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AN EXPERIMENTAL INVESTIGATION ON FRICTION COEFFICIENT IN PLAIN JOURNAL BEARINGS DURING START-UP

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ABSTRACT

The start-up friction coefficient is a parameter which is very useful for engineers who design hydrodynamic bearings. It 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. Some studies are more general and deal with the transient thermal behaviour in journal bearings as well as in thrust bearings. Other studies deal with the friction coefficient during running, at a fixed rotational speed. The aim of this study is to provide experimental measurements of the bush torque during start-up of plain journal bearings, varying the specific pressure. Thus, the friction coefficient at start-up (deduced from measurements) is obtained for four different bearings varying feeding conditions, radial clearance and length.

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 related to this topic.

In 1989, Dufrane et al. [1] analyzed seizure of journal bearings due to the insufficient lubrication by a simple numerical model, relating the seizure time to the bearing operating parameters. They concluded that, in high-speed bearings, the seizures could occur within seconds of operation. Sun et al. [2] presented in 1994 a paper on the starting characteristics of an unlubricated journal bearing. They showed numerically that the motion of the journal in an unlubricated bearing consists of various combinations of rolling and sliding. Harnoy [3] presented in 1995 a model to evaluate friction during start-up in hydrodynamic 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 [4] studied in 1996 several examples of industrial bearing pads that have wiped during a cold start-up. He demonstrated that the face temperatures of the thrust shoes increase rapidly during start-up, causing a catastrophic deflection of the pads which induced the rupture of the oil film. He concluded that this problem could be avoided by preheating the bearing pads before start-up. Ettles et al. [5] studied in 2003 the effects of start-up and shut-down on hydro-generator thrust bearings. Based on the observations made by Pistner [4], 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] 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 and also that the film temperature is lower in a rapid start-up than in a slower start-up. Lu et al. [7] presented in 2006 a study on the lift-off speed in journal bearings. They analyzed the characteristics of the Stribeck curve varying load, oil temperature and oil type. They continued in 2007 [8] with an experimental investigation of grease lubricated journal bearings. They discussed the advantages of grease instead of oil as a lubricant for heavily loaded bearings operating at relatively low speeds. Ünlü et al. [9] conducted the same year a study which deals with the determination of the friction coefficient in journal bearings. They developed a new test rig and a method to measure friction coefficient of journal bearing, but they were not interested in the friction at start-up.

Most of these works are numerical while the published experimental data focuses mainly on the consequences of seizures caused by high friction during start-up or on friction in steady-state operating conditions.

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, varying static load and for four different bearings.

EXPERIMENTAL RIG

The experimental apparatus being well described in a previous work [10], only some details on the test bearing will be given here.

The equipment of the bearings was reduced in order to not disturb torque measurement. Only few thermocouples were inserted at the film/bush interface. The torque measurement device is composed of a shaft which is attached on each side of the bush supporting assembly and allows the frictional torque to be transmitted to the torque transducer, placed in the front of the test bearing. A view of the measurement apparatus is given in Fig. 1.

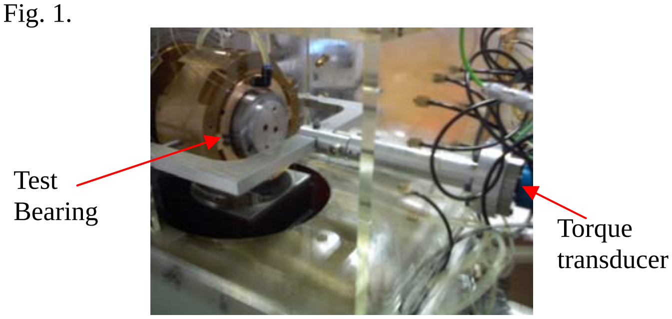

Figure 1. Torque measurement device.

Table 1 gives the geometrical parameters of the different bearings. The bearings were all driven with the same stainless steel shaft. The operating conditions are given in Tab. 2 just below. The test where performed the same way for all the bearings: a 10s acceleration from 0 to nominal speed followed by a 50s maintain at nominal speed and stop.

Bearing	#1	#2	#3	#4
Grooves	2	1	1	1
Length(mm)	0.08	0.08	0.06	0.04
Bush diameter(mm)	99.96	100.015	99.96	99.96
Radial clearance(µm)	90	117.5	90	90

Table 1. Geometrical parameters.

Parameters	Units	Value
Nominal speed	Rpm	1000
Applied load	kN	1 – 8
Supply pressure	MPa	0.1
Supply temperature	ºC	39
Lubricant grade	-	ISO VG 32

Table 2. Operating conditions.

RESULTS

In order to ensure a good repeatability of the measurements, the tests presented here were performed after the thermal equilibrium of the test rig was achieved. Each test was repeated at least six times. Moreover, according to the results presented in a previous work [11], the acceleration time and the feeding temperature, which have no influence on the maximum torque measurement, are fixed and respectively equal to 10s and 39°C for all the tests.

The uncertainty on specific pressure is constant, whatever the value considered. Hence, being ±0.0125 MPa, it ranges from ±10% for the low specific pressure to only ±1% for the higher value. The uncertainty on torque measurement varies from ±0.7 N.m for the low specific pressures to ±1.8 N.m for the higher ones.

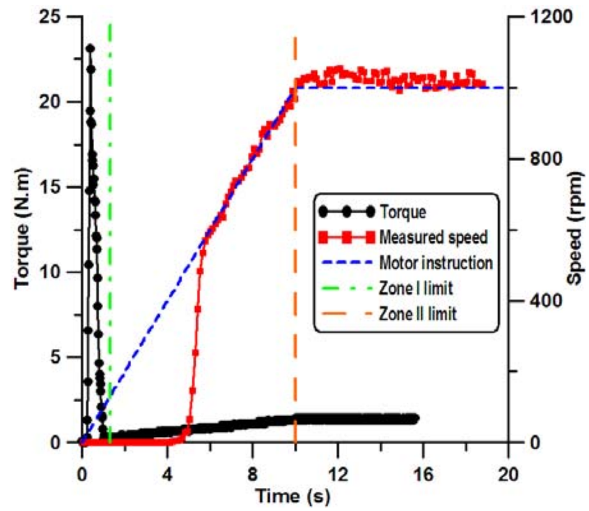

Figure 2. Torque during start-up under 3 kN static load.

The measurements presented in fig. 2 are obtained on bearing #1 for a static load of 3 kN, i.e. a 0.375 MPa specific pressure. The evolution of torque during start-up can be separated in three zones. The first zone (from 0 to vertical green dash doted line) corresponds to a peak which is due to direct contact between the shaft and the housing. The torque evolution is from 0 to 23 N.m followed by a fall towards 0.22 N.m which corresponds to the introduction of oil in the contact. The second zone (between vertical dashed lines) shows the increase in torque due to the acceleration of the shaft. Once the nominal speed is attained (third zone), the torque remains nearly constant and equal to 1.3 N.m.

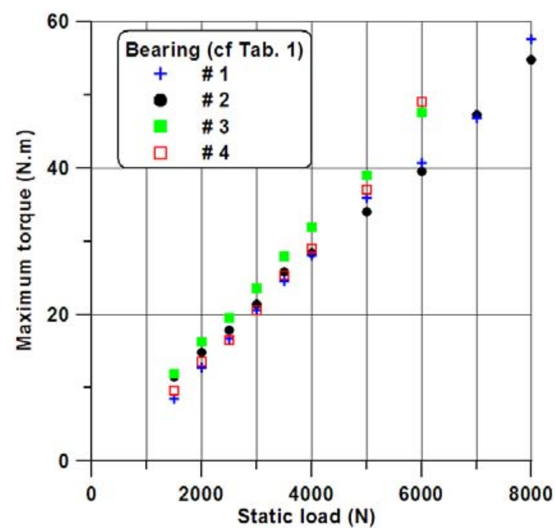

Figure 3. Maximum torque *vs* static load

Figure 3 shows the results obtained for the four bearings at several values of static load. Whatever the bearing considered the maximum torque at start-up increases linearly with the increase in static load. One can note that the slope increases for the shorter bearings, and decreases for the bearing with higher radial clearance.

The evolution of the friction coefficient as a function of specific pressure is presented in Fig. 4. The friction coefficient for bearing #1 increases with the specific pressure towards a value around 0.14, and remains almost constant with increasing specific pressure from 0.4 to 1 MPa. For bearing #2, which has the higher radial clearance, there is a slight decrease of friction coefficient around the same value of 0.14. The bearing #3 shows higher friction coefficient but quite constant, around 0.16, whatever the specific pressure considered. The shortest bearing friction coefficient slightly increases with specific pressure from 0.13 to 0.16 but the mean value remains around 0.14.

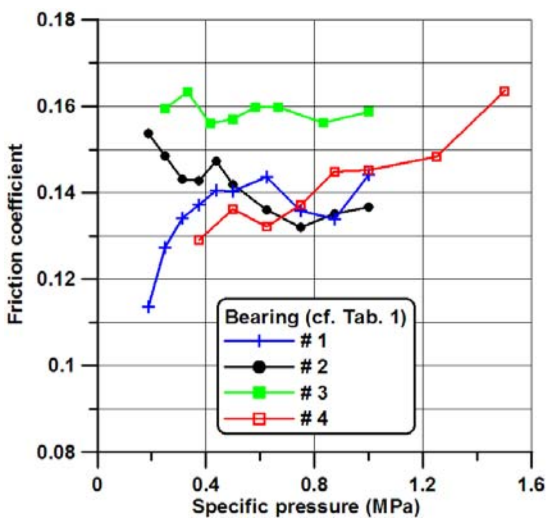

Figure 4. Friction coefficient *vs* specific pressure

We can then conclude that the friction coefficient at start-up doesn't depend on the specific pressure applied to the bearing. Its value remains in a domain which varies around 0.15 ± 0.01. This value is very close to that which can be found in the literature for the friction coefficient between steel and bronze. Indeed, the value of the friction coefficient for bronze/steel contact is 0.15-0.18 for dry contact and 0.1-0.13 for lubricated contact. At start-up, there is a direct contact between the shaft and the bearing which can be neither dry with a high friction coefficient nor partially lubricated with a lower friction coefficient.

CONCLUSION

The measurement of the torque at start-up is not an easy task. Nevertheless, the tests showed that the torque increases linearly with the specific pressure and that the friction coefficient remains almost constant whatever the specific pressure considered.

In order to understand exactly what occurs in the first revolution of the shaft, future work will be conducted to determine the mechanisms of the transition between contact and hydrodynamic lubrication.

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