

# Failures of Vibration Supervisory Systems for Turbine-Generator Sets

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Manufacturers of turbine controls and instruments since 1977.

## Forward

Turbine Supervisory Instrument (TSI) systems should in the least provide operators with adequate warning that the monitored machines are experiencing damage from excessive vibration. TSI systems should also warn operators of non-vibration operating parameter relationships known to contribute to excessive vibration or other dangers.

A review of recent TSI system installations suggests that although the electronic equipment is of high quality and good design, the adaptation of the instrumentation to turbine-generator sets has failed in several key areas and the systems are not providing adequate machinery protection

These failures are for the most part attributable to one common fault, the misapplication of small rotor, anti-friction bearing monitoring techniques on large journal bearing turbines. What results is an instrument system which functions, i.e., electronically operates, acquires data, presents impressive displays etc., but does not accomplish the original objective.

This paper provides the basis for identifying and correcting turbine TSI shortcomings in order to obtain a truly useful and not just electronically functioning system.

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6. Summary

# 1. Vibration TSI Design Ancestry

Most of the TSI market is dominated by firms which established and developed their expertise on antifriction (ball or roller) bearing supported shafts, and they do fairly well on these applications to date. Small to medium motor driven equipment utilizing antifriction bearings were the design target, and the process industries (petroleum, pulp, and others) were the target customers. In many cases systems were devised to successfully monitor and provide operators with abnormal condition alarms for dozens of such small machines.

It was a natural extrapolation to apply the new low-maintenance non-contact proximity probes, signal conditioning, and data display systems to turbine-generator sets and other rotating equipment mounted in journal bearings. The same *hardware* can be applied, but *how* the instrumentation is installed and *what other* monitoring sensors are needed is radically different. Far too often the same techniques for such application engineering choices as sensor type, sensor mount type, sensor location, alert level determination, and alert conditions are simply extrapolated from small machine systems with dangerous results.

If you find you have such a "small machine" setup on a turbine-generator, don't fret, because a few additional sensors and correct alarm setpoint determinations can easily establish the sort of true machinery protection you thought you had.

# 2. Major Differences Between Anti-Friction (Ball/Roller) Bearings and Turbine Journal Bearings.

The key differences significant in supervisory monitoring can be summarized as:

Clearance...	Antifriction bearings have very small and circumferentially uniform clearance. Rotor forces due to unbalance or upset are applied near-equally (as they rotate) to the full bearing circumference.
	Turbine journal bearings have comparatively large, non-uniform clearances, including "top" clearances typically designed as one mil (.001 -inch) per inch of shaft diameter. Larger journal bearings may use multiple pads or when two-halves may have additional side clearance termed "cheeking".
Lubrication...	Antifriction bearings ride on very thin oil/grease films maintained between the bearing rolling elements and the bearing race. These films have little if any damping effect on the shaft.

Turbine Journal bearing supported shafts at speed ride on a relatively thick (.003 inch typical) hydrostatic oil "wedge" existing between the shaft and the lower half or pad/s only. This "wedge" of oil has considerable damping properties.

Stiffness...

Anti-friction bearing machine cases mounts are of generally low stiffness permitting vertical and horizontal deflections under modest vibration forces.

Journal bearing support structures are very rigid with high stiffness in the vertical and horizontal directions, but low stiffness in the axial direction. A flexible beam system couples the bearing support structures to the rigid foundation.

Rotor Rigidity...

Anti-friction supported rotors may be either rigid or flexible depending upon the machine.

Journal bearing supported turbine rotors are very flexible at speed and are easily deflected according to combinations of static and dynamic deflections.

### **3. Three Most Common TSI Errors:**

#### **a. Probe Locations Near Bearings (within 16-inches) Are at Nodal Points**

The small machine practice of monitoring shaft position x-y "orbits" at bearings is widely accepted. To understand why this does not work on TSI requires some basic large turbine rotor dynamics.

With a few rare exceptions, large turbine vibration consists of either imbalance forces or "bends" creating shaft deflections away from the center of rotation. This can be likened to bending a paper clip, then twirling it between the thumb and index fingers of two hands. When the clip "high spots" rotate they cyclically apply forces to the fingers. The turbine generator rotors also have such bends or deflections and vibration results as the "high spots" apply forces to the bearings as they rotate under machine operation. Note that the entire rotors do not "shake", nor do they vibrate like a guitar string. The bending of the rotors creates a longitudinally alternating displacement which varies with amplitude between the bearings according to modal pattern.

The deflection shapes of turbine and generator rotors follow modal patterns possible at the rotor speed compared to its resonant speed (or critical speed).

Steam turbine rotors are designed to operate between first and second critical speeds which permits only two types (and any combination thereof) of deflections, static (single bow) or dynamic (sinusoidal double bow). Most generator rotors and large gas turbine rotors are designed to run between the second and third criticals. Examining the properties of these modal deflections we can spot two very important characteristics:

The deflections are reduced to zero at some location within the bearing length. The zero-deflection location is termed the nodal point.

The magnitude of the deflections increase with travel inboard longitudinally from each bearing. In fact, we can find any deflection (and thus vibration) magnitude we want from zero to the shaft maximum deflection if we can move our monitoring location inboard along the rotor span.

Let's hold here a moment and digest these characteristics with respect to supervisory monitoring. If the main purpose of the instrumentation is to *alert operators of potentially damaging conditions to the machinery, wouldn't it be a good idea to place the monitoring probes in a location where they will detect representative deflections?* If we use anti-friction bearing monitoring techniques and mount displacement probes in axial locations at or within the bearing and mounting between the bearing face or bracket and the shaft we will be selecting a location at or near the nodal points. This is being done on many turbine generator installations for reasons of "tradition" or "because that's the way the instrument company says to do it". The results provide a system which not only fails its primary objective, but also provides a false sense of security that vibration levels are satisfactory and that the machinery may be safely operated.

When a significant unbalance exists only a very small portion of the rotor vibration is evidenced on probes mounted in bearings (near or at nodal points). As the case examples (Section 5) will demonstrate, this misconception of severity has resulted in major forced outages and heavy equipment damage.

How is this error corrected? One solution is to move the displacement probes inboard as far as reasonably possible. In most applications this involves mounting the probes on the bearing inboard oil seals. There is a major practical drawback to moving probes, however, and that is the often poor condition of the rotor shaft surface. The rotor oil seal locations are used for lifting slings and outage maintenance stand supports and are often full of dents and scratches will be perceived as vibration by the probes and monitor. A quality clean up using in place machining solves this problem but is expensive and hard to fit in reduced outage schedules. A second, and much more economical solution is to add a calibrated spring/mass sensor system to detect rotor bow and unbalance at each bearing to supplement the existing proximity probe system. This retains the typical X-Y 90-degree aligned proximity probes within 16-inches of the bearings

as inputs to detect non-synchronous vibrations in which the rotor actually does lift off the bearing (like oil whip and steam whirl, and impacting) while detecting synchronous vibration phenomena (rotor unbalance and rotor bows) using the new tuned and calibrated sensors. The LCC ALERT Supplementary TSI System provides the required sensors and analyzers to accomplish this in an easy to install package.

## **b. "Absolute" Monitoring Probe System Attempts**

Major supervisory instrument manufacturers have come up with what they term as "Absolute" vibration monitoring sensor systems consisting of a displacement probe reading "relative" displacement between the probe bracket and the shaft and a velocity probe monitoring the motions of the bracket itself. By vectorially adding the two signals (taking into account the 90-phase shift of displacement to velocity) a new "Absolute" reading can be calculated which represents, according to the manufacturer, the true shaft motion. This sounds pretty good on the face of it, but alas, is again not correct for large journal bearing applications.

This system can work very well on anti-friction bearing supported machines where the bearing clearance is very small and circumferentially uniform and the oil film is very thin. Any shaft displacement will apply immediate forces to the frame. Also, since the frame structures are themselves not very stiff we would expect bracket motions. This system should work to compensate for the resulting motions and the *absolute* resultants can be argued to have real value.

We hit snags, however, when applying this system to turbine journal bearings. The obvious first problem is that the bearing cap motion is strictly the flexure of the bearing support beams set on the foundation, quite low in amplitude due to support stiffness, so the "velocity correction" is very low and although it has a relationship to the actual rotor deflection this function is not used. Further, due to the closeness of the zero deflection node within the bearing the relative signal value obtained by the proximity probe is also very low and most probably dominated by probe electrical runout, not true displacement. The combination yields a low level muddled waveform.

## c. **Lack of Adequate Calibration**

It is generally accepted that instrument systems employed in power plants should be calibrated to the best accuracy obtainable. When operators must make judgements on TSI data, that data should not harbor a cloud of uncertainty over the calibration of the individual signal sensors. Unfortunately, this is exactly the case with the "standard" vibration probe systems in widespread use today due to the following drawbacks:

### **Installed Gain Factor**

The proximity probe calibration does not take into consideration key application factors which have a great influence upon the installed probe gain factor. Rotor circumference, local eddy-current reflections, and rotor local surface metallurgical differences have been shown to influence a particular proximity probe output by +/- 30% in gain factor. Since the current "accepted" calibration procedure removes the probe from the turbine and positions it against a small flat target (even if the target material is the exact same alloy), none of these factors are accounted for and none of the probable errors are reduced.

### **No Provision for Gain Adjustment**

If careful measurements are made to determine gain adjustments (such as dial indicator monitored fixed probe position changes in place), the probe amplifier (or "proximator") is a potted, sealed canister with a factory preset gain. Any attempt to adjust the amplifier gain requires breaching the potting, de-soldering a fixed resistor, and re-soldering a calculated replacement... a procedure not advised by the factory.

The solution to correct these shortcomings is to provide for variable-gain probe amplifiers or splice in such amplifiers if the current system does not have them. Be sure to select a unit with at least a 10 kHz bandpass to allow 10x over sampling of the highest expected turbine-generator phenomena vibration frequency of 1 KHz. The correct calibration procedure then physically moves each probe in its installed position by a fixed displacement and recording the net change in gap voltage output. Compare this change (usually expressed in millivolts output per mil displacement with most 200 mV/mil) with the design input of the monitoring system and adjust the new amplifier gain until the monitoring system receives the expected change.

Lovejoy Controls has designed Step Test Probe Mounts to facilitate this calibration, which induce precise .007-inch probe gap steps *in place* by turning a lever which applies an offset cam to the probe position. Alternatively, a dial

indicator may be mounted on the probe which can be threaded in and out to enact the test gap change at least 600 RPM speed. Lower speed or turning gear or stationary proximity probe gapping is very inaccurate due to lateral motion freedom in the bearing until the supporting oil wedge is developed.

To complete a full loop calibration, substitute the amplified analog probe signal with a good composite waveform generator (like the HP 33120A) and apply a sinusoidal test signal to the monitor inputs of each channel, with a tracking square wave of known phase offset connected to the monitor phase reference. By varying the sinewave amplitude and frequency, and phase (offset to square wave) you can test the monitor channels over the design ranges of each to verify correct values are obtained.

After in-place probe gain and monitor input verifications we are assured that the system provides accurate data. It has always amazed me that the same engineering management who order five-point deadweight tests on all pressure gauges in a plant approve of sloppy calibration methods to vibration monitors without question while the data is used for critical operation decisions.

#### **d. Lack of Vital, Non-Vibration Instrumentation**

Some current TSI systems do not include sensors nor useable data points for non-vibration parameter measurements which are either directly linked to vibration phenomena (see 4.c., Cause-Effect Relationships Ignored) or are worthy of monitoring in their own right. These parameters include:

- Journal Bearing Metal Temperatures
- Thrust Bearing Metal Temperatures
- Thrust Bearing Oil Drain Temperature
- Torsional Vibration Sensors
- Generator Seal Oil Delivery Temperature
- Lube Oil Delivery Temperature
- Gland Steam Delivery Temperature

Most TSIs do monitor eccentricity, zero speed, casing expansion, and differential expansion adequately.

### **4. Data Management Errors**

#### **a. Alarm Severity Limits and the *Rathbone Chart***

In 1939 T.C. Rathbone drew a *Vibration Severity Chart* as part of his analysis of small machinery vibration standards. The Rathbone Chart, as it has become known, provides a *Very Rough* level line as the default standard for scalar alarm setpoints for small sub-fractional horsepower motors. In a classic extrapolation error this chart has been used on up to 1500 Megawatt steam turbine-generator sets. Vibration alarm setpoints of conventional proximity probe TSI can be traced to the Rathbone Chart if exceeding a single scalar amplitude in vibration velocity or displacement "at each bearing" is presented as the alarm function.

A quick check can determine whether your existing alarm setpoints have been taken from the chart, which yields the following for the respective operating speed vibration:

1800 RPM Machines	7.0 mils (.21 In/Sec)
3600 RPM Machines	3.5 mils (.21 In/Sec)

TSI monitors using the above limits on steam turbines are woefully inadequate and are in immediate need of reconsideration. It does not take a genius to notice that following the modal pattern of the deflection over the rotor span between bearings we can find a wide range of mils displacement. The best determination of alarm setpoints is based upon the rotating element clearances (available from the manufacturer's Spindle Clearance Drawing). Plotting potential modal deflection combinations vs. rotor clearances will provide a good "deflection to contact" measure which can establish safe operating limits of 75% of contact range. Note that the actual alarm setpoint is not a fixed number (as in the Rathbone Chart) but variable according to the modal deflection pattern.

For example, let's look at a typical nuclear low pressure steam turbine rotor which has a 1800 RPM running speed and very large and long-spanned journal bearings. This rotor is symmetrical, i.e. each longitudinal side of the rotor is identical and steam flows from the center out each end to the condenser below. The Spindle Clearance Drawing indicates that the seal clearances axially are set to the following at outage inspections:



Clearance at near mid-span:

.035 inch

Clearance at near 3/8-span:	.040 inch
Clearance at near 1/4-span:	.045 inch
Clearance at near 1/8-span:	.050 inch

These clearances are plotted (red) on Figure 1 along with the first (blue) and second (orange), range of amplitude mode deflection curves. Several important observations are revealed in Figure 1:

- (1) If contact is to be avoided between rotating and stationary turbine elements (a good practice!), the deflection due to unbalance or rotor bow must not reach the clearance at any longitudinal position.
- (2) The relative deflection and therefore safe operating vibration levels of the rotor highly depend upon the modal shape of the deflection. For example, the mid-span .035-inch clearance is not even approached by the second mode at high amplitude but is easily exceeded by first mode at rather low amplitude.
- (3) Using the simplistic Rathbone severity of single-valued displacement alarm level is totally inadequate to warn of hard internal contact and turbine rotor damage.

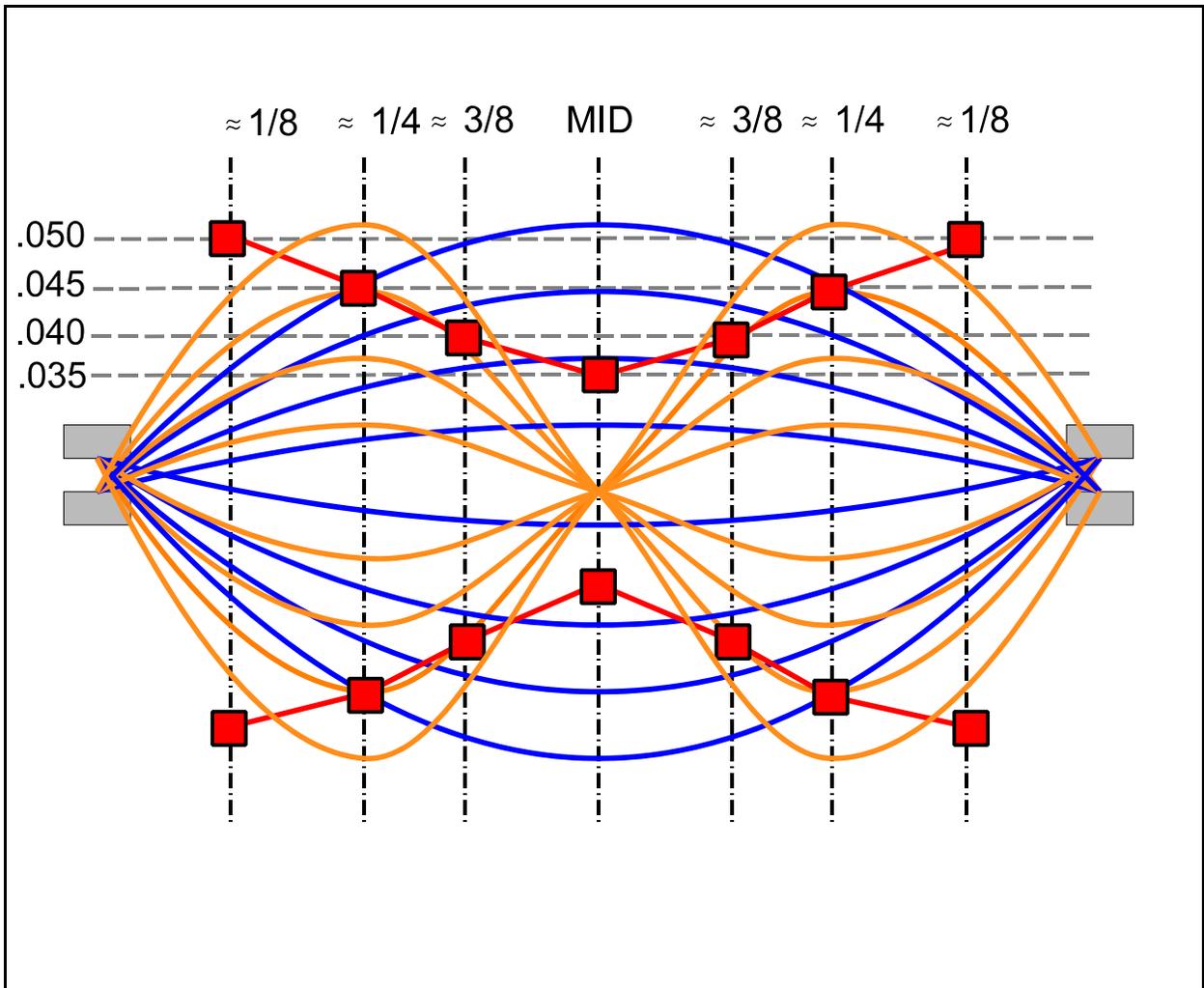


Figure 1

*How are these rotors correctly monitored for unbalance and rotor bows?*

The LCC ALERT Supplementary TSI System Signal Processor (Alert SP) determines the modal deflection pattern based upon the amplitude and phase data from the calibrated Tuned Vibration Monitor (TVM) sensor inputs. The Alert SP is also programmed with the rotor's longitudinal clearance function. Alarms are then determined at run time when the modal deflection at any longitudinal point reaches 75% of the clearance at that point. The arbitrary alarm setpoint determination is eliminated, and the vibration TSI performs exactly the intended function, to warn operators of impending damage, unlike the conventional X-Y proximity probe TSI systems limited to rotor/bearing upset phenomena detections.

## **b. Magnitude Envelope Monitoring vs. Vector Tracking**

A popular technique employed by some supervisory instrument systems is to produce alarm notifications on changes of vibration magnitude rather than absolute levels. The idea promoted is that as long as the magnitudes do not change, the machine remains healthy and there is no cause for concern with respect to vibration. *This is incorrect.*

We have discussed how measured vibration is actually shaft deflections from rotating center applying forces to bearings. These deflections are not scalar quantities. The shaft deflections can only be accurately described (at any particular longitudinal location) as a vector with both amplitude and phase components. Simplifying these vectors to one-dimensional magnitudes for comparison can be likened to advising airline navigation based upon a flat-earth belief, i.e., one-dimension short! The true impact of turbine generator vibration is the force being applied both to the rotor (in bending moments) and the bearing system. Changes in the applied forces, especially sudden, signify catastrophic events or failures which the most modest of TSI systems should detect. The flat-earth logic of comparing only magnitudes without regard to vector positions has a history of missing catastrophic failures as shown in Case Example 5.d. It is quite possible and even likely that major turbine damage such as shroud or blade loss will enact balance changes with applied vector changes resulting in no significant magnitude change or even less magnitude after the event.

## **c. Cause-Effect Relationships Ignored**

Most turbine generator sets have characteristic operating restraints based upon careful control of operating parameters. A useful TSI will incorporate these parameters, and diagnostic monitoring, into the overall protection scheme. Several examples of these cause-effect relationships follow.

Large Westinghouse hydrogen-cooled generators with brushless exciters tend to exhibit large and alarming exciter shaft vibrations when the hydrogen seal oil temperature drops in relationship to the turbine lube oil temperature. The explanation for this phenomena is the shrinking of the babbitted hydrogen seals under cooler oil results in generator rotor upsets which "wag the tail" of the exciter shaft which is rigidly coupled and much lighter. This problem is totally avoided by maintaining hydrogen seal oil temperatures 5 degrees F above the lube oil at cooler exit locations. A valuable TSI feature for these machines is to monitor the two temperatures and alert operators if the difference is approached. The alert can thus *preclude* a vibration excursion.

Large fossil units operating on 3000+ psi steam boilers often must use steam de-superheaters to feed turbine steam gland seal cases. In the case of a malfunction or incorrect operation of the de-superheaters which permits elevated gland steam temperatures, the turbine steam glands will induce thermal "bows" and high resulting vibration on the turbine rotors. The monitoring of the gland steam temperature with alert alarms upon excess temperature can detect these excursions before they induce vibration problems.

Turbine generator sets which are shut down and after brief cooling but with turbine rotors still heated are frequently placed off turning (or barring) gear for a variety of short-term maintenance reasons. If a hot turbine rotor is stopped for several minutes it will acquire a "thermal bow" which if not corrected will result in excessive first-critical vibration and potential seal damage upon startup. The normal correction is to roll the rotor a half-turn from stopped position and hold that position for half the original stopped period. A TSI can be set up to direct the timing and phase angles of the corrective actions, preventing a problem on startup.

After a major overhaul inspection which includes tandem-compound unit alignment moves, it is common to find that after normal operating temperatures are reached during operation a bearing is found to be too low and the resulting decrease in reaction forces and damping causes exaggerated deflections and resulting higher vibration. A simple but effective detection of this phenomena is the comparison of the bearing metal temperatures at each end of all compound rotors. As a rule, the bearing metal temperatures should not differ by greater than 10 degrees F, at which point the cooler bearing is not providing equal support and will likely experience elevated vibration levels. A TSI system specifically monitoring the difference between the rotor bearings will detect and solve this phenomena immediately.

A good source of information to locate such cause-effect relationships for a particular turbine generator set is to examine all known service bulletins issued by the turbine OEM since original installation. Reviewing each bulletin for parameter effect relationships and designing TSI monitoring to provide operators with advisories will add both practicality and convenience for better return on investment.

In some instances the installed TSI system design is too rigid, without the capability of adding new monitoring points nor comparison algorithms to detect and alert cause-effect turbine relationships. In these situations it may be more cost affective to import the functions into an existing plant data computer system or to add a small PLC to handle the additional monitoring functions.

## 5. Field Cases

### 5a. Entergy ANO Unit 1 Main Turbine

In 2000 the Unit 1 turbine-generator set at Entergy's Arkansas Nuclear One plant near Russelville, Arkansas had been suffering what had been diagnosed by the plant vibration analyst as a "strange flooring resonance" since the last unit startup.

The floor was indeed vibrating, so badly that the machine shop below could not operate lathes or milling machines for precision work. The advice given the plant management was that structural experts would be needed to analyze the flooring and perform detailed modal studies to solve the mystery, because the problem was not caused by rotor vibration.

The evidence that the extreme floor shaking was not from the turbine rotor was based upon data acquired by the B-N TSI system, a recent plant upgrade, which monitored no rotor levels greater than 2.0 mils at the 1800 RPM synchronous speed. Management at this point contacted both the OEM (Westinghouse) and Lovejoy Controls Corp. (LCC) for "second opinions", desiring use of their machine shop and not fully accepting the floor resonance theory.

Both Westinghouse and LCC turbine engineers suggested rotor unbalance, which was met with disagreement at the plant due to the low TSI rotor vibration levels. B-N system "orbit" plots were produced showing the low amplitude diameters and accompanying "not evidencing unbalance" determination using the standard B-N analysis techniques.

The LCC engineer, who does not value orbit plots on turbine bearings much, proceeded to take a manual vibration reading using a wooden dowel accessing the shaft of the #2 LP rotor *at the inboard oil seal location*. The resulting reading of 14 mils stunned the local observers. An ANO operator summed up the situation well by stating...

"It's the same old story. Have expert people at the site that know everything and no one to question them. Well they know everything except how to solve the real problem. All the management at ANO walked by the turbine for over a year with it vibrating 10 to 14 mils but the vibration monitoring system said it was only 2 mils so it had to be okay."

The LCC engineer took vector readings and advised an immediate balance correction for the large dynamic couple present. Unit operation continued, however, despite the now apparent rotor vibration cause and a generator hydrogen feed line was sheared causing a forced unit outage.

The #2 LP rotor was balanced on the subsequent startup eliminating both the severe rotor vibration *and* the floor shaking, permitting use of the machine shop and saving further damage to hydrogen lines..

Why did this happen? *Small Machine TSI Design.*

The saddest outcome of the incident is the new concocted explanation, which identifies the low B-N probe readings as due to bearing motion! The explanation given is that the bearing was moving 12 mils, the rotor was moving 14 mils, so the proximity probe relative reading of 2 mils was "correct". The solution proclaimed is the installation of B-N absolute probe mounts and the use of the absolute data for future alarm comparisons.

The fact that the B-N proximity probes were located near nodal points within the bearings and therefore could not read representative shaft deflections remains elusive to the "expert people" the operator referred to. Since the bearing top clearance is greater than 14 mils, the rotor could not have been "picking up" the bearing cap to obtain this reading so we must conclude that *the entire bearing support structure was being compressed 12 mils thirty times a second!* Let's take a look at the mechanics of such a forced compressive motion and review the free body system to see if it is more or less likely than the nodal point monitoring explanation.

The best vibration analysis returns to force analysis to judge the true magnitude of phenomena. The balance correction installed in this turbine which effectively eliminated the deflection was approximately five pounds working at a balance plane radius of about thirty inches. This provides that the deflection force can be accurately calculated as the corrective weight centripetal force:

$$F = mV^2/r$$

where :      F = Force (lbf)      m = mass (slugs)  
                 V = velocity (ft./sec.)      r = radius (feet)

Using the balance move numbers yields:

$$\begin{aligned} V &= 30 \text{ cycles/second} \times \text{radius in feet} \times 2 \times \pi \\ &= 30 \times (30/12) \times 2 \times \pi \\ &= 471 \text{ ft/second} \end{aligned}$$

$$\begin{aligned} m &= 5.0 \text{ lbs scale} / 32.1 \text{ lbs scale/slug} \\ &= .156 \text{ slug} \end{aligned}$$

$$\begin{aligned} r &= 30 \text{ inches} / 12 \text{ in/ft} \\ &= 2.5 \text{ feet} \end{aligned}$$

Solving for force:

$$\begin{aligned}
F &= mV^2/r \\
&= (.156 \text{ slug}) \times (471 \text{ ft/second})^2 / (2.5 \text{ feet}) \\
&= 13,843 \text{ lbf.}
\end{aligned}$$

Since half this restoring force was applied to each bearing, approximately 7000 lbf in rotating force was being applied at each rotor end.

We now have the force available by the case unbalance (7000 lbf) at each end bearing to evaluate. First, let's review how much compression this force could induce into the bearing support structure. We could apply 7000 lbf with a hydraulic jack (albeit with a difficult support apparatus!) and measure the bearing pedestal deflection, but a baseline calculation using the support system stiffness modulus should suffice.

Where A = Pedestal Support Cross Sectional Area , approx. 200 in<sup>2</sup>

m = Pedestal Stiffness Modulus, 3 x 10<sup>6</sup> lbf/in<sup>2</sup> for steel

d = Deflection, inch

s = Pedestal Support Height, approx. 40 in.

$$\begin{aligned}
d &= ((F / A) / m) \times s \\
&= ((7000 \text{ lbs} / 200 \text{ in}^2) / 3 \times 10^6 \text{ lbf/in}^2) \times 40 \text{ in} \\
&= 4.66 \times 10^{-5} \text{ in, or .0466 mils}
\end{aligned}$$

This leaves us short in deflection of the "12-mil explanation" by a factor of (12 / .0466) or 257!

Another "De-Bunking" calculation is available by Free Body Force Analysis. This calculation finds that the 115 Ton, 230,000 lb LP rotor is supported at each end by the journal bearing reacting with 115,000 lbs of force. This force balance remains whether the shaft is spinning or not, provided we assume that the rotor is held within the earth's gravitational field. Since Russelville, Arkansas so qualifies, we have a force balance that requires a rotating force in excess of 115,000 pounds to lift the rotor off the lower bearing halves. With 7000 lbs available from the unbalance we are a factor of 16 times short of a lifting force

We can safely conclude from these calculations that this is case of incorrect probe location near nodal points and that an "absolute" probe system will be "absolutely" useless.

## 5b. Bently Nevada's "Rotor Modeling" paper

Although it seems odd, Bently Nevada has published a paper which very well documents the failure of their typical X-Y proximity probe at bearings TSI system. Published in the Bently Nevada company's own *Orbit* experience journal, fourth quarter 1999 by Ron Bosmans and having the windy title "Our rotor modeling service defines a new balancing standard for a repaired turbine rotor" the paper provides great insights to what B-N thinks a TSI should accomplish.

Let's take some verbatim quotations from this paper and follow each with a realistic evaluation of what is happening...

"A Combined cycle power plant recently experienced two incidents of major rub malfunctions on their steam turbine. The first rub occurred when a turbine blade broke off the rotor during full power operation. The resulting high unbalance forces caused a high deflection in the midspan area of the rotor and caused a rotor rub."

*...The B-N TSI system with proximity probes mounted in the bearings provided minimal amplitude detection and no warning of the rotor rub and subsequent seal damage which was found during disassembly, i.e., the TSI failed.*

"After the rotor was returned to the power plant and installed, an attempt was made to recommission the turbine. During the initial startup plant personnel followed their normal startup and heat-soaking procedures. They accelerated the turbine to 2080 RPM and then held the rotor speed constant to observe the vibration levels. The 2080 RPM speed is just below the frequency of the first balance resonance."

*...The turbine was repaired off site, returned, and restarted but no corrections were made to the blind TSI system.*

"The vibration levels at each turbine bearing were approximately 35.6  $\mu\text{m}$  (1.4 mil) pp. This was considered acceptable to permit acceleration through the balance resonance region."

*...For the second time in a row the large first mode unbalance deflection was not detected by the incorrectly located proximity probes at nodal points and the operators were given a false sense of an acceptable operating condition.*

"However, as they attempted to accelerate through this region, the rotor midspan again made contact with the seals in another major rub event. The turbine was shut down immediately, and the rotor and seals were inspected. Significant damage to the rotor seals had occurred, which necessitated their removal and repair."

*...For the second time in a row the TSI failed to perform as a machinery protection system, resulting in another damaged turbine, major repair, and lost generation. B-N eventually*

*discovered the vibration levels were higher than their system reported, but never mentioned correction of the failing points of the TSI system.*

From this paper we must look very hard to find any value in a TSI system that happily reports the machinery condition as fine, allowing operators to grind the seals off the turbine twice. It also is odd that the TSI manufacturer would be *proud* of this event!

## **5c. Rotor Bow Data Comparison**

A large fossil steam plant in the Midwest experienced an HP turbine rotor thermal bow due to a malfunction of the gland steam supply desuperheater. The turbine was instrumented with a conventional X-Y proximity probe system at the bearings. In a major coincidence of good fortune, the turbine was also being manually monitored with temporary velocity sensors on shaft riders at the inboard oil seals to acquire data for a planned balancing move. To the surprise of the balancing engineer, the balance data readings began to show a large first mode vibration excursion as data was taken.

Reporting the increasing levels to the control room, the balance engineer provided the following:

Bearing #1 Inboard Oil Seal	6.6 mils pp and increasing
Bearing #2 Inboard Oil Seal	5.9 mils pp and increasing

The operators on shift reviewed the TSI system vibration displays and found them to read:

Bearing #1	1.4 mils pp and stable
Bearing #2	1.8 mils pp and stable

Under usual operating conditions the TSI readings were trusted without question. But now a paradox existed. Which readings were correct? The operators alertly took a third reading with a hand held vibration meter at the inboard oil seals and confirmed the balance engineer's readings. Reviewing plant equipment known to cause turbine vibration upsets, they found a stuck control valve on the gland steam desuperheater, freed the valve, and continued manual monitoring. The vibration levels measured manually slowly decreased to under 3.0 mils at each inboard oil seal as the gland steam pressure was brought back under control. The TSI system, however, continued to report less than 2 mils throughout the event without a hint of trouble detection.

## **5d. Zion LP Turbine Blade Loss Incident**

A classic example of rotating mass loss occurred at the now decommissioned Zion Nuclear Plant in 1988. The Unit 2 LP-C rotor had been operating with a vibration vector level as measured at the inboard oil seals by shaft rider mounted velocity pickups as:

Bearing #3	5.5 mils @ 220 degrees referenced to exciter notch
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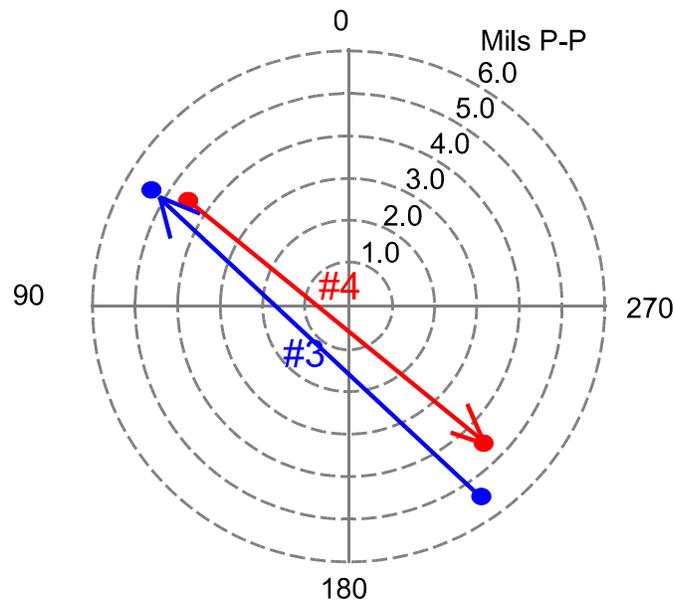
Bearing #4 4.5 mils @ 50 degrees referenced to exciter notch

A loud report was heard by operators in the turbine vicinity. The vibration levels were examined again and found to be:

Bearing #3 5.4 mils @ 72 degrees referenced to exciter notch

Bearing #4 4.4 mils @ 195 degrees referenced to exciter notch

At this point it appeared that whatever happened, the turbine vibration levels slightly *decreased*, and therefore shouldn't be cause for concern. If we look at a polar vector plot of the data, however, the true magnitude of the event becomes clear.



Although the before and after vibration amplitudes (distance from plot center) didn't change much, the vector moves "crossed center" indicating a sudden and large change in rotating mass. The resulting inspection revealed the cause to be three lost shroud on the L-2 blade row, an event undetected by conventional monitoring even with amplitude envelope monitoring.

## 6.0 Summary

Summarizing the differences between conventional X-Y proximity probe systems and the LCC Alert Supplementary TSI System:

### Conventional X-Y Proximity Probe Systems:

- Have probes mounted in the wrong longitudinal location, too close to bearings, unable to measure the most common turbine problems of unbalance or bow.
- Have error-prone proximity probe calibrations which do not account for shaft radius nor reflection targets plus false electrical runouts which in total lead to +/- 30% displacement measurement error.
- Use a simplistic single scalar alarm level value from the ancient Rathbone Chart and do not account for actual machinery internal clearances nor the modal deflections of the rotor.
- DO successfully detect Steam Whirl and Oil Whip phenomena.

### LCC Alert Supplementary TSI System:

- Use precision Tuned Vibration Monitor sensors which detect unbalance and bowed rotor conditions immediately.
- Use a calibration procedure defining vibration levels at all longitudinal rotor positions to within 5%.
- Use rotor clearance vs. modal deflection pattern to determine alarm status based upon internal contact approach.

In the early days of auto theft alarm systems many people joked that the systems fell into two categories, those that never alarmed and those that went off all the time. If a conventional vibration TSI is used it meets the former category, perhaps protecting against the now exceedingly rare oil whip and steam whirl, but failing in the vast majority of turbine vibration phenomena detection. To resolve this liability arrange the installation of an LCC ALERT Supplementary TSI System and begin true turbine protection.