HEAVY-DUTY GAS TURBINE HOT-END BEARING FAILURE

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A number of technological and scientific advances through the ages have helped us reach the standard of living we enjoy today. One of the most important was the development of anti-friction bearings. These bearings allowed the design, construction and operation of innumerable mechanical devices for countless mechanical purposes. For example, rotating machinery relies on anti-friction bearings to support and align their rotating components during operation. They also help prevent catastrophic clashes between the rotating and stationary parts.

The rotors of heavy-duty gas turbines are exposed to the heat of combustion during their operation. This thermal loading, in addition to the mechanical loading due to self-weight and rotation, provides a challenge to the designers of gas turbines when aiming for the reliability demanded by the commercial requirements of operation.

This article briefly introduces the particular issues of bearings in heavy-duty gas turbines, and then reviews a case study of a gas turbine bearing that could no longer hold its load and the situation surrounding its failure.

GAS TURBINE BEARINGS

Large gas turbines (greater than 20 MW power output) can be generally split between the aero-derivative models and the heavy-duty or frame-type industrial turbines. Aero-derivatives, because of their aviation heritage, size and general multiple-shaft layout, tend to use typical rolling-element bearings to support and locate the shafts.

Because of the lower restrictions on weight and size of components, heavy-duty gas turbines exclusively use large and rigid fluid-film plain journal bearings to support and locate the rotor radially within the casings. Some older designs required three bearings to maintain the required clearances along the shaft (e.g. MS7001EA, MS9001E). Smaller, more modern designs with stiffer rotors reduced this to two bearings (e.g. MS6001B, GT13E2, etc.). Plain-journal bearings or tilt-pad bearings are used, depending on the particular manufacturer’s preference and requirement for accommodating misalignment due to rotor flex or casing distortion during operation of the turbine. Although it is not addressed further in this article, a thrust bearing is always utilised to locate the shaft axially.

All modern heavy-duty gas turbines follow the same form factor: ambient air enters an axial-flow compressor and then exits into one or many combustion chambers at high pressure and temperature due to the compression. Then, due to combustion, a further rise in temperature occurs, and the resultant hot-gas stream is expanded through an axial-flow turbine, running on the same shaft as the compressor, to extract power. The hot combustion gases flow through the turbine, between the shaft and casing (both protected from most of the heat by a combination of heat shielding and cooling air) before exiting the rear of the machine. The exhaust temperatures of most heavy-duty gas turbines are generally between 500 °C and 600 °C, depending on the duty, fuel and efficiency of the turbine.

It is in this hot exhaust gas that the hot-end bearing must sit, relying on extensive thermal management of the exhaust frame, lubrication oil and cooling air to maintain its position (and that of the spinning rotor) and to maintain its own integrity. Many design solutions from the major manufacturers have been incorporated to minimise the impact of thermal distortion on the hot-end bearing (e.g. MHI’s tangential struts for their MS901 and M701 models or Alstom’s adjustable bearing housing supports in their GT13E2 model).

CASE STUDY: MS6001B NUMBER 2 BEARING FAILURE

Background

The MS6001B (or Frame 6B) is a widely utilised General Electric heavy-duty design. It is generally used for power generation, at both 50 and 60 Hz, outputting between 38 and 42 MW depending on the model. The design is “E-class”, derived from 1970s technology, but has been continuously upgraded since its introduction in 1978. Although it uses a reasonably flexible built-up rotor, it requires only two radial bearings for support (Figure 1). Both are plain elliptical journal bearings. Power is taken off the hot (turbine) end of the machine.

The particular unit in question was in a co-generation role, supplying steam and power to neighbouring industry. The unit had seen approximately 120,000 fired hours and an average of five starts per year after commissioning. Due to demand fluctuations, the unit was running base load during the day and part load at night.

The unit did have a history of vibration and alignment issues during commissioning and while in service. Alarm levels of vibration (approximately 13 mm/s) were often exceeded during start-up. Base load levels of vibration were within the OEM limits but always close to alarm. Two bearing temperatures for the hot-end bearing were measured as part of the control package. These temperatures were always noted to be high, but below alarm. A number of bearings had been replaced during the life of the unit, based on their condition during scheduled or opportunistic maintenance.
The replacement cost of the bearing was low, so there were little to no cost pressures to solve what appeared to be an acceptably managed “quirk” of the unit.

The Incident
One afternoon the alarm temperature (130 °C) on the hot-end bearing was reached. The load on the unit was reduced immediately. A subsequent reduction in the bearing temperature was observed. Plant operations decided to monitor and shutdown the unit if the temperature exceeded 150 °C. It is important to note that there was no trip limit on bearing temperature in the OEM package.

Later in the evening, the alarm temperature on the bearing was reached again. Thirty seconds later 150 °C was reached and a manual shutdown was initiated. Two minutes after the shutdown was initiated, the bearing temperature was measured at 200 °C. The vibration levels increased, causing an automatic trip of the unit five minutes after the shutdown was initiated. During the rundown, bearing temperatures up to 420 °C were measured. The tin-based white metal used in journal bearings melts between 180 and 240 °C (depending on the lead content).

The Aftermath
Disassembly of the bearing housing revealed an extensively scored shaft journal, also deposited with remains of the bearing white metal. The top half of the bearing was relatively unaffected. The bottom half had almost all the white metal removed from the steel shell of the bearing, which was itself severely rubbed, overheated and distorted.

The material condition of the rotor journal was of great concern. The journal was part of a stub shaft bolted to the turbine section of the rotor. Replacement of the stub shaft would require shipping to a facility capable of disassembly, replacement of the shaft, and balancing the reassembled rotor before shipping back to site. Metallurgical inspection of the journal surface revealed localised regions of high hardness (exceeding 600 HV) and transformation microstructures that showed that parts of the journal surface had exceeded 720 °C during the incident. The hardening extended to a depth beyond the re-machining limit of the journal. The bearing housing was nose-up relative to the axis of the machine.

The latter findings were quite significant, as numerous alignment checks performed at outages showed no indications of serious misalignment. Further inspection of the bearing housing and exhaust frame revealed the following:
- The bearing housing was nose-up relative to the axis of the machine.
- Fibre insulation in the outer skin of the exhaust frame was almost completely missing.
- Internal insulating pads (Figure 3) inside the exhaust frame had dropped in the lower half of the exhaust frame, partially blocking the entry of cooling air on one side on the machine (Figure 4).

The combination of all factors resulted in an increased loading on – and higher temperature of – the hot-end bearing, directly resulting in a decreased service life and ultimately failure during operation.

During the investigations, a number of other examples of MS6001B hot-end bearings were found with similar indications of distress, revealing that the problems with this unit were not unique. The later GE “F-class” designs have a completely different bearing arrangement using two tilt-pad bearings, and driving from the cold (compressor) end of the machine, which “…eliminates the alignment issue at turbine end bearing” according to the Electric

Diagnosis
The consideration of shipping and repair lead times led to the decision to replace the entire rotor, as this would return the machine to service much quicker than any other option. As a replacement rotor was sourced, an investigation into the physical cause of the bearing failure was undertaken. Since most of the physical evidence on the failed bearing itself was missing, examination of bearings previously removed from the unit was undertaken (e.g. Figure 2). The following was noted:
- Features typical of long-term fatigue failure of the bearing white metal were observed in a number of the replaced bearings.
- There were indications of edge loading due to slight angular misalignment with respect to the shaft.
- Extrusion of white metal over the edge of the bearing indicated creep of the white metal during service due to high operating temperatures of the bearing.

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The incident was extensive. The physical evidence on the failed bearing itself was missing, examination of bearings previously removed from the unit was undertaken (e.g. Figure 2). The following was noted:
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The latter findings were quite significant, as the manual for the turbine [General Electric, MS6001B Gas Turbine Manual, Volume II, 1995] states (emphasis added):
- Exhaust frame radial struts cross the exhaust gas stream. The struts must be

Figure 2 Lower half of the hot-end bearing previously removed from machine. The damage seen was typical of edge-loading causing fatigue in the white metal.

Figure 3 Insulation pads inside exhaust frame protect cooling air for struts (flow path arrowed). Lower half pads had dropped, blocking the flow of cooling air to (bearing supporting) lower struts.

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Recovery
During reassembly, the insulation and cooling flow was restored to the exhaust frame. The housing alignment was corrected using angled shims. Laser alignment of the bearings was undertaken to ensure the required tolerances were reached in the fully assembled state.

A replacement rotor was installed in the machine. Vibration levels and bearing temperatures were considerably lower with the new rotor. The plant installed new instrumentation, alarm and trip limits related to bearing performance monitoring. The unit has been running with little issue for more than a year since the incident.

Epilogue
The damaged rotor was recently shipped to a repair facility for refurbishment. During de-stacking of the built-up rotor, it was apparent that there had been loss of interference fit on a number of compressor wheels that made up the compressor rotor. It is speculative at this stage, but such a loss of interference fit may have influenced both the rigidity and heat transfer through the rotor during thermal transient conditions (such as start-up), leading to distortion (e.g. bowing) during the thermal transients and resulting in increased vibration. It also may have contributed to the noted thermal sensitivity of the rotor and high vibration levels during base-load operation.

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Reliable Online Monitoring of SF₆
Operational efficiency can be maximised while risks are minimised, maintenance can be optimised, and equipment lifetime can be extended, all of which translates into considerable savings.

Condition-monitoring systems are increasingly incorporating online measurements. The ability to gather data online not only minimises the time needed for field operations, it also provides constantly up-to-date trending data and enables rapid decision-making.

In the power industry, online monitoring offers a wide variety of benefits for critical equipment such as power transformers or switchgears. However, the factors that affect the reliability of online measurement, especially regarding online monitoring of static SF₆ gas, are not so well known.

If you are keen on knowing more about the topic, Vaisala has created a non-commercial white paper around the SF₆ online dew point monitoring. The paper examines the online dew point measurement of static SF₆ gas which is used as an insulator in switchgears and circuit breakers. Moisture in SF₆ can cause arcing and equipment failure.

The critical factors in obtaining reliable data include: 1) where the sensor is installed, 2) how the sensor is installed, and 3) what the required sensor features are. One key phenomenon, which is not very well understood but which greatly impacts these factors, is related to moisture transients between solid materials and SF₆.

The paper concludes with a clear set of recommendations that will help to ensure optimal dew point measurement performance. The complete white paper is available at www.vaisala.com/power

TechCon® WELCOMED NEW IDEAS
SF₆ online dew point monitoring was also one topic in the recent TechCon® Asia-Pacific conference in Sydney, where the author of the Vaisala white paper, Mrs. Senja Leivo spoke about the measurement challenges. Mostly Australian and Indian grid operators were interested in discussing and finding out more about online measurements which are becoming increasingly popular in the industry.

As labor shortage is becoming a challenge and unmanned substations popular, online measurements are naturally welcomed, so Vaisala’s recently launched new multiparameter measurement product, proved somewhat of an eyebrow-raiser. The DPT145 transmitter for SF₆ provides online data of up to seven parameters at once including dew point, density and pressure. Combining online measurement capability with multiparameter information seems the way to prepare for the future.

See further at www.vaisala.com