Hydrodynamic film thickness measurements and CFD analysis identify the root causes of repetitive thrust bearing failures on a 45 MW hydro generating unit at Hydro-Québec.

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Abstract. High temperature level recorded on the thrust bearing of a 45 MW hydro generating unit was resulting in frequent production stoppage. In spite of improvements brought to the oil cooling system since the rehabilitation in 2008, the operator had to activate the bearing oil lift system to keep the temperature below acceptable limits. Primary root cause analysis first pointed to the design of the shoe that was centrally pivoted, not allowing the formation of a thick hydrodynamic film. The removal of a strip of the soft metal layer near the trailing edge of the shoe resulted in a significant surface temperature reduction (about 15 deg. C), as predicted by a CFD model of the oil film. The goal of this machining was to increase the pivoting angle by moving the centre of hydrodynamic pressure. Proximity sensors were installed at each corner of the redesigned shoe to measure the film thickness and the bearing attitude. Signal analysis revealed a step of a magnitude close to the oil film thickness between the two halves of the rotating thrust block. This was the cause of another failure few hours since restarting the unit.

The lessons learnt through these measurements and analyses were carefully applied to the ultimate build. The unit now runs with a robust thrust bearing and even survived a significant cooling flow reduction event. This paper presents the CFD analysis results and the measurements acquired during these events.

1. Introduction

The Beauharnois Power House is a major plant on the St. Lawrence River located 25 km upstream of Montreal. This power house hosts 26 Francis units rated 45 MW and 10 Propeller units supplying between 55 and 75 MW each. Despite their similar dimensions, the units of this power house show surprisingly different thrust bearing designs. These reflect the evolution of the bearing technology within the various suppliers that designed and built the alternator part of each unit.

The oldest units, commissioned between 1930 et 1950 use a spring-mounted segmented stationary ring supporting a rotating thrust block. This design had not yet integrated the outcomes of the theory of hydrodynamic film. The relatively good behavior of these bearings results from the slightly convex shape that each segment adopts when run at operating temperature. About 15 years
ago, one of these units underwent many events of thermal runaway. This triggered a fine analysis of the hydrodynamic film that considered the distortion of the segment [1]. This analysis was predicting a 13 micrometers film thickness, much less than the current 50 micrometers acceptance of modern designs.

Recent units, commissioned since 1950, include shoes that can pivot individually, whether it be on an array of springs or on the tip of a supporting pin.

2. Historical background.

The bearing of a unit commissioned in 1953 consists of 8 individual shoes that tilt on a pin. The height of each pin is adjusted to achieve equal load sharing. The tip of each pin is exactly centered on an arc drawn between the leading and the trailing edge of the shoe. It is well known by today that such central location does not lead to the creation of a convergent hydrodynamic oil film. Figure 1, from early theoretical research [2], clearly shows that the film thickness approaches zero when the pivot is located at the center of a sliding bearing. A thin hydrodynamic film makes the thrust bearing more sensitive to surface imperfections and to oil contamination.

![Figure 1 Non-dimensional film thickness vs pivot position arc ratio, from “Analysis of Sliding Bearings” Raimondi and Boyd](image)

The shoes also feature a complex system of grooves and channels machined close to the trailing edge, all of these ending up in the oil tank. It is presumed that this machining was meant to favour the transfer of the hot oil from the shoe surface to the surrounding oil tank.

By 2008, the rehabilitation of the turbine runner gave an opportunity to provide this bearing with an oil lift system. In such system, a volumetric oil pump sends oil at high pressure in a distribution pocket at the surface of the shoe during turbine start-up and run-down sequences, when the rotational speed is too small to promote creation of a hydrodynamic oil film. Besides, this auxiliary system simplifies the maintenance steps that require slow rotations of the rotor.

Before this rehabilitation, the bearing temperature was about 20°C above that measured on the adjacent units with shoes mounted individually on an array of springs. This difference was rightly or wrongly ascribed to a deficient heat transfer between the coils of the cooler and the oil bath. To remedy the suspected deficiency, the number of coils had been increased and those were fit with cylindrical fins.

The rehabilitation did not reduce the bearing temperature. In fact, this parameter showed a slow progressive increase at the average rate of about 3°C by year. The first year, one could typically measure 72°C for a water cooling flow of 240 litres/minute, which is nearly double that required on the adjacent units.
In 2013, a failure occurred. Figure 2 shows that the soft metal layer fused. The circular pocket of the oil lift system, centered on the supporting pin, can still be discerned. One can also see the machined groove and channels near the trailing edge. The rotational direction of the thrust block is shown by the blue arrow and the “6” refers to the number of this shoe, with 533mm internal and 1117 mm external radii.

Following the replacement of the shoes by an identical model, the unit is back to production in July 2013. Then the temperature stabilizes out at 77°C for a cooling flow of 314 litres/minute. In order to limit the uplift of temperature, limitations are set on the power of the unit, for it is observed that every power increase has a direct impact on the temperature due to the suspected incremental hydraulic load. When the temperature of the shoes soars close to the 85°C threshold, the bearing oil lift system must be activated. This exceptional procedure temporarily reduces the temperature and allows the unit to cross the peak demand of hydro-electric power during the winter of 2013-2014. Nevertheless, early in spring 2014, the bearing fails again and the unit must be dismantled one more time. On figure 3, one can observe severe shredding of the soft metal layer.
3. Finite element analysis of the existing design.

From then on, the search for a definitive solution to the misreliability of this bearing is entrusted to the “Direction Expertise de Centrales” at Hydro-Québec Production. Questions were then risen regarding the role of the channels machined on the shoes. Besides, some modifications to the design suggested by observing other designs deserved closer examination, for an example, the addition of scrapers to prevent the transfer of the hot oil from a shoe to its following.

The hydrodynamic science of the pivoted shoes is well documented when it deals with shoes with a smooth surface [3][4]. However, since the existent shoes were showing several singularities, an analysis by computational fluid dynamics (CFD) was deemed necessary to properly take those into account.

The engineers entrusted to the solution currently use the CFD solver CFX-5 in a very wide spectrum of applications. Whether it be the turbulent flow between the blades of a Francis runner [5] or in the tubes of a heat exchanger, or the ventilation and natural convection prediction around control cabinets, this software has demonstrated its usefulness. The need to know the conditions for operation of the existent bearing (e.g. the thickness of the film and the angle of inclination of the shoe) justified the implementation of a CFD model.

3.1. CFD model.

The fluid domain includes the complete environment of one shoe. In a first step, the solid model of the shoe is withdrawn from that of a 45 degrees angular segment (remember: 8 shoes) formed by the outlines of the oil bath and the thrust block by using the boolean function of a geometrical design software.

Initially, the surface of the shoe is positioned parallel and at 50 micrometers from the thrust block, for one does not know in advance neither the thickness of the film nor the attitude of the shoe when submitted to the load and speed conditions. The geometric file is used to produce a structured mesh of a typical size of 500 000 elements with the software ICEM-HEXA. The choice of a hexahedral mesh, rather than tetrahedral, becomes evident if one considers the very large number of tetrahedrons that would be needed in the thin hydrodynamic film to keep geometric determinants within a range usable by the CFD solver.

Within the film and in the vicinity of the shoe, the Reynolds number of the problem is small enough to use the option "laminar flow" in CFX-5. However, the friction between fluid layers being the ultimate source of warming, the option "energy equation" is activated.

Although these are considered as simplifications, all surfaces are treated as adiabatic and the oil dynamic viscosity is given by the Vogel equation which follows:

\[ \mu = 0.00018 \times \exp((159.56/(T+95)-0.181913) \times \ln(870 \times \text{VG}/(1000000 \times 0.00018))) \]

Where:
- \( \mu \) : dynamic viscosity (Pa.s)
- \( T \) : oil bath temperature, 40°C.
- \( \text{VG} \) : ISO grade of the mineral oil (46)

To launch the analysis, the rotational velocity (75 RPM) is imposed at the surface of the thrust block. Periodic boundary conditions are imposed on either end of the angular segment.
The search for the operating condition is an iterative process which consists of adjusting, by trial and error, the following parameters:

- The minimal oil film thickness (which is the minimal distance between the shoe and the thrust block).
- The attitude angle of the shoe in the circumferential direction.
- The attitude angle of the shoe in the radial direction.

The geometry rotational and translational tools of ICEM-Hexa are extensively used to modify the position of the shoe boundaries within the domain, however, while keeping the same blocking for every mesh.

The process is completed when the following criteria are fulfilled:

- The force computed by area integration of the pressure field in the film is equal to the thrust load. In the case presented here, this is the weight of the rotating unit plus an additional estimated 25% contribution from the hydraulic load.
- The final angular attitude of the shoe is such that there is no rotational moment around the pin about the two main axis.

3.2. Results.

This table shows the results of the basic design and for a configuration proposed for a re-build.

| Configuration                               | Base  | Proposed |
|--------------------------------------------|-------|----------|
| Minimum film thickness (micrometers)       | 46    | 53       |
| Attitude of shoe in circumferential direction (degrees) | 0.0036 | 0.0064   |
| Max temperature rise with respect to oil bath temperature (°C) | 74.9  | 55.3     |
| Frictional losses (KW) or power absorbed by the bearing | 84.9  | 68.3     |
| Maximum film pressure (MPa)                | 7.04  | 7.09     |

Despite of the central position of the support, the basic shoe pivots by a slight 0.0036 degree angle. The minimal film thickness, 46 micrometers, is barely less than the currently accepted 50 micrometers criterion. The analysis of the pressure field, (figure 4), suggests that the interruption of the film by the machined channels effectively moved the pressure center away from the trailing edge, although this might not have been the primary purpose of these. However, the same analysis shows that the presence of these channels does not have much effect on the heat exchanged with the oil bath.

The analysis also predicts a temperature field, (figure 5). The present temperature sensor location is far from the predicted high spot which is at the trailing edge and outwards. In addition, the temperature at the leading edge is already 15°C above the oil bath, which would give enough justification to add a scraper between two shoes. A subsequent analysis effectively shows that such item would effectively reduce the maximal temperature by 10°C without any adverse effects on film thickness. This option was not retained due to excessive delays in procurement. Finally, the predicted friction losses match reasonably well the heat rejection that could be deduced from measurements of the rise of water temperature through the cooler.
3.3. Proposed design

The specific pressure of the base configuration, 2.39 MPa, is low in comparison to the limitation of 4 MPa that is currently used for the soft metal, which is ASTM grade 2 Babbitt. There was intuition that the beneficial effect of moving the effective pivot position to 60% of arc length would by far compensate for any increase of specific pressure. The removal by machining of a strip of the soft metal layer near the trailing edge of the shoe, such as circled on figure 7, is a straightforward method to bring the effective pivot center at 60% of the arc length.

Though this was done at the cost of increasing the specific pressure to 2.74 MPa, the maximum pressure was expected to increase very slightly from 7.04 MPa to 7.09 MPa. The proposed configuration was expected to perform significantly better: increased thickness of oil film, reduction of heat generation by friction and reduction of the maximal temperature as can be seen in the rightmost column of table 1. All these benefits are a result of a simple machining operation that
effectively increases the attitude angle of the shoe. This design change was first applied to an existing set of shoes of the previous configuration.

4. First start with the proposed shoe configuration.

In September 2014 the unit is re-started with the proposed configuration. Four precision proximity sensors working on the principle of magnetic permeability were installed at each corner of a shoe (figures 8 and 9), facing the thrust block, to provide a live measurement of the hydrodynamic film thickness and of the shoe attitude.

At Beauharnois, the usual starting procedure of a unit with a new bearing is very cautious, due to past experiences of early failure. It includes a 24 hours of operation at “speed no load” followed by a gradual power increase by steps of 5 or 10 MW at each stabilization of the bearing temperature. Figure 10 compares the temperature rise of the new thrust bearing (in pink) with that of 2013 (in blue). While the 2013 bearing was showing an erratic behavior and needed frequent re-adjustments of the cooling flow, the new bearing reached a steady temperature plateau after less than 10 hours. Moreover, this was obtained at the expense of a much lower cooling flow (144 vs 314 litres/min).
Despite those good results that seem to confirm the analysis, the signal of the proximity sensors was giving rise to some pre-occupations. Figure 11 obtained during a start-up from rest, shows the distance measured by the sensor at the trailing edge and inwards position. By mistake, the oil lift system was not activated, but this allowed to monitor the creation of an oil film close to 55 micrometers as the unit reached synchronous speed. The film thickness was found to oscillate between two values (48 et 62 micrometers) at each passage of the joint between the two halves of the thrust block. Although the positive peaks can be explained by the presence of a chamfer between the two halves of the thrust block, the negative peaks can hardly be explained by anything else than a brief metal-to-metal contact between the shoe and the block.

5. Root cause identification of bearing failure

About 40 hours after having re-started the unit, the bearing temperatures quickly ran away and provoked the stoppage of the unit. This unexpected event triggered one more unscheduled disassembly.

The root cause identification process is a tool that helps to identify the possible cause(s) of an event, after elimination of the least plausible. On the tree of bearing failure causes, the only causes that could not be eliminated after examination of the data and hardware are the thrust block lack of flatness and the Babbitt deposition process.

At the strip of the bearing, a step of 0.001 inch (25 micrometers) was measured between the two halves of the thrust block. Figure 12 shows some parallel scratches in the radial direction, which were attributed to the metal-to-metal contact also suggested from the proximity sensor signal analysis.

About 0.001 inch (25 micrometers) thick material removal was also found on each shoe. A metallographic analysis suggested that this could have been caused by a deficient soft metal spraying process that prevented adhesion between layers. A later examination of the proximity sensor signals confirmed a gradual diminishing of the Babbitt coating thickness estimated to 0.001 inch.

The lessons learnt from this event were applied to the last build. The soft metal layer was cast. Both halves of the thrust block were carefully ground together after their assembly with tight fit dowel pins. The step measured at assembly was less than 8 micrometers. Some more CFD analysis was done to validate minor changes to the oil lift distribution pocket. The proximity sensors were re-installed, for having demonstrated their usefulness.
6. Measurements taken on last build.

6.1. Measurements taken at the last start-up March 3 2015.

This time, at the nominal power, 45 MW, the proximity sensor signals do not display any metal-to-metal contact. As expected, positive peaks correspond to the passage of a recessed chamfer, which does not seem to negatively impact bearing performance. As could also be expected, the minimal distance to the thrust block is found at the outwards corner on the downstream side, which confirms that the shoe tilts in the right direction.

![Figure 12 Two halves of the thrust block, prior to disassembly](image)

A 3D Graphical analysis of the time-averaged measurements show that the shoe tilting angle in circumferential direction is 0.0035 degree, which is less than the prediction (0.0064 degree). However, the measured minimal film thickness, 76 micrometers is above the prediction.

![Figure 13 Proximity sensors temporal signal](image)
There might be some explanations for the discrepancy between measurements and predictions. The differential thermal expansion between the steel body of the sensors and the thick 10 mm soft metal layer introduces a non-negligible negative correction factor estimated to 16 +/- 5 micrometers. Besides, the tip of the 4 sensors do not seem to align into a perfect plane, which may be a result of minor shoe distortion, estimated to less than 5 micrometers.

On the other hand, it is understood that the CFD model can be improved. For an example, the local viscosity can be recomputed based on local mesh temperature and pressure. Multiphysics coupling with a structural analysis of the shoe can also account for potential distortion effect of the shoe surface.

6.2. Cooling flow reduction: event of 12th March 2015.

Few days after this successful re-start, the water cooling flow was accidentally reduced from 134 to 20 litres per minute for a full day. Despite this unhappy operational mistake, the shoe temperature (in green) came back to its initial value after re-establishment of cooling flow. This demonstrates the ruggedness of the new configuration.

7. Conclusions

Many lessons were learnt from all these events. First, engineers should not hesitate to question and revisit original designs with the numerical tools that are now currently available. Despite some limitations, these provide the necessary insight needed for solving problems and evaluating the potential of design modifications.

In addition, quality problems encountered with the soft metal has triggered a study on quality control of this process.

Finally, instrumentation has demonstrated a major role in root cause identification and have supported the improvement in design and assembly practices.

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