Case Study
Analysis of Two Stage Planetary Gearbox Vibration

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Abstract:

A two stage planetary gearbox used in underground coal mining experienced an overload in service which caused bearing and bolting failures. The gearbox was repaired and underwent a no load spin test. A very audible noise was present in the vicinity of the 1st stage gear set. Vibration analysis was used to determine the source of the vibration. The equations for calculating the planetary gear shaft speeds, gear meshing frequencies, and bearing frequencies in the gearbox are provided.

Background:

Gearboxes used in underground coal mining are of compact design. A typical two stage planetary gearbox, 800 HP, 40.173:1 Ratio with 1800 RPM input is shown in **Figure 1**. The unit was received by a repair facility for rebuild following failure from an overload incident. It was reported that the bearings were replaced and that one bearing had broken into many fragments. Following repairs a no load spin test of the gearbox was performed as a check for bearing faults, **Figure 2**. There was an audible impacting type noise from the input planetary section.

Analysis:

During the spin test, vibration data were measured using an accelerometer with rare earth magnetic mount. Initial inspection of the data indicated impacting and ringing of natural frequencies of the gearbox, **Figure 3**. The impacts measured a 221.87 mSec period or 4.507 Hz ~ 704.6 CPM. The FFT of the time domain data showed harmonics of 704.6 CPM and indication of excitation of several natural frequencies of the gearbox.

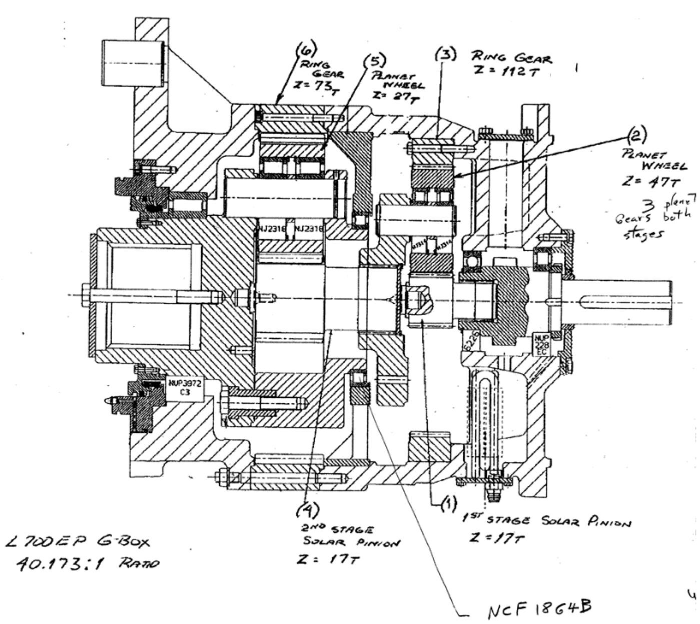

Figure 2. Cutaway View of 2-Stage Planetary Gearbox, 40.173:1 Reduction.

Figure 2. Gearbox On Test Stand For No Load Spin Test.

Before a determination of the source of the vibration could be made, an understanding of the gearbox design was required and calculation of the excitation frequencies. Based on the information provided by the drawing shown in **Figure 1**, several calculations were made to obtain the shaft speeds, bearing fault frequencies and gear meshing frequencies.

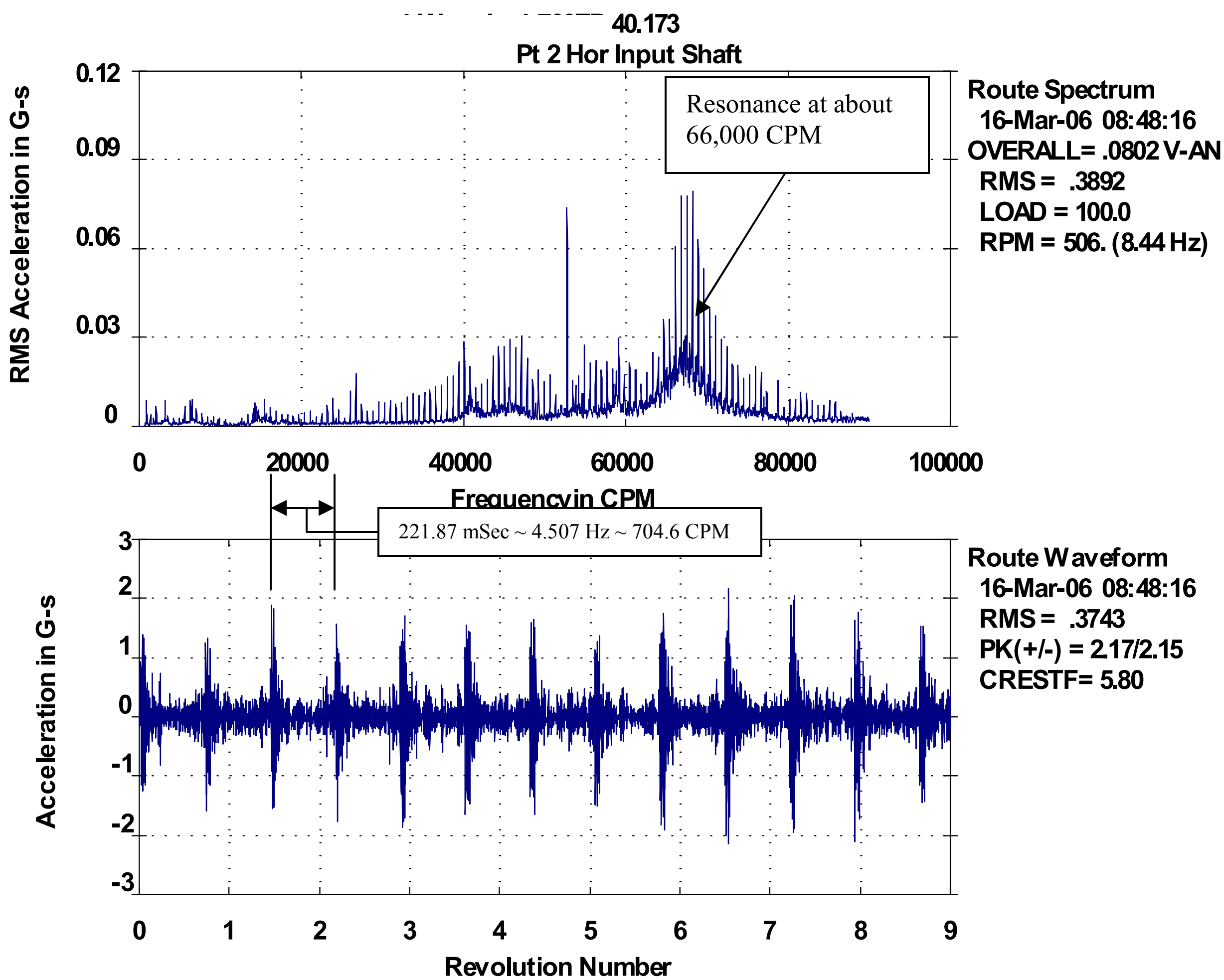

Figure 3. Vibration Signal Measured at Gearbox Input Section Showed Impacts at 221.87 mSec Interval ~ 4.507 Hz ~ 704.6 CPM. The FFT (Top Plot) Indicated Excitation of Several Resonant Frequencies Including A Very Response One at About 66,000 CPM ~ 1,100 Hz.

Epicyclic gear boxes derive their name from the epicyclodial curves that the planet gears produce during rotation. There are three general types of epicyclic arrangements, 1) planetary which consists of a stationary ring gear combined with a rotating sun gear and moving planet carrier, 2) star configuration which consists of a stationary planet carrier coupled with a rotating sun gear, and 3) solar gear that has a fixed sun gear combined with a moving ring gear and planet carrier. The planetary arrangement is most common and is shown by the schematic in **Figure 4**. The subject gearbox had the planetary arrangement for the 1st and 2nd stages. Input was from the sun with three planets supported by a carrier revolving about the sun pinion and the ring gear fixed.

		1st Stage	2nd Stage
S_S	Sun Gear RPM (Input Speed)	1782	-234.838
R_T	Ring Gear Teeth	112	73
P_T	Planet Gear Teeth	47	27
S_T	Sun Gear Teeth	17	17
T_{value}	Train Value	0.151786	0.2328767
C_S	Carrier RPM	-234.838	-44.358
P_S	Planet RPM	559.612	119.931
$P_{Sabsolute}$	Planet RPM Absolute	-324.775	-75.573
R_S	Ring Gear RPM	0	0
P_{GMF}	Planet Gear Meshing Freq CPM	26,301.76	3,238.14
$F_{GMF-Sun}$	Sun Gear Meshing Freq CPM	30,294.00	3,992.23
Ratio	Stage Ratio	7.5882	5.2941

Table 1: Summary of The Gearbox Shaft Speeds and Gear Meshing frequencies.

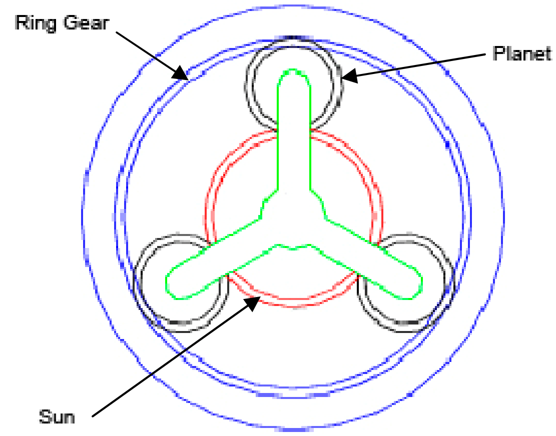

Figure 4. Gear Arrangement Of 1st Stage Planetary Input Section.

Step 1: Carrier Speed

The 1st stage carrier speed can be calculated as follows:
Train value:

$$T_{Value} = \frac{S_T \times P_T}{P_T \times R_T} = \frac{17 \times 47}{47 \times 112} = 0.151786$$

The 1st stage Carrier Speed then calculates to:

$$C_S = \frac{R_S - T_{value} \times S_S}{1 - T_{value}} = \frac{-0.151786 \times 1782}{1-(-0.151786)} = \frac{-270.48265}{1.151786} = -234.8376\,RPM$$

The negative sign "-" indicates the carrier is rotating in the opposite direction to the sun gear.

The 2nd stage carrier speed which is also the output of the gearbox calculated to:

$$T_{Value} = \frac{S_T \times P_T}{P_T \times R_T} = \frac{17 \times 27}{27 \times 73} = 0.2328767$$

$$C_S = \frac{R_S - T_{value} \times S_S}{1 - T_{value}} = \frac{-0.2328767 \times 234.8376}{1-(-0.2328767)} = \frac{-54.6882}{1.2328767} = -44.3582\,RPM$$

The gearbox ratio calculated to:

$$Ratio = \frac{Input_{RPM}}{Output_{RPM}} = \frac{1782}{44.3586} = 40.173$$

The calculated ratio agreed with the ratio provided by the gearbox manufacture of 40.173.

The carrier speeds can also be calculated as follows:

The 1st stage carrier speed:

$$R_O = \frac{R_t + S_t}{S_t} = \frac{112+17}{17} = 7.5882$$

$$C_S = \frac{S_S}{R_O} = \frac{1782}{7.588} = 234.838\,RPM$$

The 2nd stage carrier speed:

$$R_O = \frac{R_t + S_t}{S_t} = \frac{73+17}{17} = 5.2941$$

$$C_S = \frac{S_S}{R_O} = \frac{234.838}{5.2941} = 44.358\,RPM$$

Step 2: Planet Speed

The 1st stage planet rotational frequency or planet spin speed was calculated as follows:

$$P_S = C_S \bullet \frac{R_T}{P_T} = 234.838 \bullet \frac{112}{47} = 559.613\,RPM$$

The 1st stage absolute planet rotational frequency can be determined by summing the carrier and planet rotational frequencies algebraically. Note that this frequency seldom appears in vibration data.

$$P_{S\,Absolute} = C_s + P_s = -234.838 + 559.612 = 324.775\,RPM$$

The 2nd stage planet spin speed calculated as follows:

$$P_S = C_S \bullet \frac{R_T}{P_T} = 44.3594 \bullet \frac{73}{27} = 119.935\,RPM$$

The 2nd stage absolute planet rotational frequency is then determined:

$$P_{s_{Absolute}} = C_s + P_s = -44.358 + 119.935 = 75.577\,RPM$$

The planet speed can also be calculated as follows:

1st stage planet RPM:

$$P_R = \frac{R_t}{P_t} \bullet (R_S - C_S) = \frac{112}{47} \bullet (-234.838) = -559.614\,RPM$$

2nd stage planet RPM:

$$P_R = \frac{R_t}{P_t} \bullet (R_S - C_S) = \frac{73}{27} \bullet (-44.3594) = -119.935\,RPM$$

Step 3: Gear Meshing Frequencies

The planet gear meshing frequencies were then determined for stage 1 as follows:

$$P_{GMF} = P_S \times P_T = 559.612 \times 47 = 26,301.76\, CPM$$

The higher frequency sun gear meshing frequency was calculated:

$$S_{GMF} = S_s \times S_T = 1782 \times 17 = 30,294\ CPM$$

The 2nd stage planet gear meshing frequencies were then determined as follows:

$$P_{GMF} = P_S \times P_T = 119.931 \times 27 = 3,238.14\ CPM$$

The 2nd stage sun gear meshing frequency was then calculated:

$$S_{GMF} = S_s \times S_T = 234.8376 \times 17 = 3,992.24\ CPM$$

Step 4: Bearing Fault Frequencies

After the gearbox shaft speeds were determined, the bearing fault frequencies were calculated and listed in **Table 2**. For purposes of calculating the bearing fault frequencies of the planet bearings, the spin frequency of the planets must be summed to the carrier rotational frequency. Since the outer race was turning faster the calculations were made as if the bearing inner race was not rotating.

Stage 1 Planet Spin Freq 559.612 + 234.838 = 794.45 RPM
Stage 2 Planet Spin Freq 119.931 + 44.359 = 164.29 RPM

Note that dimensions were not located in the time allowed for the cylindrical roller bearing NUP 3972 .

		1X Brg Fault Frequencies CPM			
Component	**Brg Inner Race RPM (Relative to Outer Race)**	**FTF**	**BSF**	**BPFO**	**BPFI**
Sun NU228E	1782.00	764.48	6142.55	14523.30	19334.70
Sun 6226	1782.00	746.66	5326.40	7459.45	10360.55
1st Stage Planet NJ314	794.45	453.63	2738.31	6474.50	8619.62
Output Carrier NCF 1864B	44.36	20.89	381.04	1128.20	1267.13
Output Carrier NUP 3972	44.36	0.00	0.00	0.00	0.00
2nd Stage Planet JN2318	164.29	65.88	398.89	856.60	1279.16

Table 2: Listing of Gearbox Bearings and Fault Frequencies CPM.

The bearing fault frequencies were calculated using Machinery Health Manager software (CSI RBMware). Equations from Reference 2 are provided below.

$$Cage_{Hz} = \frac{n}{2} \bullet \left[1 - (\frac{d}{P.D.}) \bullet \cos\alpha \right]$$

$$Ball\ Spin_{Hz} = \frac{P.D.}{d} \bullet \frac{n}{2} \bullet \{1 - (\frac{d}{P.D})^2 \bullet \cos^2 \alpha\}$$

$$Ball\ Pass_{Outer\ Race_{Hz}} = Z \bullet \frac{n}{2} \bullet \{1 - (\frac{d}{P.D.}) \bullet \cos\alpha\}$$

$$Ball\ Pass_{Inner\ Race_{Hz}} = Z \bullet \frac{n}{2} \bullet \{1 + (\frac{d}{P.D.}) \bullet \cos\alpha\}$$

Where:

$d = Rolling\ Element\ Diameter$

$n = Shaft\ Freqeuncy \frac{Cycles}{\sec}\ or\ (\frac{RPM}{60}), Hz$

$P.D. = Pitch\ Diameter\ (For\ ball\ bearings\ P.D. = \frac{O.D. + bore}{2})$

$Z = Number\ of\ balls\ or\ rollers\ (per\ row)$

$\alpha = Bearing\ contact\ angle\ 0\ \deg ree\ for\ pure\ radial\ load$

$\alpha = 15\ to\ 20\ \deg\ (thinner - \sec tion\ bearings)$

$\alpha = 37\ to\ 40\ \deg\ (73, 74\ Series)$

$\alpha = 10\ to\ 15\ \deg\ spherical\ roller\ typical\ range$

Step 5: Determination of Vibration Source

Referring to the data plots in **Figure 3**, it was readily determined using the vibration software cursors that the impulse frequency was 4.407 Hz ~ 704.6 CPM. A check of the frequencies in **Table 2** showed that this frequency does not match any of the bearing fault frequencies.

A check of the gearbox shaft speeds and gear meshing frequencies in **Table 1** also did not immediately identify a forcing frequency. The pulses in the time domain measured 85.42 mSec ~ 11.7 Hz ~ 702.4 CPM. Since the gearbox has three planets in each stage an impulse could occur at three times the 1st stage carrier frequency of 234.838 CPM if there were damage to the ring gear teeth. This frequency was calculated as follows:

$$3 \times 234.838 = 704.5\,CPM \sim 11.74\,Hz. \qquad \frac{1}{11.74} = 0.08517\,Sec \sim 85.17\,mSec$$

The source of the pulses was related to rotation of the carrier and the three planets in the 1st stage also called the planet passing frequency.

Updating **Table 1** to include the planet passing frequency, **Table 1A**:

		1st Stage	**2nd Stage**
S_S	Sun Gear RPM (Input Speed)	1782	-234.838
R_T	Ring Gear Teeth	112	73
P_T	Planet Gear Teeth	47	27
S_T	Sun Gear Teeth	17	17
T_{value}	Train Value	0.151786	0.2328767
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R_S	Ring Gear RPM	0	0
P_{GMF}	Planet Gear Meshing Freq CPM	26,301.77	3,238.14
$F_{GMF-Sun}$	Sun Gear Meshing Freq CPM	30,294.00	3,992.23
$\mathbf{P_{Pass}}$	**Planet Passing Freq**	**704.514**	**226.719**
Ratio	Stage Ratio	7.5882	5.2941

Table 1A: Summary of The Gearbox Shaft Speeds, Gear Meshing and Planet Passing Frequencies.

Expanding the time domain plot to show only two pulses, **Figure 5**, the pulses ring down which is typical response of structural resonance. The spectrum data also provided clear indication of resonance excitation.

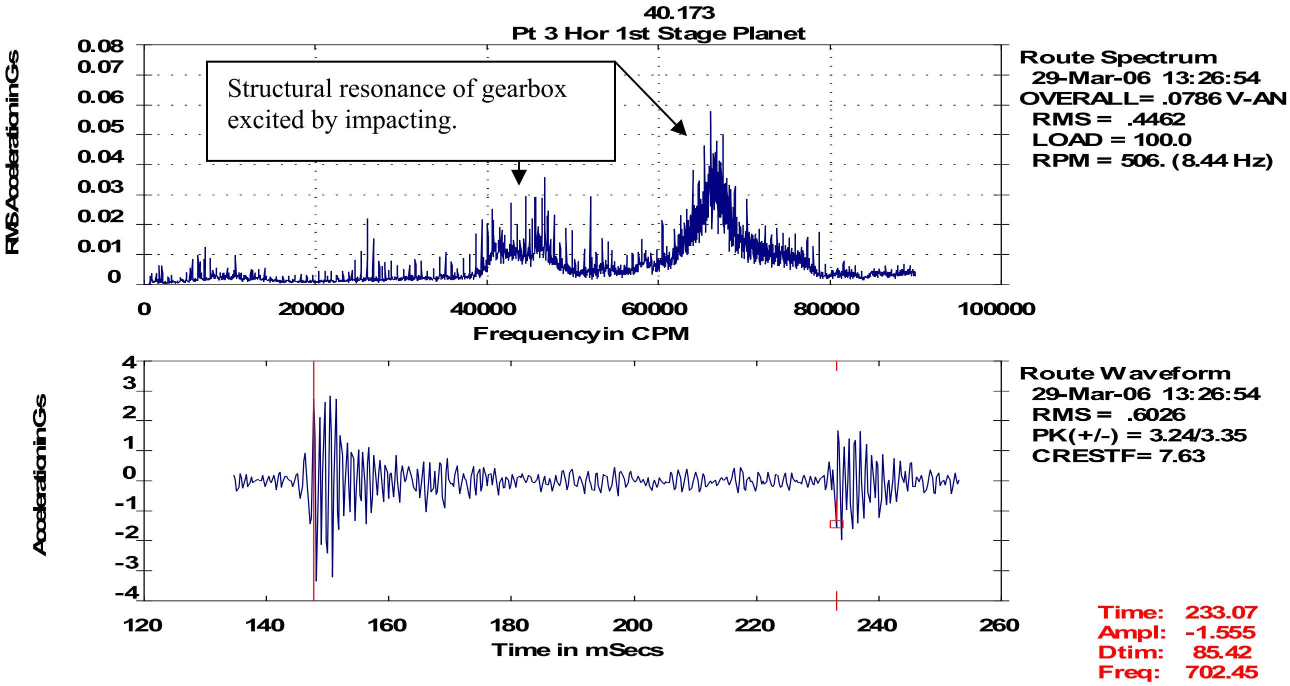

Figure 5. Time Data Expanded To Show Impacting and Ring Down.

Plotting a single pulse in **Figure 6** showed more clearly the time between oscillations was about 1.042 mSec or 57,600 CPM. Note that spectrum analyzers don't make good oscilloscopes due to the rather course sampling at 2.56 times the maximum frequency to be displayed in the frequency spectrum.

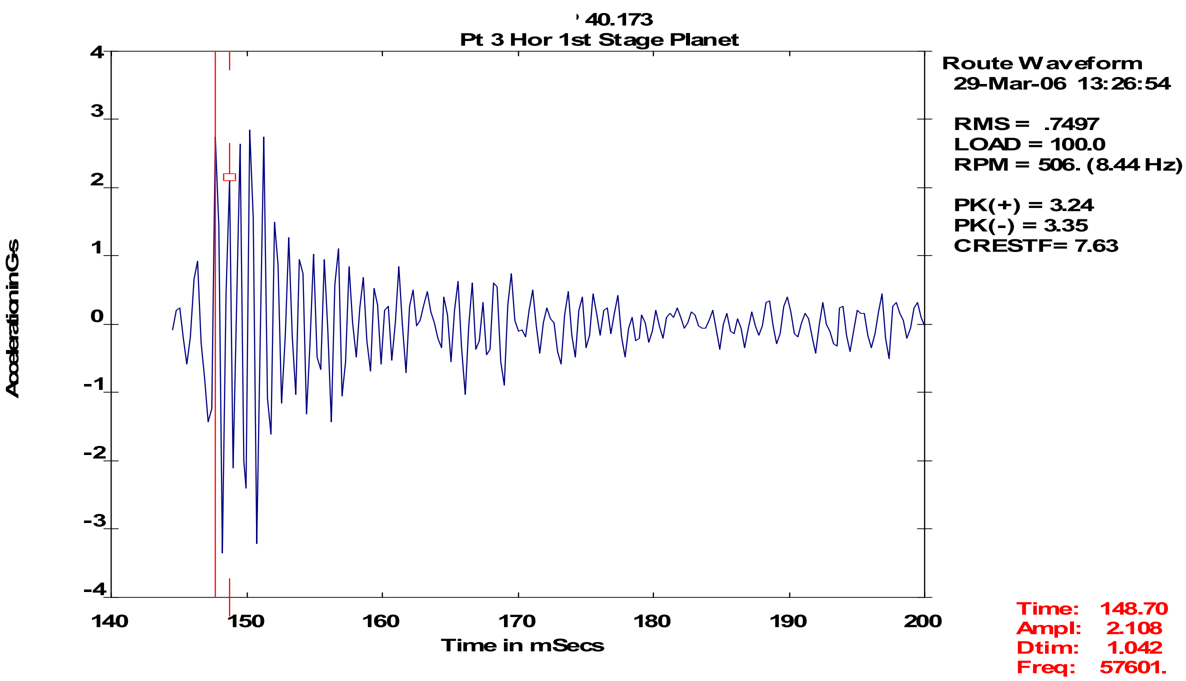

Figure 6. Expanded Plot of One of The Impact Events in The Time Domain Shows The Ring Down Frequency Is About 57,600 CPM.

Plotting the time data in a circular plot, **Figure 7**, clearly shows three periodic impacts per revolution of the carrier. The impacts occurred as each planet rolled over a damaged ring gear tooth.

Peakvue spectrum and time domain data are plotted in **Figure 8** and shows impacting to about 5g's with the same frequency content as the normal vibration data. The auto correlation plot of Peakvue time data is plotted in **Figure 9** in a circular plot format.

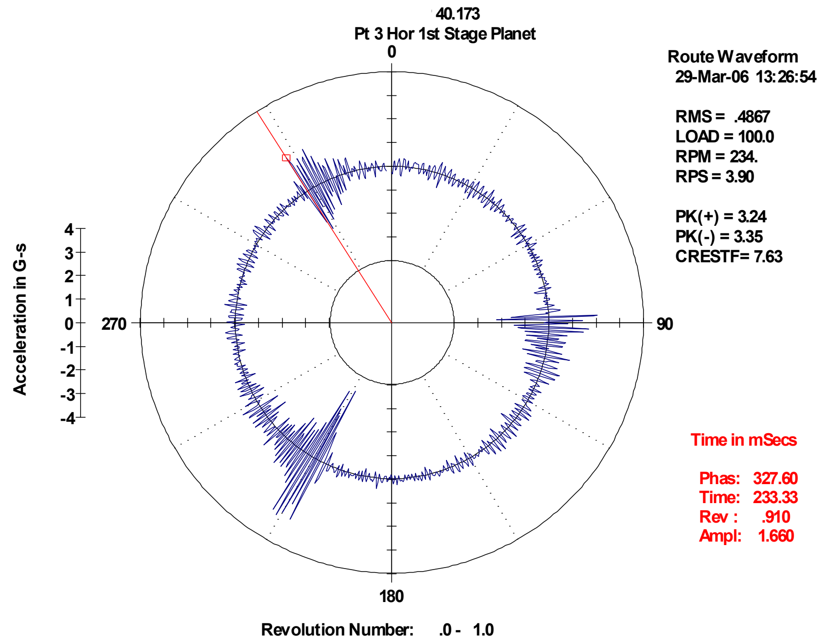

Figure 7. Time Domain Data Plotted in Circular Format.

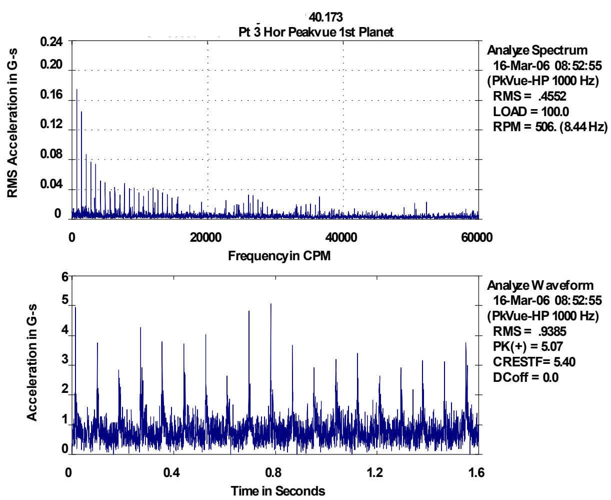

Figure 8. PeakVue Data Also Contained Impacting Data At the 3X the Carrier RPM.

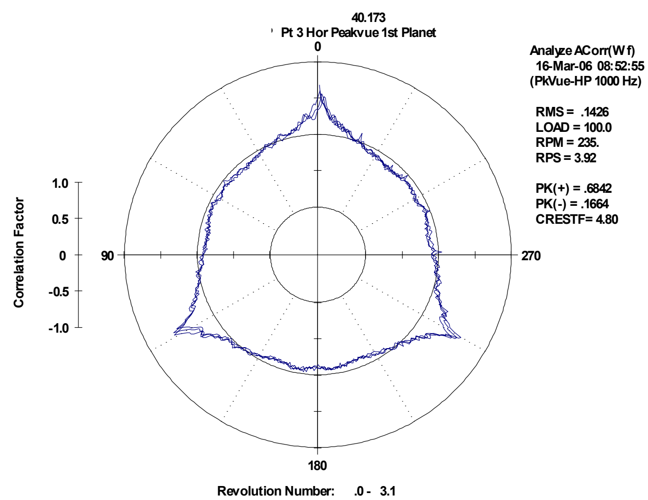

Figure 9. Auto Correlation of Peakvue Time Data.

After reviewing the data and calculations, the conclusions were:

1) The 1st stage carrier was impacting a stationary object three times each revolution, or
2) Each planet gear in the 1st stage was rolling over damaged teeth on the ring gear.

No gear meshing frequencies were evident in the data. Opening of the gearbox for inspection of the 1st stage was recommended.

Gearbox Inspection:

With the likely problem area in the gearbox identified, the gearbox was dissembled for inspection. A small fragment of the disintegrated bearing was found imbedded in the unloaded side of one tooth of the ring gear. The bearing fragment was removed, the damaged tooth dressed, the gearbox reassembled and spin tested again.

Before and after vibration data are plotted in **Figure 9 & 10**. The periodic impacting caused by the planet teeth rolling over the damaged ring gear tooth was reduced.

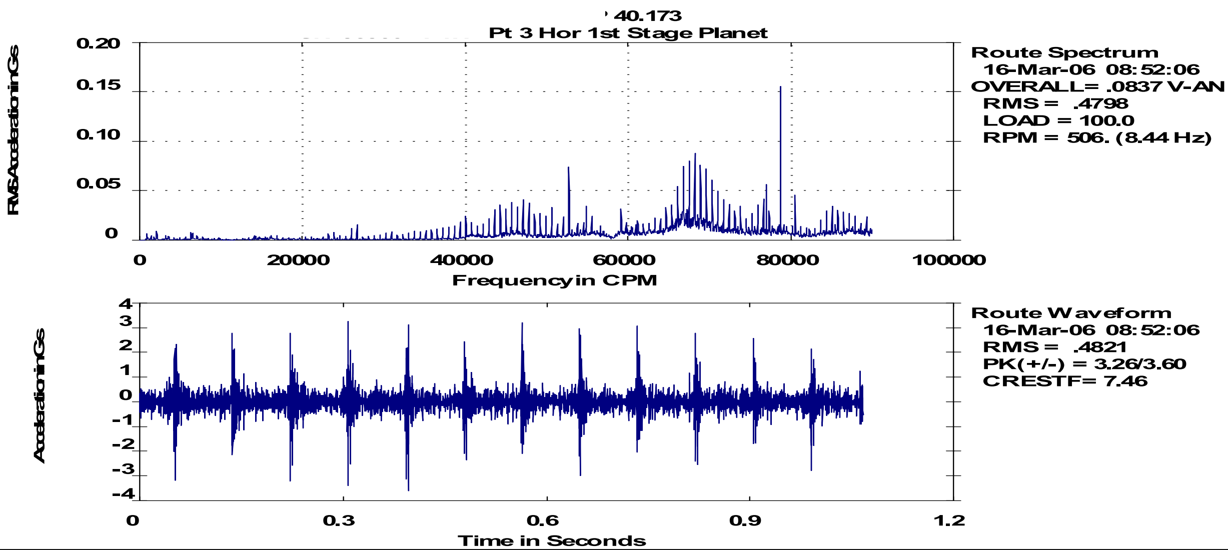

Figure 9. Initial Spectrum And Time Domain Data With Brg Fragment Imbedded In The Ring Gear.

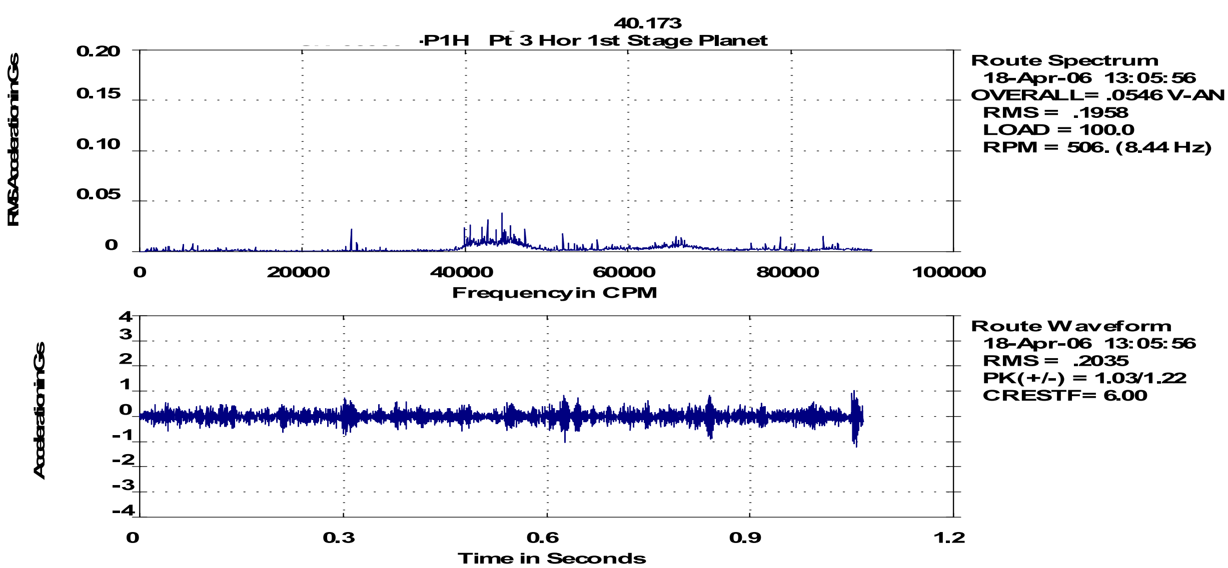

Figure 10. Spectrum And Time Domain Data After Dressing Ring Gear Damaged Tooth.

A photo of the damaged ring gear tooth is shown in **Figure 11** after dressing. Indentations can be seen where the tooth material was compressed by the bearing fragments.

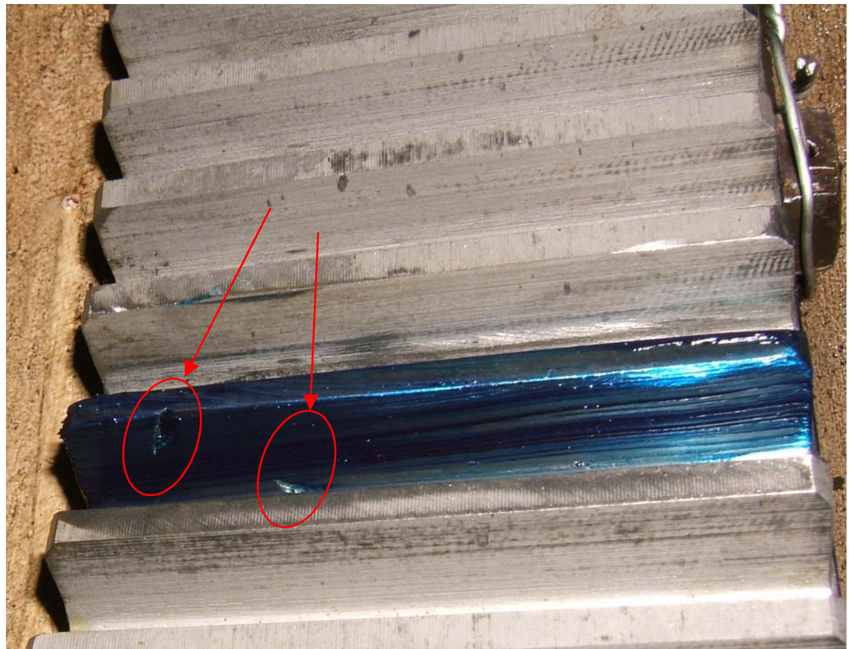

Figure 11. Ring Gear Tooth Afer Removing Brg Fragment & Dressing Raised Metal.

Conclusions:

This article describes the process that was used to analyze impacting type vibration of a two stage epicyclic planetary gearbox during a post-repair unloaded spin test. The forcing frequencies were calculated and identified the probably source of the vibration. Inspection of the ring gear identified fragments of a bearing race embedded in the unloaded side of a ring gear tooth.

References:

1. Eisenmann, Sr., P.E., Eisenmann, Robert, C., Jr. **Machinery Malfunction Diagnosis and Correction**, Prentice Hall PRT, ISBN 0-13-240946-1, PP 470-477.
2. Guyer, Raymond A. Jr., Rolling Bearings Handbook And Troubleshooting Guide, ISBN 0-8019-88761-3, PP 108.

Author

Ken Singleton is Manager of KSC Consulting LLC with over 40 years industrial experience. He retired from Eastman Chemical Company in 1999 as a Senior Engineering Technologist in the Rotating Machinery Technology Group after 32 years service. He has presented technical papers at the Vibration Institute National Meetings, P/PM Conferences, ASME Joint Power Conferences, and Piedmont Chapter of the Vibration Institute. Education includes an AAS Electronic Engineering Technology, Mechanical Engineering ICS, Journeyman Machinist, Washington Co Technical School.