Vibration Monitoring Identifies Steam Turbine Seal Rub

Bently Nevada* Asset Condition Monitoring team helps petrochemical plant to continue operations for 32 months!

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In 2005, the Bently Nevada team entered into a long term service agreement (sometimes known as Supporting Services Agreement, or SSA) with a petrochemical plant in India. Scope of work included monthly visits for machinery diagnostics and regular optimization of data collected in the machinery management software – which included both Data Manager* 2000 & System 1* installations. The events described in this article took place on a compressor unit in one of the main plants at the facility.

Monitored Assets
The machine of interest is a centrifugal compressor driven through a flexible disc pack coupling by a 14-stage 30 MW steam turbine (Figure 1) with rated running speed of 4068 rpm (maximum 4271 rpm). The turbine and compressor both have fluid-film bearings, and both are instrumented with XY proximity transducers for radial vibration monitoring, axial probes for thrust position monitoring, and a Keyphasor* transducer for phase and speed measurements. The machine train is protected using a Bently Nevada 3300 System, while Data Manager 2000 software facilitates vibration condition monitoring and diagnostics.

FIGURE 1: Simplified machine train diagram of the refrigeration compressor
Event History
In July and August, 2007, the steam turbine NDE bearing started showing brief high vibration excursions on both probes (Figure 2). Although peak amplitudes were less than the Alert setpoints, these vibration excursions started becoming more frequent in the month of September.

Frequency domain analysis showed that the vibration excursions were mainly due to synchronous (1X) excitation (Figure 3).
To study the event in more detail, we used the high resolution trend capabilities of Data Manager 2000 software, and connected an ADRE* unit to collect additional data. Along with 1X vibration amplitude changes, 1X phase changes were negligible during any single event of elevated vibration amplitude. However, the phase changed significantly from one event to the next (Figure 4). These observations indicated the possibility of an intermittent rotor thermal bow, with the added characteristic of a “hot spot” location that did not remain constant. Orbit size and shape (Figure 5) confirmed the rotor bow due to a rub.

FIGURE 4: September 2007 ADRE polar plots showing widely varying phase angles for different high vibration events.

FIGURE 5: September 2007: ADRE plots show changes in orbit size and vibration phase [Keyphasor dot] indicating intermittent rotor bow symptoms.
Rub-Induced Thermal Bow

Figure 6 shows a local heating mechanism that can cause thermal bow of a rotor. Due to friction & impact forces caused by a rub, localized heating of the rotor creates a hot spot at the point of contact between the rotor and a stationary component of the machine. When combined with the rotor’s original heavy spot (residual unbalance), the “high spot” at the heated area produces a new effective unbalance vector and a new heavy spot resulting in a new phase angle for the 1X vibration. If the location of the hot spot doesn’t change over a rub cycle, the thermal vector is repeatable and the resultant phase angle is constant. However, the amplitude and phase change due to new effective unbalance vector.

With an intermittent rub, the hot spot – and its associated thermal vector – disappear when the physical contact between the rotor and the stationary component is eliminated. The hot spot cools, the rotor bow subsides, and vibration amplitudes and vectors return to original values based on residual unbalance, without the influence of a thermal bow. However, if the root cause remains and the rub reappears, causing a new hot spot with a different location, the vibration excursion may happen again – with a completely different vector. This appeared to be what was happening during the observed events of 2007.

Annular rub is another variety which can destroy the machine in a matter of hours when the rotor contacts stationary components such as the inner surface of a steam seal or bearing for a full 360 degrees. With this type of rub, the orbit is typically very circular, since the rotor is rolling all the way around the inner surface of the stationary component. If adequate lubrication exists between the rotor and the stationary component, precession of the rotor vibration will continue to be
forward (in the same direction as shaft rotation). But if inadequate lubrication exists, it produces a dry rub, which results in reverse precession, and in its most extreme form, can destroy the machine in a matter of minutes.

Note: A rub is always a secondary effect that is produced when another malfunction causes the average or dynamic shaft centerline to be too close to the limits of available clearance between rotating and stationary components. Excessive radial loads caused by problems such as misalignment can force the average shaft centerline to shift too close to the clearance limits, while high vibration caused by unbalance, instabilities or rotor bows can allow contact to occur even if the average shaft centerline position is normal.

Data Evaluation
As discussed, the observed orbits were not circular, and observed precession was predominately forward. Combined with the fact that the episodes of elevated vibration were not continuous, these symptoms indicated the presence of a light, intermittent rub. The changing phase angles corresponded to a classic shaft rub with a wandering hot spot causing a thermal bow.

Vibration data pointed to a localized rub close to bearing 1 (steam turbine NDE, the governor end of the turbine). The most suspected locations were the bearing 1 oil seal – where accumulation and hardening of lube oil in the labyrinth seal may have had the chance to create hard coke deposits – and the HP packing (front labyrinth seal of the casing itself), which is made of a very hard stainless steel alloy.

Action Plan
Since the high vibration amplitudes were less than the alert setpoint, and the intermittent events were of short duration, plant staff decided to continue running the compressor while closely observing the data available in the Data Manager 2000 software. They controlled operations to minimize changes in process conditions in order to limit the occurrences of high vibration. This allowed them to meet plant production requirements while maintaining a heightened watch on compressor condition.

Additionally, the staff planned ahead for corrective maintenance in case the compressor needed to be shut down before the next scheduled outage. They staged a spare turbine rotor and related spare parts for this contingency, and were standing by to take action at short notice if needed. Operations continued as before, with occasional instances where vibration amplitude briefly exceeded the Alert setpoint (Figure 7).

Eventually, at two different times, vibration amplitude exceeded the Danger setpoint and caused the machine to trip (20JUL2008 & 13AUG2009). One of these instances happened late on a Sunday evening. Immediately following the trip, the Bently Nevada Machinery Diagnostic Services (MDS) Engineer remotely analyzed the data from Data Manager 2000 over a dial-up network.

The MDS Engineer verified that the steam turbine had tripped due to excessive high vibration during another brief episode of rotor thermal bow, which was similar to what had been observed for several months. Plant staff allowed the rotor to cool and the bow to straighten by operating the machine train at slow-roll conditions. After verifying that vectors were within acceptable values, they continued with a very cautious startup. Operation continued under close observation by the plant condition monitoring team and Bently Nevada MDS engineers.

The first occurrence of rub had been witnessed in the month of July, 2007 and the refrigeration unit was successfully run until March, 2010 with heightened monitoring. As 2010 progressed, the observed time between the episodes of high vibration was increasing. It was apparent that there was a definite trend of small peaks appearing in clusters, building up to the highest amplitude peak over a period of about 6 to 8 days. Once past the high amplitude peak, the vibration excursions would cease for a few days and then gradually reappear (Figure 8).
FIGURE 7: May 2008 trend shows persisting vibration excursions, occasionally crossing the Alert set point (yellow line, emphasized by a dashed line overlay at 100 microns pp).

FIGURE 8: January-March 2010 trends show that vibration excursions continued, and that the period between episodes of peak amplitude was increasing.
The Shutdown

Although the plant engineers knew that they needed to perform an internal inspection of the steam turbine – and most likely replace some damaged seal components – they were able to avoid shutting down the refrigeration unit for more than 32 months after the first occurrence of the rub. They mitigated the risk of operation and built their confidence in managing the plant by continuing close observation of vibration condition monitoring data using the continuous data collection of the 3300 System and the diagnostic features of the Data Manager 2000 software.

Everything was fine until 13MAR2010, when there was a sudden increase in vibration on the turbine DE bearing (Figure 9). This vibration spike was very sudden, as observed in 4 second fast trends captured during a high vibration Alert event. Interestingly, no elevated vibration amplitudes were recorded on the steam turbine NDE bearing. Vibration rise was just 15 microns pp on the NDE bearing.

At the same time as the sudden increase in vibration occurred, a sudden small (1 micron) shift in average shaft centerline position was recorded at both the DE and the NDE bearings of the steam turbine. As we will see in the inspection results photos (Figure 12), this small but significant change was caused by an interesting mechanical event...

Also, a significant shift in rotor precession occurred at the time of the elevated vibration. Figures 10 & 11 show that the orbit shapes changed and the forward and reverse vibration components changed – as shown in the full waterfall plots.

FIGURE 9: 11-13MAR2010 trend shows sudden vibration increase on steam turbine DE bearing.
FIGURE 10: Orbits and full waterfall for steam turbine NDE bearing. Reverse 1X frequency components dominated, and change in orbit shape indicated a change in radial loading caused by the increase in rub severity.

FIGURE 11: Orbits and full waterfall for steam turbine DE bearing. Forward 1X frequency components dominated. As we will see in Figure 12, there was a very good mechanical cause for increased unbalance at the DE bearing end of the turbine rotor.
The DE bearings at both the compressor and the steam turbine experienced the maximum changes in amplitudes of vibration with corresponding 1X phase changes. These vibration changes at the DE bearings persisted. Changes on the steam turbine NDE bearings were smaller and did not persist. After the initial step changes, 1X vibration amplitudes and phase angles at NDE bearings stabilized.

**Synchronous Rotor Response**

Synchronous or 1X vibration response is the resultant of both unbalance force and dynamic stiffness of the rotor system. Changes in either one of them (or both) can cause changes in 1X amplitudes and phase.

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\text{Synchronous response motion (d)} = \frac{\text{Unbalance Force}}{\text{Dynamic Stiffness}}
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A sudden change in 1X amplitudes, as was observed in this event, can be caused by a sudden change in unbalance force. Such changes can be caused by the addition of mass (anything becoming loose and trapped in the rotor) or removal of mass (typically by a blade failure on LP stages of the steam turbine). Damage to the coupling (such as a bolt that breaks and flies out) can also cause changes in 1X amplitude on DE bearings of both compressor and turbine.

Based on these possibilities, our MDS Engineer recommended shutting down the compressor at the earliest opportunity to inspect the coupling and the steam turbine internals. Over the next few days, increasing vibration trends were noted, along with higher than normal structural vibration in associated piping and supports. At this time, plant staff shut down the refrigeration unit for planned inspection and any required corrective maintenance.

**Inspection Results**

**RUPTURED TURBINE BLADE**

Visual inspection revealed that the machine coupling was intact but one of the steam turbine blades had ruptured at about 100 mm from the tip (overall blade is about 325 mm long). The unbalanced force caused by this “liberated” blade (Figure 12) explains the observed sudden step change in vibration.

**STEAM SEAL COKING**

The cause of the intermittent light rub was identified as formation of coke deposits on the NDE oil labyrinth seal. Rubbing against the hard carbon deposits was observed to cause deeply scored grooves on the corresponding rotor shaft area (Figure 13). The affected area was about 100 mm wide and the grooves were up to 35 mm deep in the rotor shaft!

**FIGURE 12:** One blade had ruptured at the last row of the 14 stage steam turbine (DE end, lowest pressure stage).
FIGURE 13: Rubbing against hard coke deposits on the high-pressure labyrinth seal has caused deep scoring (shiny bands) on the corresponding shaft area.

FIGURE 14: Oil catcher labyrinths were filled with coke (including the oil drain ports, which needed to be cleaned).

FIGURE 15: Simplified drawing of steam turbine labyrinth seal, before (left) and after (right) the nitrogen purge injection feature (red) was added. In both cases, oil from the bearing flows from left to right through the labyrinth to the drain, while steam flows from right to left.
The observed coking in the labyrinth seals was apparently caused by heating of leaking lubricating oil and vapors by gland sealing steam. Over a period of time, the coke deposits built up in the clearance between the seal and rotor. Eventually, the soft coke started touching the rotor – causing symptoms similar to seal “run-in.”

As the soft coke was removed, normal clearance was restored, eliminating the rub and reducing the rub-induced vibration. But after a while, more coke would form and the whole cycle would repeat. The lubricating oil drain ports were also discovered to be plugged with coke, which contributed to further buildup of oil in the NDE labyrinth seals (Figure 14).

**Labyrinth Seal Modification**

A proven technique to prevent lube oil from being overheated by steam in the labyrinth seals is to provide a “curtain” of cool nitrogen purge gas on the steam side of the seal (Figure 15). In addition to keeping the oil from forming coke deposits, injecting a purge gas can also reduce the need for dewatering in the lube oil system by reducing the ingress of water into the oil in the first place. Plant engineers worked closely with the steam turbine manufacturer to design, manufacture, and install a suitable barrier plate and implement a nitrogen purge system for the affected labyrinth seal. After these modifications and the installation of a new rotor and seal assembly, the compressor was restarted and has run successfully without any problem recurrence.

**Conclusions**

Machinery management systems such as Data Manager 2000 and System 1 software can be a valuable and integral part of a comprehensive plant condition monitoring program. They provide not only the necessary proactive information on changes in machinery behavior, but also high resolution diagnostic data captured during unanticipated machine trips and other surprise events. These systems can provide added confidence for operating the plant during periods of known machine degradation.

Note: Data Manager 2000 software is obsolete. The plant where this event occurred will be upgrading to the latest version of System 1 software in an upcoming outage.

As described in this article, a petrochemical plant was faced with a high vibration situation on one of its highly critical steam turbine drivers. With our ongoing long term service agreement facilitating the help of the Bently Nevada team, the plant staff received not only pinpoint diagnostics response (often at odd hours) but they were also able to continue vital production by safely extending the required plant outage for almost 32 months. Accessibility of data over a remote connection made it very easy for the Bently Nevada MDS Engineer to promptly review the vibration data for the high vibration turbine trip events and subsequent startup – without the time delays or cost of travelling to the site in person.

**References**


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