EXPERIMENTAL VERIFICATION OF A MODEL OF THE LOSS OF FLUID FRICTION IN A HYBRID THRUST BEARING

Key words:
thrust bearing, hydrostatic lubrication, pressure distribution measurement

Summary

One of the known solutions of thrust bearings for heavy machines is a hybrid slide axial bearing composed of a set of bearing pads supported on sets of compression springs. During operation of such a system, a series of failures have been observed. The failures were caused by inappropriate...
parameters of the supporting springs and the appearance of an adverse system of forces acting on the bearing pad.

The paper presents the results of experiments in which pressure distribution in a bearing interspace of a model hydrostatic thrust bearing was measured. The research goal was to verify an earlier formulated quantitative model, describing the conditions at which the mentioned failure occurred. The results of the experiment confirm the legitimacy of the proposed model.

INTRODUCTION

One of the known British firms has developed an original design of water turbine thrust bearing. Significant axial forces are borne by a flanged bush, whose flange mates a sliding ring rested on a set of self-inclining bearing pads. The required self-inclination is assured by settling each bearing pad on a set of high rate compression springs. The bearing is built as a hybrid one. During start-up and run-down, at low rotational speed, required fluid friction conditions are assured by a hydrostatic lubrication system. Each bearing pad has an oil groove, supplied with oil of pressure high enough to generate continuous oil film.

Once the necessary rotational speed is achieved, hydrodynamic lubrication conditions appear. Such bearing system is applied in one of the Polish hydro-electric power stations. The bearing’s sliding ring mates sixteen bearing pads supported by sixteen springs each – Fig. 1.

In the initial period of turbines operation, a few failures have been recorded, one of those followed by serious consequences. The analysis of the failures’ sequence and their outcomes allowed drawing a conclusion, that during the start-up, individual pads self-inclination was incorrect. This led to oil film continuity loss, metallic contact of the mating sliding surfaces, friction parameters change and, in consequence, destabilisation of bearing operation parameters.

The most probable reason for the failure was a dispersion of technical parameters of the supporting springs. This caused non-uniform pads support, which in turn caused uncontrolled inclination of the pads, loaded from above with the hydrostatic oil film. After examination of the springs characteristics, every single of them was assigned strictly specified location. Thanks to this, a required bearing pad supporting forces distribution was obtained. Ever since, the bearings performance was failure-free, which confirms that the conclusions were correct.
QUALITATIVE MODEL

The mechanism of loss of stability of the bearing pad in the hydrostatic start-up phase is presented in Fig. 2. If all the supporting springs have the same characteristics, the resultant of the spring compression forces should have its application point in the centre of the bearing pad – Fig. 2a. Otherwise, the springs share the load unequally. This causes the resultant force application point move. In consequence, the bearing pad tilts, and the geometry of the bearing interspace changes, which in turn makes the oil film pressure distribution change. The oil film acts on the bearing pad with a resultant force, applied in the centre of gravity of an imaginary solid depicting the pressure distribution. As long as the two points lie on a common vertical line, the bearing pad remains stable, although tilted – Fig. 2b. If the bearing pad tilt doesn’t result in enough large change of the interspace geometry, the two forces $F_{\text{oil}}$ and $F_{\text{spr}}$ will
create a moment, which will further tilt the bearing pad, until physical contact of the bearing pad’s edge and mating surface – Fig. 2c [L. 1, 2].

**Fig. 2.** Behaviour of a bearing pad under the forces of oil film total pressure and springs support resultant: a) resultant spring support force $F_{spr}$ acting in the centre of the bearing pad; b) slightly eccentric resultant causes bearing pad tilt and bearing interspace geometry change; c) significantly eccentric spring support force causes local loss of fluid friction – 1 – bearing pad; 2 – sliding ring

Rys. 2. Zachowanie segmentu łożyska pod wpływem działania sprężyn i hydrostatycznego filmu olejowego: a) wypadkowa siła podparcia sprężynami działa w środku segmentu; b) nieznacznie przesunięta wypadkowa działania siły podparcia, powoduje pochylenie segmentu i zmianę kształtu szczeliny smarowej; c) znacznie przesunięta siła podparcia sprężynami, powoduje lokalną utratę tarcia płynnego – 1 – segment łożyska; 2 – pierścień ślizgowy

**RESEARCH STAND**

In order to verify the presented quantitative model of fluid friction loss on the bearing pad, a measurement of pressure distribution in the oil film was conducted. The measurement was taken in a hydrostatic oil film generated on an active face of ring-shaped model bearing pad – Fig. 3.

It had been assumed, that the ring-shaped face is for the experiment’s purpose, a sufficient approximation of a real bearing pad with circular oil groove in the centre. The experiment was performed on a stand, whose essential scheme is presented in Fig. 4. The examined bearing pad model (1) rested on a measurement slab (2). Between the mating faces of the model and the slab, a hydrostatic oil film was generated. In the point “p”, the measurement slab has a hole, connected to an NPx200 pressure sensor. The measurement slab rests on an additional bearing slab (3), and thanks to the oil film generated between them, has the ability of free movement, so that the point “p” can be placed within 120 mm away from...
that point, but only in one direction – the measurement slab has one degree of freedom. The measurement slab location is recorded with a PJx300 linear translation sensor. The data are acquired with an NI USB-6009 data acquisition card. It is also possible to adjust the location of the bearing pad model so, that the load $G$, simulating the springs support resultant force, can be applied within $\pm 16$ mm from the centre of the model. In the corners of the bearing pad model, dial gauges are fixed to measure the oil film thickness.

**Fig. 3. Sketch of a slide thrust bearing pad model**

Rys. 3. Szkic modelu segmentu ślizgowego łoża wzdłużnego

Pressure distribution measurements were taken at load values $G$ of 40 kN, 60 kN, 80 kN, and 100 kN, and at eccentricity $e$ of the bearing pad model axis and load force between -16 mm, and +16 mm, every 4 mm. Due to mechanical limitations of the stand, during data processing it has been assumed, that the pressure distribution recorded at the same load value and opposite eccentricity values, put together represents pressure distribution along the diameter of the model at the given load and eccentricity.

Additionally, for every pressure distribution, location of the centre of gravity of the area bounded by the pressure distribution plot was calculated, using the trapezoidal rule.

The pressure distribution measurement results were also compared with the isothermal theoretical model of the pressure distribution in a rigid interspace, described by the Reynolds’ equation [L. 3, 4]. To calculate the theoretical pressure distribution, finite differences method, described elsewhere [L. 5, 6] was applied.
Fig. 4. Essential scheme of the research stand: 1 – examined bearing pad model; 2 – movable measurement slab; 3 – fixed hydrostatic bearing slab; 4 – ball, transferring load. Phantom line represents the range of possible positions of the model during the experiment

Rys. 4. Schemat stanowiska badawczego: 1 – badany segment łożyska (model), 2 – ruchoma płyta pomiarowa; 3 – hydrostatyczna płyta nośna; 4 – kulka przekazująca obciążenie. Linią dwupunktową oznaczono zakres położenia modelu segmentu podczas badań

Table 1. Selected results of measurement of the distance of the bearing pad model corners from the measurement slab face

<table>
<thead>
<tr>
<th>Load ( e ) (mm)</th>
<th>40 kN</th>
<th>60 kN</th>
<th>80 kN</th>
<th>100 kN</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10 µm</td>
<td>-10 µm</td>
<td>-10 µm</td>
<td>-10 µm</td>
<td>-10 µm</td>
</tr>
<tr>
<td>250 µm</td>
<td>260 µm</td>
<td>160 µm</td>
<td>140 µm</td>
<td>140 µm</td>
</tr>
<tr>
<td>20 µm</td>
<td>20 µm</td>
<td>20 µm</td>
<td>20 µm</td>
<td>20 µm</td>
</tr>
<tr>
<td>180 µm</td>
<td>180 µm</td>
<td>140 µm</td>
<td>140 µm</td>
<td>140 µm</td>
</tr>
<tr>
<td>0 µm</td>
<td>100 µm</td>
<td>90 µm</td>
<td>80 µm</td>
<td>80 µm</td>
</tr>
<tr>
<td>90 µm</td>
<td>90 µm</td>
<td>80 µm</td>
<td>80 µm</td>
<td>80 µm</td>
</tr>
<tr>
<td>160 µm</td>
<td>160 µm</td>
<td>120 µm</td>
<td>130 µm</td>
<td>130 µm</td>
</tr>
<tr>
<td>30 µm</td>
<td>30 µm</td>
<td>20 µm</td>
<td>20 µm</td>
<td>20 µm</td>
</tr>
<tr>
<td>190 µm</td>
<td>190 µm</td>
<td>170 µm</td>
<td>160 µm</td>
<td>160 µm</td>
</tr>
<tr>
<td>200 µm</td>
<td>200 µm</td>
<td>140 µm</td>
<td>140 µm</td>
<td>140 µm</td>
</tr>
<tr>
<td>-20 µm</td>
<td>-20 µm</td>
<td>-20 µm</td>
<td>-20 µm</td>
<td>-20 µm</td>
</tr>
<tr>
<td>-30 µm</td>
<td>-30 µm</td>
<td>-30 µm</td>
<td>-30 µm</td>
<td>-30 µm</td>
</tr>
</tbody>
</table>
EXPERIMENT RESULTS

Table 1 presents the results of measurement of the distance of the bearing pad model corners from the measurement slab face (dial gauge readouts). These, indirectly, inform on the oil film thickness, as well as on the mating faces inclination.

![Diagrams]

Fig. 5. Selected plots of oil film pressure distribution along the diameter of the model bearing pad; continuous line – measured pressure distribution; dash line – theoretical pressure distribution derived from isothermal model. $G$ – load; $e$ – load eccentricity; $P_0$ – mean oil groove pressure.

It is important to mention, that during measurements at extreme eccentricity values ($\pm 16$ mm), translation of the measurement slab was significantly more difficult, which may indicate local loss of fluid friction. These cases are the ones shaded in Table 1. Fig. 5 presents selected plots.
of oil film pressure distribution along the diameter of the model bearing pad. The calculated abscissa coordinates of the centres of gravity, compared with the eccentricity setting are provided in the Table 2.

Table 2. Comparison between the set load eccentricity and oil film total pressure application point
Tabela 2. Porównanie ekscentryczności przyłożonego obciążenia z położeniem środka naporu filmu olejowego

<table>
<thead>
<tr>
<th>Load $G$</th>
<th>load eccentricity $e$</th>
<th>0 mm</th>
<th>4 mm</th>
<th>8 mm</th>
<th>12 mm</th>
<th>16 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>10 kN</td>
<td>0.046</td>
<td>3.935</td>
<td>7.706</td>
<td>11.339</td>
<td>14.823</td>
<td></td>
</tr>
<tr>
<td>20 kN</td>
<td>-0.102</td>
<td>3.779</td>
<td>7.555</td>
<td>11.153</td>
<td>14.617</td>
<td></td>
</tr>
<tr>
<td>30 kN</td>
<td>0.045</td>
<td>3.909</td>
<td>7.686</td>
<td>11.24</td>
<td>14.626</td>
<td></td>
</tr>
<tr>
<td>40 kN</td>
<td>-0.281</td>
<td>3.604</td>
<td>7.574</td>
<td>11.36</td>
<td>14.774</td>
<td></td>
</tr>
</tbody>
</table>

RESULTS DISCUSSION

In the middle part of the plots, constant value of the pressure corresponds with the oil groove pressure. Offsets, visible in the very midpoint of some plots are results of inaccuracy in applying the load $G$ before measurements – eccentricity change required reducing the load to zero. Despite these irrelevant faults, following conclusions can be drawn:

1. As the eccentricity of the load and the bearing pad axis grows, pressure distribution becomes more asymmetric. The pressure decreases with the growing distance from the oil groove faster on the side, where the interspace widens. This observation is consistent with the conclusions drawn from the previously presented quantitative model.

2. Although for small loads (40 kN), the shape of the plot hardly varies from the theoretical pressure distribution described by a model for a footstep bearing [L. 7], it is clear that for greater loads the difference is more significant. In the proximity of the oil groove, the pressure decreases slower, than it should according to the theoretical model, later the decrease rate is faster than the mathematical model describes. Most probably it is caused by deflection of either the bearing pad model, or the measurement slab under the acting pressure. Where the pressure is high, that is near the oil groove, greater deflection
causes interspace thickness increase and less bounded oil flow. This explains slower pressure decrease in that area. As the distance from the oil groove increases, the deflection decreases, as well as the oil film thickness. The oil, flowing through a tapering interspace experiences increasing resistance, which consumes its energy.

3. Location of the numerically calculated centre of gravity of each plot insignificantly deviates from the set eccentricity of the load. This indicates the situation, when support resultant and total pressure force act along a common line (Fig. 2a and 2b). For eccentricity set to 16 mm the centre of gravity is about 2 mm closer to the centre of the bearing pad, than the line of direction of the load \( G \). This represents the situation presented in Fig. 2c, where the support resultant force, and total pressure force create a moment, countered by mechanical reaction force on the edge of the bearing pad.

**CONCLUSIONS**

The experiments carried out confirm the quantitative model of loss of oil film continuity on the active face of an axial bearing pad. This refers both to the pressure distribution and the system of forces acting on the bearing pad at hydrostatic stage of performance. Simplification of the bearing pad, by modelling it with a ring-shaped face allowed clear observation of results.

Certain inconsistency has been observed between the measurement results and the mathematical description of pressure distribution in the bearing interspace. Increasing deviations at larger loads suggest, that one of the factors, neglected in mathematical model is elastic deflection of parts. Most likely, the experiment was mainly affected by relatively small rigidity of the measurement slab, loaded not only with the researched oil film from above, but also the supporting hydrostatic film from below. This supposition is confirmed by negative recorded values of distance of the model corners from the slab surface.

**REFERENCES**

Streszczenie

Jednym ze stosowanych rozwiązań łożyskowania wzdłużnego ciężkich maszyn jest łożyskowanie w hybrydowym ślizgowym łożysku wzdłużnym, złożonym z segmentów wspartych na elementach sprężystych. W trakcie eksploatacji takiego układu zaobserwowano serię awarii, polegających na niezamierzonym metalicznym kontaktie powierzchni ślizgowych. Awarie były spowodowane niewłaściwymi parametrami sprężyn wspierających segmenty i wystąpieniem niekorzystnego układu sił działających na segmenty łożyska.

W pracy zaprezentowano wyniki badań standowiskowych rozkładu ciśnienia w szczelinie smarnej modelu hydrostatycznego łożyska wzdłużnego. Badania służyły weryfikacji, sformułowanego wcześniej, modelu jakościowego opisującego warunki wystąpienia opisanej awarii. Wyniki badań potwierdzają słuszność zaproponowanego modelu.