MACHINERY VIBRATION RELATED TO POOR BOLTING PRACTICES

Ken Singleton, Manager
KSC Consulting L.L.C.
Bristol VA
ksingleton@vibrationconsulting.com

Bob McGinnis P.E.
McGinnis Mech Engineering L.L.C
Kingsport TN
mmengineering@charter.net

Abstract: Poor bolting practices are often identified as a source of machinery excessive vibration. The design and selection of foundation anchor bolts, skid hold down bolts, bearing housing and machine feet mounting bolts often do not receive the attention necessary to achieve reliable bolted connections. Problems such as bolt failure in fatigue, loose bolting, changes in machine alignment, etc., are common. Various techniques are discussed to identify bolting issues which include visual inspection, vibration data, and operating deflection shape analysis (ODS). Examples of machinery with poor bolting practices are provided. Definitions pertaining to bolting and general guidelines are provided.

Keywords: Alignment; anchor bolts; fatigue; Junker machine; ODS; vibration.

Introduction: The authors have spent many years analyzing machinery failures, most exhibiting high vibration or rapidly failing components. An on-site analysis typically begins with a bottom to top visual inspection of the machine or machines in question capturing areas of concern with notes, photos, and video. One issue seems to almost always be present when this in-depth inspection of the machine is done. Inadequate and downright poor bolting practices are identified in virtually every case. Examples of a broken anchor bolt is shown in **Figure 1**, missing shims, see **Figure 2**, thin flat washer yielded into a large hole, see **Figure 3**, and only the corners of a bolt bearing against a washer, see **Figure 4**. These findings are common. This may not always be the primary problem prompting the investigation, but it is one that can be prevented with knowledge and application of proper bolting techniques. Attention to this overlooked issue will also be a valuable tool in improving machinery reliability.

Let's begin by saying that you will find little to no specific bolting information provided in your installation and operating documents for the machine. Very little practical information is provided in machine design, maintenance reliability, or other technical reference books and publications. It is our goal, that at the end of this presentation, you will leave with new knowledge and an approach to solve some of the machinery reliability problems related to bolting you are so often confronted with or perhaps were even unaware that you have.

Examples of Problems Found

- **a.** Thin flat washers.
- **b.** Washers with over-sized inside diameters
- **c.** Bolts and nuts installed in oversize holes with improper flat washers or no washers
- **d.** Missing fasteners
- **e.** Broken fasteners
- **f.** Inadequate quantity of anchor and/or hold-down bolts
- **g.** Wrong grade of bolt and/or nut

h. Improper design to allow for bolt "free stretch"
i. Improper bolt/nut/washer design to prevent self-loosening
j. Improper design of attachment points for bolts
k. Poorly designed flexible machine and bearing frame assemblies
l. Improper size/length/installation foundation anchor bolts

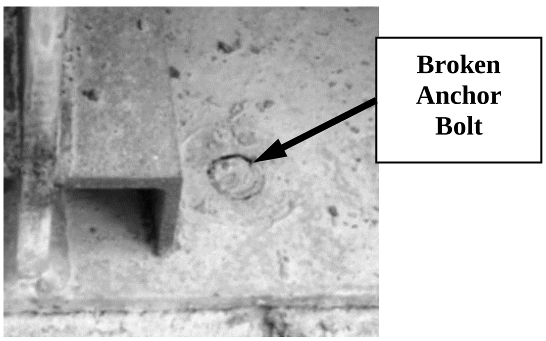


Figure 1. Broken Anchor Bolts on Fan Base.

Figure 2. No Grout Under Frame. Relative Movement Between Frame

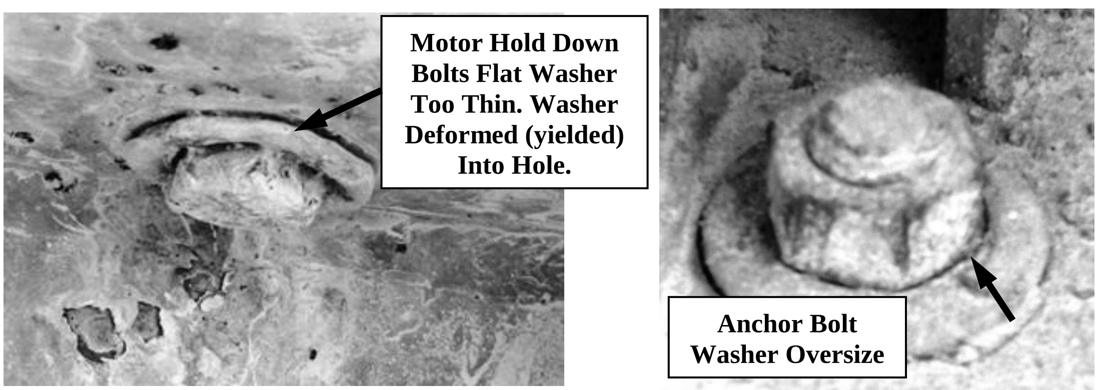


Figure 3. Washers Too Thin and the Washer ID Too Large (Only Corners of Nut Contacting the Washer).

Wrong bolt or nuts for the application - A bolted joint may load the fastener in tension or shear or both. The correct selection of the bolt, washers, nut or preloading device must be properly specified.

- All-thread rod should never be used for any foundation, baseplate, or soleplate fastener application.
- Do not use stud bolts as defined by ASME B16.5 standard for mounting or hold-down bolts. These are specifically designed for pipe flanges.
- Do use S.A.E. Grade 5, 8 or Metric Class 8.8, 10.9 bolts and nuts, unless otherwise specified. Bolts and nuts should be the same grade. Never use grade 2 or unmarked fasteners.
- Grade 5 and Grade 8 bolts should not be exposed to temperatures above 450°F or below -50°F. Expect a reduction in yield strength at elevated temperatures.
- Do not substitute ASTM A325, A490, F568M, or F1852 for S.A. E. fasteners. Fasteners made to these ASTM specs are specifically for structural applications.

- Fully threaded fasteners should not be used except on thin frame assemblies as shown in **Figure 4**.
- Threaded holes in base and sole plates for hold-down bolting should be used only as a last resort. An example is a 3,950 HP Motor bolted to the base plate using 1” bolts in tapped holes, see **Figure 5**. Cast, drawn, and rolled plates and machine components typically have but a fraction of the yield strength of grade 5 and 8 fasteners. Stripped threads are common.
- Coarse threads should typically be selected over fine threads. Fine threads have a tendency to strip.
- The largest bolt which will fit the assembly holes should be used. Where motor alignment is required the motor bolt holes should be larger than the base plate holes.

Figure 4. Full Tread Bolt for Thin Mounting Structures

- Thick hardened washers should always be used under the fastener head or nut. Place the washer under both or at minimum under the one which will rotate during tightening.
- Bolt threads should not extend into both the sole or baseplate and the machine mounting foot.
- The shear plane is the plane between two or more pieces under load where the pieces may move parallel to each other, but in opposite directions.
- The bolt threads may be included in the shear plane or excluded as shown in **Figure 6**.

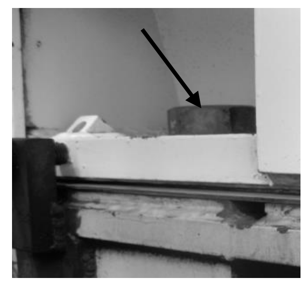


Figure 5. 3950 HP Motor, 1” Bolts in Base Tapped Holes.

- The capacity of a bolt is greater with excluded threads
- In addition to the clamping force to hold machines in position, shifting due to thermal growth or loading may impose axial and shear loads.

Fastening design fails to allow for bolt “free stretch” - It is important that bolted joints be clamped with enough force to prevent relative movement which can cause loosening of the joint as well as fretting corrosion. A bolt must also have adequate preload (stretched) to effectively clamp the joint. In order for the bolt to sustain the proper preload without failure in tension a minimum free stretch length is required. For anchor bolts, reference [9 pg 19], states minimum free stretch length of 7 bolt diameters for rotating machinery and 12 bolt diameters for reciprocating machinery. Adequate free stretch length is also important for hold down bolts on motors, bearing

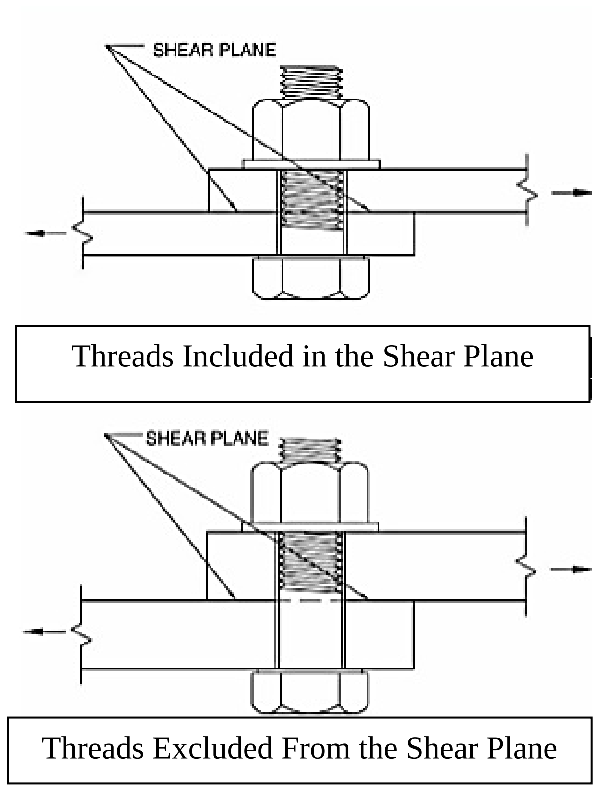


Figure 6. Bolt Threads May Be Included In Shear Plane or Excluded.

housings, machine cases, etc. If the bolt does not have adequate free strength length, the bolt may fail in tension or fatigue; see examples in **Figures 1 & 7**.

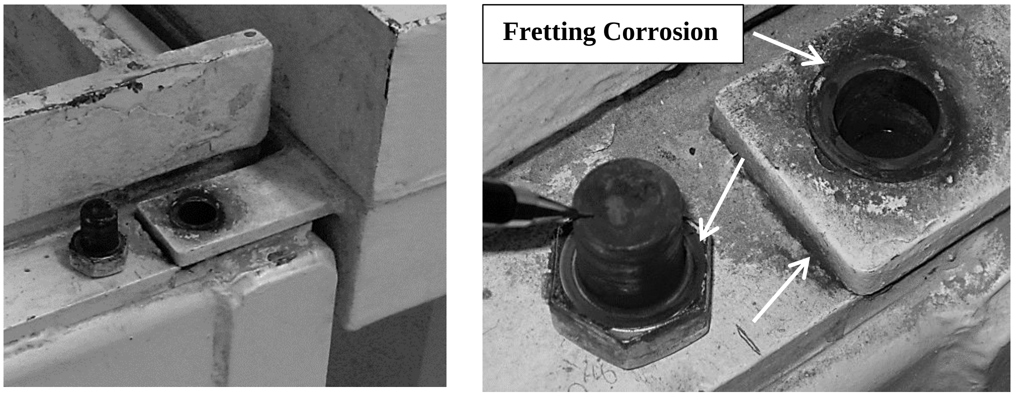


Figure 7. Machine Frames Bolted Together. Bolt and Frame Fretting Corrosion Indicated Looseness (Arrows). No Flat Washer. Bolt Failed in Fatigue Which Was Indicated by Beach Marks and Final Rupture Zone.

A bolt's free-stretch length is the distance between the nut (actually several threads into the nut or tapped hole) and the head, see the illustrations in **Figures 8 & 9**. The length of the thread engagement in the nut or tapped hole should be sufficient to break the bolt in tension, reference [10 pg 52]. The nut wall thickness and length must also be taken into consideration.

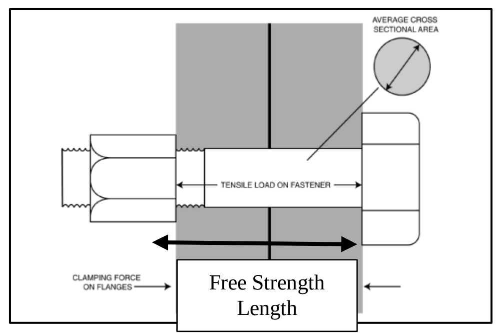


Figure 8. Free Stretch Length of Bolt, Reference [16 pg 3].

Tensile strength is the mechanical property most widely associated with standard threaded fasteners, reference [15]. The tensile strength is the maximum load in tension that the bolt can support prior to fracture. The load in tension that a bolt can withstand is given by equation **(1)**.

$$\boldsymbol{P = St\ X\ As} \qquad \textbf{(1)}$$

Where:
P = Tensile load or Clamp Load lb_f
St = Tensile Strength psi
As = Tensile Stress Area in^2

For example, for a 3/4" SAE J429 Grade 5 HCS bolt:
St = 120,000 psi
As = 0.3340 in^2

P = 120,000 psi x 0.3340 in^2 = 40,080 lb_f

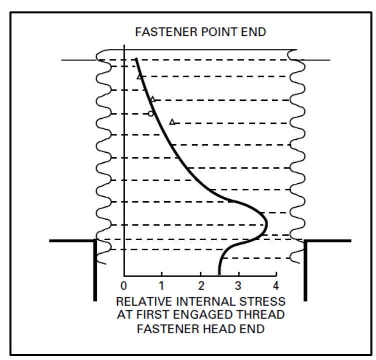


Figure 9. Relative Internal Stress Of Fastener Threads, Reference [10 pg 55].

A tensile stress-strain diagram is shown in **Figure 10**, [15 & 16]. If the bolt is deformed or stretched by applying load in tension (stress), as long as the load does not exceed the Yield Point, the bolt will return to its original shape after the load is removed. The Typical Clamp Load is taken as 75% of the Proof Load. The Proof Load is typically 85-95% of the material Yield Point. If the bolt is subjected to a load exceeding the material Yield Point (in threaded section of bolt), it will neck down and elongate further with a reduction in stress. It will not return to its original shape once the load is removed. Strain is the amount of elongation divided by the original length of the stressed section.

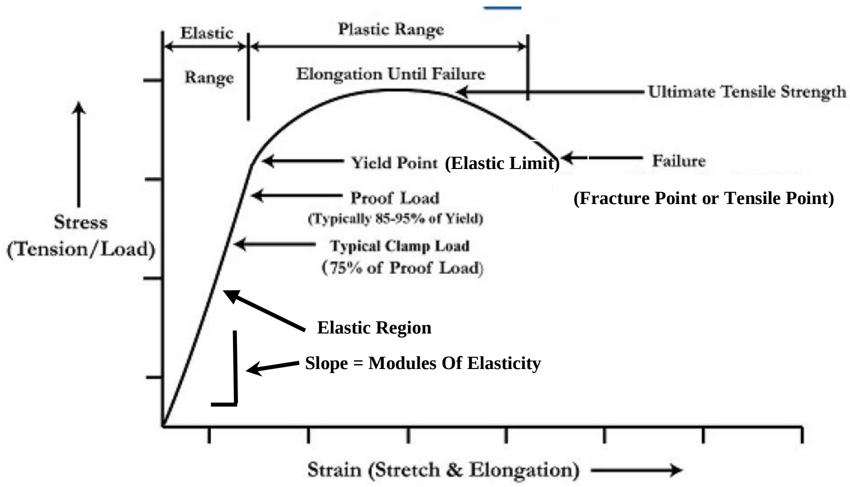


Figure 10. Tensile Stress-Strain Diagram, Ref [15, 16].

Stress/strain curves for some materials used for bolts are shown in **Figure 11**, [15]. Note that harder, higher tensile strength fasteners such as ASTM A574 Socket Head Cap Screw (SHCS) and SAE J429 Grade 8 are less ductile than lower strength SAE J429 Grade 5 and ASTM A307 Grade A. Although the ASTM A574 and SAE J429 materials have high tensile strength, the overall length of the strain curve is decreased when compared to materials J429 and A307.

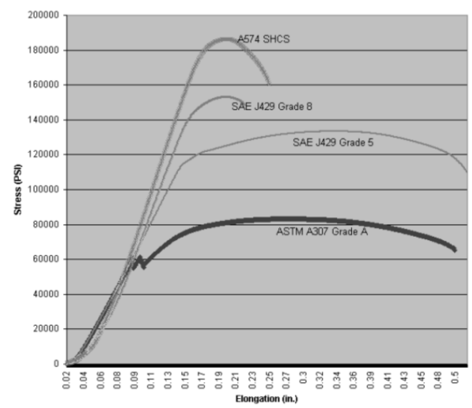


Figure 11. Stress/Strain Curve for Different Materials, Reference [15].

Washers, thick wall pipe with ends machined, etc., may be used to provide the specified bolt free-stretch length. Some examples are shown in **Figures 12 & 13**.

Very high local pressures in the contact areas on the threads and the nut and bolt head contact surfaces occur when the bolt is tightened. Local plastic deformation at these interfaces results in flattening of surfaces. The <u>flattening is also called embedding</u>. The amount of embedding influences the reduction of bolt stretch or preload, [14].

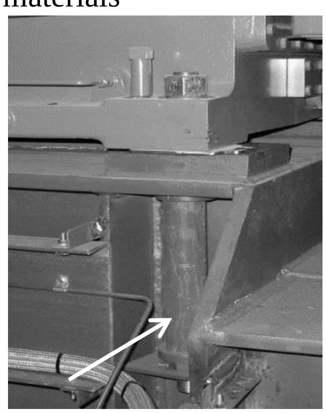


Figure 12. Recip Compressor Crankcase Hold-Down Bolts Using Pipe to Provide Adequate Free-Stretch Length.

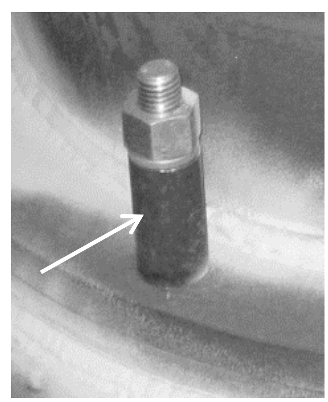


Figure 13. Pressure Vessel Hold-down Bolt on Recip Compressor Skid, Use of Pipe to Provide Adequate Bolt Free-Stretch Length.

Improper torque applied to bolt or nut - Torque is probably the most often used method of tightening a bolted joint after the “tighten by feel” method. The common torque wrench is quick and easily implemented. However, according to [Ref. 31, 32 & 33] a torque wrench may only be accurate to +/-35%. Torque generates parasite torsion stress, see **Figure 14** [33 pg 9], in the bolt or stud which may result in failure of the bolt before adquate preload occurs. Only about 10% of the torque typically is used to tension or stretch the bolt, see **Figure 15**. Most of the torque is expended overcoming friction, 50% under head and 40% threads according to reference [36 pg 5].

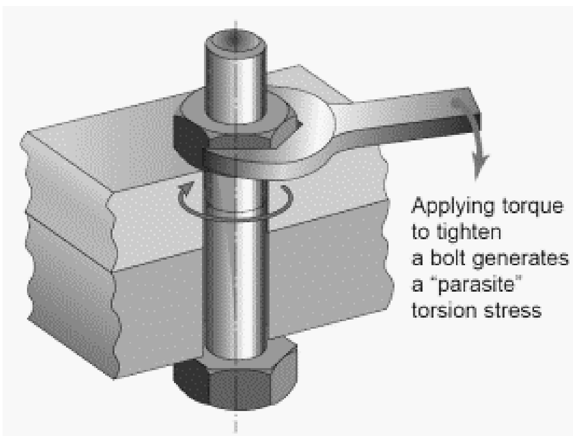


Figure 14. Applying Torque Generates a Parasite Torsion Stress [33 pg 9]

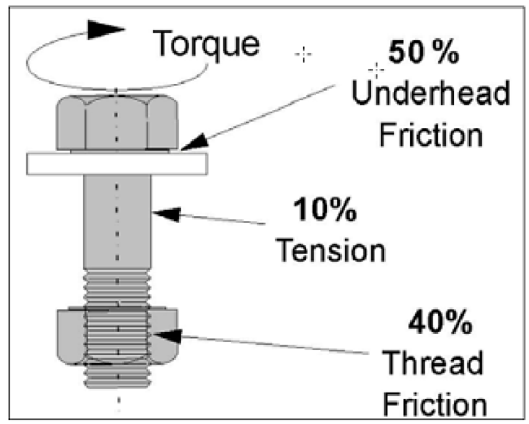


Figure 15. Where Does The Applied Torque Go? [36 pg 5]

There are many portable torque wrenches available which include hydraulic, impact, tools with pneumatic or electric adjustment, impact wrenches with stored energy via a torsion bar or other means. Many of the portable torque wrenches can be calibrated using a Skidmore-Wilhelm gage, reference [34] or Torque-Tension Research Head (load cell), reference [36 pg 9]. The Skidmore-Wilhelm gage and Torque-Tension Research Head measure the tension or pre-load of the bolted assembly. For critical applications, testing several bolt-nut-washer assembles provides a method of calibrating the torque wrench or tool to significantly reduce bolt tightening error.

Another method of achieving bolt preload is Torque-Angle Monitoring With Control, references [31, 36], see **Figure 16**. Torque is applied until a specified threshold level is obtained then the joint is tightened by rotating the nut a specified angle. This method is reportedly accurate <=5% in the joint elastic clamping zone per reference [36, pg 12-13]. Calculations to determine the nut rotation angle in the elastic clamping zone for a ¾-10 bolt, see **Figure 17**, showed that if the bolt free-stretch length was only 1 bolt diameter, the nut rotation angle was less than 2 degrees. If bolt preload relaxation was 0.002 inch, the bolted assembly would become loose (all preload lost) for

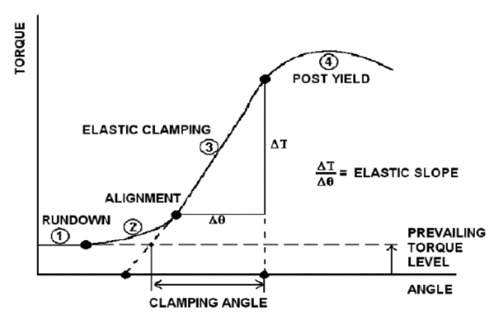


Figure 16. Torque-Angle Monitoring With Control, [31, 36 pg 15].

bolt diameters 1 thru 3. Therefore, an advantage to long free-stretch length of 7 or more bolt diameters is that the nut angle of rotation is greater and thus easier to measure and the potential error decreases. The longer bolt also retains more preload if the joint relaxes.

3/4-10 Bolt	Bolt Grip Length Inches	Preload psi	Bolt Stretch Inch	Remaining Pre-Load lb_f Relaxation= 0.002 inch	Nut Advance in Degrees (In The Elastic Clamping Zone)
1 Bolt Diameter	**0.75**	**60,000.00**	**0.0005**	**None**	**1.6**
3 Bolt Diameters	**2.25**	**60,000.00**	**0.0014**	**None**	**4.9**
7 Bolt Diameters	**5.25**	**60,000.00**	**0.0032**	**6,832**	**11.5**
12 Bolt Diameters	**9**	**60,000.00**	**0.0055**	**11,594**	**19.7**

Figure 17. Calculations for Nut Rotation Angle to Achieve 60,000 PSI

Absence of flat washers where oversize holes exist – Although this may seem improbable, we have found bearing housing and motor hold down bolts installed with no flat washers. In a few instances, only the corners of the nut were actually contacting the base plate as shown in **Figure 18**.

Figure 18. Motor Hold down Bolt, No Washer.

Wrong type of flat washers – Flat washers are typically used as the contact surface for the bolt head or nut. As the nut is tightened, galling may occur so the washer serves as a sacrificial surface. The washer also spans the hole through which the bolt is inserted. One problem we have documented is that thin flat washers often yield into the hole (dish), as shown in **Figure 19**, which results in loss of bolt pre-load. The dishing also has a centering effect (side load on the bolt) which has a tendency to cause the motor, bearing housing, etc., to move. This movement can result in the machine moving out of alignment. Consider using hardened extra thick flat washers, reference ASTM [45] see table in **Figure 20**. Up through 1 ½ in, the washers are through hardened. A stiff washer with large diameter increases joint stiffness. The strength of the washer must be at least equal to the strength of the bolt. These are available from many sources such as Grainger and Fastenal.

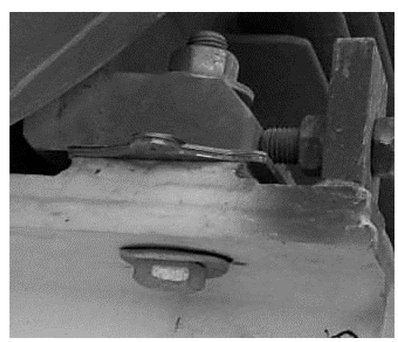


Figure 19. Motor Hold Down Bolts, Oversize Flat Washers Deformed Into the Base Plate Holes.

Nominal Size	Nominal OD	Nominal ID	Min Thickness (in)	Max Thickness (in)
½	1.063	0.531	0.305	0.375
5/8	1.313	0.688	0.305	0.375
3/5	1.468	0.813	0.305	0.375
1	2.000	1.063	0.305	0.375
1.125	2.250	1.188	0.305	0.375
1.250	2.500	1.375	0.305	0.375

Figure 20. Extra Thick Hardened Flat Washers, Ref ASTM [45].

These are available from many sources such as Grainger, Fastenal, etc.

Figure 21 Split Lock Washer Reference [12].

Use of split or lock washers –According to reference [12], work completed during the 1960's in Germany found that the effectiveness of locking devices such a split lock washers actually aided loosening. As a result, the conventional spring lock washer, shown in **Figure 21**, is no longer specified. Consider removing split lock washers from your stores inventory.

Expecting bolting to remain properly tensioned when high vibration is normal – In a paper published by Junker reference [4], it was discovered that lateral vibration applied to a bolted assembly would cause self-loosening. The Junker Machine (named after the inventor) was developed in Germany in the 1960s and is still used today to test various fasteners. Under high vibration, most fasteners will self-loosen; see reference [22] which has a link to a YouTube video. To address self-loosening, thread locking devices were developed and can be grouped into three categories: 1) Free Spinning, 2) Friction Locking and 3) Chemical Locking.

- **a.** Free Spinning type uses a plain bolt with a circumferential row of teeth under the washer head. These teeth are ramped and allow the bolt to rotate in the clamping direction but lock into the bearing surface when rotated in the loosening direction. Brand names include Nord-Lock [21], Oglaend, Disc-Lock, etc.
- **b.** Friction Locking is divided into two groups; metallic friction locking and distorted thread. The Philidas nut and Nyloc nut are included in this group. The distorted thread includes Stanley Spiralock, reference [24].
- **c.** Chemical Locking includes adhesives which fill the gaps between the male and female threads bonding them together. Chemical Locking provides the greatest resistance to vibration loosening, reference [12 pg 3]. Loctite, reference 42, is a well-known brand.

Excessive quantity of shims – Stainless steel shims are now commonly used (since the 1980's) when aligning machines. Occasionally, primarily in older plants, we still find carbon steel shims in use. The carbon steel shims are typically heavily corroded. Shims act as springs and thus tend to soften the bolted joint especially a stack of shims like that shown in **Figure 22**. The bolt should always be the weaker element of a joint in order to maintain bolt preload. Consider using as few stainless steel shims as possible under machine feet (3 to 5) and a single solid shim under cast bearing housings.

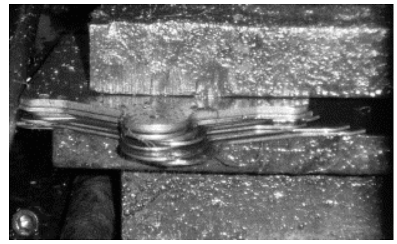


Figure 22. Too Many Shims at Corner of Gearbox.

Poor grouting practice related to anchor bolts – This remains a serious issue even today after many years of seminars and published papers providing detailed information on proper grouting. There are primarily two types of grout, cementitious and epoxy. Cementitious is typically used as filler which holds the shims in place upon which the soleplate or skid rests. The soleplate or skid is clamped by anchor bolts to the concrete foundation with shims at each anchor bolt. Epoxy grout, reference [9] (when properly used), glues or bonds the soleplate or skid to the concrete foundation. Where epoxy grout is suitable, (not for use in high temperature area such as exhaust end of gas turbine) it is considered superior to cementitious grout due to compressive strength of 14,000 psi and 2,100 psi tensile strength per reference [38]. But, whichever type grout is used, the anchor bolt must have adequate free stretch length, reference [9, pg 19]. The grout, shims and soleplates should handle the compressive stresses imposed on them, reference [39, pg 24].

Thin base plates or soleplates - For most non-API equipment, the soleplates and base plates were deficient in size and design. Charlie Jackson, reference [41] stated that steam turbine soleplate thickness should be minimum 3” thick, epoxy grouted to a monolithic poured re-bar re-enforced concrete mezzanine. The soleplate should be larger than the turbine base by 6” or more on the outer perimeters. Coat tops of soleplates with 0.5 mil electroless nickel, reference [43], against corrosion. .

Bolt Tightening Methods – Generally there are two methods, torque or tension. Torque is used to rotate a nut, stretch the bolt and clamp the joint. Tension is implemented using mechanical devices, reference [20], or hydraulically actuated tools, reference [33], to stretch the bolt without rotation. A nut is typically spun down and snugged while the bolt is under tension and stretched. When the force applied to the bolt is removed, the nut maintains the bolt stretch. There are many problems with torque because the majority of the torque is expended in friction. Tension is more accurate but also more expensive due to the tooling or the type of nut used

Mechanical Chocks: Vibracon reference [13], see **Figure 22**, and ExactAlign reference [44]. There are adjustable supports for use under diesel engines, natural gas engines, motors, etc. The advantage is easier alignment changes since shimming is not required

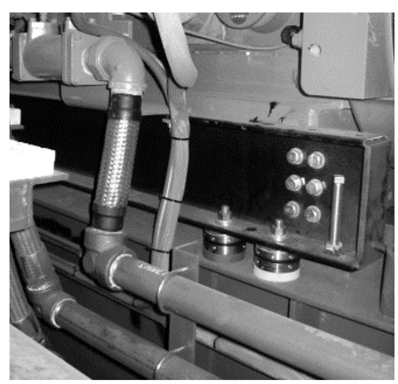


Figure 23. Vibracon Chocks, Reference [13].

Wedge Locking and Friction Based: Ref [17]: The principle of friction locking is based on the increased friction in the thread and the surface of the bolt head/nut. The primary disadvantage of assembling dry or unlubricated bolts is the increased torsion stress in the joint. The high torsion may cause the fastener to yield at a lower preload than expected. Since friction conditions are uneven, the bolt may not reach the necessary preload. Insufficient preload is the most common cause of bolting failure due to fatigue.

Thread lubricants are often used to minimize the friction and to obtain more uniform clamp load. However, lubrication will significantly reduce the locking ability of any friction based method.

Figure 24. Nord-Lock X-Series Washers Ref [25].

The wedge locking method offered by Nord-Loc, see **Figure 24 & 25**, Disc-Lock, etc., is based on tension instead of friction. The most common example of the wedge locking system is a pair of washers which have cams with a rise greater than the thread pitch of the bolt. The washer pair is installed cam-face to cam-face. When the bolt/nut is tightened, teeth grip and lock the mating surfaces, allowing movement only across the cam faces.

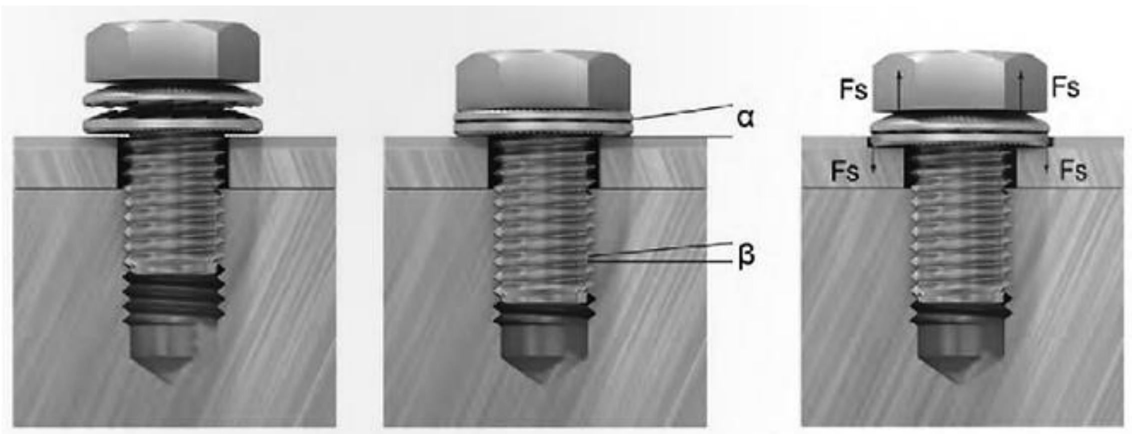


Figure 25. Nord-Lock X Series Washers Illustrating Installation, Ref [25].

Any rotation of the bolt/nut is blocked by the wedge effect of the cams. The wedge locking ability is not affected by lubrication.

The Spiralock by Stanley Ref [24] uses a wedge ramp at the root of the internal thread which only engages when the joint starts to build clamp load during tightening, see **Figure 26**. The crest of the standard external threads draws tightly against the wedge ramp which is reported to eliminate all radial clearance and create a continuous spiral line of contact between the internal and external threads. One of KSC's clients has reported successfully using the Spiralock in assembly of underground coal mining equipment.

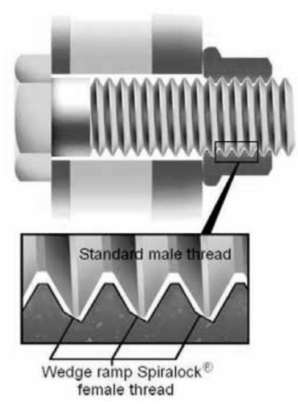


Figure 26. Stanley Spiralock, Ref [24].

Since the Spiralock has special internal thread geometry, special taps and gages are required. The locking action is in one direction only. Identifying holes tapped with Spiralock taps is difficult so a means of tracking assembles using this type fastener should be implemented.

Bolting Criteria

- **a.** Washers: (Flat, Flat Extra Thick Hardened, Split Lock, Bellville, Wedge Locking, Direct Tension Indicating, etc.)
- **b.** Thread Lubrication Effect:
- **c.** Secure Bolting
  - **i.** Double Nuts, Serrated, Ribbed (Rip-Lock, Nord Lock, DISC-LOCK WASHER
  - **ii.** Chemical Thread Lockers, Polymer Patches, Adhesive Patches
  - **iii.** Nuts and Bolts with Thread Locking Features

Proper Bolting and Potential Solutions Summary

- **a.** Develop and implement standards and specifications regarding hold-down and anchor bolting
- **b.** Bolt Grades: grade 5 vs grade 8 (8.8 vs 10.9 metric) Grade 1, 2 and unmarked don't use!!
- **c.** Verify proper materials, design and installation at initial installation
- **d.** Always check fasteners for proper "Grade", especially after maintenance is performed. Be aware of imported counterfeit fasteners.
- **e.** Insist on exta thick, hardened washers with proper inside diameter be installed on motor, bearing housing and machine frame hold-down bolts.
- **f.** Eliminate split lock washers from your bolt assemblies and inventory.
- **g.** Determine proper bolt torque and insist that proper torqueing procedures be followed and document in procedures.
- **h.** Using "Torque-Angle Monitoring with Control" as previously described will provide more accurate tensioning provided the bolt diameter to length is 1 to 7 or greater.
- **i.** Use specialty type nut locking devices where needed, especially where high vibration is present.

Example 1: AMCA Arrangement 8 Fan. High Amplitude Vibration, Anchor Bolts Breaking:

Vibration Analysis: Vibration data showed high amplitude 1X and orders or harmonics. Spectra and time domain data measured on the fan end bearing housing with loose hold down bolts are shown in **Figure 27**. Multiples of the run speed frequency in the spectrum and relative motion of the bearing housing are indicated in the time domain data.

The 3D model of the fan and foundation, see **Figure 28**, was developed in ME'scopeVES, reference [30]. Cross channel vibration data were measured at various locations on the motor, fan base and concrete foundation for the ODS. The vibration amplitudes at various points are shown in **Figure 29**. The vibrating shape or pattern of the fan was rocking side-to-side, pivoting about

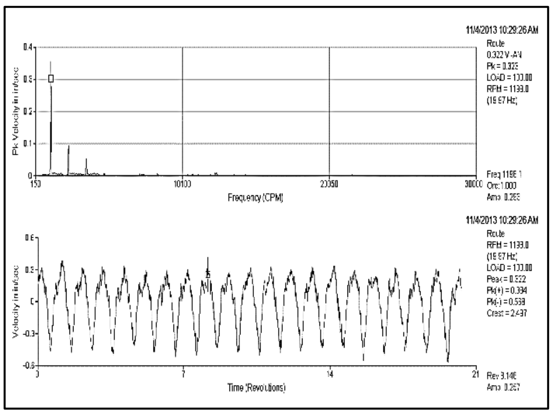


Figure 27. Fan End Brg Housing Ver, Time Data in Velocity in/sec pk.

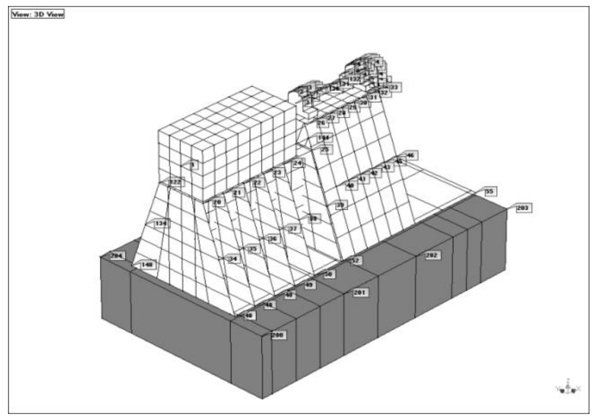


Figure 28. Fan Model Developed in ME'scopeVES, Reference [30].

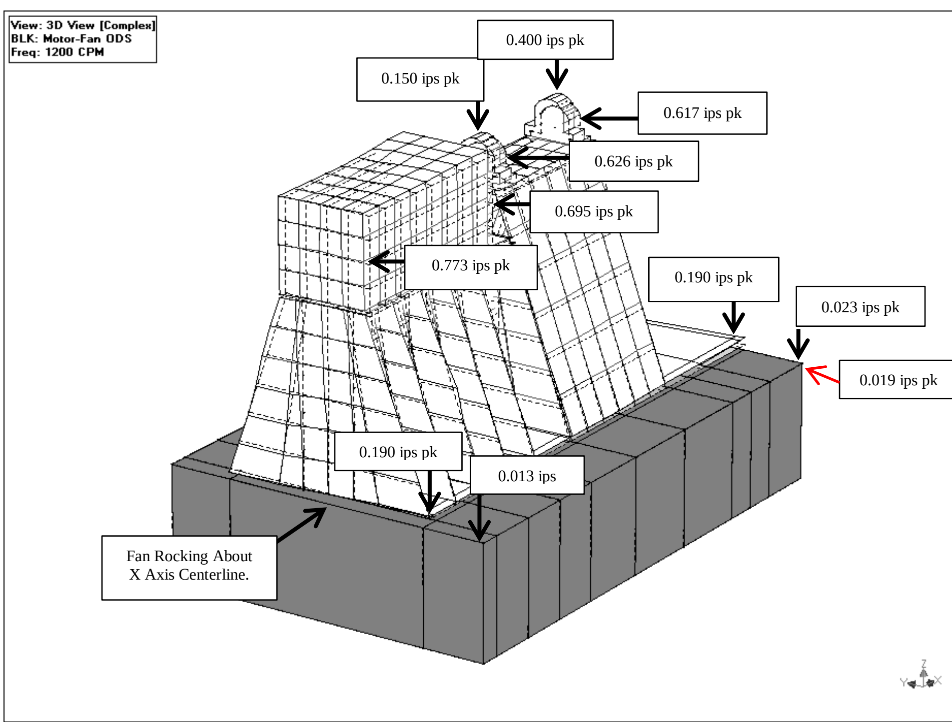


Figure 29. Animated ODS at Fan Run Speed Frequency.

the X Axis just above the surface of the concrete base parallel to the shaft's centerline. The severe rocking motion of the entire fan assembly was a result of fan wheel unbalance (forcing function) amplified by a rigid body mode natural frequency near running speed and fan rotor 1st critical. The rigid body mode was caused by loose mounting of the fan base to the concrete foundation (failed anchor bolts). The failed anchor bolts also lowered support stiffness which lowered the rotor 1st critical down to run speed frequency.

Fan frame anchor bolts had only ½ inch free stretch length. Some of the anchor bolts were loose and other bolts had experienced fatigue failure. The anchor bolt washers were too thin resulting in deformation of the washers and inability of the bolted assembly to remain properly tensioned. Typically a sleeve is installed around the bolts to prevent concrete and grout from contacting the bolts. The sleeve allows the bolt to stretch over an adequate length of at least 7 diameters to prevent over stressing during tightening. This long stretch length also allows the bolt to be more elastic than the other components in the joint.

The motor hold down bolts had flat washers which were too thin and had deformed into the base holes. This created a loose mounting and allowed relative motion or vibration of the motor to the base. The fan bearing housing hold down bolts also used flat washers. Both bearing housings exhibited looseness but the fan end bearing had significant relative motion to the bearing support plate. There was very low amplitude vibration of the concrete pad indicating it had very little participation in the fan vibration.

Several recommendations were made to address the bolting issues that included installing anchor bolts, as shown in **Figure 30**, full grouting and stiffening of the fan base, see **Figure 31**, and installing Grade 8 bolts with thick washers under the motor and bearing housing hold-down bolts and nuts. In the end, the decision was made to replace the fan.

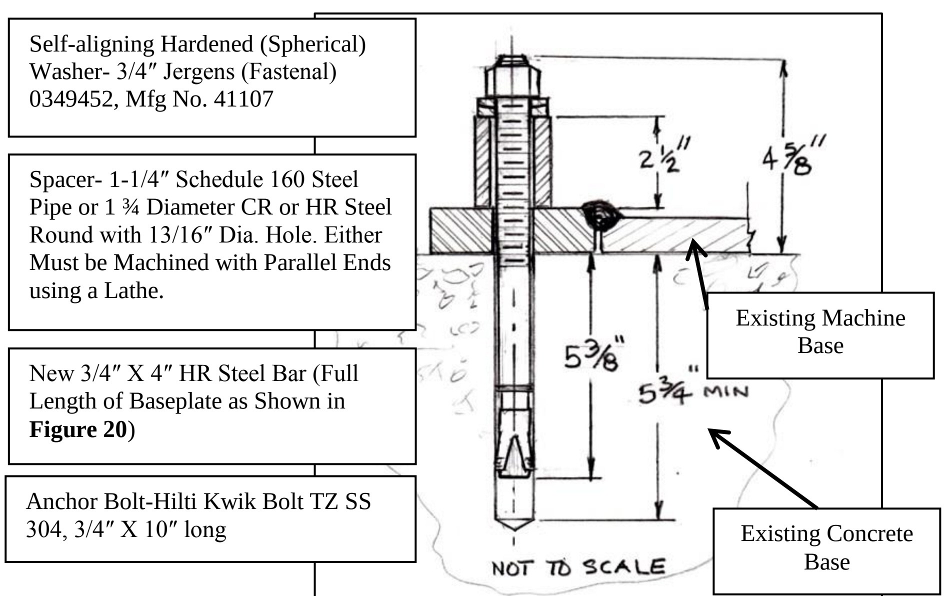


Figure 30. Anchor Bolt Installation Detail.

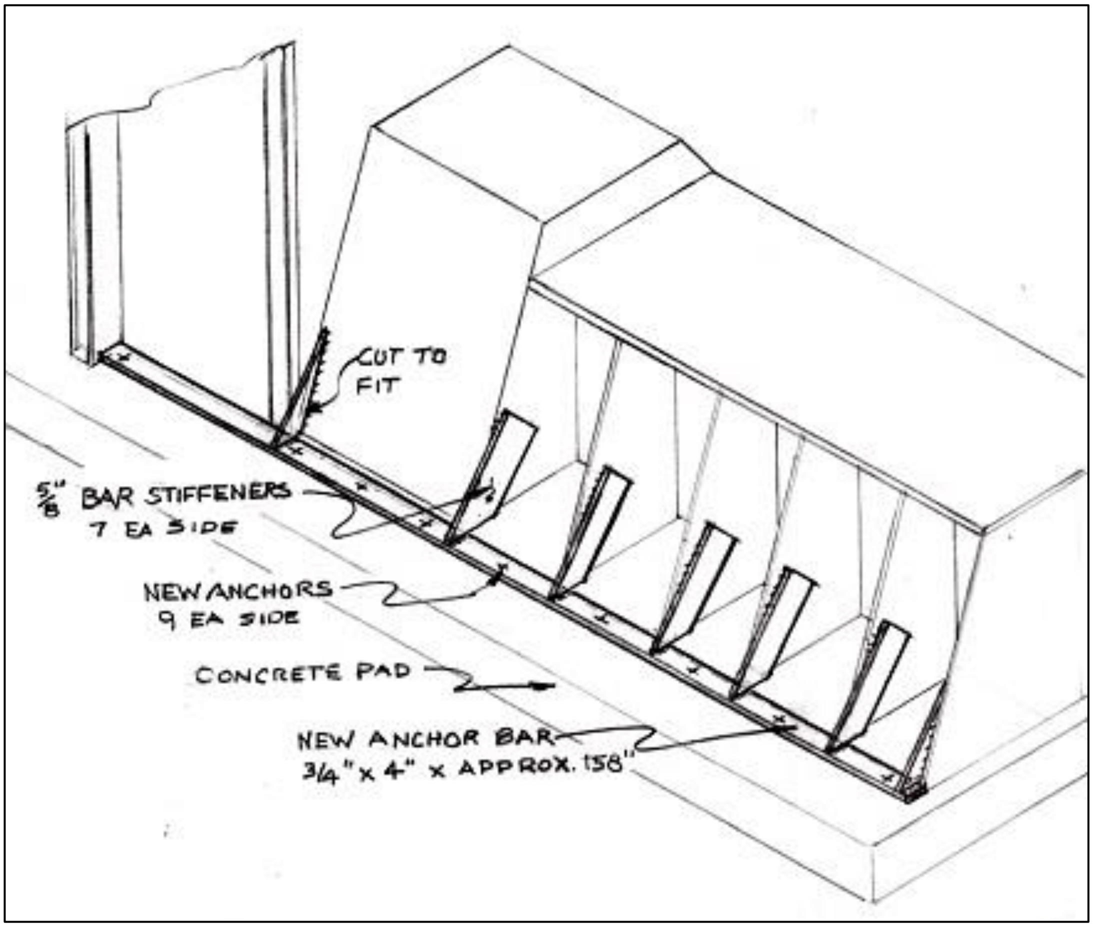


Figure 31. Anchor Bar and Bracing Isometric Sketch.

Example 2 AMCA Two Arrangement 8 Process Fans

The reported reliability problems for two process fans (A & B) were short bearing life, high amplitude vibration and previous failure that resulted in a bent fan shaft. The fans were 500 HP, 56,433 ACFM, AMCA Arrangement 8. Our initial findings verified elevated vibration amplitudes plus loose fan bearing housing and motor hold down bolts and bearing defect frequencies. Vibration data measured during coastdown indicated fan rotor unbalance as the primary forcing function. An ODS clearly showed motor relative motion of the bearing housings and motors to the support plates and foundation rocking of Fan A.

The vibration data at Fan A inboard bearing housing, see **Figure 32**, showed 1X, 2X and 3X as the primary frequencies. The overall amplitude measured 0.152 in/sec pk. Time waveform data measured 2 g's.

Fan bearings were spherical roller type with four Grade 5 hold down bolts. The hold down bolt washers were dished with only the corners of the bolt heads contacting the washers, see **Figure 33**. Split shims had been inserted under the bearing housings instead of solid shims resulting in the center of the bearing cast housing being unsupported. The bearing housing support surfaces had been machined to 2 to 3 mils/ft flatness.

Both fan's bearing housing hold down bolts (under the base plate), see **Figure 34**, had a stack of washers that included a flat washer, split lock washer and square washer burned from bar stock by cutting torch. All bolt assemblies were found loose.

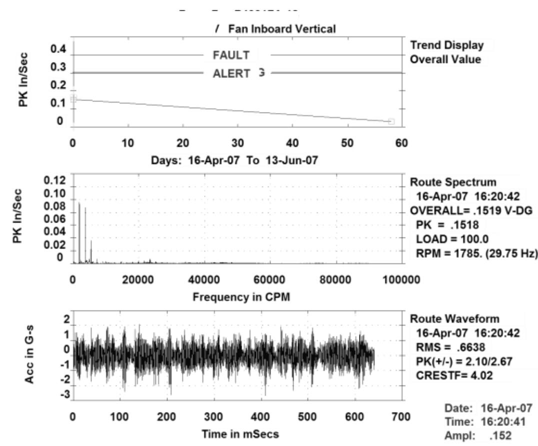


Figure 32 Initial Vibration Data at Fan Inboard Bearing Housing Vertical Direction.

Some of the motor hold down bolts had low carbon steel flat washers which had dished into a large hole. Some bolts had no washer with only the nut corners contacting the support plate, see **Figures 35-37**. The holes in the base plate supporting the motor were much too large.

Figure 33. Bearing Housing Hold Down Bolt Washers Dished, Full Shim not Used Under Housing.

Figure 34. Fan A, Bearing Housing Hold Down Bolts.

Figure 35. Motor Hold Down Bolt, Corners of Nut

Figure 36. Motor Hold Down Bolt, Oversize Hole, No Washer.

Figure 37. Fan A, Motor Hold Down Bolt With Deformed Flat Washer.

Coastdown data on one fan is plotted in Bode format is shown in **Figure 38**. The vibration response was typical of unbalance with no indication of structural resonance.

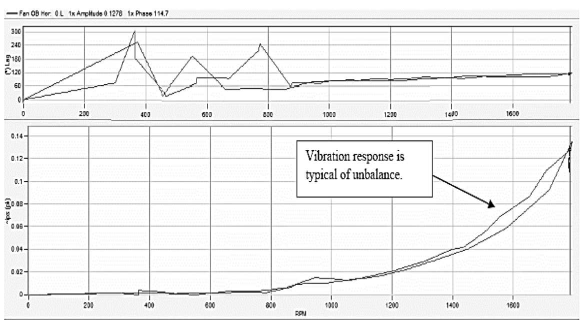


Figure 38. Fan A, Coastdown Data, Bode Plot, Fan OB Brg Hor.

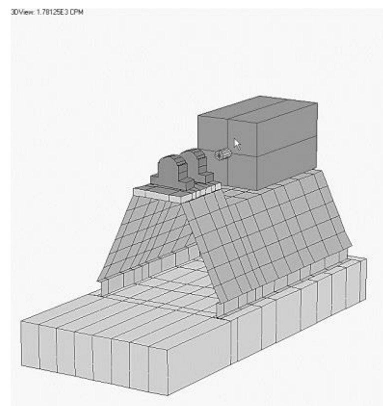


Figure 39. Fan Foundation, Base, Motor & Bearings ME'scopeVES [30] Model.

ODS data was acquired using one reference accelerometer and five roving accelerometers. The model of the fan foundation, fabricated steel base, bearing housings and motor was developed in ME'scopeVES, [30] see **Figure 39**

The ODS animations at running speed showed bearing housing and motor relative movement to the base plate. Vibration data measured at the motor right-front foot of Fan A at 1X was 0.050 in/sec pk and 0.039 in/sec pk on the support plate, see **Figure 40**. Amplitudes were lower at Fan A Wheel End Bearing Housing measuring 0.025 in/sec pk on the top of the housing and 0.016 in/sec pk on the base plate, see **Figure 41**.

Corrective actions were as follows:

1. The fan rotor was balanced per ISO 1941-1 for over hung rotor to G2.5.
2. Wheel hub advance on the tapered fit was calculated to maintain interference fit allowing for differential thermal growth of the shaft, fan hub and tapered sleeve.
3. The fan bearings were replaced.
4. Bearing housing hold down bolts with hardened washers torqued per specification.

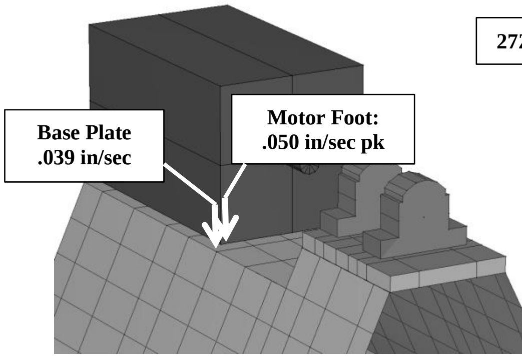


Figure 40. ODS Vibration Data at Fan A Motor RF Foot & Support Plate 1X Running Speed.

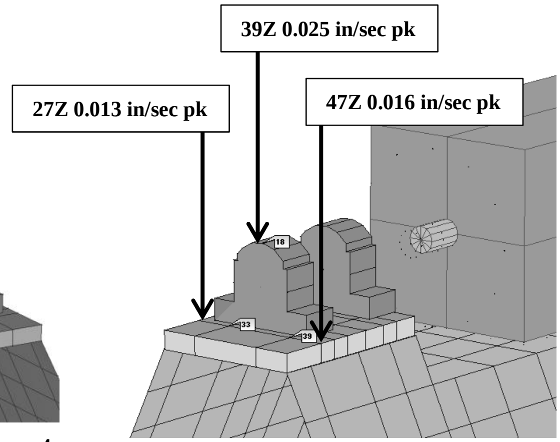


Figure 41. ODS Vibration Data at Fan A, Wheel End Bearing Housing and Base Plate.

Vibration was reduced 88% at 1X and 90% at 2X.

Example 3: New Installation of Slurry Pumps, Excessive Vibration

Three pumps, 500 Hp, 1170 RPM , 5508 GPM, 212 ft head, had been installed at a mining facility. [Ref 8] The units were direct coupled, variable frequency drive in a waste disposal system. One of the units is shown in **Figure 42**. The client was concerned about the very high amplitude vibration of the motors and the skids reported by the consulting company monitoring vibration periodically. Resonance was suspected.

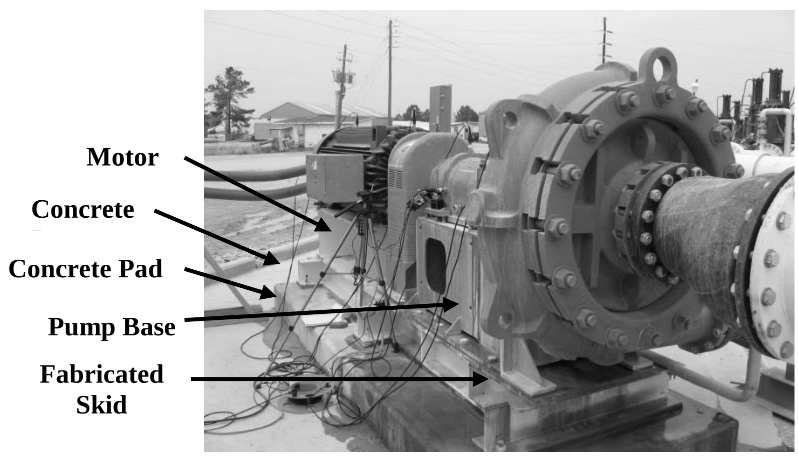


Figure 42. Pump Assembly Naming Convention.

The motors and pumps were mounted on fabricated skids which were bolted and epoxy grouted to individual concrete bases. Visual inspection, vibration data, ODS and Experimental Modal Analysis (EMA) were conducted and identified the following:

- Motor hold down bolts – <u>flat washers yielded into base plate holes</u>.
- The pump base mounting foot underneath the <u>pump housing did not have a hold down bolt to the skid</u>.
- The <u>six (6) anchor bolts on the bottom of the skid were fastened through a very flexible</u> flange or "wing". The thin flanges of this design allow excessive flexure.
- The <u>anchor bolt washer's inside diameters were too large and the washers were too thin. This resulted in a small clamping area for the nuts, and deformation of the washers</u>.
- There were an <u>inadequate number of anchor bolts</u> on the lower section of the skid frame.

- There was lack of bonding of the epoxy grout to the concrete and the structural steel frame at some locations.
- Lack of bonding of the concrete mounting base to the concrete mat The mass of the concrete mounting base was undersized according to the information provided for the weight of the motor, pump, fabricated frame and the dimensions of the concrete. Typically, the mass of the concrete should be a minimum of three times the mass of the rotating machinery mounted to the concrete base.
- Resonant frequencies of the fabricated skid and motor pedestal near 1X the motor run speed frequency and pump blade passing frequency (4X).
  - The resonant frequency at 1X the motor run speed was side-to-side rocking mode shape of the motor and flexure of the motor pedestal.
  - The resonant frequency at 4X the motor run speed was a twisting motion about the Z axis of the motor and motor pedestal.

The skid was fabricated of wide flange beams and flat plate. The flanges of the wide flange beams were used to install the skid hold down bolting see **Figure 43 & 44**. This attachment design lacked adequate rigidity.

The skid hold-down bolts were carbon steel and had already started corroding. The <u>washer holes were oversize with only the corners of the nuts make contact</u>. The <u>motor hold-down bolt washers were low carbon steel with oversized holes which had deformed into the holes in the motor support base</u>, see **Figure 45**. All bolts were found loose.

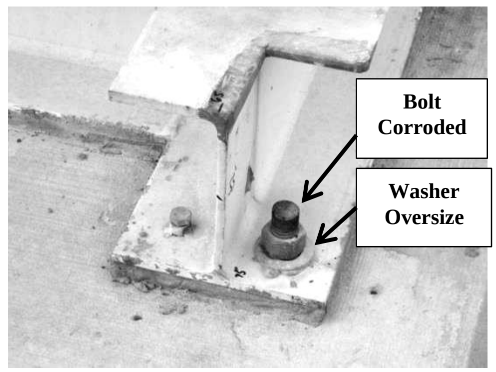


Figure 43. Motor End of Skid, Caron Steel Anchor Bolts, Over

Carbon steel shims had been inserted under some motor feet and had started corroding.

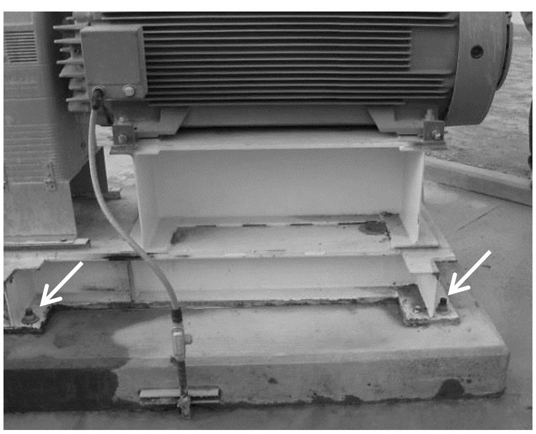


Figure 44. Flanges of the Skid Wide Flange Beams Were Used For Anchor Bolt Hold Down.

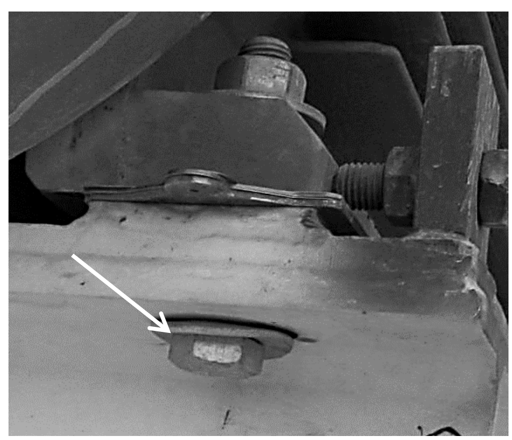


Figure 45. Motor Hold Down Bolts Had Oversize Washers Which Had Deformed Into the Holes.

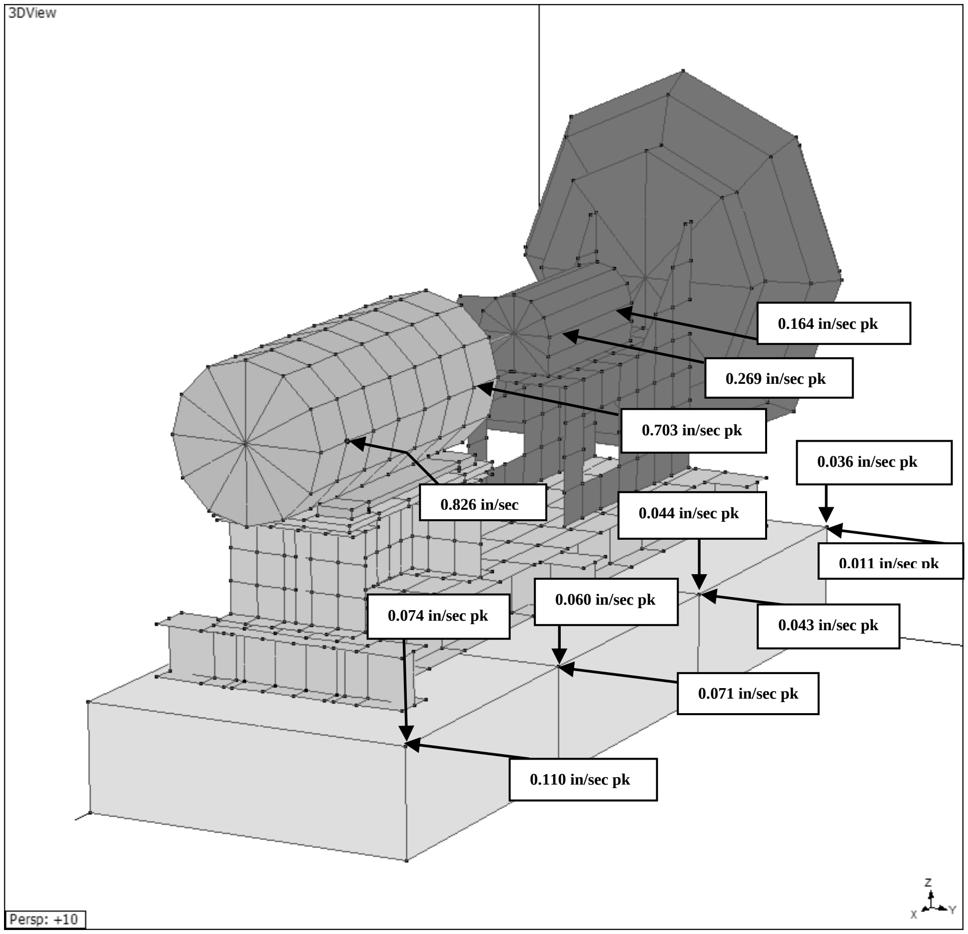


Figure 46. Average Vibration Amplitudes for Operating Deflection Shape Model At 1X Run Speed Frequency.

The ODS Model developed in ME'scopeVES Ref [30] is shown in **Figure 46**. Vibration amplitudes in/sec pk for 1X run speed are labeled for some locations. The motor outboard bearing housing had the highest amplitude at 0.826 in/sec pk.

ODS spectra data measured at the motor right-front foot and base vertically are shown in **Figure 47**. The data is plotted with log magnitude scaling to more clearly show the indications of resonance near 1X and 4X run speed (pump vane passing).

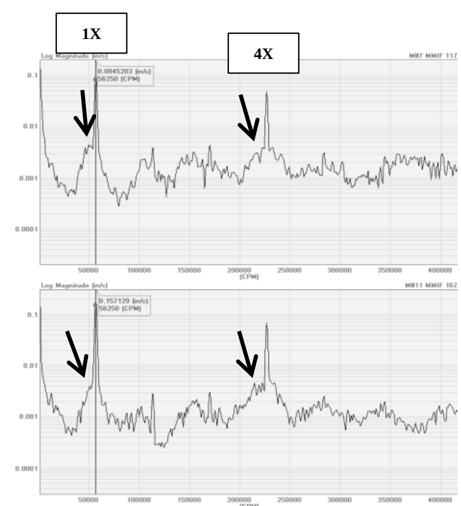


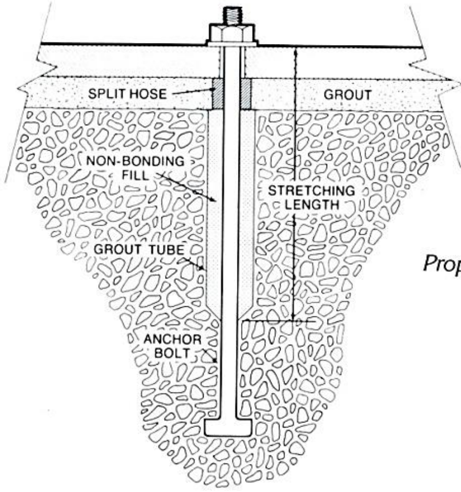
Figure 47. Spectrum at Motor IB Right Front Foot and Support, Log Magnitude Scaling Is Better Showing Indication of Resonance.

The vertical vibration was much lower amplitude than horizontal. As shown in **Figure 47**, the Motor Right-Front foot measured 0.157 in/sec pk while the base plate under the foot measured 0.085 in/sec pk.

Corrective Recommendations: There were many problems identified with the installation of the pumps but this article focuses on bolting problems. A drawing, see **Figure 48**, was provided Ref 9 to show the proper installation of anchor bolts, Ref [9].

Recommended modifications to the skid and motor support are shown in **Figure 49**. This drawing showed the proper method of skid design for the hold down bolts. Stiffeners were recommended to significantly increase the rigidly of the motor support.

Thicker washers (1/2" thick) and Grade 5 bolts were recommended for the motor hold down bolts.



Proper method of installing anchor bolts.

Figure 48. Anchor Bolt Installation Ref [9 & 38] Escoweld 7505E/7530 Machinery Grout.

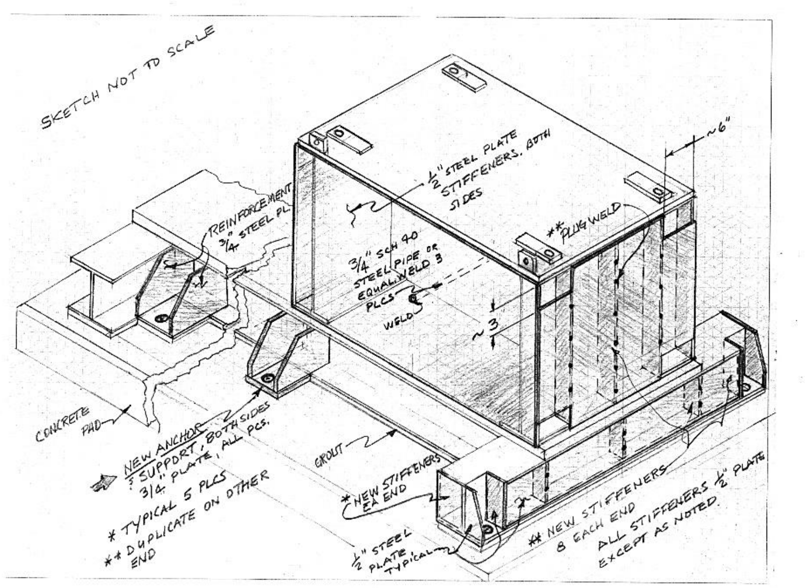


Figure 49. Drawing Showing Modification of Skid.

Example 4: Turbine-Generator Train Bearing Housing & Pedestal Excessive Vibration

Following an outage of a turbine-generator train, vibration of some of the bearing housings and pedestals was found to be very high amplitude. There were other issues identified during the investigation but the focues of this article is on bolting. The unit consisted of three turbines, see **Figures 50-51**, high pressure turbine (H1P), two low pressure turbines (LP1) & (LP2), Generator and Exciter. The bearing housings and pedestals of LPA & LPB, bearings 4 and 5, see **Figure 52**, had very high amplitude vibration primarily in the axial direction. Note that the <u>pedetal hold down bolts has structural steel angle welded to the nut and top of the bolts</u>, see **Figures 53-54**.

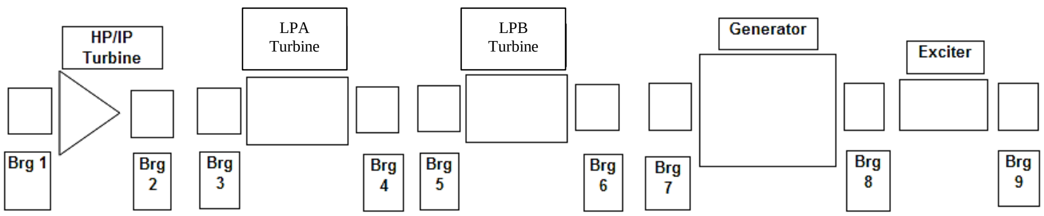


Figure 50. Layout of Turbine-Generator Train.

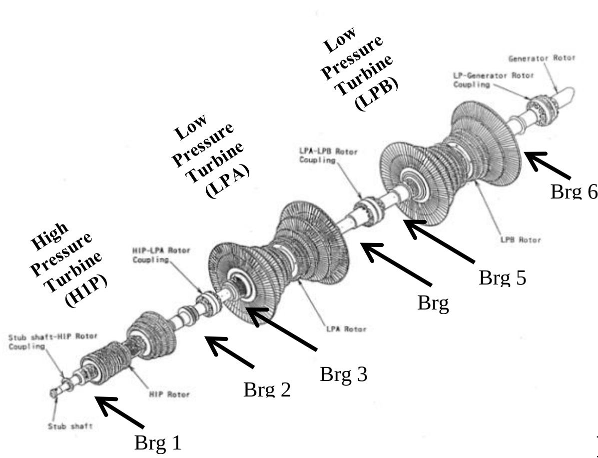


Figure 51. High and Low Pressure Turbines.

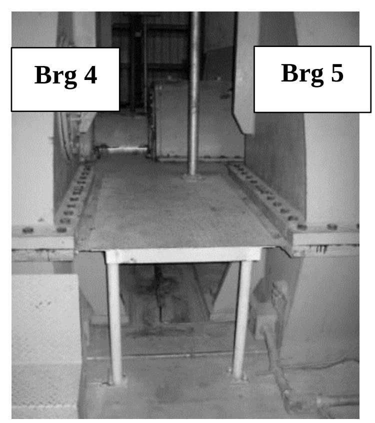


Figure 52. LP1 & LP2 Bearings 4 and 5 Pedestals Right Side.

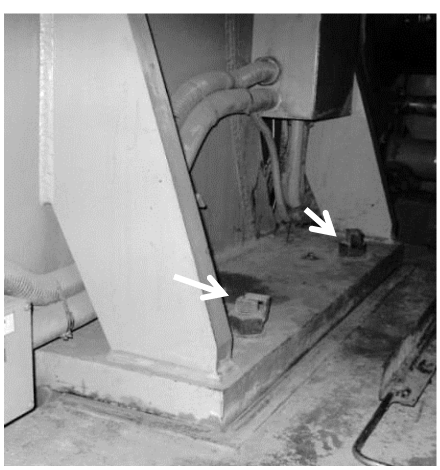


Figure 53. Bearing 4 Pedestal & Soleplate LPA Right Side.

Figure 54. Bearing 5 Pedestal & Soleplate LPB Right Side.

The vibration spectral data showed indication of structural resonance near running speed, see **Figures 55-56**. The spectra are plotted with log magnitude scaling to aid identification of natural frequencies.

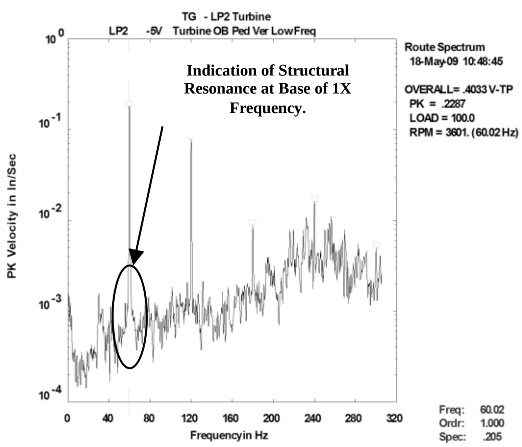


Figure 55. Frequency Spectrum, Brg 5 Casing, Vertical.

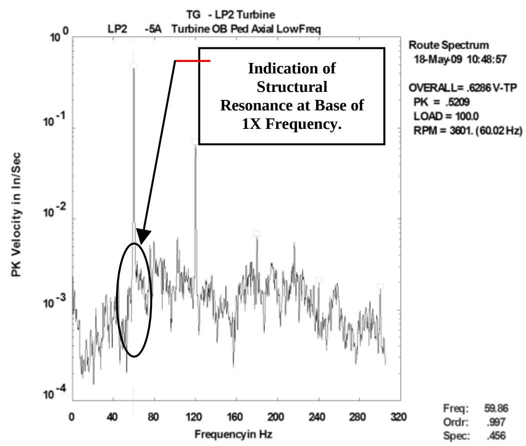


Figure 56. Frequency Spectrum at Brg 5 Casing, Axial.

The bearing housings and pedestals of the LP Turbines were modeled using ME'scopeVES, see **Figures 57-58**. ODS data were measured and imported to ME'scopeVES for animation. The ODS data measured on Brg 4 & Brg 5 pedestals showed flexure of the pedestals and relative movement of the pedestal base plates and soleplates. Vibration was highest amplitude near the central points of the casing split lines and bearing housings. Inadequate clamping by the hold down bolts (no bolt pre-load) was indicated.

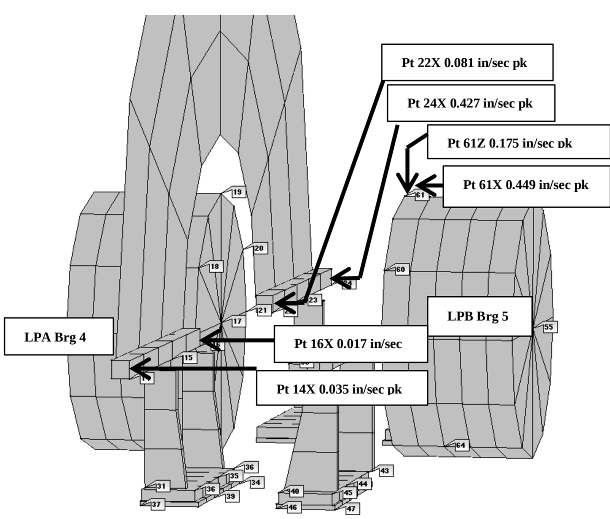


Figure 57. ME'scopeVES Model, LPA Brg 4, LPB Brg 5 Bearing Housing Pedestals & Soleplates.

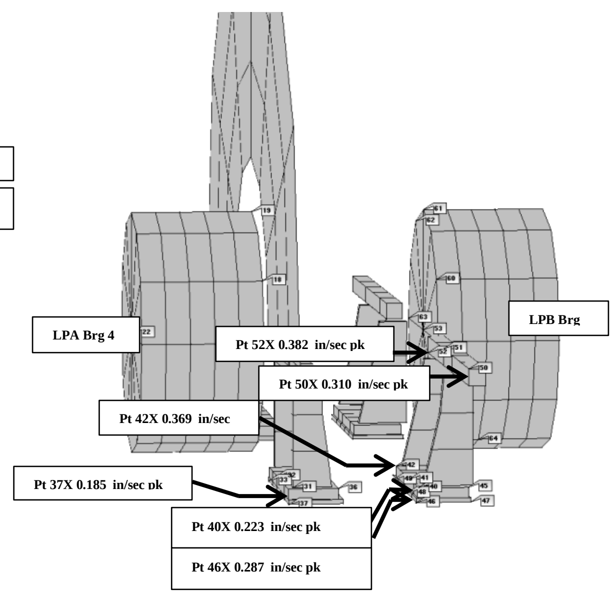


Figure 58. ME'scopeVES Model, LPA Brg 4, LPB Brg 5 Bearing Housings, Pedestals & Soleplates.

Additional testing to include checking tightness (adequate pre-load) of hold down bolts, modal testing to identify resonance, coastdown data collection from the proximity probes and casing accelerometers was recommended. After these issues and other problems were identified, the client's decision was to bring the outage contractor back on site to resolve the identified problems and collect the recommended vibration data.

References:

1. Barbee, A, Bracher, B, McGinnis, B, Singleton, K, **Case Study Analysis of Process Fan Failure And Bearing Housing/Shaft Design**, Vibration Institute National Training Seminar 2013.
2. McGinnis R, Singleton K, **Power House FD & ID Fans Analyzing Concrete Foundation Resonance,** Vibration Institute, Piedmont Chapter, May 8, 2015.
3. Monroe, Todd R., Monroe, Perry C. Jr, "**The Road To Reliable Pumps"**, 23rd International Pump Users Symposium (2007).
4. Junker, G., **New Criteria for Self-Loosening of Fasteners Under Vibration**. SAE Paper 690055, 1969.
5. Fastener transverse vibration test machine, Junker vibration test,http://www.boltscience.com/pages/junkerstest.htm.
6. Temitope, Stephen J. **Condition Monitoring of Bolted Joints**, Department of Mechanical Engineering, The University of Sheffield, June 2015.
7. Kaas, Michael, **Stop Loosening of Fasteners**, Fixing Magazine, www.fastenerandfixing.com
8. McGinnis R., Singleton, K, **New Pump Installation, Analysis of Excessive Vibration Root Cause**, Vibration Institute Piedmont Chapter, May 2008.
9. Escoweld 7505E/7530, **Anchor Bolt Installation Reference**, Escoweld 7505E/7530 Machinery Grout.
10. Unbrako Engineering Guide, http://unbrako.com/docs/engguide.pdf
11. StressTel, **Guide To Ultrasonic Inspection of Fasteners**, http://www.ge-mcs.com/download/ultrasound/bolt-extensometer/stresstel-fasteners.pdf
12. Bolt Science – **Vibration Loosening of Bolts**, http://www.boltscience.com/pages/vibloose.htm
13. SKF **Vibracon Application Guide**, https://skf-solution-factory-marine-services-674702.c.cdn77.org/downloads/SKF-Vibracon-Application-guide.pdf
14. Bolt-Science, **Tutorial on the Basics of Bolted Joints**, http://www.boltscience.com/pages/basics1.htm
15. Bolted Joint Design, Fastenal Engineering & Design Support, https://www.fastenal.com/content/feds/pdf/Article%20-%20Bolted%20Joint%20Design.pdf
16. StressTel Ultraxonic Testing Equipment, **Guide to Ultrasonic Inspection of Fasteners**, http://www.ge-mcs.com/download/ultrasound/bolt-extensometer/stresstel-fasteners.pdf
17. NORD-LOCK, http://www.nord-lock.com/wp-content/uploads/2014/03/BOL1_14_en.pdf BOLI_14_en.pdf
18. **Fastenal Technical Reference Guide**.pdf, https://www.fastenal.com/
19. NORD-LOCK, http://www.nord-lock.com/wp-content/uploads/2015/03bol1_15_3n.pdf
20. NORD-LOCK, http://www.nord-lock.com/superbolt/multi-jackbolt-tensioners/flexnuts/introduction/

21. NORD-LOCK, http://www.nord-lock.com/nord-lock/multifunctional-wedge-locking/x-series-washers/joint-guide/
22. NORD-LOCK, **Junker Machine Test Video**, https://www.youtube.com/watch?v=IKwWu2w1gGk
23. Applied Bolting Technology, **Applied Bolting Brochure**, http://www.appliedbolting.com/our-brochure.html
24. Stanley, **Spiralock Fasteners & Threading Tools Product Specifications,** http://www.stanleyengineeredfastening.com/sites/www.emhartamericas.com/files/downloads/Spiralock%20Products%20Catalog.pdf
25. NORD-LOCK, **X Series Washers 2012**, http://www.nord-lock.com/2012/09/28/nord-lock-x-series-bolt-security-without-compromise/
26. SmartBolts, **Visual Indication System Product Catalog,** www.smartbolts.com
27. **Hilti North America Product Technical Guide**, www.us.hilti.com/anchors
28. **Jergens Workholding Components.pdf** pg 260, pg 268, http://www.jergensinc.com/Spherical-Flange-Assemblies
29. RBW Manufacturing**, Helpful Hints,** www.rbwmfg.com/wp-content/uploads/2012/02/HelpfulHints2.pdf
30. Vibrant Technology, www. http://vibetech.com/
31. Oberg, E., Jones. F., Horton, H., and Ryffel, H: "**Machinery's Handbook**", 27thEdition, Industrial Press Inc., New York, 2004
32. "Criteria for Preloaded Bolts", NSTS–08307, Rev. A, NASA, 1998
33. SKF, "**Bolt-Tightening Handbook**", https://www.google.com/?gws_rd=ssl#q=skf+bolt+tightening+handbook
34. Skidmorewilhelm, "**User Manual Model MZ-100 Bolt Tension Calibrator**", http://www.skidmore-wilhelm.com/
35. Raynor-Keck, Lisa, "**Torque Vs. Tension: Is There a Clear Winner?"**, Fastorq, www.fastorq.com
36. Shobert, Ralph S. P.E., "**Engineering Fundamentals of Threaded Fastener Design and Analysis**", http://www.hexagon.de/rs/engineering%20fundamentals.pdf.
37. Grainger, https://www.grainger.com/product/TE-CO-Heavy-Duty-Washer-2YJG5?s_pp=false&picUrl=//static.grainger.com/rp/s/is/image/Grainger/2YJF8_AS01?$smthumb$
38. Escoweld 705E/7530, **Technical Bulletin # 1612D**, http://na.itwengineeredpolymers.com/downloads/tds/Escoweld-7505E-tds.pdf
39. Lee, James Pl, Golod, Yelena S., **Foundations for Dynamic Equipment**, Reported by ACIS Committee 350, http://www.inti.gob.ar/cirsoc/pdf/fundaciones/ACI-351-3R-04.pdf
40. RotoBolt Load Monitors, http://rlrowan.com/products/rotobolt-load-monitors
41. Jackson, Charlies, **Reliability Treatise Part 1**, CompressorTech, January-February 2000.
42. Loctite, **Application Wall Chart V10**, http://www.loctite.com.au/aue/content_data/311083_7129_Application_Wall_Chart_V10.pdf
43. Micro Plating Inc., http://www.microplating.com/?gclid=CKmZoY2igcsCFQhkhgod91cE4w
44. Rowan, Robert .L. & Associates, Inc., http://rlrowan.com/products/exactalign-adjustable-motor-supports-for-skids/