Technical Paper

How to Prevent Turbomachinery Thrust Failures

By Charles Jackson

Thrust failures in high speed turbomachinery can cause a considerable loss of plant production. These failures can easily be prevented if thrust bearings are installed correctly and rotor axial movement is monitored. Here are the important facts. Machines such as turbines and centrifugal compressors can be severely damaged if the thrust bearing fails unexpectedly. Failure can easily cost over USD $500 000 including lost production. Such a failure can be prevented if careful consideration is given to the machinery and its protective instrumentation.

Most machines of this type have a thrust bearing of the hydrodynamic (slider bearing) type that develops an oil wedge lifting the thrust collar or runner from a stationary bearing, which could be of the straight flange, tapered land or tilting pad type. Actually, a thrust bearing is really an assembly that has two bearings. One bearing takes the normal or expected thrust loads imposed by the rotor. The other bearing would therefore be located, in design, to take abnormal or unexpected thrust loads. Many vendors will refer to one as the active (normal) thrust bearing and the inactive (counter or abnormal) thrust bearing.

Because of thermal expansion of the bearing elements, the two bearings could grow into one another, causing severe loading and immediate failure. For this reason, the contact on each bearing is separated a certain distance based on the operating environment temperature. This separation distance is often referred to as the float zone between active and inactive thrust shoes (pads, plates, surfaces). For steam turbines, this distance might be 0,23 to 0,36 mm *(0.009 to 0.014 in.)*. Thrust bearings for steam turbines will normally be at the steam inlet end since the rotor will expand from the thrust bearing. The blading (rotor) should move away from the nozzles and stationary blading, as this is a critical and sensitive dimension.

On the other hand, centrifugal compressors are more lenient, with cold rotor floats of 0,38 to 0,56 mm *(0.015 to 0.022 in.)* typically. Total rotor float of a centrifugal compressor (with thrust bearings removed) moving axially in the case from one staging to the next could be in the order of 6,35 mm *(0.25 in.)*.

There is nothing unusual when a machine, in operation floats randomly between the active and inactive bearings. I would consider this quite healthy exposing basically no thrust bearing loads on the bearings, provided an axial vibration is not set up that could cause fatigue or efficiency swings.

Thrust loads can be measured in many ways: bearing metal temperatures, oil exit temperatures or heat differences, load cells behind the pads or bearings, or by deflection of the pads. Temperature and load measurement are recommended for actual engineering data or load distribution for attitude and alignment studies. Rotor deflection in an axial direction is recommended for machine protection monitoring with automatic shutdown to prevent severe failure damage. Thrust failure protection is the topic of this discussion, and a systematic procedure is outlined with case histories of “saves” from the worst type of machine failure.

Different types of sensors can be used on a continuous basis to measure thrust movement or deflections. Earlier developments used both air and hydraulic nozzles that varied in back pressure as the distance between the nozzle and a part on the rotor (often the thrust collar or trim balance ring) changed. Calibrations could be obtained with sensitivities of 1 psi per mil *(0.001 in.)*. However, sensing ranges were often limited to 25 to 30 mils *(0.030 in.)*. The better sensor is the eddy current or inductive probe, which can be installed in rather small tapped holes, 1/4 to 3/8 inch, and can sense movements in excess of 80 mils with sensitivities of 100 to 200 mV/mil. At this point, one must develop a proper perspective of the problem and the solution. Several things are important to consider.

- Thrust bearings are known to fail in about 30 seconds, yet immediately prior to failure, the bearing on examination may have zero defects until the supporting oil film breaks down.
- Cost of the thrust bearings may be $700 or higher for pads only and may be $5 000 for a total assembly. A machine wreck by thrust bearing failure will intermarry the rotating parts to the non-rotating parts with a repair cost of $500 000 or more. Therefore, the thrust bearing (failed) should be sacrificial to the big machine.
- Trying to save the thrust bearing will only cause severe confusion. First, if shutdown occurs before babbitt failure, inspection will indicate the thrust bearing was okay, but the sensor and related instrumentation indicated the onset of a failure! Unnecessary shutdowns will lead to a disarming of the shutdown system rendering the system incapable of proper protection. Second, the thrust bearing, being constructed of pads for example, with self-leveling articulating links and possibly shim packs of 1/8 to 1/4 in. can deflect 4 to 8 mils under load increases well within the design of a thrust bearing. Large deflections can be reduced and downtime minimized by machining single element shim plates to replace large packs of thin shim.
- Machine builders are naturally going to set short limits of rotor movement in their instructions for thrust shutdown instrumentation to protect their machine designs from possible criticisms. The builder does not care how many times you may shut down for inspections on movements of 5 to 6 mils (his suggested limits, but within expected deflections for rated loads).
- Limits for shutdown should definitely indicate that babbitt is being removed, yet the shutdown response must be sufficient to bring the machine to rest within the babbitt thickness and definitely within the rotor travel clearance before contact occurs.
- A sensor system should be correlated with normal bump checks of rotors and dial indicator measurements. This is a very good way to improve everyone's confidence in the machinery protection instrument system.

Thrust bearing fitting procedures and sensor positioning

Steam turbine example

- Specified thrust float by manufacturer = 0,23 to 0,30 mm *(0.009 to 0.012 in.)*
- Specified shroud (rotor blading) standoff to nozzle ring (closest contact) = 1,14 to 1,40 mm *(0.045 to 0.055 in.)*
- Specified total rotor float, minimum = 3,05 mm *(0.120 in.)*

Procedure

- With the thrust bearing removed, move the rotor until shrouds contact the nozzle ring ("A" clearance = 0). Indicate with a dial indicator (compressed more than 200 mils) and set indicator to read zero. Move the rotor towards the exhaust until the first stationary blading stops the rotor on axial travel contact. Record this travel on a maintenance setup sheet after assuring that more than 120 mils was obtained.
- Return the rotor to the zero reading (against the nozzle block) and install only the active thrust bearings with shims. Move the rotor towards the exhaust until contact is made with the thrust bearings (lower half only is used at this point). It is good if the indicator reads 0.055 in.; if it reads more than this, increase the active bearing shim thickness until 55 mils is obtained. Place equal amount of shims in the upper half active thrust bearing. If less than 55 mils, reduce shims until 55 mils travel is obtained.
- Place inactive thrust bearings with shims into place. With the rotor against the active thrust bearing, move the rotor towards the nozzle block shimming for a 46 to 43 reading. This yields a float zone of 9 to 12 mils.

This procedure could have been reversed, setting the inactive bearing shims for 45 mils standoff and then the active shims for 9 to 12 mils float. With a large float tolerance, this would be preferred. Confirmation of this total float is necessary by a bump check with the top half bearings in place. Lack of confirmation, less travel, can be a real problem in troubleshooting machine fits and will not be covered here. This would fall within a maintenance paper concentrated at setting thrust bearings.

Sensor placement

A calibration curve is plotted for the eddy current sensor and its related readout device. This calibration curve will be for the metal of the shaft to be sensed and the plot will be volts (ordinate) versus gap (abscissa). Gap implies the space between the probe (eddy current sensor face) and the metal used in calibration. The calibration curve is at a sensitivity of 100 mV/mil with over 80 mils of linear range, and the voltage at the center of the linear range is 8 V. At +40 mils from the center of linear range, the output voltage is 12 V. At –40 mils from the center, the voltage is 4 V.

Note: At 5 mils on either side of the center, the voltage is 8.5 V and 7.5 V, respectively.

For further convenience, assume the bump checks above yielded 45 mils to 55 mils or exactly 10 mils float. Because the center of the calibration range of the sensor system from the calibration curve occurred at 73 mils gap with an output voltage of 8 V, this point will be used for the center of float position. Two methods can be used to secure this position:

1. Set the rotor in the center of the float zone by dial indicator and adjust the sensor to thrust collar (or shaft end) gap until 8 V output is obtained at the thrust monitor output jack. The zero adjust would manually be moved until the readout meter indicates zero (center of scale). Bump the rotor to confirm both indicator and voltage agreement, –5 mils = 7.5 V, zero = 8 V, +5 mils = 8.5 V.
2. Move the rotor against the active thrust shoes and adjust the sensor gap until a voltage that corresponds to 5 mils active direction is obtained at the thrust monitor output jack, 8.5 V. The zero adjustment calibration would manually be moved until the readout meter indicates 5 mils (active). A bump check should indicate 10 mils of travel on the indicator and 10 mils (+5 to –5 mils) on the readout meter of the sensor system.

Either method will work; however, the author prefers the last method with present instrumentation. It is easier to hold the rotor against one end of the travel. Further, should you wish to set up all machines against the active bearing and have the same meter reading regardless of float zone; then the second method becomes a standard practice.

Most turbines will require that the sensor be placed on the steam inlet shaft end or at least outside (outboard) the thrust collar. Further, the thrust will normally be toward the exhaust so that on turbines an increasing probe gap will indicate a movement in the active load direction. Conversely, a decreasing gap will indicate an inactive (counter) movement. One arrangement of shaft end (near thrust bearing) dual probes used for thrust movement sensing is taken from a current task force effort of the API Subcommittee of Refining Equipment. A current proposal to API has been presented for adoption as a complete specification for vibration and thrust movement monitors.

Because a compressor's normal thrust can easily be in either a gap decreasing or increasing direction, it seems normal to use the turbine direction, calibration, polarity, etc., as standard and call out the compressor system by specific design. It is worth noting that new instrument designs may soon require limited adjustment of the zero meter indicator in favor of self-centering zero meter convention. This will commit the first method to be the preferred method and give equal travel range to active and inactive movements of the rotor. This presents no problem to the user. However, to meet the requirements of all current types of turbomachinery, it shall require a minimum of 80 mils linear range by the sensor system builder.

Centrifugal compressor

Centrifugal compressor thrust bearing shims are set up similarly. However, the compressor rotor may have 250 mils of total float and will be set up in the center of float with a spacer or shims to give 15 to 20 mils (typically) of float. Efficiency of the compressor and possible aerodynamic initiated instability would be adversely affected by sizable excursions from this position. The operating, maintenance and diagnostic people have found that it helps to place two strips of black tape on the thrust meters to indicate the limits of the free float positions when a machine has been calibrated and checked.

Thrust alarm and shutdown limits

Based on more than 20 years of experience with thrust bearing failures (over 50 by count) and use of various thrust indicators from "squallers" (electric touch method) to hydraulic back pressure indicators for gap, to pneumatic nozzles with back pressures versus gap, to present eddy current sensor systems, our limits have been well defined and countless saves have been accumulated.

Thrust **alarms** have been set up as 15 mils from the normal commissioning rotor position at design conditions (generally, with good practice, this will be within 5 mils of the active bump position). However, one can exert little force on a thrust bearing during the process of manual prying to bump the rotor.

Thrust **shutdowns** have been set up as 5 mils past the alarm point or 20 mils from normal load deflection points. Given the proper instrument system, this shutdown point can be extended 5 mils for a 25 mil set point value. At present, this has not been done with existing instrumentation systems. New instrumentation systems allow sufficient range.

Conclusions

It is hoped that this review of normal thrust setting procedures, thrust or rotor movement monitoring, plus some of the practices and beliefs in thrust failure prevention will be helpful to those establishing a failure prevention system in this area.

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PUB CM1017 EN · February 2012