors that an accelerometer might throw in. They really show just how well each analyzer does its job. You can see that coming up with any kind of acceptance standard for bearing defect amplitude would have to include the information of WHICH analyzer was used, just to make sure the amplitudes were correct.

In general, above 300 cpm most analyzers agree well so a standard for bearing defect amplitude can be done irrespective of the analyzer that took the data. This situation does NOT exist below 300 cpm since the value given can be so much in error just because of the analyzer/accelerometer combination used to take the data.

For reference, the analyzer described as B-DC coupled does the best job in Figures 6 through 11. It gives results that are close to perfect in all Figures.

CASE HISTORY

An inherent design flaw insured that some of these bearings would experience failure. In the reactors where they failed they lasted from 27 to 36 months before failure occurred. In this application FAILURE was defined as when the mechanical seal started leaking. In many cases the bearings would be considered ANNIHIALATED or completely DESTROYED at this point.

Due to internal loads the failure was always in the axial or thrust carrying direction within each of these bearings. Knowing this prompted me to install accelerometers in the axial direction of the seal cartridge as close to the failing bearing as possible. Due to the large size of the seal we had no idea if a bearing fault at one location of the outer race would be seen in the full 360 degrees or only at the point of origin of the fault. By not knowing we took the conservative approach and installed 4 accelerometers (at 90 degree spacing) around the circumference of each seal, all pointing in the axial direction as shown in Figure 2. The hot enclosed environment of the seal housing dictated that permanent mounting of accelerometers was the best way to get the data.

The bearings in question were 619/500 deep groove ball bearings. They were supplied from SKF and FAG. Bearing defect frequencies are shown in Figure 12.

Bearing Type	FTF	BSF	BPFO	BPFI
SKF 619/500	.457	5.714	9.132	10.868
FAG 619/500	.458	5.928	9.621	11.379

Figure 12. Bearing Defect Frequencies (Multipliers of Shaft Turning Speed)

Readings were taken once per month and nothing definitive was uncovered for a 6 month period. Many of the problems described earlier got together to “mask” the bearing frequencies that were already being generated. The first interesting observation that finally led to the solution was in comparing readings taken with two different types of analyzers. While reading the same accelerometers that were permanently attached to the reactor we noticed a BIG difference in the spectrums each type collector produced. The amplitudes at any frequency above 300 cpm were virtually identical. Below 300 cpm they differed significantly. We began our investigations into the WHY of this which led to much of the information published under the ANALYZER section of this paper.

Another very interesting aspect of the problem unfolded over the 3 years we have been taking data. In several cases we have monitored bearings that took over 18 months to progress from initial spalling to advanced failure with race cracking of the outer race. Even at this point the mechanical seal had not started leaking yet. Rather the seal was removed from service at an opportune time to avoid production losses.

The failure mechanism of bearings in these reactors (due to the slow speed) is different than what I am used to at speeds above 1200 rpm. At the higher speeds I have not usually seen cracked rollers and cracked races. Normally the spalling that starts at high speeds produces frictional heat within the bearing that leads to a temperature related failure of the rollers or races. This shows up in "smearing" of the surfaces or discoloration. Taken to extremes it leads to seizure of the elements to one of the races or the races to the housing or shaft. In the slow speed failure of these reactors (below 12 rpm) although the failure starts as spalling of the outer race there is no significant temperature due to friction of the rolling elements over the spalling. After the race spalling comes race cracking, then roller cracking and foreign damage debris of the races. In the most extreme cases we saw EACH ball cracked in half and the races in hundreds of cracked pieces.

After being able to detect bearing damage with conventional vibration techniques we have been able to spot the original onset of spalling. Figure 14 shows the outer race of a bearing from a seal removed in its early stage of spalling. Figure 13 is the spectrum used to make the decision to remove the seal from service. Figure 15 is the last spectrum taken before removal from service. Figure 16 shows a bearing removed with more advanced spalling. Figure 17 showed a significant increase at the 1XBPFO from the previous month and was predicted as the progression from spalling to race cracking. Figure 17 is the last spectrum taken before removing the seal from service. Figure 18 shows the spalling and cracked race of the bearing in the seal analyzed in Figure 17.

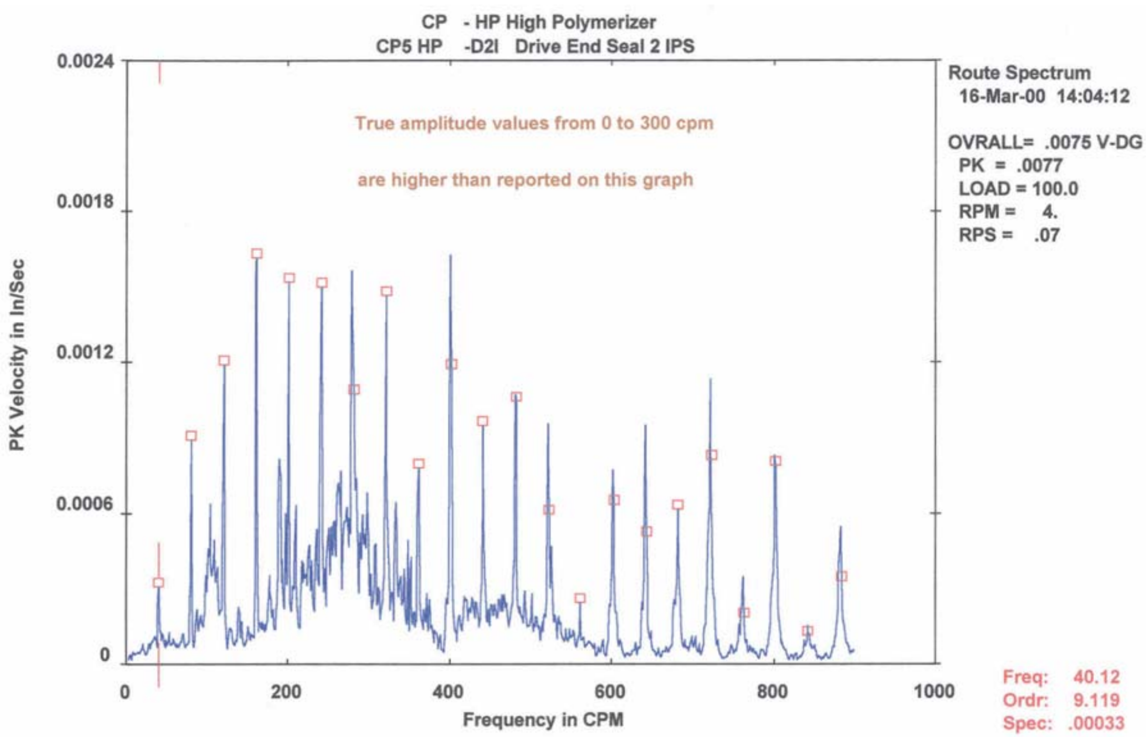


Figure 13. Last Spectrum Taken Before Bearing in Figure 14 was Removed from Service. Shaft Turning Speed of 4.4 RPM

Figure 14. Beginning of Spalling of Bearing Described in Figure 13. Spalling Over 5 Degrees of Arc on Outer Race

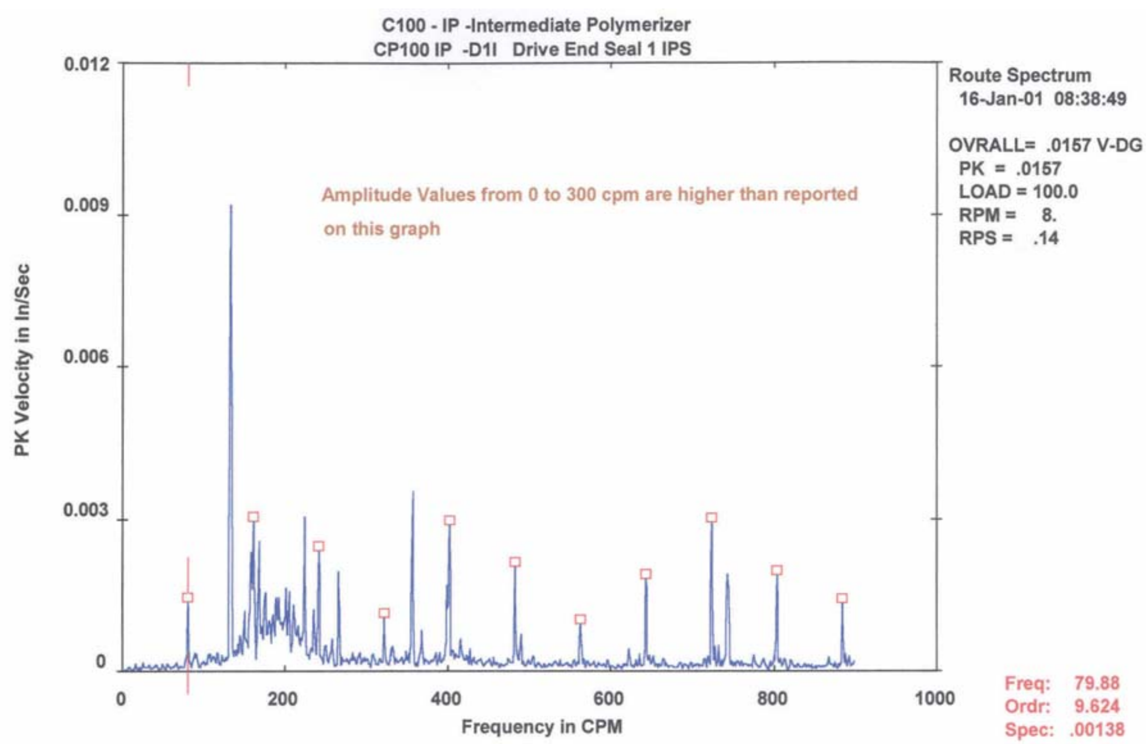


Figure 15. Last Spectrum Taken Before Bearing in Figure 16 was Removed from Service. Shaft Turning Speed of 8.3 RPM

Figure 16. Advanced Spalling of Bearing Described in Figure 15.
Spalling Over 45 Degree Arc of Outer Race

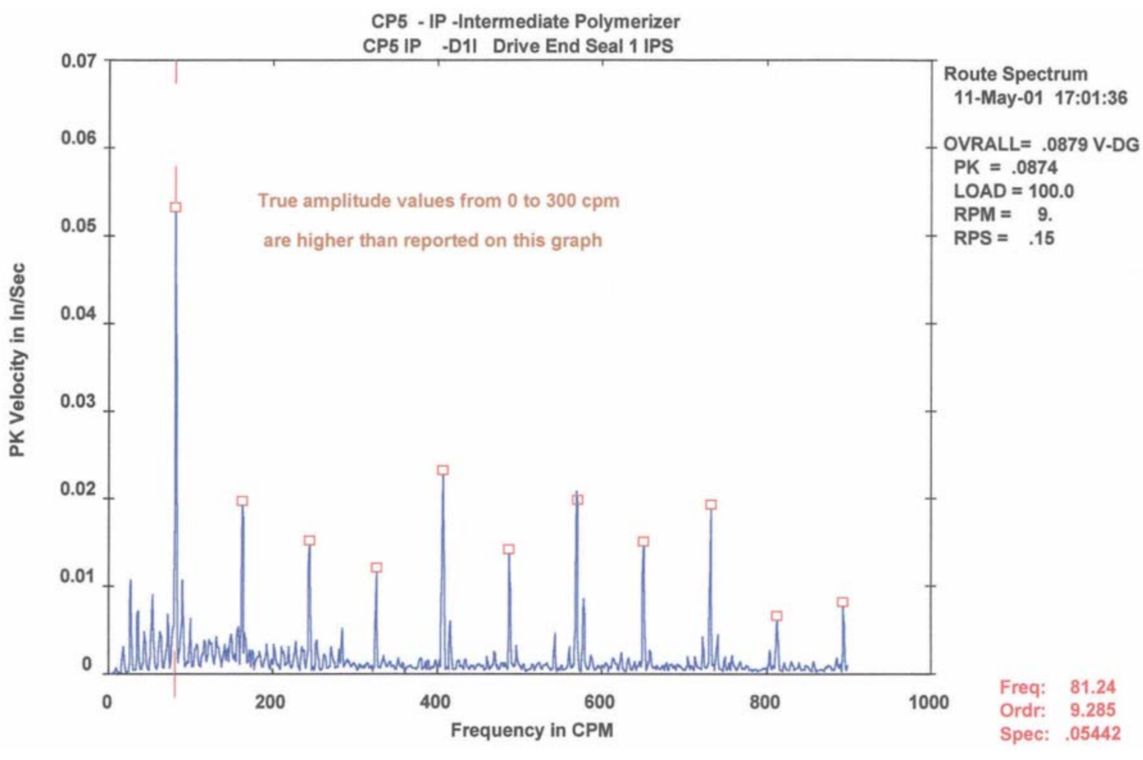


Figure 17. Last Spectrum Taken Before Bearing in Figure 18 was Removed from Service.
Shaft Turning Speed of 8.75 RPM

Figure 18. Bearing Removed from Service Referenced in Figure 16.
Outer Race Spalling Over Full 360 Degree Arc. Outer and Inner Races Cracked.
Two (2) Balls Cracked

Another lesson learned involves the use of 4 accelerometers on each seal. Normally it is fair to say that whatever bearing frequencies appear are viewable on at least 2 (and sometimes all) of the four accelerometers regardless of where the spall starts. In several cases though only the transducer nearest the spalling gives an indication. In one particular case all 4 accelerometers showed the bearing frequencies for months on a particular failure YET when the crack of the race occurred (later verified as a crack) only the transducer nearest the crack showed an increase at the BPFO. The amplitude at 1XBPFO of this accelerometer was 5 times higher than the amplitudes of any of the other accelerometers at this frequency. Due to this, we will continue to install 4 accelerometers per seal on those we choose to monitor. If you apply this to your particular problem you should give significant thought to signal transmission paths from the actual bearing to your accelerometer.

The spectrums in Figures 13, 15 and 17 are for three separate and independent occurrences. Each spectrum was taken with an analyzer in its "uncorrected" mode. For that reason all amplitudes shown from 0 to approximately 300 cpm are in some degree of error. The reason for this is best explained in Figures 8 and 9 where the "uncorrected analyzer" shows poor performance close to 0 cpm. Note that for all values above 300 cpm the amplitudes will be correct.

This particular type of bearing failure generated many harmonics of the 1X BPFO. The amplitude of these harmonics also correlate nicely with the amount of damage found when the bearing was inspected. Also notice with the extensive damage in Figure 18 the spectrum in Figure 17 shows clearly defined turning speed sidebands around harmonics of the BPFO. All cursor values shown in Figures 13, 15 and 17 are of harmonics of the BPFO.

It is important to note that we can not, and probably never will be able to, predict actual seal life. It is far too complex an issue. Even after bearing destruction, in several instances, the seal has run for a few more days. What we do know is that the bearing is the axial locator of the seal parts. When it no longer can provide this axial location function, stationary parts come in contact with rotating parts. Seal failure is only a short time away at this point. Knowing that bearing damage exists has still allowed us to make conservative predictions and keep us from incurring production losses. Just as important is the fact that when there is no bearing damage, we know that also. Before acquiring this ability we replaced some seals with perfectly good bearings, based on running time considerations only.

You must be cautious in applying the lessons learned on these reactors to your equipment because of possible differences in size, location of accelerometers, distance from accelerometer to bearing, and transmission path of the signal from bearing to accelerometer.

CONCLUSIONS:

From the few FFT analyzer/data collectors tested it is apparent that no COMMON method of reporting amplitudes at these low frequencies exists. As a generality you could conclude that of the analyzers tested, all were consistent at 300 cpm, but not much below that. If you have important equipment assessments to make below 300 cpm you should probably conduct the signal generator test explained in the text. For there to be any MEANINGFUL standards on bearing defect amplitudes at low speeds, you would have to know how, and with what analyzer/transducer combination, the data was collected. Only then could you insure that the data was accurate.

Most of the analysts using this paper for a reference point will have larger or smaller equipment than described in the text. Size of equipment is important. It can have a LOT to do with signal transmission path and physical distance from transducer to bearing. Please keep this in mind when using the text as a reference.

The design of FFT analyzers/data collectors is a dynamic field. Many changes are made within a year's time. For that reason, the graphs presented under the Sine Wave Generator Test section will become OBSOLETE quickly.

The real message of this paper is: "TO MAKE A MEANINGFUL ANALYSIS, YOU MUST KNOW HOW YOUR ANALYZER/TRANSDUCER COMBINATION PERFORMS AT LOW FREQUENCY."

Bearing defect frequencies are generated at low shaft turning speeds as well as high shaft turning speeds. The problems mentioned with instrumentation dictate that more care should be taken in obtaining the data at low speeds. If this is done, you may be able to use conventional techniques (bearing defect frequency) to assess the condition of equipment at low shaft turning speeds.

ACKNOWLEDGEMENT:

I would like to acknowledge ALL members of the KoSa reactor seal team for their involvement in the solution of this problem. I would also like to thank Dr. Art Crawford for making me aware of the problems associated with instrumentation, and for PATIENTLY helping me work around them.

REFERENCES:

1. Randall, Robert B., "Vibration Measurement Instrumentation", Chapter 13, Harris Shock and Vibration handbook, McGraw Hill Book Co. 2001, P. 13.3

2. Mitchell, John S., "Introduction to machinery Analysis and Monitoring, Second Edition"' Penn Well Books, 1993