Thrust bearing monitoring of vertical hydro-turbine-generators

Ryszard Nowicki¹, Nikolay Morozov² co-author

¹ GE Digital, Div. Bently Nevada, Poznań, Poland, Ryszard.Nowicki@ge.com
² GE Digital, Div. Bently Nevada, St. Petersburg, Russia, Nikolay.Morozov@ge.com

Abstract

A serious thrust bearing failure of a high power vertical hydro-generator is described. Analysis of the applied monitoring and protection system for the thrust bearings is provided. Slow response time of the protection system is investigated. A guideline for the more efficient protection system is provided.

1. Introduction

In the case of a vertical hydro-turbine-generator (HTG) the thrust bearing supports the mass of both the generator and turbine plus the hydraulic thrust imposed on the turbine runner. The bearing is located either above the generator rotor (suspended unit) or below the rotor (umbrella unit). Thrust bearings are constructed of oil-lubricated, segmented, and shoes lined with various materials. It can for example be traditional Babbitt(¹) offering high thermal conductivity required to absorb and carry away thermal energy generated in the bearing or a polymer (e.g. a fluoroplastic(²), PEEK(³)). The polymers provide numerous advantages as compared with other materials. They can operate at temperatures up to ~120°C (248°F) higher than Babbitt and other conventional white metals. In addition, they have a significantly lower coefficient of friction, reduced thermal expansion and contraction characteristics, and being compressible conform to the contours of interface. They also offer good bearing electrical isolation.

An important attribute such a bearing is an optimal balance between possibly low coefficient of friction, especially in initial period of movement when the amount of oil between thrust collar and bearing segment is limited, and possibly high thermal conductivity(⁴), what is ability of bearing material to dissipate heat quickly generated due to friction. The Babbitt thermal conductivity is 25,9 W/mK. For the PTFE, it is much smaller k = 0,25 W/mK (and the value can be depended on filled materials: e.g. for 25% glass filled PTFE k=0,44 W/mK or for 25% carbon filled PTFE k=0,65 W/mK). For the PEEK k is in the range 0.25 to 0.93 W/mK.

Usual methods of lubrication of large thrust bearings have not changed much since they were first invented; in vertical shaft machines, it is normal to arrange a bath lubrication with the pads immersed in oil. Although flooded lubrication is relatively simple, it results in high parasitic power loss due to turbulence at high speed. When mean sliding speeds more than 50 [m/s] are expected, these losses may be largely eliminated by employing the system of directed lubrication which reduces power loss by typically
50%, consequently decreasing the bearing temperature, and in most cases, allows a reduction in oil flow requirements [1].

Under operation, the capacity of hydrodynamic bearings is restricted by three limitations: minimum oil film thickness, mechanical limits, and temperature of material of the bearing running face [2]. The critical limit for low-speed operation of Babbitted shoes is minimum oil film thickness. For polymer-lined shoes there is no need for high pressure oil injection between surfaces to overcome the frictional effect of high loads at start-ups due to the exceptionally low coefficient of static friction. Therefore, for high speed operation of such bearings, temperature of running face material is usually the limiting criteria.

Figure 1. (A) An example of double-ring thrust bearing layout, (B) Layout of bearing pads for the discussed case

There are currently three different lubricant supply methods for thrust bearings. These are: pressurized supply (flooded), spray feed, and leading edge distribution groove. Each lubricant supply method has a different influence on bearing power loss and its temperature. Because of large amounts of heat generated in the bearing (up to ~1 MW in extreme cases) it is necessary to provide forced oil cooling. If the cooling is insufficient, or if an issue occurs during unit operation, a temperature increase can lead to melting of the thrust pad coating. It can result in considerable losses with high direct and indirect costs.

There are known various problems with thrust bearing resulting in some failures. Two examples, the first from the Sir Adam Beck complex [3]. Investigation identified quality issues with the post-overhaul refurbished thrust bearing and a glitch in the shutdown system that could, under some circumstances, allow the unit to rotate without oil lift. The second instance happened in Akosombo Hydro Power Plant (HPP) [4]. In the previous decade, due to some problems with thrust bearing (cavitation and the thermal ratcheting), the HPP did their upgrade on two units and the Babbitted pads were replaced with polymeric pads. This resulted in a reduction of the thrust bearing temperatures of approximately 7…8°C. and elimination of previously visible problems.

This paper describes a polymeric thrust bearing failure of a high-power HTG which occurred during the commissioning process of a new unit. The unit has power rating of approximately 630MW and is driven by a Francis turbine. The generator operates on a 50 Hz grid and has a 42-pole rotor. Nominal operating speed of the rotor system is 2.38 s⁻¹ (~143 RPM).

Thrust bearings with polymeric coatings have been widely used in some countries for many years. The Figure 1A presents pads coated with fluoroplastic press-fitted into the bronze wire. The composite structure is soldered to the base. For the discussed case, the unit uses thrust bearing with two rings of pads as shown in Figure 1B. The discussed unit uses a spray feeding cold oil lubrication to the thrust bearing working surfaces.

2. Condition monitoring of thrust bearings

Axial load of large vertical hydro units has been known to reach twelve mega-newtons, with the unit’s entire weight carried by the thrust bearing. For operating condition
evaluations, a thrust bearing can be fitted with temperature sensors, proximity probes, load cells and oil level indicators. An absence or even reduction in oil film thickness at the thrust pads results in breakdown of the bearing material running surface which can further lead to rotor/bearing damage. The proximity sensors can be applicable for indication of the rotor axial position as well as dynamic movement when fixed e.g. to the bearing bath, and for monitoring of the oil thickness between bearing collar and thrust bearing pads [2,3,5] if fitted with bearing shoes and observing the thrust collar. The number of temperature transducers used for thrust bearing monitoring can vary and depends on bearing load and the bearing construction. There is a suggestion provided in [6] for bearings under a specific load of more than 7 MPa the temperature sensors, both for visual monitoring and for protection should be installed in all segments, regardless of their number and type of bearing, and for HTG rated below 7 MPa not obviously, all segments must be monitored. However, in such configuration, can happen the thrust bearing problems that will be not recognized in proper time as it was presented in [7]. The mentioned above transducers can be connected to an On-Line monitoring system. In parallel to the On-Line bearing monitoring some Off-Line systems can be used additionally, e.g. oil monitoring.

When the continuous monitoring system (CMS) is correctly configured and operational problems may be identified and corrected before a thrust bearing serious failure.

3. Thrust bearing temperatures

The oil film between antifriction and steel ring thrust surface should carry the load of each HTG rotor system. If something undesirable occurs, then the HTG thrust bearing can exhibit a temperature increase. This can be due to excessive load (including overload), bearing partial fatigue, inadequate or incorrect lubrication, a rotor system misalignment or an axial clearance that is too tight. If there is incorrect circulation, the oil film can break down, causing the surfaces to contact. Lubricant starvation often offers few symptoms until damage has occurred. For some malfunctions the evidence on some thrust bearing segments might be a temperature decrease [7]. Other causes may be strong axial vibration which will cause oil film instability and tearing. This can happen when a turbine operates at partial capacity when the water flow becomes increasingly unstable (what results in variable axial load), and consequently (i) the axial clearance varies what can result in (ii) getting oil thickness smaller than expected, and (iii) surface temperature increasing over an accepted limit.

Many papers were published covering the topic of thrust bearing temperature measurements, and providing a comparison between losses of Babbitt and polymeric bearings. A comprehensive review can be found in [8]. However, the temperature measurements provided are overall for comparing performance operation of thrust bearing lined with various materials. They were mostly provided for different test stands and process variables (like rotor speed, bearing load), and were controlled enough slow to achieve stable change of temperature of chosen bearing points, and additionally for some tests oil temperature, too. However, what is good for test stands, is not obviously preferred for On-Line CMS tasks. Change in operating conditions can develop slowly, or in some cases rapidly. Therefore, CMS for large thrust bearings needs to be constructed with a good understanding of potential risks.
4. Predictive maintenance needs a diagnostic system

Various maintenance approaches need to use various condition management systems. The HPP, where a new unit was under commissioning, uses the condition based maintenance approach. This need to use not only a monitoring and protection system (which is enough for preventive maintenance) but additionally a diagnostic one. The important measurements collected by the diagnostic system are typically different kind of mechanical vibrations, air gap (over all for low speed generators), different kind of temperatures (e.g. for bearings, windings, seals), and water dynamic pulsations (among others for cavitation intensity detection). Excessive unit vibrations during commissioning may result mostly from mechanical or magnetic unbalance, and additionally from flow excitations – mainly during partial load operation. A professional monitoring and protection system (here: monitoring SYSTEM 3500), using mostly vibration measurements (vibration monitoring devices such as proximity probes for rotor vibration measurements to detect shaft runout and seismic transducers for monitoring of structural vibrations were included into the system) for condition evaluation and for alarm indications and unit shutdown when an over limit threshold is indicated. In addition to vibration, axial position and air gap monitoring, temperature monitoring is used - among other for all segments of the thrust bearing being in arrangement as presented in Figure 1.

**Figure 2. Screen shot from the diagnostic SYSTEM 1 presenting chosen measurements used for condition based maintenance of the HTG**

Beside temperatures of thrust bearing segments there are the following temperature measurements on the bearings: lube oil inlet and outlet temperatures for the thrust bearing bath, and oil bath temperature for the turbine and generator guide bearings. The temperature monitoring is provided by another system which records temperature data in database per % of value change, but not more often, than once per 1 sec. It means that in the HPP is used a distributed CMS. Both monitoring and protection systems (mechanical measurements and temperature based) are connected to the data acquisition server of the diagnostic system that runs SYSTEM 1 software. The diagnostic system is capable of collecting data from several other HTGs operating in the HPP, and can import additionally (i) condition measurements provided by other monitoring systems, (ii) process variables like e.g. active and passive powers, and (iii) environmental variables like e.g. water head, water temperature which are accessible in the unit control system (a DCS). Figure 2 presents the diagnostic system screenshot of chosen condition dedicated measurements provided for this unit which supports operational personnel and maintenance staff.
5. What did happen

The HTG during commissioning process had several start-ups already and passed through balancing. During the next start-up for fulfilling the load test the emergency shutdown device received command for emergency stop due to inner thrust bearing pad temperatures exceeding the High-High alarm.

A typical temperature sensor implementation into a pad is presented in Figure 3A, excluding detail ‘b’ which is equivalent to better temperature measurements of the backing material. For such an approach the following design is used frequently: sensor length 170 [mm], distance of its axis from the bond δ = ~22 [mm], diameter of sensor slot is Φ = ~25,4 [mm], the layer of pad coating has thickness ζ = ~10 [mm], and for the coating is used a fluoroplastic (Photo ‘a’ presents corner part of a pad coated with fluoroplastic).

![Figure 3A](image1.png)

Figure 3A. Two arrangements of a thrust bearing temperature measurement: (A) pad-fluid film in whole in coating material (B) fluid film – hole with bypass flow

The scale of oil films in such bearings is on the order of micrometres and depends the surface finish as well as other variables. In the case of accurate bearing surface without tool marks, characterized as \( R_a = 0,8...1,6 \mu m \), the allowable minimum film thickness is \( h_0 = 25 \mu m \) [9]. This is the minimum film thickness to avoid surface-to-surface contact under clean oil conditions and with no rotor system misalignment. It may be necessary in some cases to use a larger film thickness than indicated.

![Figure 4](image2.png)

Figure 4. Temperature change of nine thrust bearing segments from the:
(A) outer set of pads, (B) inner set of pads

Figures 4 presents the temperature changes of thrust bearing outer pads (Figure 4A) and inner ones (Figure 4B). The figures also include the rotor speed curve, and several the vertical continuous lines which are cursors. The red area indicates a period (~42 s) when
the HTG rotor was running at nominal speed. It was also observed that after unit startup, the oil level indicator installed on outside wall of the thrust bearing bath indicated a higher oil level which remained high until shutdown.

The inner pad temperature gradient after the emergency shutdown is ~+150°C/minute, and reduces the rate of change when the rotor speed is below 60 RPM (approximately 2 minutes after initiation of the unit shutdown signal). The maximum registered temperature of the inner pads was in the range 340…380°C, and started to reduce ~5 minutes after beginning of the shutdown process with a temperature gradient ~20°C/minute. This allows an estimate that the inner pads will get again the “normal temperature” after ~half hour.

In fact, the true maximum temperature of the operating surfaces had to be higher. There is provided information [10] that 1.5 mm thick PTFE layer leads to thermal insulation up to 23°C, what means that for such bearing which has ~10mm the insulation is probably significantly higher in temperature. In addition, the TC sensors used in [10], were located 4 mm below the bond, while for many bearings the depth exceeds 20 mm. Even for Babbitt bearings there is a significant difference between running surface temperature and temperature indicated by the sensor. There is information in [11] that temperature at a TC sensor installed ~3 mm from the surface may be 40°C (or more) cooler than the surface. The [10] fails to discuss the time delay between change of the temperature on the surface, and indication of the fact by the temperature monitoring system.

When the thrust bearing was opened, it was determined that all the inner segments were damaged due to overheating, but the outer segments were in good condition. Metal on thrust bearing surface required polishing. Lubrication oil chemical analysis showed no deviation from specifications.

6. Associated measurements for the discussed case

Figure 5 is a composite plot presenting data from vibration transducers for a similar range of time as used for Figure 4 (these are two non-contact transducers and three seismic transducers). These non-contact transducers are fixed to the thrust bearing oil bath, and measuring rotor axial relative vibrations and position⁶). The non-contact eddy current transducers provide Vae signals (which inform about dynamic vibration), and additionally the Vdc components (which inform about the axial position change of this rotor system component which is observed by these transducers) measured against point of the transducer fixing. All seismic transducers have vertical orientation, and are fixed to turbine- and generator guide bearings, and additionally to the thrust bearing support. Signals from these seismic transducers are connected to the 3500 SYSTEM and integrated to vibration displacements.

From the plot, it is apparent:

i. There are no rotor relative vibrations when rotor does not rotate.

ii. There are permanent seismic vibrations coming to the transducers from other vibration sources located in the HPP. This noise-floor is ~20 µm_p-p_.

iii. All vibration signals increase when the shaft rotates.

iv. The seismic vibrations reach the highest level just before reaching the nominal speed, and almost immediately after HTG tripping. Such vibration behaviour is
like that described in [12]. For the nominal speed the vibration levels are $\sim 60 \pm 10 \mu m_{p-p}$.

v. The rotor axial vibrations are almost twice the magnitude of the of the seismic levels. They are highest for the full speed: $\sim 100 \pm 20 \mu m_{p-p}$. They decrease to $\sim 60 \mu m_{p-p}$ after shutdown, when the rotor speed is below 100 RPM and stabilize for $\sim 210$ s. During this period the rotor speed drops down to $\sim 8$ RPM, and during the next minute this rotor relative vibration disappears.

vi. The rotor system (relative) axial position is constant before this run. During the run the rotor moves downward. The full change of rotor position is $\sim 0.45$ mm. There are various components that can contribute to this change e.g. (i) thrust bearing wearing, (ii) local thermal deformation of the thrust bearing collar due to rubbing process of the inner ring pads, (iii) a deformation of the transducer attachment (e.g. due to some temperature influences). The change of oil film thickness does not have significant influence on the mentioned possible reasons.

![Figure 5. Trends of chosen signals during the problematic run](image)

Referring to the existing standards and to the best practices guidelines for thrust bearing monitoring as described in [13]. In parallel to the bearing pad temperature monitoring must be monitored the rotor axial position. For O&G applications it is recommended to use two thrust position non-contact probes and usually preferred logic “2 out of 2” of the relay output to the machine emergency shutdown device. This approach is justified for thrust bearings where the transducers are externally accessible (there is a continuous access to the transducers if needed). However, for bigger steam turbines the thrust bearing is usually located between high pressure and medium pressure cases, what means that the transducers need to be installed inside of the bearing pedestal, and does
not exist a quick access for them. Therefore, for such arrangements three transducers are preferred and usually used for rotor axial position monitoring and relay output logic “2 out of 3” is applied. Description of the used approaches for mentioned machines can be helpful with a guide line for the HTG.

In the case of thrust bearings of vertical HTGs there are two methods of transducer mounting

A. from outside if a collar permitting such an approach exists.
B. Direct monitoring of the thrust bearing collar position in which case the transducers must be installed inside the thrust bearing bath.

It is sufficient to use two probes for the scenario A. However, if the scenario B must be used, then it is strongly advised (especially for bigger power trains) application of three transducers. Additionally, the type of thrust bearing segment support should be analysed. If the segments are spring supported, for thrust bearing condition monitoring a transducer fixing directly to the segments should be considered.

For the presented case, a change of the rotor axial position was monitored (including vibrations) by means of two transducers fixed along of scenario B. Unfortunately, the axial movement of the rotor (Vdc component of the signal from the transducers) was not included in the protection system.

**Table:**

<table>
<thead>
<tr>
<th>EVENTS AND FACTS</th>
<th>TIME: hh:mm:ss</th>
<th>From beginning of start-up:</th>
<th>No of rotor revolutions [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>ZERO SPEED / HTG START-UP</td>
<td>20:26:00</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2.5% of RAPCh (0,05mm)</td>
<td>20:26:38</td>
<td>43</td>
<td>65</td>
</tr>
<tr>
<td>0.5% of RAPCh (0,1mm)</td>
<td>20:26:48</td>
<td>48</td>
<td>77</td>
</tr>
<tr>
<td>10% of RAPCh (0,2mm)</td>
<td>20:26:56</td>
<td>56</td>
<td>99</td>
</tr>
<tr>
<td>15% of RAPCh (0,3mm)</td>
<td>20:27:11</td>
<td>71</td>
<td>131</td>
</tr>
<tr>
<td>Nominal speed</td>
<td>20:27:15</td>
<td>75</td>
<td>140</td>
</tr>
<tr>
<td>Temperature Emergency Shutdown</td>
<td>20:27:57</td>
<td>117</td>
<td>238</td>
</tr>
<tr>
<td>20% of RAPCh (0,4mm)</td>
<td>20:28:15</td>
<td>135</td>
<td>279</td>
</tr>
<tr>
<td>Temperature monitoring indicates that pads reached the maximal PTFE continuous service temperature (260°C)</td>
<td>20:29:30 ±30[s]</td>
<td>210±30</td>
<td>385</td>
</tr>
<tr>
<td>Temperature monitoring indicates that pads reached the PTFE melting temperature (327°C)</td>
<td>20:30:25±55[s]</td>
<td>265±55</td>
<td>431</td>
</tr>
<tr>
<td>ZERO SPEED</td>
<td>20:33:00</td>
<td>420</td>
<td>480</td>
</tr>
</tbody>
</table>

In the Table is presented a history of this case with indication of pertinent facts tagged in time and in number of rotor revolutions from beginning of the run. It is visible that the temperature protection initiated emergency unit shutdown, after 238 rotor rotations. Having Rotor Axial Position Change (RAPCh\(^7\)) included into the protection and its alarm setup for 0,1 mm\(^8\) it would be possible to indicate the abnormal situation and shutdown the unit after 77 rotor revolutions or 69s earlier than when it did occur.

7. **Quality of protection system installation and operation**
There are various features of a condition monitoring and protection system influencing its operation quality, for example a system ability for auto-diagnostics of connected transducers and operating channels, or an ability for alarm management improvements as it was discussed in [14], and [15]. However, neither monitoring or protection systems can recognize wrongly connected transducers and compensate errors. It is widely recognized that many temperature transducers yield information of poor quality. Some examples of such wrongly connected temperature transducers are presented in [16].

8. About temperature transducers

For temperature measurements, various transducers can be considered: Thermistors, RTDs, Thermocouples (TC), Fibre-Optic, ICs, and digital devices. Decisions concerning the type of temperature sensor should be made considering purpose of the measurement (protection versus condition monitoring), its time constant (different for various transducer constructions), and expected range of temperature indications. Some types should be disqualified immediately if condition based monitoring is expected, for example thermistors due to high non-linearity, IC and MEMS exhibit slow reaction times and insufficient operating range. Therefore, the choice is limited frequently to the most traditional ones: RTD or TC. Both are accepted and preferred by [13]. Usage of highly linear RTD elements with a stable resistance versus temperature relationship which are designed to meet the parameters of IEC 751 (DIN EN 60751), incorporating Amendments 1 and 2 is recommended. Similarly, the TCs should conform to IEC 584 (DIN EN 60584). By exhibiting high stability (for RTD, less than 0.1°C shift per year) we can expect reliable measurements, and consequently the maintenance decisions will have a lower level of uncertainty. Although RTDs can be small enough to give temperature readings in fractions of a second, generally, RTDs take up to a few seconds to provide an accurate reading – if they are correctly installed. TCs are less accurate than RTDs, requiring additional maintenance and calibration due to corrosion of the wires. However, in comparison to RTDs, they also have faster response times.

9. Temperature transducer localization

There are many standards, often company site specific, that speak about acceptable temperature limits for condition monitoring. However, there are very few standards that discuss optimal construction of the sensors used for a variety of temperature monitoring types and sensor locations. One such specification is [13] which addresses machinery typically used in the oil and gas industry. However, the specification is widely accepted in other industries, and its qualitative rules provide for bearing condition monitoring by means of temperature measurements can be used in many other areas (see 6.1.9.2.2 in [13]). The standard discusses as well temperature measurements both for radial bearings and thrust bearings. For hydro area applications, its serious limitation is lack of rules for machinery with a vertical rotor axis.

The main intention of the thrust bearing temperature monitoring is temperature observation of the pad coating or the temperature of the bearing running face. In general, two methods of temperature sensor retention can be used: the sensor can be bonded into material of the bearing layer or bonded into the backing material. It is desirable to have the transducer tip imbedded into the material of the bearing coating as
close to the working surface as possible. Such a transducer location would minimize the time delay between temperature change of the thrust bearing and indication on a monitoring system. Such a requirement is particularly desired for shoes with polymeric coating because they can have thermal conductivity lower than Babbitt from by ~4 times (PEEK) up to by ~170 times (PTFE).

Figure 6. Location of temperature transducer in thrust bearing pad: (A) position of transducer tip, (B) position of transducer axis according to direction of rotor revolutions indicated by $\omega$

Figure 6 presents requirements for temperature transducer localization in the thrust bearing pad in accordance with [13]. In this standard, the transducer tip is positioned $\zeta = \sim 0.75 \text{ mm}$ below of bottom surface of the working material. For various machines and for various bearing coatings the distance can be different. For example, there are some OEMs producing steam turbines which use the $\zeta = 0 \text{ mm}$, and even its negative values are being applied if the temperature transducer does not influence negatively on correctness of the coating layer operation and reliability of the bond of coating material with the backing. Figure 6B presents the location of the sensor axis. The recommended location for the sensor is termed the “75/75 location” on a thrust shoe face, i.e. 75% of the arc length of the shoe in the direction of rotation (as indicated by $\omega$ in this figure) and 75% of the radial width of the shoe measured from its internal diameter to the outer diameter. Such scenario is adequate for rotor systems rotating in one direction only. For the reversible hydro-units it is requested to use two sensors per pad with localizations: 75/75 and 25/75, where the first number refers to the angular position.

Guidelines for sensor installation in thrust bearing segments can be found also in some factory instructions. An example is presented in [17], however, it is only limited for the Babbitted surfaces.

Monitoring quality can be degraded by using temperature sensor housings. The primary purpose of the housing is to protect the sensing element. A selection of the housing will affect the response time of the sensor but can also affect the accuracy of the entire system. It is important that the housing be matched with the sensor to allow for good thermal transfer between the two components. Bearing temperature sensors should ensure constant contact of the sensing element with (preferably) the bearing coating layer. This is possible to achieve by using a spring-loaded design. Only the spring-loaded design ensures rapid and constant speed of heat transfer to the sensor. The spring-loaded design also compensates for thermal expansion of the bearing pad and sensor body which can influence change of thermal conductivity, and consequently a change of time delay between temperature change of bearing running face and proper reaction of the CMS.
Noting the problems previously described, some OEMs try to improve the thrust bearing temperature monitoring doing various “improvements”. One of them is an additional hole as presented in Figure 3A by detail “b”. The hole is localized over the temperature sensor. Its intention is providing oil directly to the backing material to improve the rate of change of temperature indications. Another modification is to make the vertical hole longer, which allows penetration of the sensor slot by oil, and additionally to make the second horizontal draining hole which allows oil circulation [18] as presented in Figure 3B.

10. A typical installation of temperature sensor in pads

In the case of many HTGs the temperature sensors are located far below the coating layer than it is required by [13]. The location of a bayonet temperature transducer\(^{(9)}\) in a pocket (see Figure 3) results in a significant time delay of a change in the bearing surface temperature, and recognition of this fact by a monitoring and protection system. An increasing of the distance “\(\delta\)” escalates the delay.

![Figure 7. Temperature monitoring of a thrust bearing pad: (A) before a temperature sensor installation, and (B) after](image)

Figure 7 shows a common method of installation of the temperature sensor into thrust bearing pad that corresponds to that shown in Figure 3. Figure 7A shows a pad with a sensor slot opening before, and Figure 7B – after temperature sensor installation. The transducer (with its housing) does not use spring loaded design. Therefore, it cannot assure continuous, and fully circumferential contact with surface of the pocket. In various pads the bayonet will touch the backing material in various places. It leads to various temperature indications for the same thrust bearing load.

Figure 8 shows 12-hour temperature trends from several thrust bearing pads of a ~200 MW reversible HTG. No thrust bearing failure was detected for this run. The HTG uses temperature transducers fixed to pads similarly to this approach as shown in Figure 7. It is visible, that

i. after HTG start-up, the stabilization of temperature indications exceeds 30 minutes,

ii. during unit operation, and after thrust bearing temperature stabilization, the coolest pad temperature is ~58°C, and the hottest one ~74°C, hence the temperature distribution is ~16°C;

iii. temperature decay after the HTG shutdown takes approximately 5 hours.
It is not desirable to have high distribution of temperature measurements in thrust bearing pads, because a change of running surface temperature of thrust bearing pads can be a symptom of malfunctions what is failure of several springs supporting thrust bearing pads. Very high loads (including load pulsations) can be transferred through pads to the pad supports and can destroy them. The only diagnostic indication of such support failure is decreasing temperature of less loaded pads, down to oil temperature. Therefore, monitoring of temperature deviation from normal to lower values can be helpful with recognition of such failure.

Another failure where the temperature monitoring would be helpful is reduction of flow to a segment. The malfunction usually results in a temperature increase of the segment. Such case is described in [7]: a HTG was shut down after temperature increasing of one segment over 20°C compared with the temperatures measured on the other segments.

If there is high distribution of temperature indications for normal operation condition (e.g. due to an application of low quality sensors and their poor installation), then it can be a problem to recognize this type of single segment failure.

11. Conclusions

The paper describes common practice versus the best rules of thrust bearing temperature monitoring for vertical HTGs.

I. A typical approach using the bayonet transducers and their horizontal localization in the thrust bearing pad is criticized. It presents high time delay between change of the running face temperature and the monitoring time reaction.
II. Bayonet transducers introduced in the past for Babbitted bearings are frequently used for polymeric coated segments. This is not desirable because the polymer thermal conductivity is significantly lower than Babbitt. Such protection can be used for instances of slow changes in running surface temperatures. However, it is deficient when oil starvation occurs causing rapid temperature change.

III. Based on guidelines provided by [13] the sensor located in the backing material must be enough close to the pad bound; locating the temperature sensor in the pad coating as close as possible to the pad running face decreases time delay of monitoring system reaction and provides more real temperature indications; consequently, the machine is better protected against rapid temperature excursions which may be harmful.

IV. Careful installation of high quality temperature sensors will result in smaller variances of temperature indications from identical pads, and minimize transducer sensitivity changes over time. For the fluid film bearing temperature measurements spring loaded transducers should be considered. More reliable temperature monitoring allows more reliable asset condition management. Temperature transducers without spring load are good for general purpose temperature monitoring, where quick and significant changes of temperature are not anticipated. For bearing monitoring of critical machinery spring loaded temperature transducers are preferred.

V. Various methods of oil monitoring are important for asset condition management. For thrust bearing operation it may not be sufficient to only monitor operating temperature. Oil starvation may not reliably be shown as temperature change. However, oil starvation will result in the change of the rotor system axial (vertical) position (if the monitoring system is enough sensitive. Change of oil thickness between thrust bearing components can be easy recognized by means of non-contacting position transducers. Such measurements can be easy provided inside the thrust bearing housing by means of eddy current transducers.

VI. Diversification of provided measurements for asset condition management allows formulating more reliable in depth conclusions about changes of HTG operating conditions.

VII. Requested number of temperature transducers used per thrust bearing can be dependent on the bearing load.

VIII. Requested number of temperature transducers used per bearing pad depends on unit criticality, and can depend additionally on types of the unit operating modes. Redundancy of transducers should be a factor, especially on critical machines.

IX. For the condition monitoring [13] presents the opinion that all condition dedicated measurements (for example, vibrations, positions, and temperatures) should be provided by one system. It is especially important, when condition based maintenance is scheduled for a HTG.

X. There are various techniques used for rotor system axial position and thrust bearing condition monitoring. An advance application of proximity probes allows collecting and analysing not only static measurements but additionally the dynamic measurements. There is requested waviness (surface irregularities) with particular limit for the thrust bearing collar; one can find description of a failure in [7], where the waviness from requested not more than 80 µm increased
up to 900 µm because of not proper operation of chosen thrust bearing segments; analysis of the dynamic component from proximity transducers what would be added to the thrust bearing monitoring would help with recognition of such malfunctions.

For this described case,

a) the operator of the unit was disappointed that the vibration monitoring system failed to detect the failure. It is well established that increased vibration may not occur during significant failures or the vibration symptoms react contrary to expectations. For example, instead of a vibration increase, it decreases.

b) The manufacturer of the unit, after checking all the deviations from previous start ups, found in the DSC event log that the start-up sequence had been changed. The lube oil vapor ejector was started early, prior to the start of the hydro unit. OEM engineers stated in their report that the ejector located above outer diameter pads created a funnel resulting in reducing oil level in the centre. This resulted in lack of oil during operation at the inner ring of segments and increased level at the periphery.

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References and footnotes

17. ‘Temperature Detector in Thrust Bearing, Case Style B, Babbitted Method’, MINCO, Engineering Instruction No. 180

(1) Babbitt - an antifriction metal first produced by Isaac Babbitt in 1839. In present-day usage, the term is applied to a whole class of silver-white bearing metals, or "white metals". Its static coefficient of friction (measured against polished steel) is 0.42...0.70 (with lubrication: 0.08...0.25), and kinetic 0.33...0.35 (with lubrication: 0.05...0.16); the melting point depends on composition is in range: (181°C/358°F) 223°C/433°F ...241°C/466°F (244°C/471°F).
(2) There are various fluoroplastics, e.g. PTFE, FEP, MFA, PFA, ETFE. The most popular and used for hydro thrust bearings from 1978 is PTFE = polytetrafluoroethylene. The PTFE was discovered in 1938. Its coefficient of friction (measured against polished steel) is 0.03...0.15 (0.35 - which is the third-lowest of any known solid material, and melting point is 327°C /620°F. PTFE is better known as Teflon®. Teflon® PTFE resins have a continuous service temperature of 260°C/500°F.
(3) PEEK = polyetheretherketone. These are the high-performance plastics started to be produced in 1983 by ICI and Bayer. Its coefficient of friction is 0.25...0.43, and melting point ~343°C/~650°F. PEEK lined thrust bearings can operate at higher bearing unit loads than Babbitt lined bearings.
(4) Thermal conductivity k is a material property describing the ability to conduct heat. Thermal conductivity is defined as the quantity of heat transmitted through a unit thickness of a material - in a direction normal to a surface of unit area - due to a unit temperature gradient under steady state conditions. Thermal conductivity units is W/(m K) in the SI system and Btu/(hr ft. °F) in the Imperial system.
(5) Such distributed CMS is not necessarily in compliance with API STANDARD [13]. The standard requests for rotating machines to have vibration, rotor axial position and temperature monitoring provided by one system. The distributed CMS presents lower quality of diagnostic inference usually than available from the integrated CMS.
(6) In some countries, it is still not a frequent approach to use rotor axial positioning monitoring for HTGs. In [11] is provided an in-depth review of reasons of HTG failures due to the thrust bearing polymeric segments. Using Eddy current transducer systems, it is not a difficult task to implement such monitoring as presented in [5].
One of problems discussed for polymeric thrust bearings in [11] is change of shape of the segment from side of the trailing edge resulting from partial wearing of the segment surface. Application of oil thickness monitoring for the trailing edge can quite easy recognize such malfunction.
(7) In polymeric coated pads used in Russia, there are wear marks that are typically provided with 2 or 4 shoes of the set’s PTFE friction surface. Depth of the wear marks are: 0.05, 0.10, 0.15 and 0.20 mm.
(8) Some liners have up to 1,5 mm of wearable thickness. Longevity of polymeric pads is expected to be between 40 to 70 years. After 20 years of operation of such bearings in Ust-Ilim HPP the wear was ~0,3 mm. In the described case during one run the RAPCh was almost 0,5mm. Keeping in mind the range of wearable thickness, we may need to do alarm level refinement in the monitoring and protection system after a number years.
(9) In general, it is possible to find “Bayonet Spring Loaded Sensor Assembly” for temperature measurements. However, in this paper, the term "bayonet temperature transducer" is does not refer to a spring-loaded transducer.