BEHAVIOUR OF A HYDRODYNAMIC JOURNAL BEARING: TORQUE MEASUREMENT DURING START-UP

Jean BOUYER, Michel FILLON

Laboratoire de Mécanique des Solides, Université de Poitiers, UMR CNRS 6610, SP2MI, 86962 Futuroscope Cedex, FRANCE.

bouyer@lms.univ-poitiers.fr

ABSTRACT

The start-up friction coefficient in hydrodynamic bearings, used by engineers in the design of bearings, is most often issued from an approximation. Several studies can be found in the literature on this topic but most of them are concerned with air bearings or are only numerical. Other studies are more general and deal with the transient thermal behaviour in journal bearings as well as in thrust bearings. Only few studies are focused on experimental investigations of the friction coefficient or bush torque measurement. The aim of this study is to provide experimental measurements of the bush torque during start-up for various static loads.

Keywords: journal bearing, experimental, torque, start-up.

INTRODUCTION

The start-up friction coefficient in hydrodynamic bearings has been the subject of several published studies, but most of them are dedicated to air bearings or are only numerical. We will present here a brief review of papers on this topic.

In 1989, Dufrane et al. [1] studied the seizure times of journal bearings. They analyzed seizure due to the insufficient lubrication by a simple numerical model, relating the seizure time to the bearing operating parameters. They also conducted a series of experiments to determine the seizure time. They concluded that, in high-speed bearings, the seizures could occur very rapidly, within seconds of operation. Harnoy [2] presented in 1995 the modelling of the time-variable friction during the start-up of a journal bearing. He showed that it is possible to reduce the wear of the bush by increasing start-up acceleration and that a flexible support could reduce the maximum friction force and energy losses. Pistner [3] studied in 1996 several examples of industrial bearing pads that have wiped during a cold start-up. He demonstrated that, due to the large size of pump/turbine thrust bearings, and due to the rate at which the units are accelerated to the running speed, the face temperatures of the thrust shoes increase rapidly during start-up. This caused a catastrophic deflection of the pads which induced the rupture of the oil film and thus caused the wiping of the babbitt surface. He concluded that this problem could be avoided by preheating the bearing pads before start-up. More recently, Krithivasan et al. [4] developed a finite element model for predicting thermally induced seizure during start-up and transient flow disturbance. Ettles et al. [5] presented in the same year a work on the effects of start-up and shut-down on hydro-generator thrust bearings. Based on the observations made by Pistner [3], they developed a two dimensional model of transient thermoelastic effects in bearing assemblies. They proposed several procedures using jacking oil lifts as to avoid seizure during the start-up and shut-down of thrust bearings. In 2005, Wang et al. [6] presented a numerical analysis of a tilting-pad thrust bearing during start-up. They studied both the effects of start-up time and start-up load on the temperature distribution in the oil film. They showed that the fluid film forms quickly at the beginning of the start-up procedure and also that the film temperature is lower in a rapid start-up than in a slower start-up. Most of these works are numerical while the published experimental data only focuses on the consequences of seizures caused by high friction during start-up.
Thus, the aim of this study is to provide experimental measurements of the bush torque during start-up. The experiments presented here have been made under transient regime, for various static loads and start-up accelerations.

**TEST RIG**

The experimental apparatus is well described in a previous work [7], and only some details on the test bearing will be given here. The bearing is a two axial groove bearing which has a diameter of 100 mm and a length of 80 mm. The radial clearance is 171 µm.

As shown in figure 1, the bearing was equipped with 30 thermocouples and 15 pressure probes. The bush supporting assembly is attached to a shaft which allows transmitting the frictional torque to the torque measurement device. A view of the measurement apparatus is given in figure 2 which also shows the twelve pressure probes in the front of the test bearing. The operating conditions are given in Table 1.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Units</th>
<th>Value</th>
</tr>
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<tbody>
<tr>
<td>Nominal speed</td>
<td>rpm</td>
<td>2000</td>
</tr>
<tr>
<td>Applied load</td>
<td>kN</td>
<td>1 – 3.5</td>
</tr>
<tr>
<td>Supply pressure</td>
<td>MPa</td>
<td>0.1</td>
</tr>
<tr>
<td>Supply temperature</td>
<td>ºC</td>
<td>39</td>
</tr>
<tr>
<td>Lubricant grade</td>
<td></td>
<td>ISO VG 32</td>
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</table>

Table 1. Operating conditions.

![Fig. 2. View of the couple meter and pressure probes.](image)

**RESULTS**

Figure 3 shows the evolution of the friction torque during 20 seconds start-ups for several feeding temperatures. The tests were composed of a 20 seconds start-up followed by a 100 seconds maintain at nominal speed. These measurements were performed at the beginning of the day, before the test rig temperature increased. We can note that the torque is significantly dependant of the feeding temperature: the more significant differences are noted at nominal speed, where the gap can be greater than 100% for the extreme cases. The torque at nominal speed was 4.2 N·m and 2.1 N·m, respectively, for the feeding temperatures of 19°C and 40°C. For all the tests, the discrepancy on the maximum torque (at start-up) is around 1 N·m.

In order to ensure a good repeatability of the measurements, the tests presented below were performed after the thermal equilibrium of the test rig was achieved.

The tests were divided in two periods: the first one was the speed increase from 0 to the nominal speed, while the second one consisted in maintaining the nominal speed for a chosen time period. Several parameters were measured during this time, such as the hydrodynamic pressure, the temperature at the film/bush interface, the feeding temperature, and the shaft position in the bearing.

![Fig. 3. Torque measurement for a 20s start-up at several feeding temperatures (2.5 kN static load).](image)

![Fig. 1. Angular location of the thermocouples (♀) and pressure holes (♀) at the inner surface of the bush.](image)
Figure 4 presents the results obtained for an acceleration time of 120s, followed by 30 minutes of operation at a constant nominal speed. The evolution of the torque can be divided in three periods. In the first one, a torque peak due to the high friction between surfaces is observed before the shaft begins to rotate, followed by a sudden decrease in frictional torque which indicates that the hydrodynamic regime is attained. In the second zone, a gradual increase of the torque occurs, corresponding to the increase of the rotational speed. In the last zone, a slight decrease in friction can be observed: it is due to the evolution of the feeding temperature during the 30 minutes of the test. It has to be noted that the value of the torque before the shutdown was 1.79 N.m which is close to the results obtained by numerical simulation, which predicted a torque of 1.66 N.m.

Figure 5 presents a more detailed evolution of the torque during the first seconds of the start-up. The dotted line shows the motor instruction, and the points show the measured speed during the test. One can note that, during the first few seconds of the test corresponding to the peak of the torque (zone I), the shaft performs less than one revolution, the speed increasing only afterwards, corresponding to the beginning of the hydrodynamic zone. The shaft begins to rotate at the beginning of zone II and then the speed quickly increases, leading to an increase of the torque. When the nominal speed is attained (zone III), the torque becomes constant.

The evolution of the torque during start-up is given in Figures 6a and 6b for several acceleration times. The shape of the torque evolution during start-up is the same as the one observed in Figures 3 and 4. The torque at start-up is not affected by the shape of the speed increase curve used to reach nominal speed. The four curves give nearly the same maximum torque and the shape of the four plots is similar. Table 2 gives the value of the maximum torque obtained during these tests. The discrepancies between the measured values of the torque are less than 1 N.m and can be explained by the uncertainties of the measurement procedure. It can be noted in Figure 6b that, whatever the start-up time, the measured torque at nominal speed is nearly constant which indicates a good repeatability of the tests.

Concerning the shutdown period, one can note that the bearing stops in the same way for all the tests. When the motor is shut down, the bearing is free to stop and the decrease of the torque during this period is nearly linear, varying from the running value to zero.

These results are in contradiction with what has been shown by Harnoy [2]. This is because the configuration studied here is a lightly loaded setup. In case of high loads, greater acceleration could indeed reduce the maximum torque at start-up. The authors will investigate these aspects in the further work.
Figure 7 shows the experimental results for the maximum start-up torque, obtained for several static loads varying from 1 kN to 3.5 kN. In order to ensure a good repeatability of the results, each test has been performed at least eight times. The results presented here are obtained from a total of 68 measurements. Error bars give the uncertainty obtained within the experiments. This figure shows that, as it could be expected, the maximum torque at start-up increases linearly with the static load. Nevertheless, this tendency would have to be confirmed by other experiments for higher static loads.

<table>
<thead>
<tr>
<th>Start-up duration</th>
<th>Maximum torque</th>
</tr>
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<tbody>
<tr>
<td>10</td>
<td>13.54</td>
</tr>
<tr>
<td>20</td>
<td>14.16</td>
</tr>
<tr>
<td>30</td>
<td>13.36</td>
</tr>
<tr>
<td>40</td>
<td>13.55</td>
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<tr>
<td>80</td>
<td>13.87</td>
</tr>
<tr>
<td>120</td>
<td>13.12</td>
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</table>

Figure 8 gives a representation of the friction coefficient as a function of the bearing specific pressure.

The values of the friction coefficient presented here are average values calculated from the results presented in Figure 7. One can note that the specific pressures investigated in this work are quite small, and further experiments are necessary in order to improve the quality of these observations.
CONCLUSION

An experimental investigation of the evolution of the friction torque during the start-up of a journal bearing has been achieved. The results showed good repeatability, and good agreement with numerical simulations was obtained at nominal operating conditions. This work led to the following conclusions:

- The torque is very sensitive to oil temperature. The measurement has to be performed after the thermal equilibrium of the system is attained.
- For low static loads, such as the ones considered in this work, the acceleration time has no influence on the maximum torque recorded during start-up.
- The maximum torque varies linearly as a function of the applied static load.

This study is the first part of a very complete work which will be achieved in a few months. It will be completed with other experimental results for higher specific pressures. Furthermore, the very first seconds of the start-up and shut-down will be analyzed with greater accuracy in order to try to explain what happens during this particular moment which reflects the transition between the mixed and hydrodynamic lubrication regimes.

REFERENCES