COAL MINING EQUIPMENT VIBRATION ANALYSIS
RESULTS OF ELEVEN YEARS SHOP TESTING

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Abstract: Rotating equipment used in underground coal mining such as stage loaders, crushers and conveyer power units are exposed to severe operating conditions. The conditions for the equipment are much different than for rotating equipment used in above ground plants such as petro-chemical, refineries, power houses, paper mills, etc. Above ground, critical rotating equipment is typically mounted on thick sole plates grouted to concrete foundations. Equipment may also be mounted on building structural steel using isolated inertial bases or vibration isolators. Underground, stage loaders, crushers, motors, gearboxes and fluid couplings of conveyer power units are mounted in steel frames which are sitting on the floor of the mine. In some mines, the equipment may even operate partially submerged in mud.

The typical practice of mines is to overhaul equipment is to replace bearings, seals, motors and fluid couplings after mining a room or panel of coal. After overhaul, the equipment is typically no-load run tested at the repair facility. Motor repair shops typically measure vibration at each bearing housing of the motors running unloaded before shipping either to the mine or to a repair facility. After assembly, the mining equipment is run with no-load to check for potential defects such as oil leaks, rolling element bearing faults, rotor unbalance, mis-alignment, seal rubs, fluid coupling unbalance, etc.

It is important that defects are detected before the equipment is returned to service. This article discusses findings after testing 555 machines over eleven years. The defects identified are grouped and charts are provided showing defect distribution. Case studies are provided to illustrate some of the more common and interesting problems.

Keywords: Balance, flexible coupling, fluid coupling, gearbox, motor, power unit, rotor critical speed, shaft breakage, stage loader.

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A. Vibration Limits: Prior to beginning vibration analysis of coal mining equipment, a review was made of existing vibration standards. No vibration standards or limits were identified that specifically applied to mining equipment. The vibration limits agreed upon were determined after review of generally accepted industry guidelines and standards. During vibration run tests, the equipment was run unloaded in repair shop facilities or in some cases at the mine shop. The equipment tested included stage loader power units, transfer drives, sprocket, crushers, and conveyer power units (consisting of a motor, flexible coupling, fluid coupling and gearbox).

Test Conditions

- Units under test were running unloaded and either resilient mounted or bolted to their massive support frames. The resilient mounting consisted of wood beams and 3/4 " to 1.5" thick rubber sheets on concrete floor.
- Coal crushers were run unloaded and supported on 1.5" rubber sheets on concrete floor, (resilient mounted).
- Vibration data were collected using 100mV/g accelerometer attached to the bearing housings (where accessible) with a 50-60 lb_f pull flat magnet and CSI 2120 or 2130 Analyzers. Data were uploaded to CSI AMS software for analysis.
- Bearing faults were identified using standard vibration data and Peakvue alarming on the time waveform per CSI's published recommendations.
- The vibration limits were as follows:

Units	Frequency Span	Overall Level	Frequency 1X	Frequency 2X	Comment
In/Sec Pk	0 to 2000 Hz	0 to 0.15	0 to 0.10	0 to 0.07	**Acceptable**
In/Sec Pk	0 to 2000 Hz	0.15 to 0.40	0.1 to 0.25	0.07 to 0.15	**Marginal** May require alignment, balancing bearing replacement, seal replacement, or other corrections.
In/Sec Pk	0 to 2000 Hz	=>0.40	=> 0.25	=>0.15	**Unacceptable** Corrective actions may include alignment, balancing, gear, bearing, seal replacement, etc.

Peakvue

Units	Frequency Span	Time Waveform Shaft RPM 1800	Time Waveform Shaft RPM 1800	Time Waveform Shaft RPM 450	Time Waveform Shaft RPM 450
G's	0-1000 Hz	Alert 5 g's	Action >10 g's	Alert 3 g's	Action >6 g's

References:

1. AGMA Standard 6000-A88, Specification for Measurement of Linear Vibration on Gear Units.
2. GM Specification V1.0a, GM-1761: General Motors / Delphi Vibration Standard for the purchase of new and rebuilt Machinery and Equipment.
3. ISO 10816-3 for flexible Mounted Machines

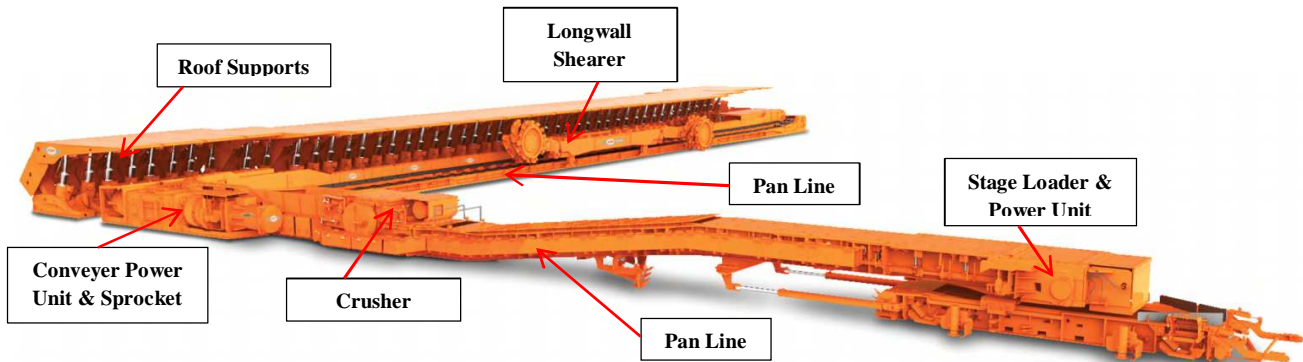## Longwall Mining Machinery Definitions

Figure 1. Stage Loader, Crusher, Conveyer Drive. [Ref 6]

Longwall Shearer: The coal shearer is mounted on a conveyor operating underneath a series of self-advancing hydraulic roof supports, see **Figure 1**. Longwall shearers have a cutting height of 1.5 to 7 meters (5 ft to 23 ft). The width of the cut is typically 1000 ft up to 1500 ft. Panels of coal are extracted - rectangular blocks of coal as wide as the face the equipment is installed in and as long as several kilometers. Shearers cut coal from the face, which falls onto an armored face conveyor for removal. Longwalls can advance into an area of coal, or more commonly, retreat back between development tunnels (called "gateroads").

Roof Supports: Hydraulic powered, the roof supports hold up the mine roof as the coal is removed, see **Figure 1**. The roof supports are typically 10-12 ft high in the US but up to 25 ft in China and Australia.

As the longwall miner retreats back along a panel, the roof behind the supports is allowed to collapse in a planned controlled manner.   Each roof support is connected by a relay bar to a segment of the conveyor called a pan. As the shearing machine passes each pan, the roof support pushes the pan forward in the void left behind by the removed coal. The support then lowers and pulls itself up to the pan and resets against the roof forming a "snake" in the conveyor.

Figure 2. Conveyer Power Unit, Sprocket and Chain.

Conveyer Power Units & Sprocket: The coal is conveyed by chains and bars from the shearer, see **Figure 1 & 2**. The chain is driven by the head drive and tail drive power units and sprocket. The term "head" and "tail" refers to the location of the drive relative to the shearer. Motor size may range from 800 to 2,200 HP. Typical arrangement is (2) power units on the head drive and (1) power unit on the tail drive.

Crusher: High speed crushers, see **Figure 3**, typically have two hammers in four rows. The crusher frame is fabricated from heavy steel plate. The crushers handle lump coal and oversize rock, crushing the material into small particles for easier conveying and improves system productivity. The crushers are typically of two designs, 1)

Figure 3. Coal Crusher, Belt Driven.

crusher shaft belt driven or 2) crusher shaft driven by a right angle gearbox, fluid coupling and motor. Motor size is typically 300 HP.

Stageloader: A stage loader or beam stage loader (BSL) is a chain conveyor that transports coal cut by the shearer from the shearer's armored face conveyor (AFC) to a main belt conveyor, see **Figure 4**. The crusher is incorporated into the stage loader's design.

Pan Line: Pan Line modules, see **Figure 5**, that make up the face conveyor are connected to each other by connectors called dumb bells allowing each module to move horizontally and vertically. This allows the conveyor to be flexible and move with the undulations of the mine floor.

Figure 5. Stageloader.

Figure 4. One Pan Module With Twin Chains. [Ref 9]

C. Major Problem Category Distribution Chart

The problems identified during the equipment test runs were grouped in major categories such as No Corrective Action, Bearings, Unbalance, Resonance, Seals, etc., see chart in **Figure 6**. The major categories were then sub-divided, for example, the Unbalance Category was sub-divided into unbalance resulting from:
- Motor Test Sheave (test sheave used for belt driven test of gearboxes)
- Missing key, Short/Long or Mushroomed Key (keys in motors & gearbox input shafts)
- Fluid coupling
- Motor Coupling Boss (hub) or Sheave
- Crusher Shaft (Crusher shaft rotating assembly including sheaves and flywheels)

1. No Corrective Actions: Of the 555 machines tested, no recommendations for corrective actions were made for 45%. This does not mean vibration amplitudes were always in the acceptable range. Some of the equipment, especially crushers, had motor mounting design issues which resulted in very high amplitude vibration. However, the scope of work for the repair facility may be weld repair of the hammers, replacement of bearings and seals but no work on the motor mounting design. In this situation, based on the information provided by the client no recommendations would be made for corrective actions.
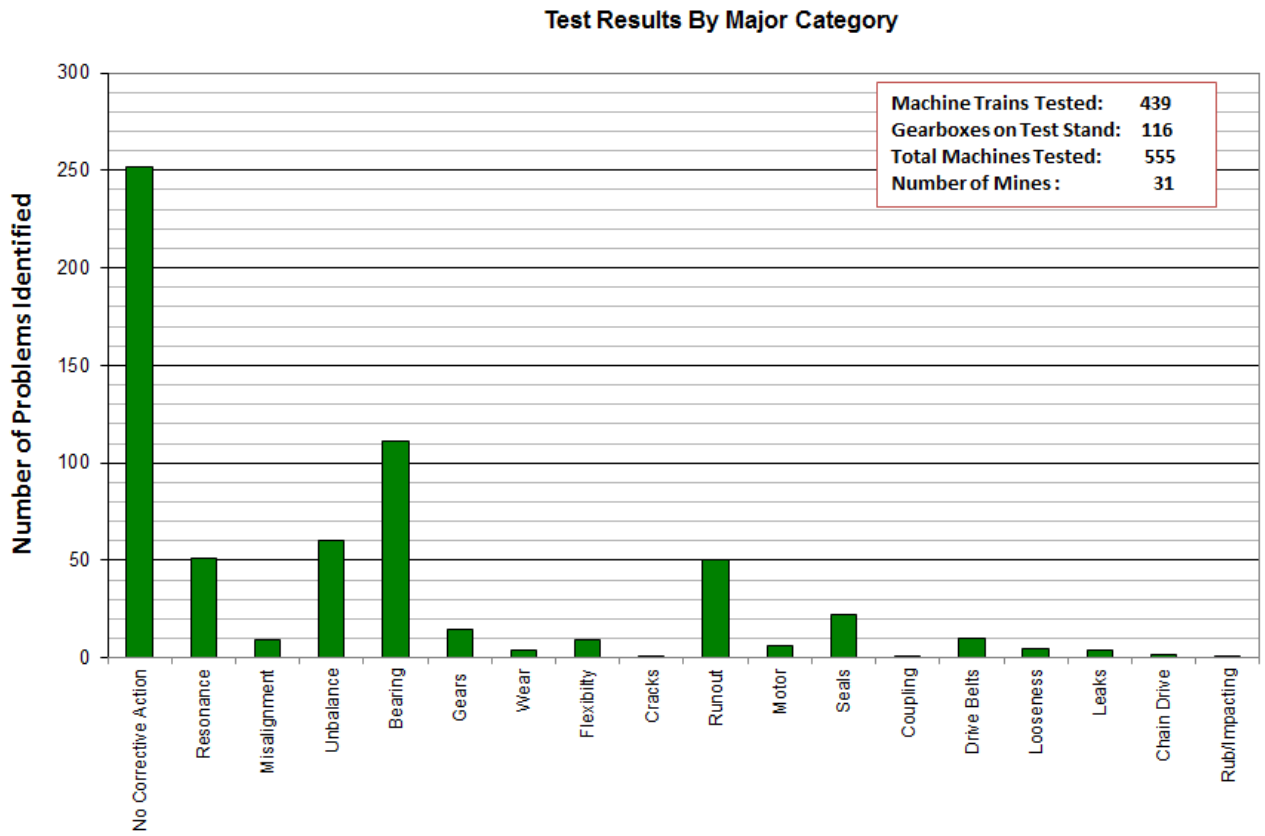
Test Results By Major Category

Machine Trains Tested: 439
Gearboxes on Test Stand: 116
Total Machines Tested: 555
Number of Mines: 31

Figure 6. Chart of Problem Major Category.

2. Bearings: Defective bearings were the most common defect identified by analysis of vibration data. Bearings documented to be generating defect frequencies were 111. The distribution of bearing problems is shown in **Figure 7**.

By far, motor bearing were the most common bearing problem identified followed by Gearbox bearing and Bearing Fit to Housing. Many of the motor bearing flaws were false brinelling and brinelling occurring during transport or improper handling. The motor shaft should be locked or preloaded axially before the motor is moved or transported. However, even though the drive end bearing is locked to prevent movement the outboard bearing is still free to move and is more susceptible to false brinelling during transport. Motors should always be stored on wood or elastomeric material to prevent false brinelling. The motor should never touch concrete to avoid brinelling of the bearings. An example of false brinelling is provided in this article that documents a motor returned to the motor OEM three times for bearing replacement.

Gearbox bearing flaws were most commonly identified for the input shaft bearings. This is the highest speed shaft typically operating at 1795 RPM. Gearbox bearing flaws also include the Bearing Fit to Housing category. A cartridge is inserted in the case bore and normally carries one or two of the input shaft bearings. This cartridge can be distorted as the retaining bolts are tightened causing the bore to go out of round and also forcing the bearing outer race out of round. A horizontal split gear case can be misaligned during assembly due to worn dowels which can pinch or distort a bearing outer race.

Bearing Problem Distribution Identified During Vibration Tests
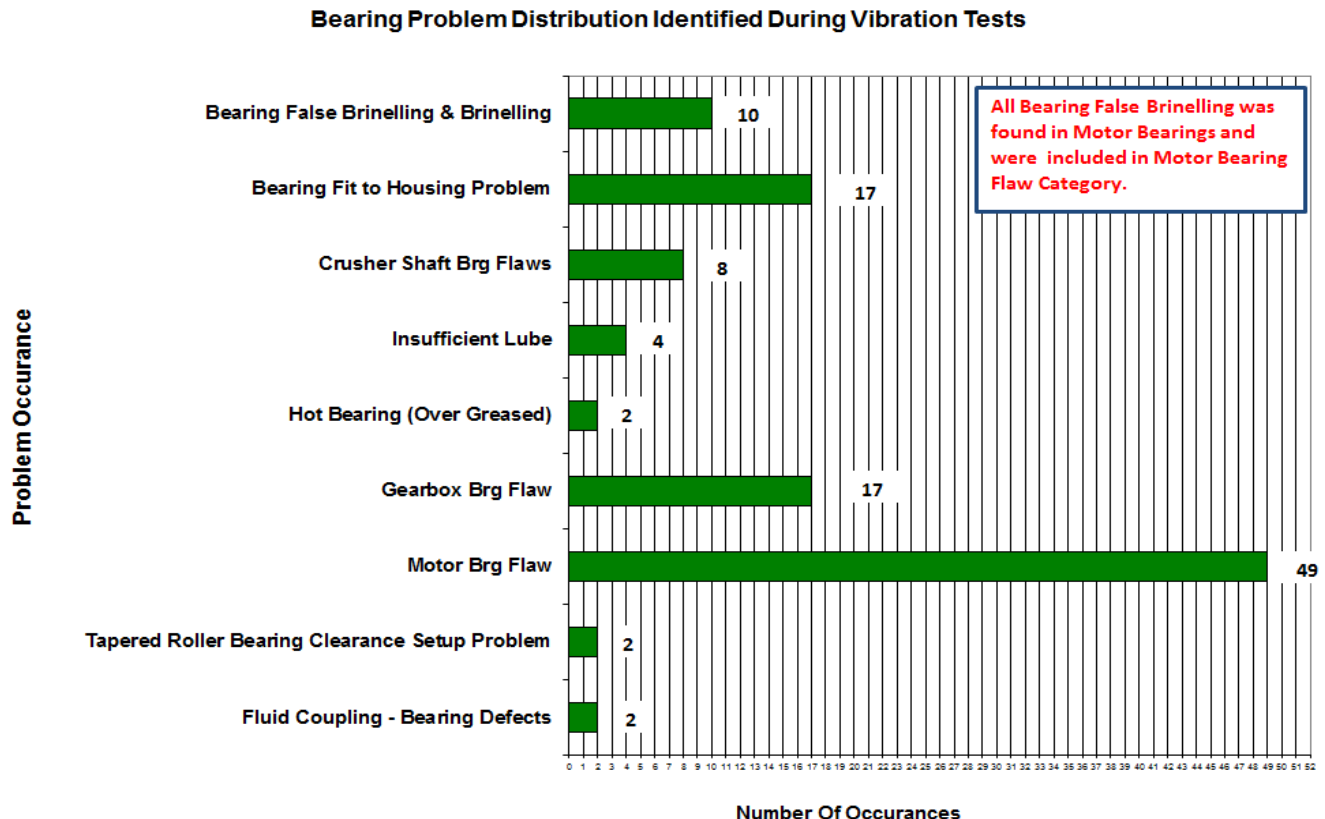
Figure 7. Bearing Problem Distribution.

3. Unbalance: Excessive residual unbalance was the 2nd most frequently identified problem with 60 occurrences documented. The distribution of unbalance is shown in **Figure 8**. Fluid coupling and gearbox input shaft rotating assembly unbalance was the most common. The gearbox input shaft rotating assembly includes the input shaft and the output section of the fluid coupling. Excessive vibration at 1X the gearbox input shaft rotating assembly can result from unbalance of the fluid coupling, coupling eccentricity to the shaft fit, bent input shaft, fluid coupling cocked on the shaft fit, coupling hub eccentricity, long/short keys, etc.

Running unloaded, the typical slip of the fluid couplings was 3 to 4 RPM. Therefore, a 6400 line spectrum with an fmax of 5,100 CPM was measured to display the motor and gearbox input shaft rotational frequencies. (5,100 CPM / 6400 Lines) * 1.5 Window Factor = 1.195 CPM Resolution.

Rotating Assemblies With Excessive Residual Unbalance
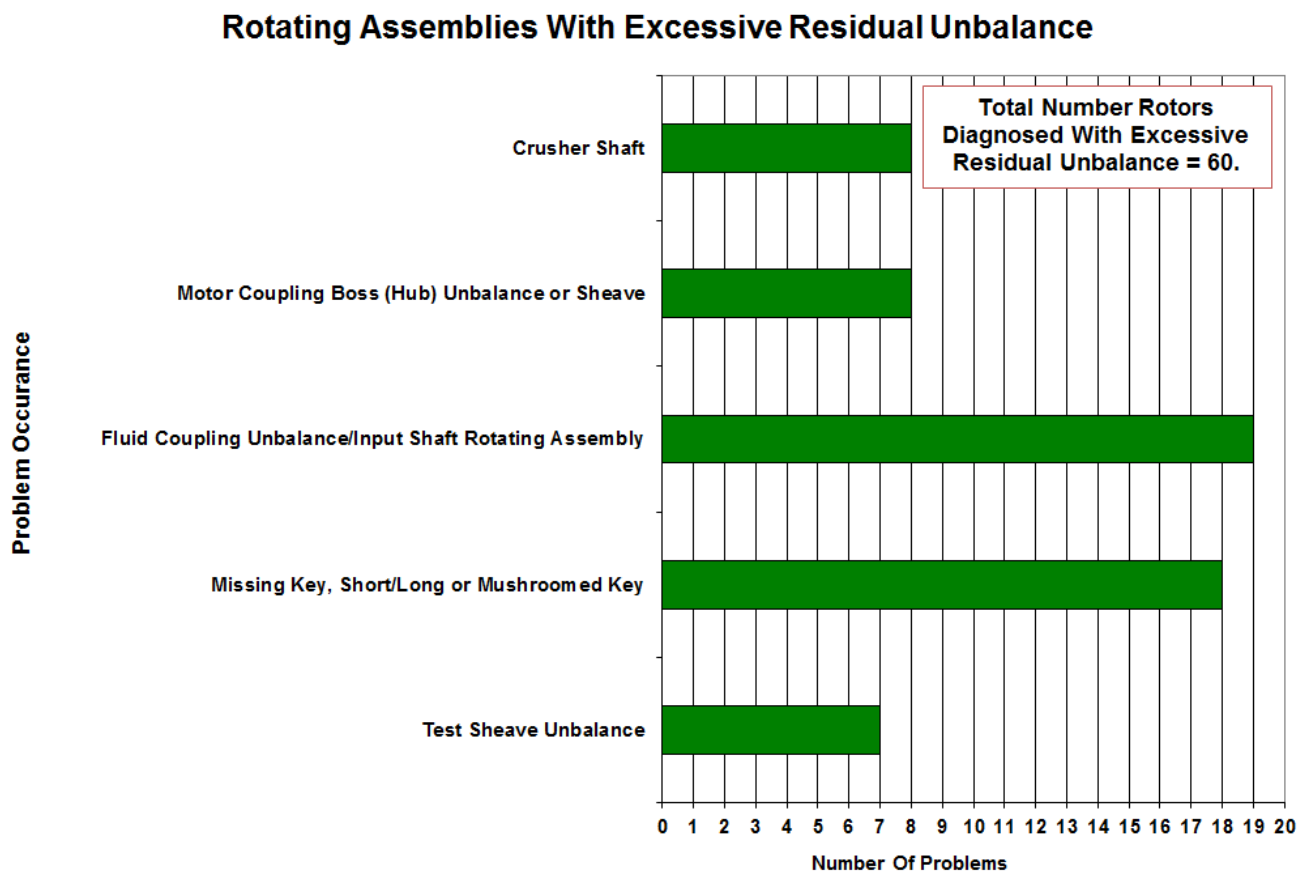
Figure 8. Unbalance Category Distribution.

Sometimes it was possible to trim balance a fluid coupling in-situ. A process and case study is provided in this article. However, one must perform runout measurements and vibration analysis before attempting in-situ balance correction. In addition, since most spectrum analyzers have fairly wide

tracking filter windows some fluid must be drained from the coupling to provide at least 40 RPM slip between the coupling input and output.

Missing keys, short keys, long keys or damaged keys generate unbalance which is a problem in many industries. For a 300 HP motor on a crusher, the typical short key calculated to generate about 100 lb_f unbalance.

Motor belt sheaves were typically installed using a taper-loc bushing. If the bushing bore is worn, radial runout of the sheave can result which generates unbalance force. Vibration at 1X shaft rotational frequency would also be generated by the sheave radial runout and fluctuating belt load.

4. Resonance: Resonance (excited natural frequency) was the 3rd most commonly identified problem with 51 occurrences, see distribution in **Figure 9**. Structural resonance was the most common and included motor mounting resonance on crushers, crusher motor support plate resonance, power unit (motor-fluid coupling housing-gearbox) flexural mode resonance, etc. The test stand resonance at gearbox input shaft speed was documented for certain size gearboxes.

The Rotor Critical Speed category was a very interesting problem area. The gearbox input shaft rotating assembly of several manufactures gearboxes operated near the 1st critical speed. Rotor modeling and modal analysis were used to investigate this problem. A case study is provided in this article.

Crusher drive belt resonance was only documented twice. The belt 2X RPM aligned with 1X the crusher shaft sheave rotational frequency exciting the belt resonance.
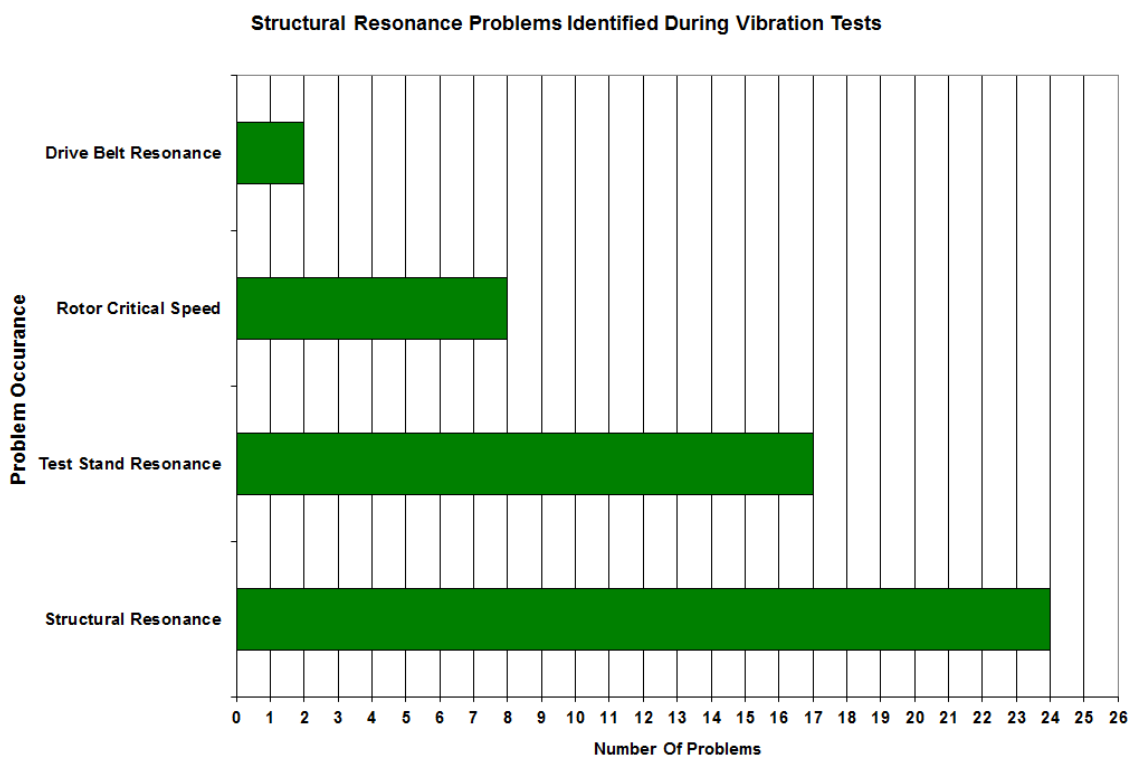
Figure 9. Resonance Category.

5. Runout: Excessive runout was the 4th most identified problem as show by the chart in **Figure 11**. The most frequently identified component with excessive runout was belt sheaves on either the motors or crusher shafts. Crusher sheave runout is indicated by the vibration data when motor vibration is primarily at the rotational frequency of the crusher shaft. Vibration at 1X the crusher shaft rotational frequency was elevated especially if the shaft was bent. Due to the environment in the mines, damage to the crusher sheave belt grooves frequently occurs; see the example in **Figure 10**. Since the sheaves, which also act as fly wheels) are at the edge of the crusher frame, the crusher shaft may be bent during movement in the mine if the crusher sheaves contacts the mine wall.
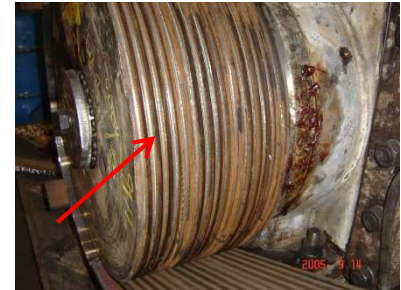
Figure 10. Coal Crusher Damaged Belt Sheave (Fly Wheel).

Excessive fluid coupling runout can generate vibration either at the drive end (motor rotational frequency) or output end (gearbox input shaft rotational frequency). The process of dial indicating the motor-fluid coupling rotating assembly and the gearbox input-shaft rotating assembly is described in this article in the case study – Fluid Coupling In-Situ Balancing. Since the fluid coupling input/output sections rotate at different RPM on their internal bearings, several problems can cause excessive fluid coupling runout and vibration as follows:

- Bent gearbox input shaft
- Fluid coupling output section cocked on the gearbox input-shaft fit
- Use of a two piece gearbox input shaft, i.e., shaft extension
- Worn bearings in the fluid coupling
- Fluid coupling internal runout
- Worn gearbox housing input shaft bores
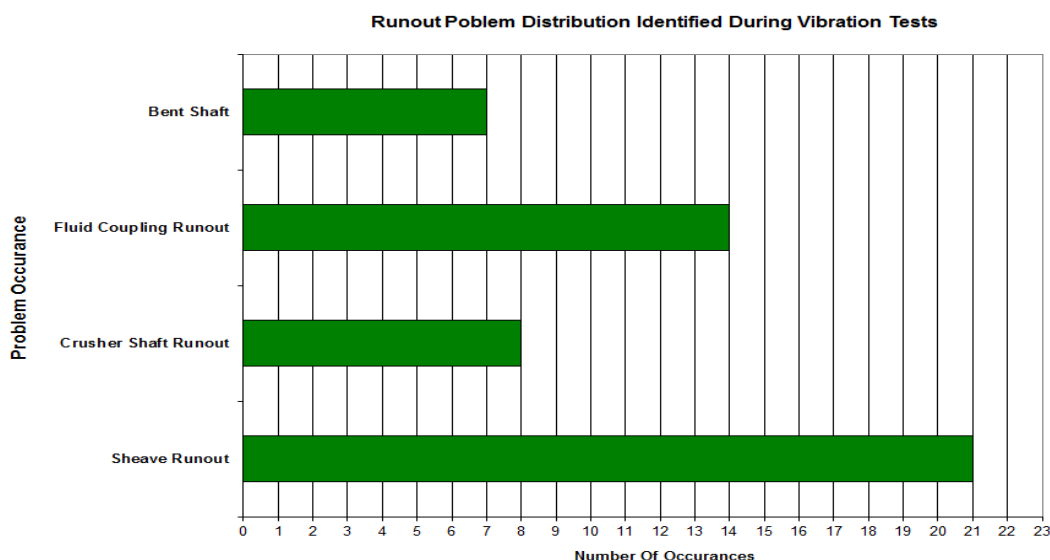
Figure 11. Runout Category.

6. Seals: Seal rubs were the 5th most identified problem, documented 22 times. Rubs were identified by vibration data (Peakvue is very sensitive to rubbing), as well as audible squealing. Inpro-Seals are typically used in Reliance motors and some crusher bearing housings. The motor belt-drive setup on crushers requires belt tension over 2000 lb$_f$. This high load on the end of the motor shaft causes deflection and can result in the seal rubbing if there is not adequate clearance between the seal rotor and seal stator, see **Figure 12**. Although the seals are advertised as non-contacting, internal clearances are small. Reliance was responsive in increasing clearance of the Inpro-Seals used in their motors for crusher application to reduce or eliminate the rubbing problem.

The crusher bearing housings are adjustable vertically using hydraulic cylinders and steel shims, see **Figure 13**. If the bearing housing guides are not perpendicular, an axial force on the seal can result in rubbing. If the crusher bearing housings are not accurately aligned in the horizontal plane so that the shaft is perpendicular to the bearing housings this can also result in the seal stator and seal rotor rubbing. The rubbing produces audible squealing and sometimes smoke. This is not necessarily a seal problem but a shaft to bearing housing alignment issue.
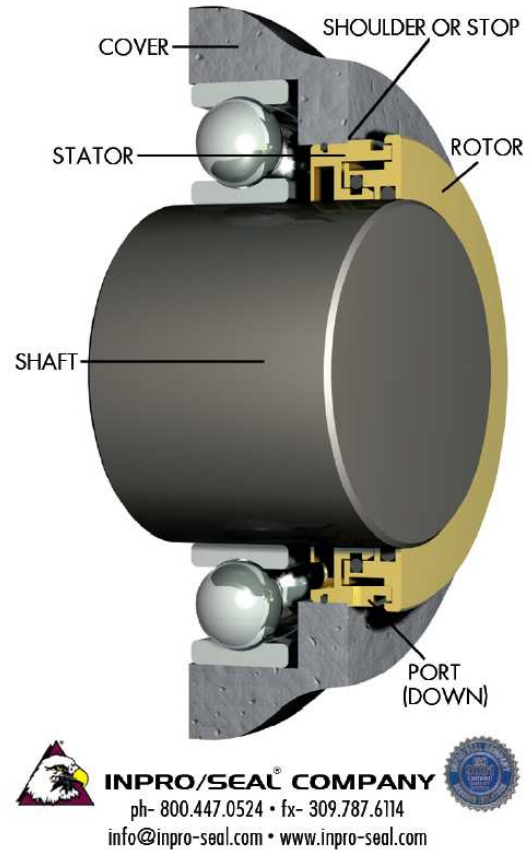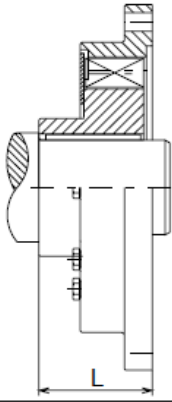

**Figure 12. Inpro-Seal Illustration.**[Ref 7]
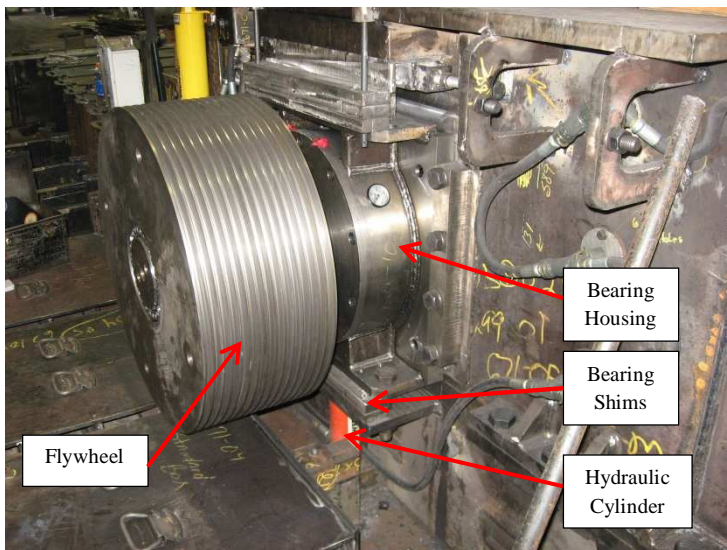
Figure 13. Crusher Drive End Bearing Showing Hydraulic Cylinder and Bearing Vertical Height Adjustment Shims.

7. Couplings: The Voith Fluid Coupling is most commonly used on mining equipment. It's connection on the input side (motor) per Voith is by a flexible coupling. A commonly used flexible coupling is the EPK coupling, **Figure 14**, **17 & 18**, which has elastomeric blocks in shear. Another flexible coupling in common use is by Tschan.[1] see **Figure 15**, which has elastomer in compressor. Falk Wrap Flex is also in common use.

The Voith turbo coupling [4] is a hydrodynamic coupling that works on the Fottinger principle. The main components consist of two bladed wheels, the pump impeller and turbine wheel, which are enclosed by a shell, see **Figure 15**. Both wheels are supported on bearings relative to each other. Power is transmitted virtually without wear since there is no mechanical contact between the power-transmitting parts. A constant amount of fluid is in the coupling. Mechanical energy provided by the drive motor is converted to kinetic energy of the operating fluid in the connected pump impeller. In the turbine wheel this kinetic energy is converted back to mechanical energy.

Figure 14: EPK (Flexible Pad Coupling). [4]
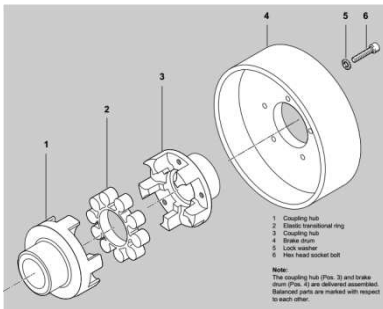
Figure 15. Tschan Elastomeric Coupling [1]
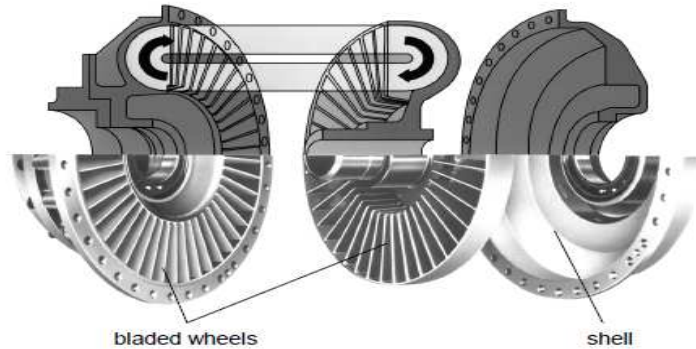
Figure 16. Voith Fluid Coupling Showing Pump Section and Turbine Section. [4]

**Figue 17. Voith Fluid Coupling Input Showing EPK Type Flexible Coupling Hub. [4]**

Figure 18. EPK Coupling Elastomeric Blocks Showing Measured Thickness.

D. Case Studies

Case Study: Motor with SKF 6224 Opposite Drive End Bearing False Brinelling. A motor bearing problem was identified from vibration data measured during an unloaded run test of a Gob Side Tail Drive power unit. The motor was 233/700 HP with ball bearings OB 6224 and IB 6226. Vibration data on the motor OB end showed Brg 6224 BPFO frequency 0.117 in/sec pk, see **Figure 19**. The 6224 BPFO was not present in Peakvue data, see **Figure 20**. Autocorrelation of the vibration time domain data showed the 6224 BPFO, see **Figure 21**. The motor is shown in **Figure 22** during one test supported on rubber belting. The motor shop advised that bearing frequencies were low amplitude before shipment and suspected shipping damage causing either brinelling or false brinelling. The motor was returned to the motor shop, the opposite drive end bearing inspected, confirmed damaged and a new bearing installed. The motor was shipped by truck back to the assembly shop.

The motor shop requested that the motor be tested as received on the shipping crate. The opposite drive end bearing again showed 6224 BPFO in the standard vibration data with amplitude 0.030 in/sec pk. This procedure (returning the motor to OEM) was repeated three times with each test showing BPFO of the outboard bearing in the vibration data. The conclusion was that false brinelling to the outboard bearing was occurring during transport by truck. The inability to lockdown the outboard bearing was allowing the bearing to move axially. Images of the damaged bearings are shown in **Figures 23-27**.
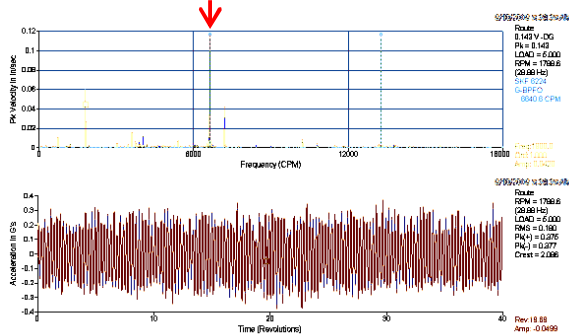
Figure 19. Motor Opposite Drive End Bearing, SKF 6224 BPFO 0.117 in/sec pk.
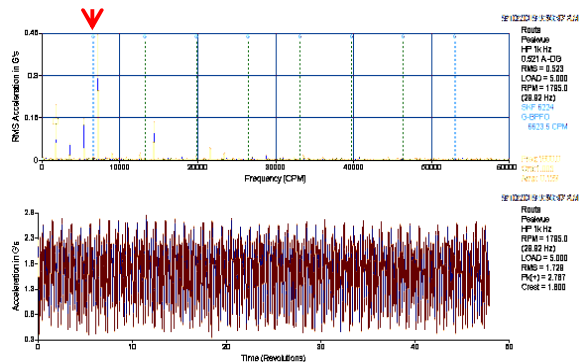
Figure 20. Motor Opposite Drive End Bearing SKF 6224 BPFO Not Present in PeakVue Data. 2X Electrical Line Frequency Highest Amplitude.
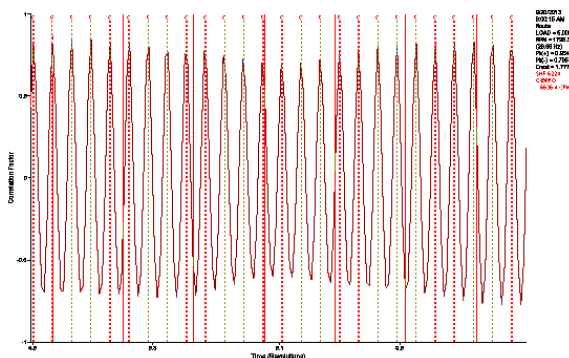

Figure 21. Autocorrelation Standard Vibration Waveform Showing SKF 6224 BPFO Primary Contributor.

Figure 22. Motor On Test Viewed From Outboard End.

According to SKF Publication *13459 Rolling Bearings and Seals in Electric Motor and Generator* [Ref 3], if a stationary bearing is subjected to vibrations, there is a risk that false brinelling will occur. False brinelling is the formation of shallow depressions in the raceways that will eventually lead to spalling and premature bearing failure. In cases where steel rolling elements were replaced by ceramic rolling elements, the bearings were found to be significantly less susceptible to false brinelling.
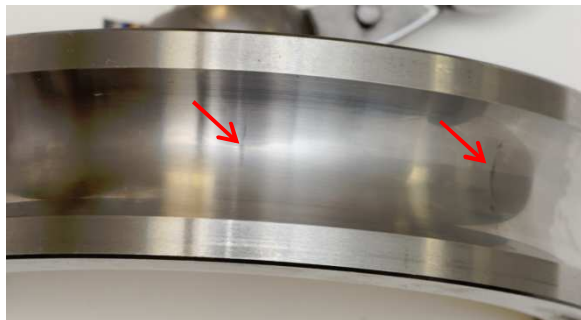
Figure 23. Motor OB Bearing, Inner Race False Brinelling.

Figure 24. SKF 6224 Outer Race Showing False Brinelling Damage.

Figure 25. Motor OB Bearing Inner Race False Brinelling.

Figure 26. Motor OB Bearing, Outer Race False Brinelling and Pitting.

Figure 27. Motor OB Bearing, Inner Race False Brinelling.

Case Study: Balancing Fluid Coupling In-Situ: Sometimes a fluid coupling can be balanced in-situ. The gearbox input shaft must not be bent and coupling runout should be less than 0.005 in TIR or 0.0025 inch eccentric. Based on test and inspection results, eccentricity of fluid couplings have measured as much as 0.030 inch TIR or 0.015 inch eccentric. The approximate forces resulting from eccentricity of the center of mass of the fluid coupling 0.015 inch from the center of rotation calculated as follows:

$$Fluid\ Clp_{Ounces} = 946lb \bullet 16oz\ /\ lbf = 15,136$$

$$Oz\ In_{Unbalance} = 15,136\ oz \bullet 0.015\ inch = 227\ Oz\ In$$

$$F_{lbf} = 1.775 \bullet 227 \bullet (\frac{1795}{1000})^2 = 1,298\ lbf$$

The life of the input shaft bearing (coupling end) carrying most of this dynamic load would be shortened.

Using the typical spectrum analyzer, it may be necessary to drain oil from the coupling to increase the slip frequency between the input and output sections. Typically at least 40 RPM is required for a spectrum analyzer tracking filter to accurately measure the 1X amplitude and phase vector.

A power unit consisting of the motor, fluid coupling and 2-stage planetary gearbox is shown in **Figure 27**. A laser tachometer senses reflective tape on the coupling input section, **Figure 28**. The power unit was supported on rubber and wood beams to lower the mounting resonance well below running speed, see **Figure 30**.

Washers were used as balance weights and inserted under the coupling bolt heads #3, #5 and #6 as shown in **Figures 28 & 29**.
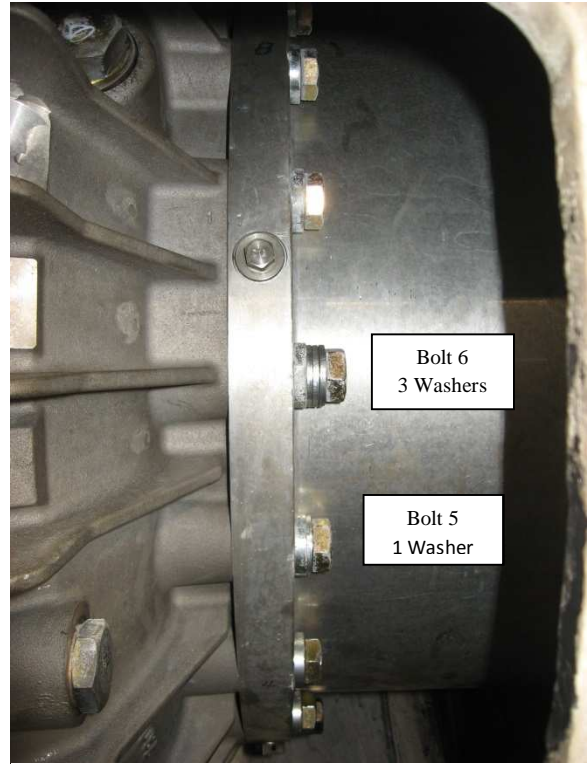

Figure 28. A Power Unit Setup to Trim Balance The Fluid Coupling In-Situ.
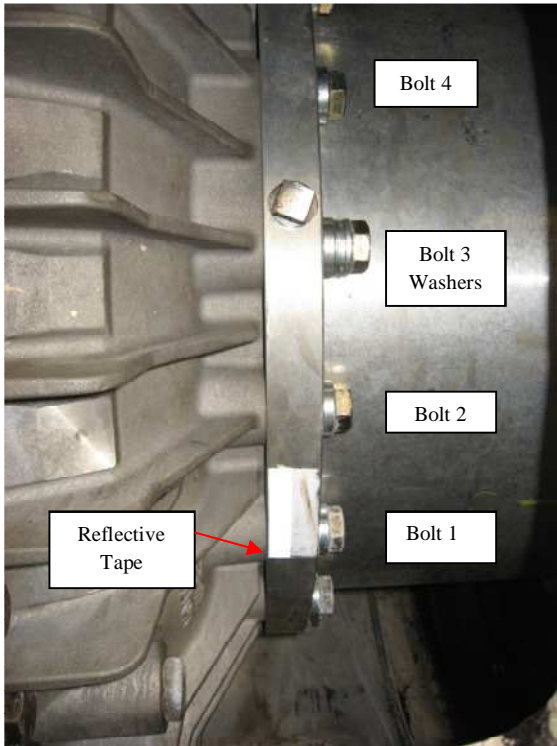
Photo 29. Reflective Tape Used as Laser Tachometer Trigger. Balance Weights at Bolt #3 (20 Bolts Total).

Photo 30. Balance Weights at Bolts 5 & 6.
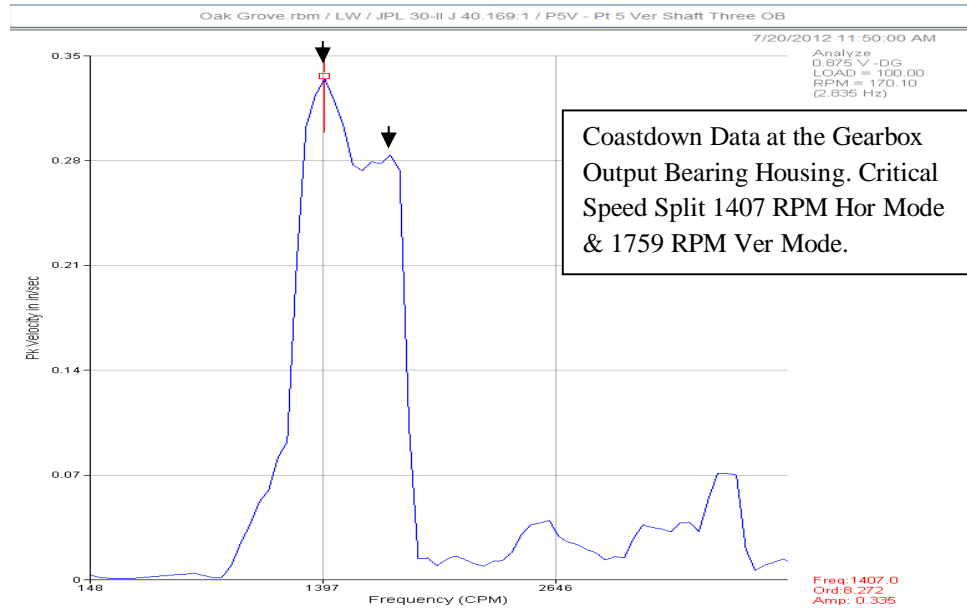
The balancing process was as follows:

    a. Drain about 75% of the fluid from the coupling to achieve at least 40 RPM slip.

    b. Mount the power unit on very soft support so that the mounting natural frequency is very low and does not act to amplify vibration at running speed.

    c. Install reflective tape adjacent to a bolt on the 21.65 inch bolt circle.

    d. Number the bolts against rotation beginning at the reflective tape with #1.

    e. Run the power unit and allow vibration to stabilize. This may require 10 to 15 minutes. Record the initial vibration amplitude and phase lag angle.

    f. Shut the unit down and install a trial weight (one to three washers under one bolt).

Figure 31. Two-Stage Planetary Gearbox Supported On Rubber Sheets and Wood Beams to Lower the Mounting Frequency for Balancing.

g. Run the power unit and allow vibration to stabilize. Record the vibration data for the trial weight.

h. Shut the unit down. Calculate the balance correction based on the trial weight influence. Install correction weight or weights.

i. Run the unit and measure vibration data. If additional trim balance is required repeat steps **e-h**.

Peak-hold data was measured during coast down to identify the input shaft critical speeds as shown in **Figure 32**. The 1st critical vertical mode was about 1759 RPM and the horizontal mode 1407 RPM. Split critical speeds are relatively common and are caused by asymmetry of the rotor support stiffness.

Coastdown Data at the Gearbox Output Bearing Housing. Critical Speed Split 1407 RPM Hor Mode & 1759 RPM Ver Mode.

Figure 32. Frequency Spectrum During Coastdown Showing Two Modes of the 1st Critical Speed of Gearbox Input Shaft/Fluid Coupling Rotating Assembly.

Frequency spectrum measured on the gearbox output bearing housing in the vertical direction before balancing and draining the fluid coupling is shown in **Figure 32**. After trim balancing on the fluid coupling, the vibration spectrum is shown in **Figure 33**. Vibration at 1X the gearbox input shaft rotating assembly measured 0.0189 in/sec pk. The motor and coupling input section 1X measured 0.0789 in/sec pk.

Note that fluid coupling runout can be measured as follows:

1) Dial indicate the motor and gearbox input shaft – maximum 0.001 inch allowable.

2) With dial indicator measuring the round diameter of the fluid coupling input section:

a. Holding the input section – rotate the output section of the coupling and record the TIR movement of the input section – 0.005 inch maximum allowed. This measures runout of the gearbox input shaft and the coupling bore to shaft fit.

b. Holding the output section – rotate the input section and record the runout TIR. This measures the coupling internal runout. This should not exceed 0.001 inch.
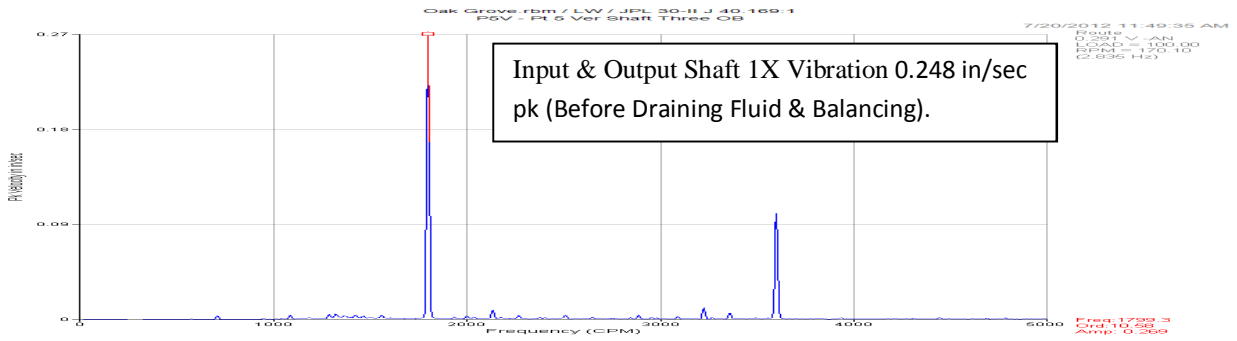
Figure 33. Frequency Spectrum at Gearbox Output Vertical Direction Before Draining Fluid From the Coupling and Balancing the Coupling Input Section.
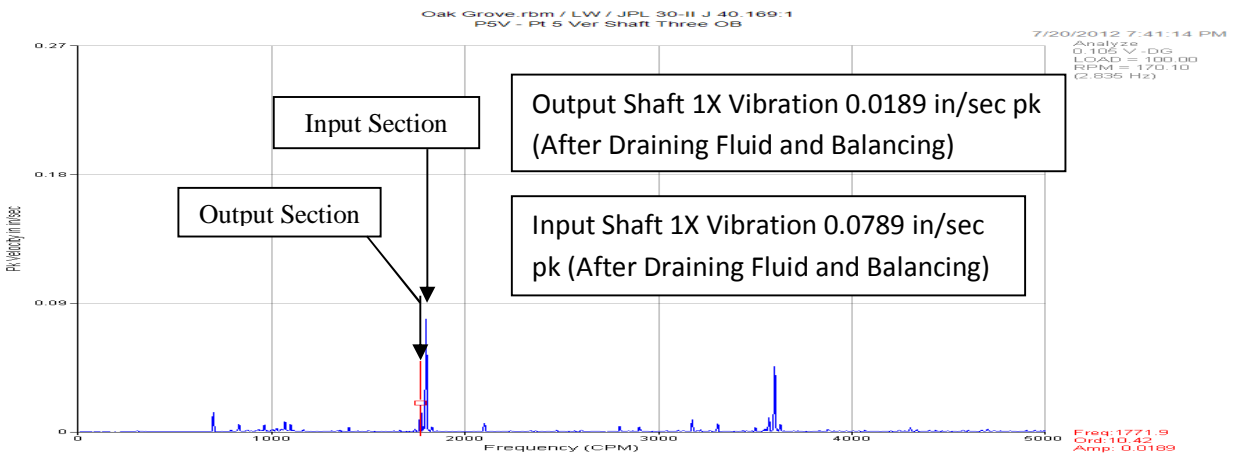

Figure 34. Frequency Spectrum at Gearbox Output Vertical Direction After Balancing the Coupling Input Section.

Case Study: Gearbox Input Shaft & Fluid Coupling 1st Critical Speed: The gearboxes for the power-units are primarily two types, 1) right angle drive parallel shaft with or without a single stage planetary stage and 2) two-stage planetary. The fluid coupling output section is an interference fit on the gearbox input shaft. The fluid coupling weight is typically >900 lb for 1000 Hp drive. The heavy fluid coupling combined with long overhang can create a 1st critical speed of the input shaft rotating assembly near the operating speed frequency. The result is elevated vibration of the power unit and increased sensitivity to unbalance and/or shaft runout. This problem has been documented on several brands of gearboxes.

The modal frequency response function (FRF), shown in **Figure 35**, for a L700BP Gearbox measured the 1st critical at 1725 RPM. The FRF, see **Figure 36**, on this double planetary gearbox coupling measured 1500 RPM. This unit would operate super critical.
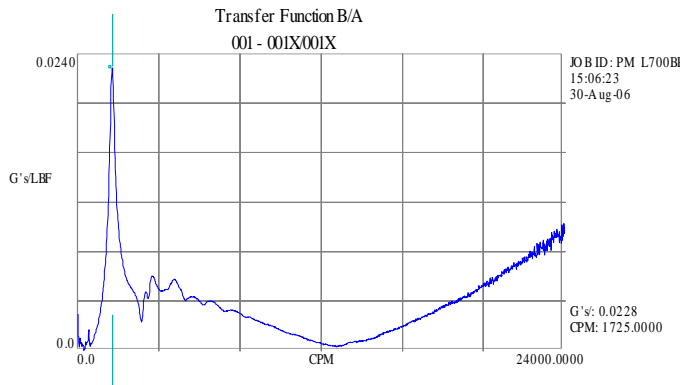
Figure 35: Modal Test Point at End Of Fluid Coupling, Input End, Horizontal Direction.
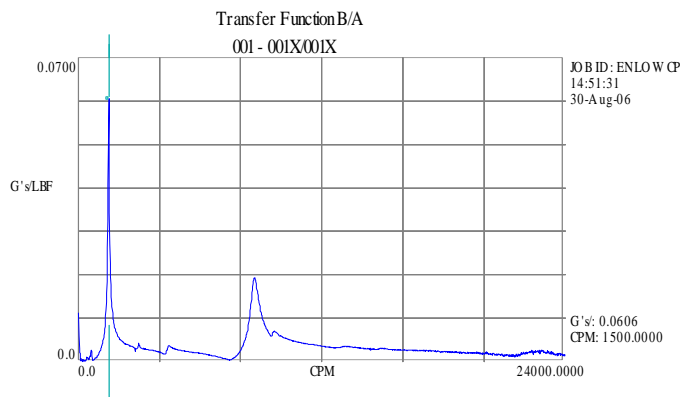

Figure 36. Modal Test Point at End Of Fluid Coupling, Input End, Horizontal Direction.

Rotor dynamic analysis was conducted using DyRoBeS FE software. Two models were developed as shown in **Figure 37**. The original shaft design used a shaft extension and two spherical roller bearings. The fluid coupling has much greater overhang with this design than the improved design shown in **Figure 37** right side. The undamped critical speed map is shown in **Figure 38** for the original and modified shaft design.
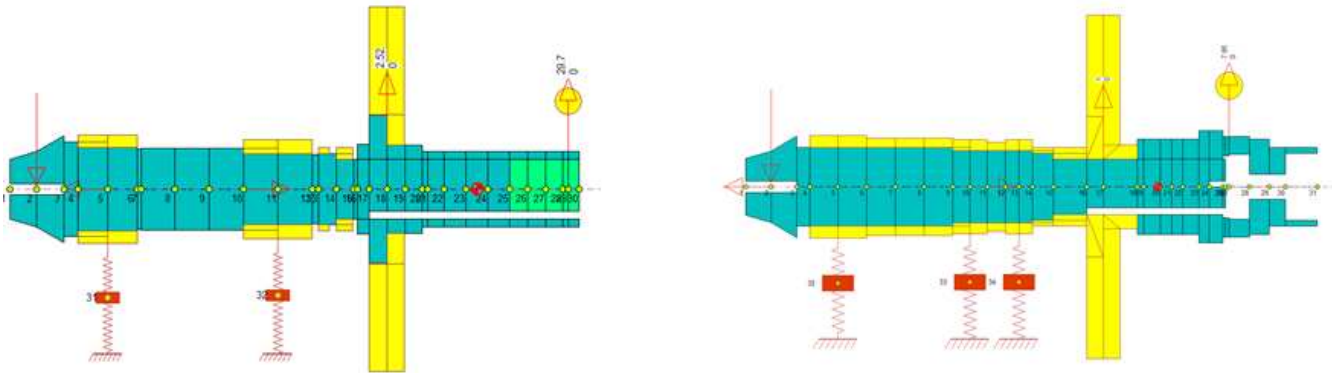
Figure 37. Model On Left – Original Design, Model on Right – Modified Shaft and Bearing Design.
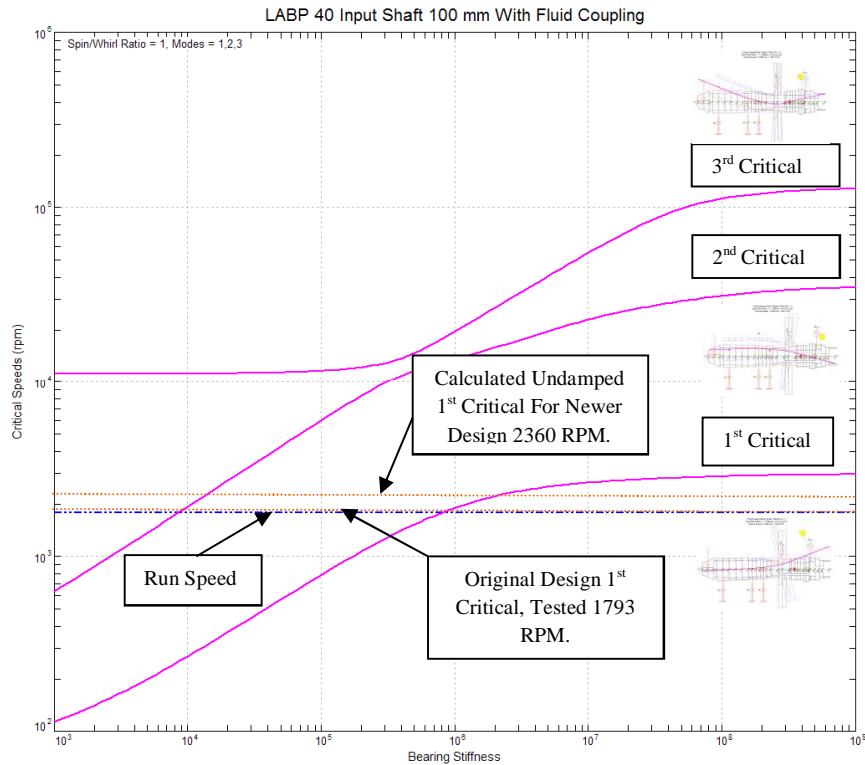
Figure 38. Input Shaft, 100 mm Diameter With Fluid Coupling, Undamped Critical Speed Map for Original and Modified Design.

The undamped 1st critical speed mode shape for the original rotor design is shown in **Figure 39**. Note that there is more flexure in the gearbox casing and less bending of the shaft (lower strain energy in the shaft). The 1st critical calculated to 1793 RPM. Since the 1st critical was almost exactly at run speed very high amplitue vibration resulted.

The undamped 1st critical speed mode shape for the modified design is shown in **Figure 40**. The stiffer bearings and gearbox case results in more bending of the shaft (higher strain energy in the shaft) primarily in the area of the spur gear fit. However, the 1st critical calculated to 2360 RPM or 565 RPM above the operating speed. The result was much lower vibration.

The damped critical speeds (which include the damping of the bearing oil film) will typically be higher frequency than the undamped calculations.

**Figure 39. Undamped 1st Critical Speed Mode Shape Shape 1793 RPM, Original Design.**

Figure 40. Undamped 1st Critical Speed Mode Shape 2360 RPM, Modified Design.

Comparison of modal FRF's measured on the fluid coupling of each rotor are shown in **Figure 41**. The static 1st natural frequency of the original rotor design measured 1734 CPM and 2334 CPM for the improved rotor design.
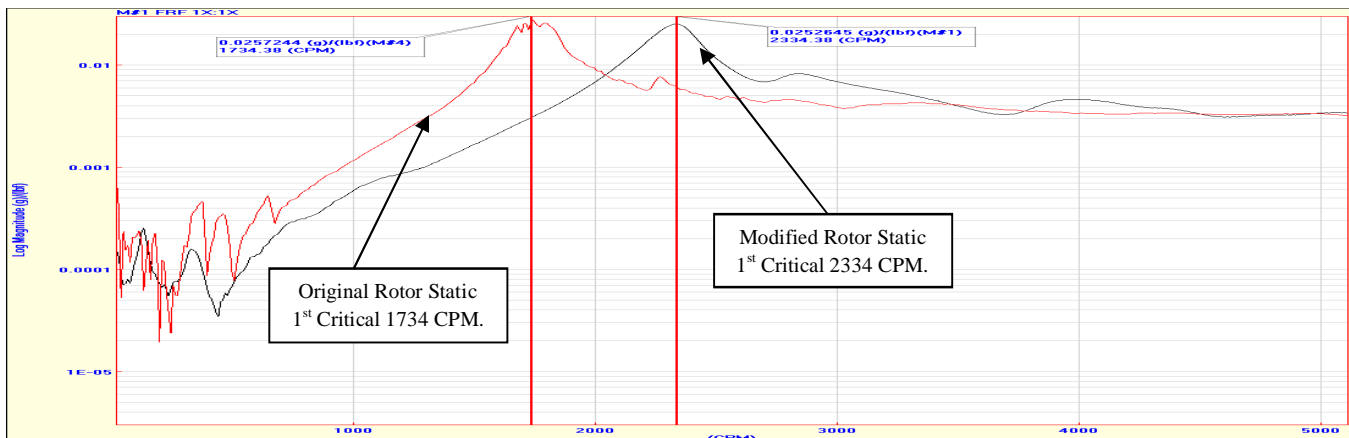

Figure 41. Modal Test Data Plotted Overlaid Original Rotor and Modified Rotor Design.

Comparison vibration data is shown in **Figure 42**, original shaft design and **Figure 43**, modified shaft design. The plots are log magnitude to enhance the ability to display the rotor 1st critical frequency.
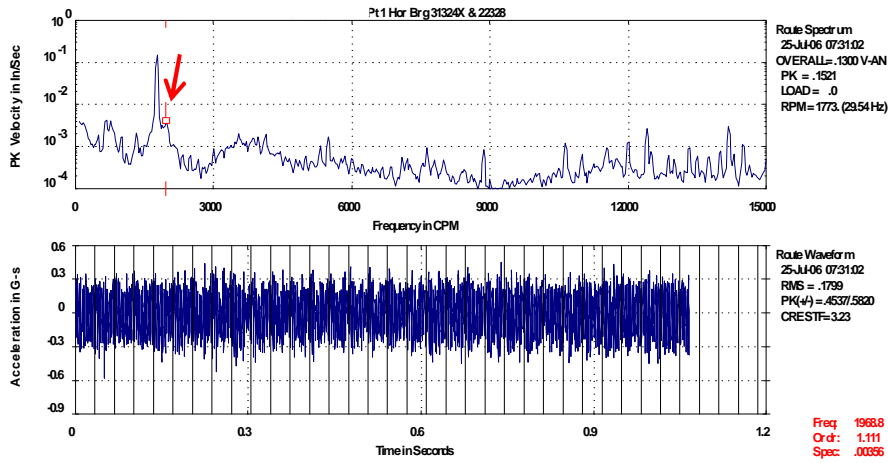
Figure 42. Original Gearbox Shaft Deign, Input Shaft Brg Housing Horizontal. 1st Critical Indicated at 1969 RPM. Rotor 1X = 1795 RPM
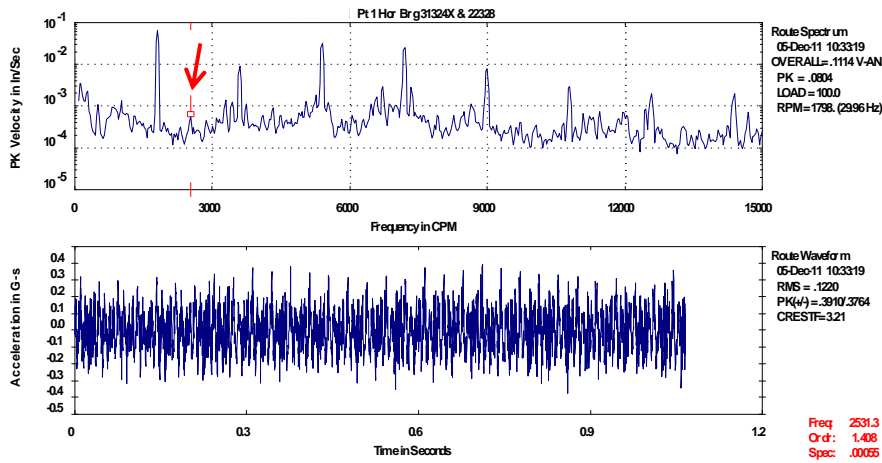
Figure 43. Improved Gearbox Shaft Design, Input Shaft Brg Housing Horizontal. 1st Critical Indicated at 2531 RPM.

Crusher Motor Mounting Designs: There are a variety of motor mounting designs for coal crushers. The most flexible motor mounting is <u>face mounted</u> to a cradle which is hinged, as shown in **Figure 46**. Sections of pipe are welded to the crusher frame and motor cradle with a piece of shafting inserted to provide a hinge. The drive belts are tensioned by a hydraulic cylinder and turn buckles. Very high amplitude motor vibration often exceeding 1.0 in/sec pk is common.

Another design bolts the <u>footed mounted</u> motor to a base plate; see **Figure 44**, which is rigidly bolted to the crusher frame. This design typically has very low amplitude motor vibration.

A motor <u>face mounted</u> to a cradle which is bolted to the crusher frame is shown in **Figure 46**. The base plate is moved by hydraulic cylinders to tension the belt, see **Figure 43**, then hold-down bolts tightened. This design has relatively low motor vibration even though face mounted. Mines often require crusher motors be face mounted rather than foot mounted so that the motor can be also be used on stage loader power units.
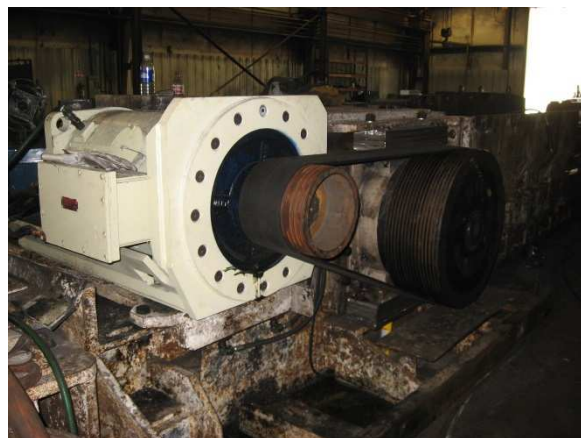

Figure 44. Crusher Motor Belt Tension Adjustment Using Hydraulic Cylinder.

**Figure 45. Motor Foot Mount Rigidly Bolted to Crusher Frame.**

Figure 46. Motor Face Mounted to Hinged Cradle.

Figure 47. Motor Face Mount to Cradle Which is Rigidly Bolted to Crusher Frame.

A less commonly seen crusher design supports the motor-coupling-gearbox on the end of the crusher shaft. The crusher shaft is splined and inserted into the hollow gearbox output shaft, see **Figures 48-50**.
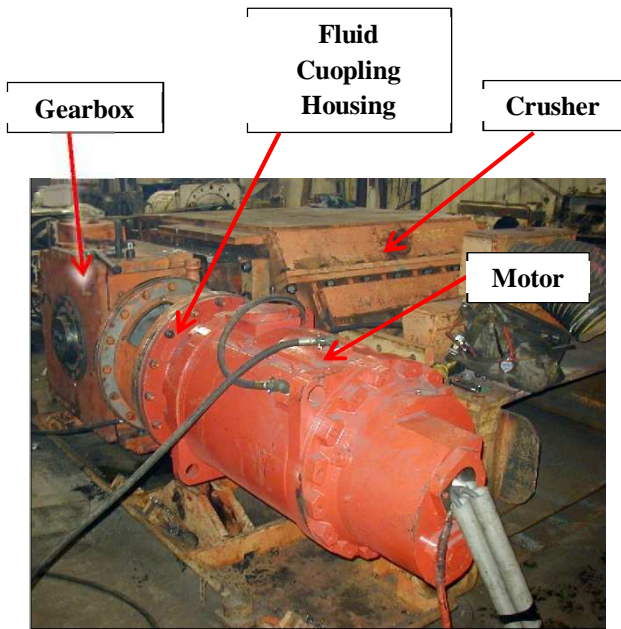
Figure 48. Crusher With Motor-Coupling Housing & Gearbox Supported on End of Crusher Shaft.

Figure 49. View Along the Motor Looking At Gearbox/Crusher Shaft Mounting.
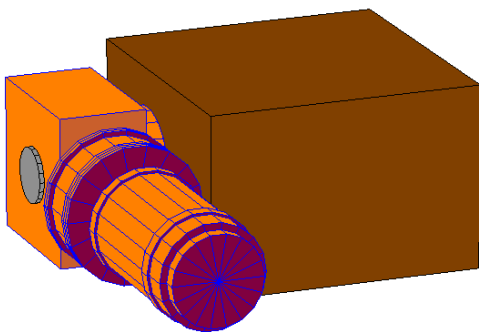
Figure 51. ME'scopeVES ODS Model of Crusher.

The crusher had very high amplitude vibration on test measuring 2.122 in/sec pk at 1X on the motor OB Horizontal. Motor OB End vertical measured 0.755 in/sec pk overall, see spectrum and waveforms in **Figures 53 & 54**.

Figure 50. Link Connecting Coupling Housing to Crusher Frame to Resist Rotation Torque.

A modal driving point measurement on the motor outboard end, **Figure 52**, identified two natural frequencies at 844 and 1107 CPM. The crusher motor 1X was 1795 RPM and the gearbox output shaft & crusher Shaft 450 RPM. This placed the crusher shaft 2X at 900 RPM or near the 1st natural frequency. ODS measurements were taken, see ME'scopeVES model **Figure 51 & 55**, although one

could visually observe the ODS since vibration amplitude of motor vibrating horizontally was 89.9 mils or about 3/32 inch. Bottom line, this design does not appear to be as robust as the belt driven designs.
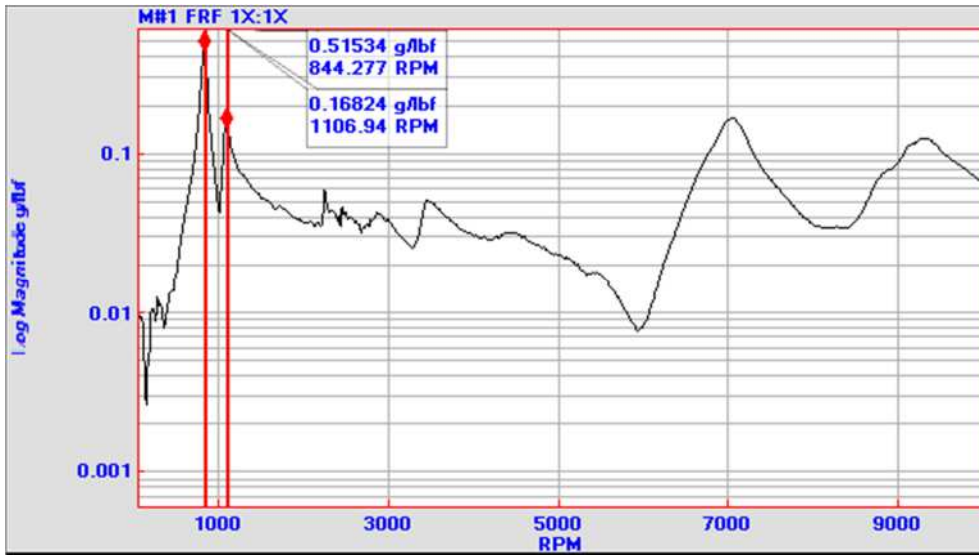

Figure 52. Driving Point FRF at Motor OB Horizontal Measured Two Natural Frequencies Within Operating Speed Range.
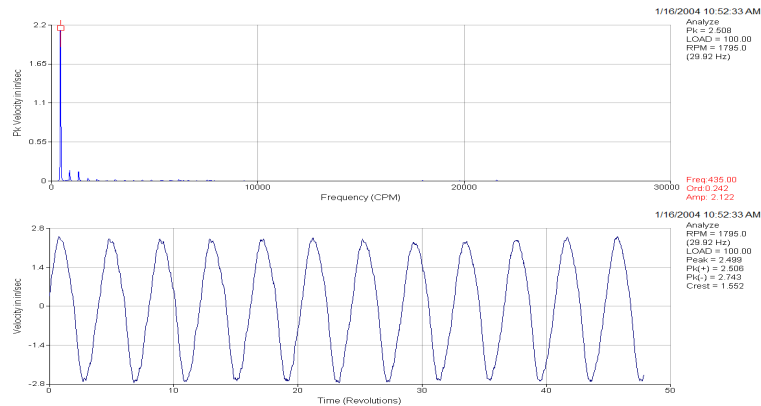

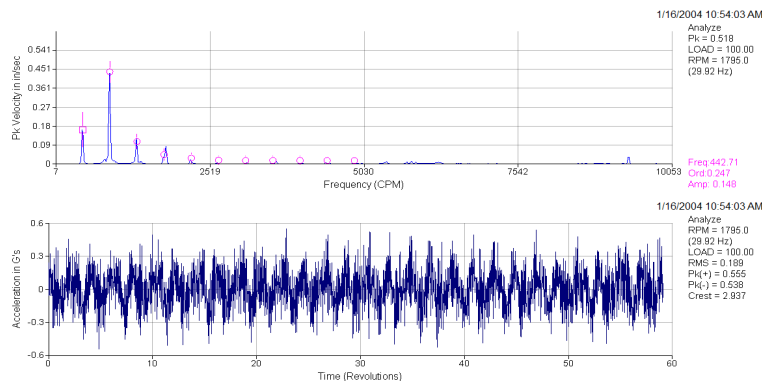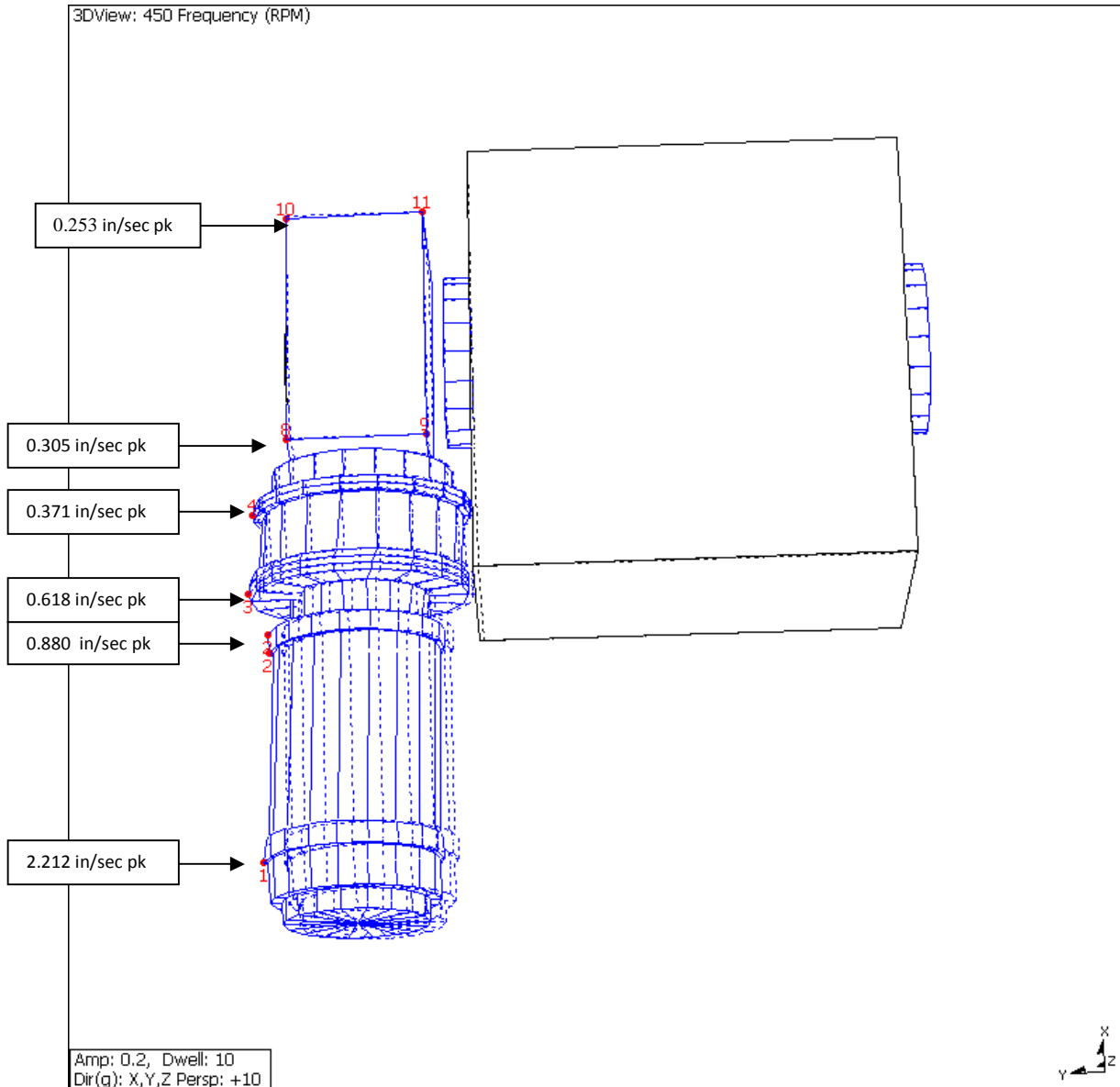**Figure 53. Motor OB Horizontal. Most Vibration at 1X=450 RPM, 89.90 mils or 2.122 in/sec pk.**

Figure 54. Motor OB Vertical, Most Vibration at 2X=900 CPM.

0.253 in/sec pk

0.305 in/sec pk

0.371 in/sec pk

0.618 in/sec pk

0.880 in/sec pk

2.212 in/sec pk

Amp: 0.2, Dwell: 10
Dir(g): X,Y,Z Persp: +10

Figure 55: The 2ⁿᵈ Run of Crusher With Overhead View of Deflected Shape at Frequency of Highest Vibration 1X Crusher Shaft 450 RPM. Most Flexure in the Coupling Housing and Flanges Connecting the Motor to the Gearbox.

Cast Study: Spirolox Retaining Ring Rub: During a run test of a coal crusher, a clicking sound was audible at the crusher opposite drive end bearing. The design used a Spirolox Retaining Ring, see **Figure 56**. The Peakvue data measured on the crusher opposite drive end bearing housing, see **Figure 57**, indicated impacting each revolution of the shaft. There was a primary impact once each revolution and a weaker impact twice each revolution. As discussed with the design engineer, the source was the Spirolox Retaining Ring impacting the split line of the bearing housing. The circular plot shown in **Figure 58**, indicates the once per revolution impact occurs about the same place each revolution. The secondary impact location occurring twice per revolutions does not appear to be as consistent in location.

Figure 56. Spirolox Retaining Ring (www.smalley.com)
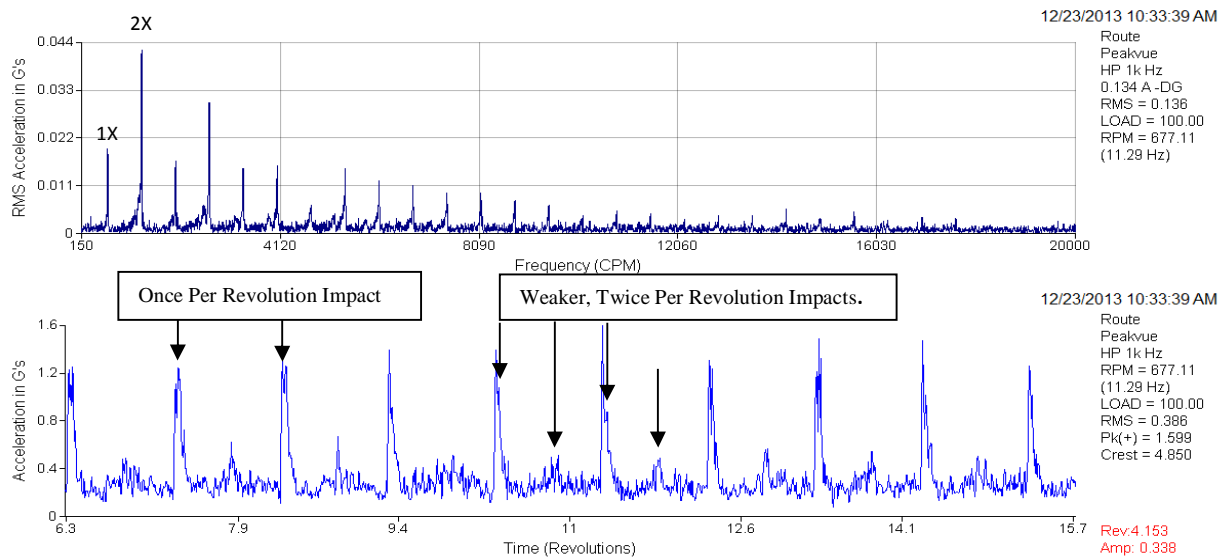
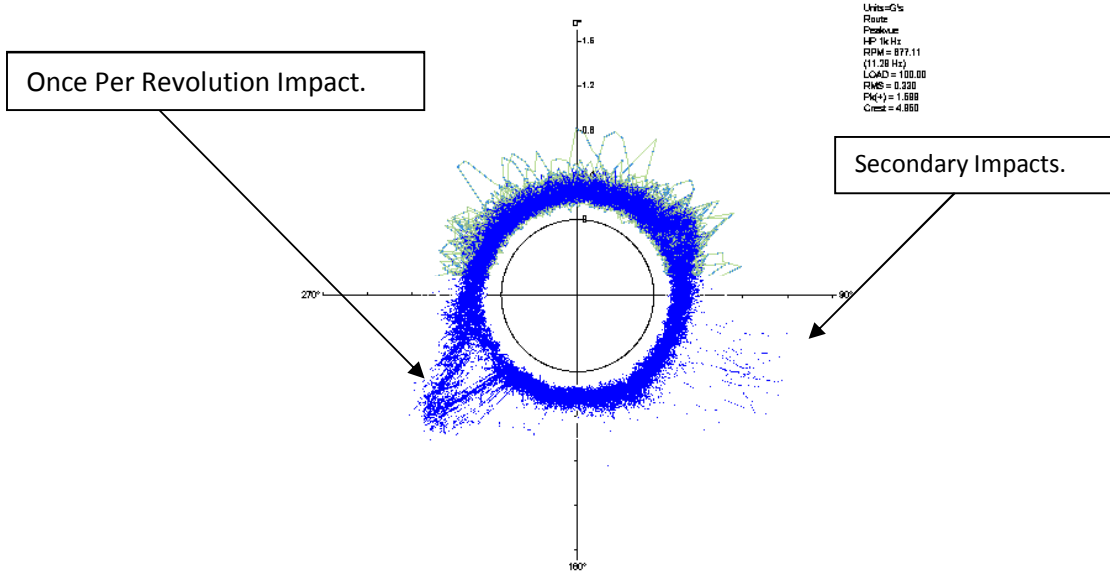**Figure 57. Crusher OB Bearing Housing, PeakVue Data Showing Impacting Twice per Revolution.**

Units=G's
Route
Peakvue
HP 1kHz
RPM = 677.11
(11.28 Hz)
LOAD = 100.00
RMS = 0.330
Pk(+) = 1.588
Crest = 4.850

Once Per Revolution Impact.

Secondary Impacts.

Figure 58. PeakVue Data on OB Bearing Housing Plotted in Circular Plot.

Crusher Shaft In-Situ Balancing: Coal crushers take tremendous abuse. The coal and rock coming from the longwall miner passes through the crusher for size reduction. The typical crusher has four rows of hammers with two hammer faces in each row. The rotor is adjustable vertically using hydraulic cylinders then locked in place using steel one inch thick shims, see **Figure 59**.

Occasionally a crusher will exhibit excessive vibration amplitude at 1X run crusher shaft during testing and will require balancing. The crusher shown in **Figure 59** required balancing due to very high vibration at the non-drive end bearing housing. During balancing, the crusher was supported on sheets of 3/4" thick rubber. The thickness of rubber sheets were adjusted to locate the machine mounting resonance well below the run speed, as shown by the bode plot in **Figure 61**.

Initial vibration data at the non-drive end was Hor 0.438 in/sec pk and Ver 0.168 in/sec pk. After balancing, vibration measured Hor 0.167 in/sec pk and Ver 0.085 in/sec pk, see **Figure 62**. The large weight welded to the outboard end was 2" X 4" X 10" = 22.64 lb_f. The weight calculated to generate 3036 lb_f. Trim balance weights of 2 lbf were welded to each end of the rotor, see **Figure 60**.
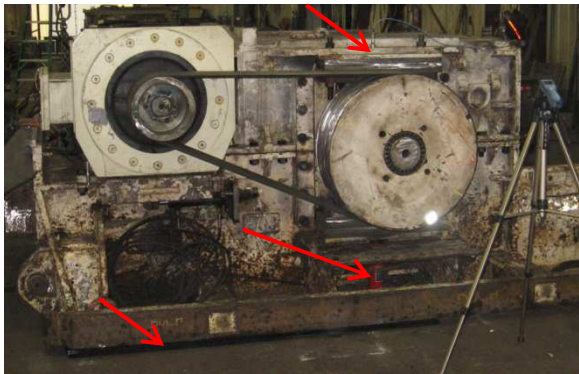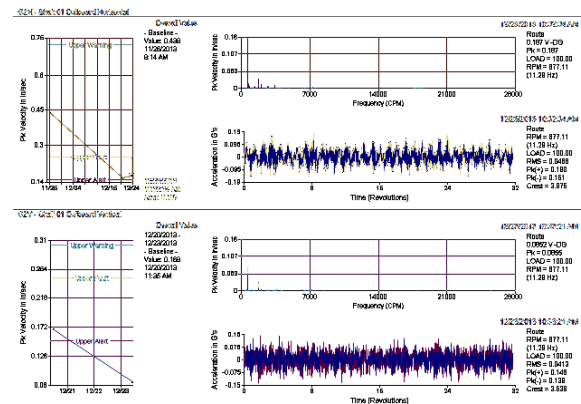

Figure 59. View of Crusher Drive End. Arrows at Steel Shims, Hydraulic Cylinder and Rubber Sheets.
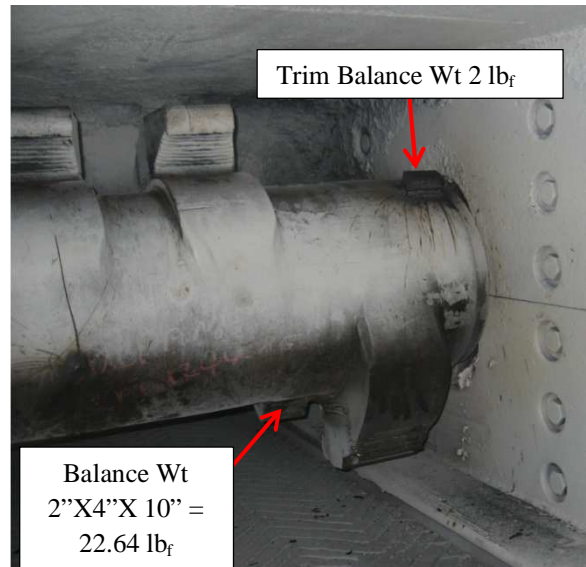
Figure 60. Crusher Rotor Showing Balance Weights Welded to the Rotor.
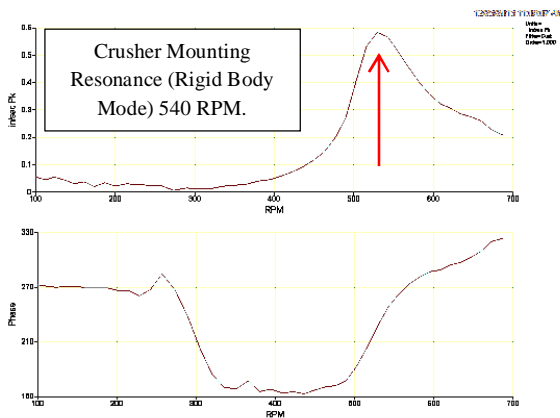
Figure 61. Rubber Sheets Placed Under the Crusher Shift the Rigid Body Mode Well Below Crusher Shaft Run Speed.

Figure 62. Crusher OB Bearing Housing Vibration Hor & Ver Before and After Balancing Rotor.

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Ken retired from Eastman Chemical Company December 1999 with over 32 years' service. At retirement he was Senior Engineering Technologist in the Rotating Machinery Group. Since 2000, he has provided consulting to over 70 plant sites in the US involving a variety of work including vibration analysis, experimental modal and ODS analysis, establishing condition monitoring programs, structural vibration analysis, rotor dynamic modeling, witnessed testing of reciprocating and rotating machinery, monitoring startup of rotating equipment trains and purchase specification development. Ken also teaches Applied Modal & ODS Analysis and Advanced Vibration Analysis with Jim Berry, President Technical Associates of Charlotte.

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4. http://www.rsa.voithturbo.com/media/011000en.pdf
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