Turbine Supervisory Instrumentation Systems

Principles, Design, and Operation

Technical Brief — *Steam Turbines—Generators and Auxiliary Systems*

Introduction

The turbine supervisory instrumentation (TSI) systems provide valuable information to operators and system owners about the state of the equipment. These systems are implemented using a wide range of techniques that vary by vintage, original equipment manufacturer (OEM), and plant, with their own inherent reliability and uncertainty. A representative TSI system layout is shown in Figure 1. The operator relies on TSI to identify turbine-generator anomalies and mitigate risks during transient loading operations. A system engineer relies on the information provided by the TSI for a variety of tasks, including setting alarm and trip points, diagnosing equipment degradation issues, managing risk, and proactively determining a maintenance strategy. To make appropriate decisions based on this information and to set up any individual system so that the output is both valid and actionable, an understanding of the techniques used by turbine designers is critical. This technical brief introduces the intent and function of the supervisory instrumentation systems and serves as a primer for plant staff.

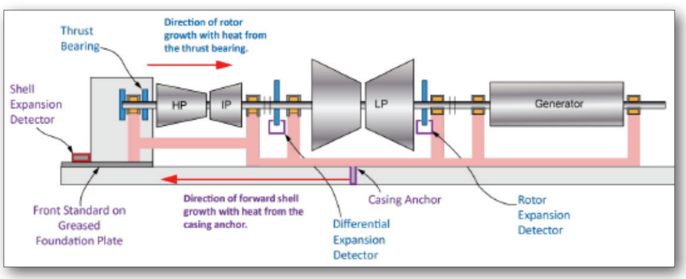

Figure 1 – Typical TSI turbine-generator locations

Thrust Wear Protection

Every turbine has a thrust bearing assembly responsible for keeping the stationary and rotating elements in their proper axial relationships. The thrust wear protection system monitors the rotor axial position for evidence of thrust bearing wear and will alarm and possibly shut the turbine down when the thrust bearing fails or might be expected to fail within a short period. The turbine thrust bearing assembly holds the rotor and the stationary components in their correct relative axial position.

For most turbine units, a single thrust assembly holds all the rotors axially. Basically, one turbine rotor has the thrust assembly, and all the other rotors are interconnected through rigid couplings. Some less common designs also have a thrust assembly for the driven device (generator or pump), and these designs need some form of flexible coupling to allow the rotors to expand independently. To maximize turbine efficiency, the designers usually maintain the inlet and high-pressure section axial sealing clearances at close to ideal levels. Therefore, the thrust assembly is typically placed in a bearing enclosure or a standard on one end of the high-pressure section or the other.

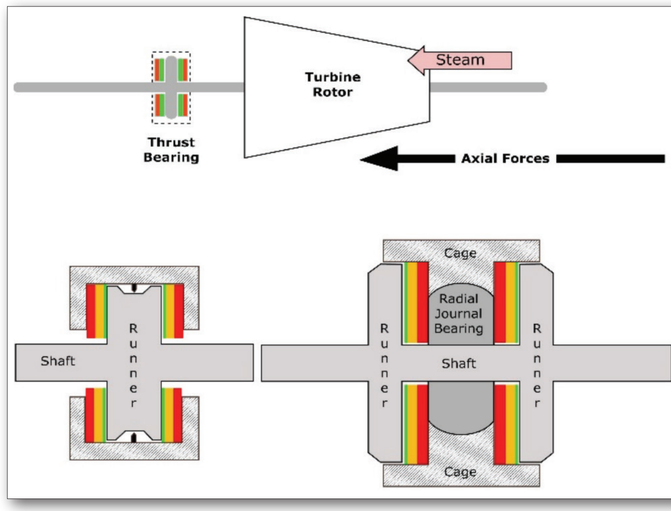

Figure 2 – Thrust bearing configurations

As shown in Figure 2, the thrust bearing assembly uses two nonrotating thrust bearings that are configured in opposite directions—either facing each other or facing away from each other. These thrust bearing plates work through a lube oil film wedge, against two rotor thrust runners. These runners (or collars) are finely machined rotor surfaces that are perpendicular to the rotor centerline and that obviously spin with the rotor. The oil film wedges, therefore, hold the rotor in the correct axial position against all of the forces that would try to push or thrust the rotor. Increasing the rotor thrust will compress the oil wedge until the oil forces match the steam forces.

The discussions and recommendations on protection systems are unaffected by whether the stationary thrust plates are designed and built as tilting pad, tapered land, or even flat land type. In all of these designs, there is an observable and limited rotor axial motion, called *rotor float*. Thrust bearing axial plate liners are flat-machined to set the mechanical float during unit erection and commissioning. This thrust float happens partly because the thickness of the lube oil film changes with the direction and magnitude of the thrust forces. Also, one can observe some rotor float from thrust bearing axial assembly clearances and some limited elastic deflections.

The thrust bearing wear protection involves one or more position-sensing devices, sometimes called *rotor axial position probes*, to observe the rotor axial position inside or, according to American Petroleum Institute (API) Standard 670, within 12 in. (304.8 mm) of the thrust bearing assembly. Because the thrust bearing is the anchor for the rotor inside the shell or casing, differential thermal expansion will not be seen at this location. One would expect to see only the normal rotor float, as previously described. Beyond the normal rotor float, the thrust wear monitoring system should allow for a reasonable amount of thrust bearing wear before issuing an alarm. How much wear to tolerate is more a matter of opinion than of fact, and settings might vary from one apparently identical unit to another. The following is a rule-of-thumb formula for calculating rotor float: Allowable rotor float = 0.0009 × shaft diameter (in.) + 0.006 in. (0.0009 × shaft diameter [mm] + 0.15 mm). The clearance within the rotating equipment dictates the amount of allowable end play. The alarm limit basically needs to keep the thrust bearing from totally failing, such as would happen if the oil wedge collapsed.

With excessive rotor motion, such as when a thrust bearing fails from being overloaded in a water induction incident, the unit should be shut down to eliminate or at least minimize the axial rubbing damage.

An appropriate location of thrust position probes is illustrated in Figure 3.

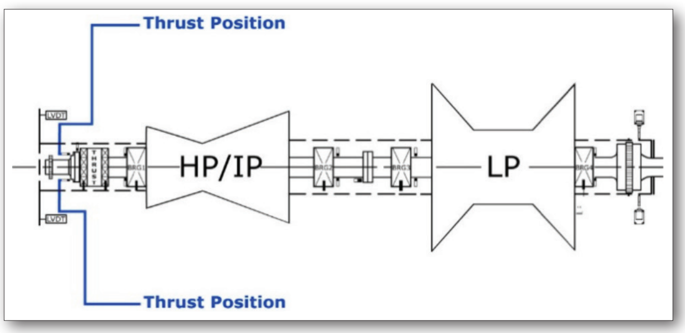

Figure 3 – Thrust bearing axial position

Note that where the rotor thrust force is always in the same direction during operation, the thrust assembly that carries the load is usually called the active thrust, and the other is called the inactive thrust. On many units, the net rotor thrust changes direction at some operational point, and the active thrust role will change from one thrust assembly to the other. In these designs, it is preferable to identify the thrust assemblies as generator end/turbine end (GE/TE) or drive end/non-drive end (DE/NDE), based on their locations.

In addition to the actual rotor thrust bearing wear indication, embedded thermocouples in the thrust plates/shoes will indicate the temperature related to the bearing load. Excessive load could be identified by high-temperature alarm points.

Figure 4 shows an arrangement where differential oil pressure is used to detect the effect of axial loading at the thrust bearing.

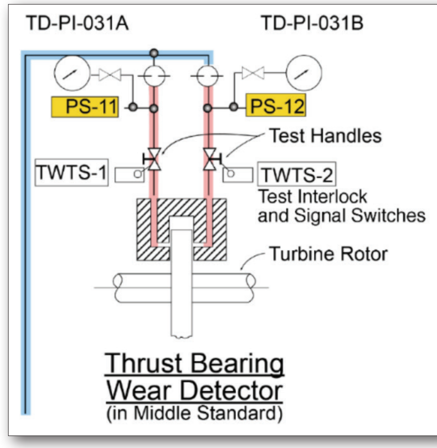

Figure 4 – Typical thrust bearing wear detector (oil pressure)

Eccentricity

When a turbine rotor has temporary bowing, it is likely to have high vibration during startup and until it straightens out with heat. The eccentricity probe system basically shows whether the eccentricity is consistent with previous startups and whether the rotor bowing is stable or changing while on turning gear. Therefore, the eccentricity probe provides information on the rotor startup preparation. The word concentric literally means "having the same centers." If one had a "perfect rotor," it would have perfectly round journals and perfectly uniform round sections everywhere. It could then be called a perfectly concentric rotor because the center of rotation and mechanical centerlines would agree exactly or be concentric. In this perfect rotor, the dial indictors in runout checks would read zero everywhere.

In reality, if one looks with enough precision at real rotors, one will find that there is always some eccentricity, which is the opposite of concentricity. In the real world, therefore, eccentricity is the term we normally use to describe the straightness of the turbine rotor. A perfectly straight rotor is said to have zero eccentricity, meaning that the rotor surface is perfectly concentric to the rotor centerline.

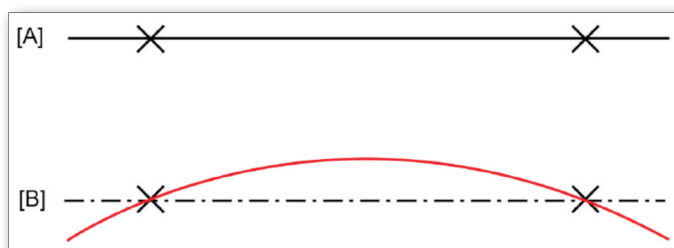

Figure 5 – Straight versus bowed rotor centerlines

Figure 5 illustrates the shape of the shaft, where [A] represents a straight shaft with no runout or eccentricity. The line drawn between the two bearings (the exes) is the rotor centerline. Part A in Figure 5 represents a straight rotor. Part B is the same rotor bowed somewhat. A bowed rotor is the most common cause of high eccentricity.

When a bowed rotor is rotated, the centerline of the shaft will generate a radial displacement, as suggested in Figure 6.

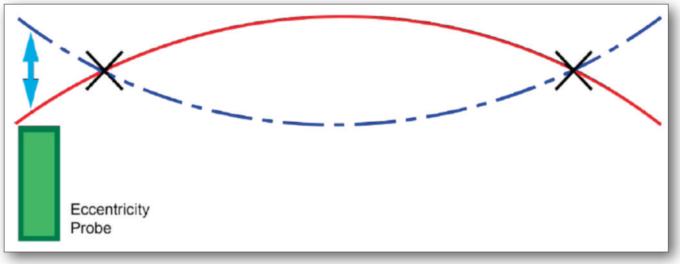

Figure 6 – Rotating bowed rotor centerline

Assuming that the rotor is round and a dial indicator is reading the rotor surfaces during rotation, an observer would witness runout everywhere but in the bearing supports. It is this eccentricity that will be covered here.

Figure 6 illustrates several important aspects of eccentricity readings:

- First, we might benefit from being able to read the eccentricity between the bearings. However, this would involve taking accurate readings inside the turbine, where heat and vibration make it almost impossible for instruments to survive.
- As an acceptable reading location, designers choose to read the rotor eccentricity outboard of the bearings, either on the rotor or on some rotor extension connected to the rotor.
- The farther from the bearings, the larger the eccentricity, and, probably, the better the readings. The actual measurement at this location is representative of the axial distance from the eccentricity sensor pickup to the bearing centerline, compared to the bearing centerline to the turbine axial centerline. As an example, 0.004 in. (0.1 mm) indicated eccentricity—depending on this distance, the ratio might be 3–15 times greater at the turbine centerline (12–60 mils). Note that some people refer to a bowed rotor as an *out-of-round* rotor, but this is not precisely accurate. The rotor might have perfectly round elements but a deviated or bowed centerline.

There are several major rotor curve classifications or shapes, as follows:

Temporary bow – A temporary rotor bow is a curve that obviously will go away with time. A temperature difference across the rotor diameter is the primary cause of a temporary rotor bow.

Any metal object will expand as the average temperature increases and shrink as the temperature decreases. Consider a high-pressure/intermediate-pressure (HP/IP) turbine rotor, such as the one illustrated in Figure 7.

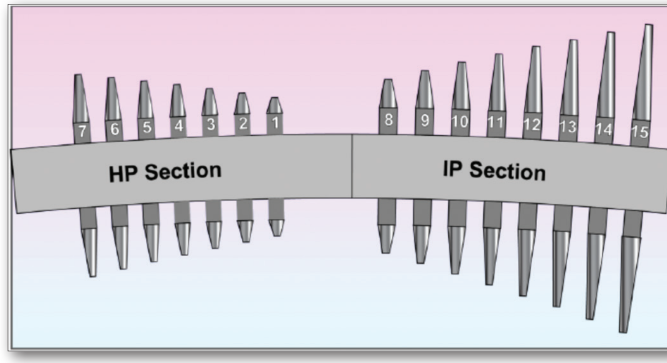

Figure 7 – Bowed turbine rotor

The turbine was recently operating at 800°F (427°C) but is now at a standstill. Heat rises, and as it does, the rotor temperature will become greatest on the top. The rotor will bow toward the hot side, just as any piece of metal would bow toward the hotter side and away from the cooler side.

While stopped, the rotor centerline changes from what is seen in Figure 5 (A) to what is seen in Figure 5 (B).

Another minor cause of temporary rotor bowing comes from temporary surface stresses that are effectively "locked in." This is a form of hysteresis that will normalize or disappear during turning gear (barring gear) or slow roll operation and when the rotor temperatures become uniform.

Permanent bow – A permanent bow happens because the rotor has substantially more locked-in stresses than can be relieved in normal operation. These stresses are from the rotor somehow yielding or plastically deforming. Such damage is typically caused by local hot spots (a severe rotor rub) or quenching (a hot rotor stopped and sitting in water) or some mechanical bending (lifting a rotor incorrectly). In each case of permanent rotor bow, the rotor was strained in tension or in compression beyond its yield point. The rotor will remain bowed forever unless corrective maintenance actions are taken.

Note that most permanently bowed rotors have kinks or bends, not the simple bowing shown in Figure 7. They might have multiple distinct kinks, in fact.

Sag – Rotors sag because of the rotor weight distribution, their elastic flexibility, and the bearing supports being located near the rotor ends. Many power plant novices think of this semi-catenary curve as some form of rotor bowing. However, for the purposes of this discussion, rotor sag is not called *bowing*. Sag is simply the curved location for the rotating centerline, and the sagging does not affect the eccentricity. In fact, rotors should spin to rated speed with their centerlines basically sagging to this same curve.

To monitor rotor straightness, an eccentricity detection system is typically installed in the turbine front standard. In smaller axial flow exhaust turbines, the generator is rigidly connected to the HP rotor, inside the front standard, and the eccentricity probe reads outboard of the LP turbine bearing instead. Figure 8 shows a typical front standard installation.

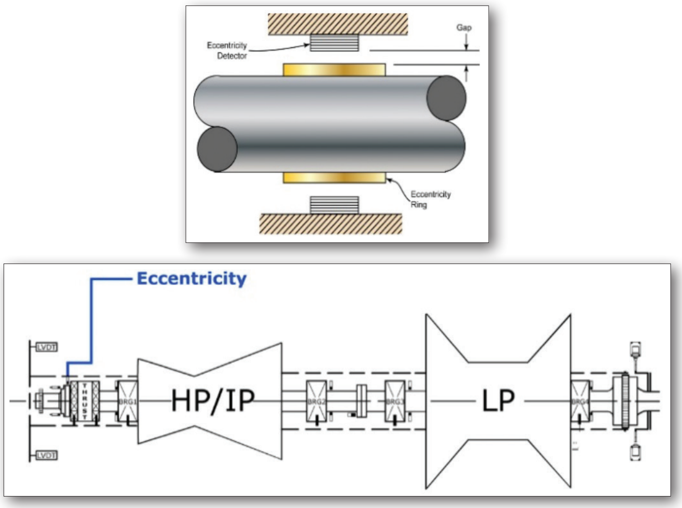

Figure 8 – Front standard eccentricity probe

Wherever the eccentricity is read, there will be a radial surface that is finely machined to get a minimum runout. When this is in the front standard, there is often a control rotor coupled to the HP turbine. Assembled onto the control rotor is an eccentricity ring that was machined according to the journal datums, resulting in a near-perfect circle.

An eccentricity detector probe is mounted on a stationary part, and it is properly aligned and gapped to the eccentricity ring. The instrumentation is then calibrated to have an output that is linear with this air gap. On turning gear, therefore, the air gap dimension will vary as a function of how straight or how bowed the adjacent rotor is.

The interpretation of this eccentricity signal is then very important, as will be explained in the following in the discussion of logic schemes. It is a good idea to consider taking dial indicator readings on the rotor surface, where the probe is mounted, to confirm that TSI eccentricity readings are accurate and to check whether readings change with axial movement, such as from thermal growth.

However, it is important for the operators to know that eccentricity should be the same as it was the last time the unit started without any problems. If the eccentricity is different—either greater or less—the rotor has a temporary bow. More precisely stated, the rotor runout vectors should be identical over time. During various maintenance activities, such as slow speed balancing or machining, it is normal practice to address the characteristics of turbine rotor bowing. It is expected that rolling the rotor slowly for a period will remove any temporary bow before taking any corrective actions.

Similarly, in a turbine startup, it is very important to remove any temporary bow before rolling the machine on steam. Obviously, a temporarily or permanently bowed rotor that is operating through the first critical speed range would result in higher vibration. If this transition is done incorrectly, the high vibration might produce rubbing and even more significant damage.

As a final note, once the turbine is rolled on steam, the eccentricity reading is of no practical value, and it can be ignored. What the eccentricity probe reads above 600 rpm is more affected by rotor dynamics than the rotor being bowed. Therefore, if the eccentricity is observed on newer units, it should be considered only as a supplement to the shaft vibration readings. For the mechanical hydraulic control and analog electrohydraulic control units that had a strip-chart recorder showing eccentricity, speed, and control valve position, the eccentricity can be interpreted only on turning gear operation.

Shell Expansion

The HP, IP, or HP/IP shells will expand significantly because the temperatures of these shells change from ambient conditions to full load. If the shell expansion were somehow restricted or limited, the resulting shell distortion could produce issues with rotor axial clearances, journal bearing alignment, vibration, or bearing loading. Therefore, turbines have a system for monitoring shell expansion.

Typical for older and larger units, the stationary shell axial locations are held or established by some type of keys or anchors near the turbine exhaust centerline. The turbine designers then make provisions, elsewhere, for the high-temperature HP and IP shells to axially expand from this anchor point. This design is driven because the condenser is large, extremely heavy when full of water, and not easily made flexible. Therefore, securing the turbine axially near the condenser centerline means that the LP turbine exhaust is aligned to this inflexible condenser throughout the load range.

An exception to this normal design is seen for axial exhaust turbines and most mechanical drive (feed pump) turbines. These might have an axial anchor at the front standard and provisions for thermal expansion on the exhaust end.

Similarly, on smaller and newer units, the shell axial earth anchor is sometimes in a bearing enclosure or standard next to the hood rather than on the hood centerline. For reference, the front standard is farthest from the exhaust hoods, and there could be a mid-standard between an HP shell and IP shell.

Also, note that LP hoods (outer cylinders) and condensers are relatively cool, and therefore they do not grow much horizontally from startup to full load. They have provisions for things like vertical thermal growth and vacuum deflections, of course.

The high-temperature shell expansion provision varies with size, year of manufacture, and turbine vendor (OEM):

- Some smaller machines might use shells that slide on fixed keys on the turbine horizontal centerline or some minimum distance below the turbine horizontal centerline.
- Older and smaller machines typically have flex legs, which are tall, thin vertical plates running from one side of the unit to the other. These flex legs are attached to the foundation below and support the shell above. The shell can then expand freely while the flex legs swing through an arc.
- Newer large machines tend to use sliding standards to support the shells and to move freely on greased plates as the shells expand. These sliding standards are axially linked to the high-temperature shells, and they use a guide key and slots to hold the transverse (side-to-side) alignment correct.
- The traditional shell expansion detector is mounted to the sides of the front standard. Figure 9 is an illustration of the shell expansion detector location relative to other steam turbine components for a unit with a combined HP and IP shell. For a unit with a separate IP shell, there could also be shell expansion detectors on the mid-standard.

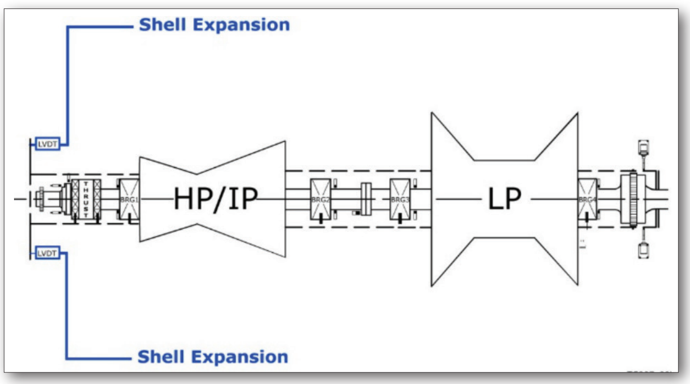

Figure 9 – Shell expansion location

Although most units have only one shell expansion detector on a given standard, Figure 9 shows one on each side. The advantage of having two shell expansion detectors is for redundancy and for detecting sticking or twisting of the standard. If the standard is sticking, the two shell expansion detectors will potentially read differently, especially if the centerline key and slot have excessive clearance.

Rotor Expansion

The rotor expansion instrumentation is an extension of the previously described differential expansion system. They both monitor the rotating element expansions so that the designed axial clearances are maintained without having any axial rubs. The main difference that distinguishes a rotor expansion instrument is that the probe support system is essentially fixed and not movable. Therefore, it measures an absolute value of rotor expansion rather than a "differential" expansion.

In most large steam turbines, a rotor expansion detector (RXD) is found near the turbine-generator coupling, which is very far from the turbine thrust bearings. As previously mentioned in the discussion of the differential expansion system, the higher-temperature HP and IP turbine section's stationary and rotating elements are designed to essentially expand together. This flexibility means that most of the axial rotor and casing thermal expansions can offset or cancel each other with very little net differential expansion to address. However, throughout the length of the LP turbine section, very few turbine designs have comparable stationary-component axial expansion compensation or correction. Therefore, most LP rotors have thermal expansion that is simply monitored by an RXD. Changes would be seen from several factors, including load, steam temperatures, startup rates, condenser cooling water (or air) temperatures, and even foundation growth.

An RXD works exactly like the previously described differential expansion detectors. In fact, not all turbine OEMs designate this LP expansion reading as *rotor expansion*; some call it *LP differential expansion*.

Finally, there are some RXDs installed into higher-temperature turbine or bearing locations, as previously covered relative to the differential expansion detector configurations. Where there is no space or practical way to directly measure the differential expansion, the turbine designers use both a shell expansion detector and RXD to produce a calculated differential expansion output. Therefore, RXDs might be found in front standards or other bearing or coupling enclosures.

Differential Expansion

Turbines are built with precisely engineered internal axial clearances between each of the stationary and rotating elements. The high-temperature sections are built so that the stationary and rotating elements basically expand to keep the axial clearances close. Turbines have differential expansion systems to monitor these relative thermal expansions so that neither the stationary nor the rotating elements expand so quickly or excessively that there is rubbing where there should be axial clearances. Differential expansion is a measure of the axial clearance between the stationary and rotating components of the turbine. High-temperature rotors are also expected to grow away from the thrust bearing. However, the rotor and shell could grow different amounts, so the internal clearances could be exceeded, especially because of a difference in thermal growth rate.

The profile of this surface varies with manufacturer and design configuration. Figure 10 shows an arrangement with probes indicating radial and axial displacement.

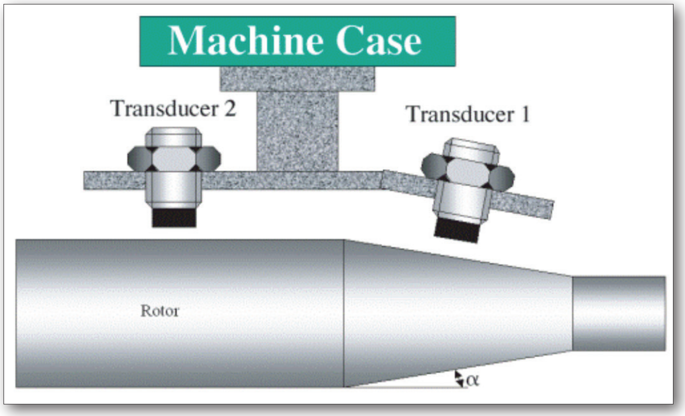

Figure 10 – Differential expansion: tapered shaft

There are many reasons why rotors and shells have different rates and amounts of thermal growth:

- The turbine shell is frequently more massive (it has more thermal mass) in comparison to the rotor cross-section. Therefore, the lighter rotor would be expected to change temperature and length at a more rapid rate than the shell.
- The turbine shell is made of a different grade of steel than the rotor and would therefore have a different thermal expansion coefficient and resulting thermal growth.
- The turbine rotor is surrounded by steam, and heat will continue to transfer into, out of, and through the rotor until it matches the steam temperature throughout. In contrast, the shell is heated from the inside and transfers heat to the outside environment through the insulation. Therefore, the shell's average temperature will be less than the rotor's average temperature, which produces thermal expansion differences.

The differential expansion detectors monitor the relative growth at least one shaft away from the thrust bearing. Depending on the unit configuration, some differential expansion detectors are mounted in components that move axially, and some are in components that are essentially fixed.

Figure 11 illustrates a configuration where the thrust bearing is in a sliding front standard and next to the journal bearing 1. Therefore, the differential expansion detector is located near journal bearings 2 or 3, which are fixed or expand very little.

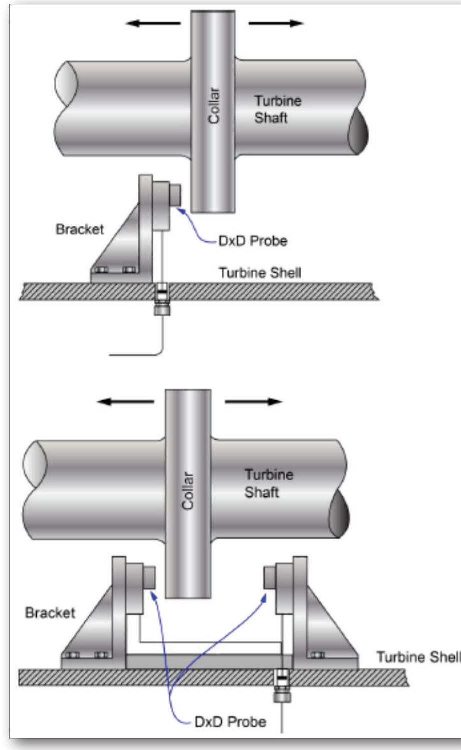

Figure 11 – Differential expansion with thrust in front standard

As discussed previously, when the shell expands, it pushes the front standard away from the axial earth anchor. In Figure 11, that would mean that the front standard and enclosed thrust bearing would move to the left. That shell growth would result in the air gap on the left side of the differential expansion detector shrinking. As the rotor expands away from the thrust bearing, it would move toward the right in Figure 11, and the differential expansion detector air gap would expand. Therefore, as the name indicates, the differential expansion detector air gap changes with the difference between the shell expansion and the rotor expansion.

Alternatively, Figure 12 illustrates a turbine configuration where the thrust bearing is located near the journal bearing number 2, and a differential expansion detector would be located inside the front standard. As the shell expands, the differential expansion detector brackets would move left with the front standard.

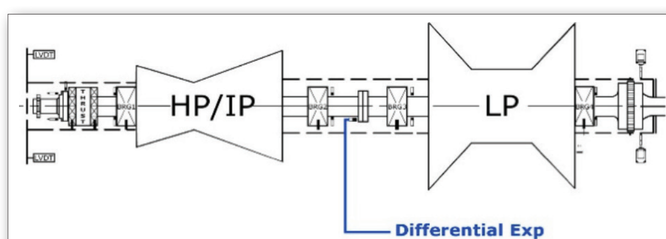

Figure 12 – Differential expansion with thrust in middle standard

As the rotor thermally expands, the differential expansion detector observes the rotor also moving to the left. The probe air gap is then a direct measure of the axial location of the thermally expanding rotor relative to the axial location of the corresponding expanding shell, or the differential expansion.

There are also units built with an IP (or reheat) shell that is separate from the HP shell. These typically have a second differential expansion detector between the IP and LP rotors. Because these units typically have the thrust bearing in the sliding middle standard between the HP and IP sections, this second differential expansion detector is very much like the one shown in Figure 13 because there is a moving thrust bearing reference point and an essentially fixed differential expansion detector probe. As a final differential expansion detector configuration, there is a group of turbine generator designs in which there was no space or practical way to directly measure the differential expansion. Here, the designers apply both a shell expansion detector and absolute RXD. The computer system combines these two inputs and displays a calculated differential expansion output, which is the difference between these two detector readings.

When the rotor has expanded more than the shell, we refer to this condition as being *rotor long*. When the rotor has contracted more than the shell (or the shell has expanded more than the rotor), we refer to this as going rotor short. A rotor long condition could be expected while the unit is being loaded (heat is being applied), and a rotor short situation might be expected when the unit is being cooled (decreasing both load and steam temperatures). Turbine designers provide limits on how much differential expansion to allow. They could supply both high alarm values, which warn operators to adjust their actions, and high-high values, which warn of damages. These limits are typically called *red bands*, and the (older) strip chart recorders would have their scales painted red at the ends. Probably most computer (HMI) screens today still follow the same convention. If the differential expansion detector were to indicate turbine operation in either of these regions, it would mean that the safety margin for axial clearance is gone. An axial rub could begin to occur at any time.

The *green band* is a similar reference mark to show how the cold turbine was assembled in the ideal axial rotor position. When a turbine cools again to ambient conditions, the differential expansion detector should indicate the rotor's return "back in the green band."

Operation *near* either red band is not normal, except during an urgent cold startup, and operation in either red band should be completely avoided. With proper use of the differential expansion data, the plant should be taking corrective action long before entering the red band.

The following list describes some problems that could produce a rotor long condition:

- **Turbine shell not expanding.** The confirmation of this is in the shell expansion reading. Does it have less-than-expected growth? Is there evidence of obstructions, such as from a steam line support or sliding fit problem?
- **Temperature mismatch.** When the steam is much hotter than the turbine metal, a temporary rotor long condition might be expected.
- **Excessive rate of temperature change.** The unit is basically being loaded too fast. With a given steam temperature mismatch, changing the steam flow changes the temperature ramp.
- **Misoperation of feedwater heaters.** This suggests water induction, which was once commonly caused by significant heater tube leakage. These events put water into the bottom of the shell, which could cool it faster than the rotor. A long rotor, in this event, is a minor indication of a major problem.
- **Thrust bearing failure.** A thrust bearing failure will result in the rotor moving relative to the shell. It is not growth, but the effect on the instrumentation is the same.

A rotor short condition occurs when the casing growth exceeds the shaft growth. This could be the result of any of the following problems:

- **Temperature mismatch.** Steam temperatures are too cold, or the metal temperature had previously gotten too hot.
- **Excessive rate of temperature change.** The machine is being unloaded much too rapidly, in effect, force-cooling the rotor.
- **Thrust bearing failure.** Again, this suggests water induction, which tends to push the rotor upstream and would more likely fail the thrust bearing that was not loaded by the steam flow.

Vibration

There is an extensive arrangement of vibration instrumentation installed to detect a variety of turbine problems. These problems include failures of rotating components; failures in the rotor support structures, including journal bearings, foundations, and others; operational problems, such as packing rubs, oil whip, steam whirl, shorted field coils, and plugged cooling passages; and general deterioration, such as from non-uniform erosion or foreign material deposition. Typically, vibration instrumentation is located at every radial journal bearing.

Normally, the vibration instrumentation is directly observing the rotor radial vibration. Sometimes, it is mounted on the bearing cover or bearing enclosure, and it measures an effect of the rotor vibration on the stationary environment.

Turbine-generator rotors are assembled from thousands of rotating elements that spin at high speed and with significant thermal, centrifugal, and other stresses.

Obviously, if any of these components suffer failures, the resulting unbalance will produce a change in vibration that should be addressed. Likewise, because all rotors have some residual unbalance, some once-per-revolution forces must be absorbed by the stationary environment. The lubrication oil system, the journal bearings, the bearing supports, and the foundations all work together to dampen these vibratory forces. Almost any changes in this energy dampening system could have effects on vibrations. These support system changes would include varying oil flows or varying oil temperatures; bearing wear and electrical pitting; foundation cracks, corrosion, and other damage; loosened foundation or bearing assembly bolts; permanent or temporary elevation alignment changes; and soft-foot/rocking supports.

Oil whip, oil whirl, and steam whirl are all unusual forms of rotor instabilities that appear almost instantaneously and can seem to disappear just as quickly. Though unusual, they can be destructive and are always disturbing. Proper analysis of the vibration signatures, especially looking at the vibration harmonics, will be useful in identifying the source of such rotor instability and could limit any repetitions. The most common rubbing (contact between rotating and stationary turbine elements) happens in the close-fitting packings. Regardless of the location, there is localized heating of the rotor surface and some resulting thermal bowing.

During loaded operation, such rubbing could be both disturbing and physically harmful to clearances. During low-speed operation, unfortunately, such rubbing can be extremely destructive. Monitoring of the vibration indications, awareness of rubs, and correct operator response can make a critical difference in the results. Finally, there is a large body of operating problems for which vibration data gathering and analysis are invaluable to problem analysis and solution. These include generator problems such as thermal sensitivity from shorted coils, plugged cooling passages, insulation or coil stick and slip, core iron resonance issues, and end-winding resonance issues. Similarly, the turbines show vibration evidence from all forms of general deterioration, such as from non-uniform erosion or foreign material deposition.

Risk-Based Considerations

The following information should be used to qualitatively determine what factors might be increasing or decreasing a unit's event risk and thus how much or how little conservatism is appropriate for the various TSI schemes. Devices with no protective function are excluded from this list. Generally, the following factors need to be evaluated:

- **Operating mode** – Flexible operation or cycling service presents a variety of challenges to equipment. The resulting transient periods increase the need for accurate instrumentation for understanding system conditions.
- **Equipment configuration** – Any given system design has an inherent risk profile for a variety of issues. In general, a more complicated system has more risks than a simpler one.
- O**perating history** – Poor ability to control operating parameters, such as steam temperature/pressure, oil temperature, and similar issues, has a direct bearing on risk. History of poor operation is a strong indicator of increased event risk and equipment damage. Finally, poorly documented unit history leads to uncertainty and thus higher risk.
- **Equipment reliability** – Units that have equipment with frequent and known issues are generally at higher risk for future problems.

These factors will contribute operational exposure to each critical supervisory parameter.

Thrust Wear – Factors that lead to higher or lower risk, respectively, include the following:

- Higher risk:
  - No bearing-metal monitoring, or drain-oil-temperature-only monitoring
  - Flat/tapered thrust plates
  - Steel-backed thrust plates
  - Shrunk-on thrust collar
  - Unmonitored thrust load
  - Poor lubrication fluid quality
  - Water induction history (feedwater system)
  - Multiple opposing high-reaction steam path sections with full load range operation
  - Peaking or cycling load
  - Poor attemperation control
- Lower risk:
  - Conformance to water induction prevention standards (ASME TDP-1 or TDP-2)
  - Bearing metal temperature monitoring
  - Tilt pad
  - Copper-backed thrust plates
  - Integral thrust collar

Differential Expansion – Factors that lead to higher or lower risk, respectively, include the following:

- Higher risk:
  - Double shells
  - Higher inlet steam pressure
  - Less prewarming
  - Faster start times
  - Peaking or cycling load
- Lower risk:
  - Single shell
  - Lower inlet pressure
  - More prewarming
  - Slower start times
  - Base load

Vibration – Factors that lead to higher or lower risk, respectively, include the following:

- Higher risk
  - Residual unbalance
  - Fast acceleration
  - Tight packing
  - Cylindrical and elliptical sleeve bearings
  - Steam whirl tendency
  - Peaking or cycling load
  - Poor attemperation control
- Lower risk
  - Loose or variable clearance packing
  - Tilting pad bearings
  - Anti-swirl packing

References

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Acknowledgments

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