Experimental investigation of the flow in a simplified model of water lubricated axial thrust bearing

O Kirschner¹, A Ruprecht² and S Riedelbauch³

Institute of Fluid Mechanics and Hydraulic Machinery, University of Stuttgart, Pfaffenwaldring 10, 70550 Stuttgart, Germany

¹ oliver.kirschner@ihs.uni-stuttgart.de
² albert.ruprecht@ihs.uni-stuttgart.de
³ stefan.riedelbauch@ihs.uni-stuttgart.de

Abstract. In hydropower plants the axial thrust bearing takes up the hydraulic axial thrust of the runner and, in case of vertical shafts, the entire weight of all rotating masses. The use of water lubricated bearings can eliminate the oil leakage risk possibly contaminating the environment. A complex flow is generated by the smaller film thickness due to the lower viscosity of water compared with oil. Measurements on a simplified hydrostatic axial thrust bearing model were accomplished for validating CFD analysis of water lubricated bearings. In this simplified model, fixed pads are implemented and the width of the gap was enlarged to create a higher resolution in space for the measurements. Most parts of the model were manufactured from acrylic glass to get optical access for measurement with PIV. The focus of these measurements is on the flow within the space between two pads. Additional to the PIV-measurement, the pressure on the wall of the rotating disk is captured by pressure transducers. The model bearing measurement results are presented for varied operating conditions.

1. Introduction

In hydropower plants with vertical shaft the weight of the rotors and the axial thrust of the runner is taken up by an axial thrust bearing. Because of the high load and the large machine size the bearings were built as pad type bearings. Depending on the load the tilting of the pads generates an optimal gap for the lubrication. Water lubricated bearings can be used to eliminate the oil leakage risk eventually contaminating the environment [1, 2]. Due to lower viscosity of water compared to oil the lubrication is characterized by a smaller film thickness [3]. A complex flow is generated by the small film thickness, meaning there could be a two phase flow with cavitation and fluid structure interaction due to the movement of the tilting-pads. This movement is in the same order as the width of the gap.

When designing water lubricated axial thrust bearings the knowledge of the flow field is helpful. The investigation of the flow at such conditions can be approached by Computational Fluid Dynamics (CFD) [4]. To assess the reliability of CFD simulation results, one must perform experimental investigation to validate the proposed CFD model. To reduce the complexity of the problem, measurements in a simplified model of a hydrostatic axial thrust bearing are carried out.

Existing investigations of the flow in sliding bearings are often investigations of journal bearings [3, 5-7]. Experimental investigations including measurements of the velocity in the gap are mostly done with nonhomologous model bearings. Those model bearings typically have larger gap sizes to...
facilitate access for velocity measurement equipment. For example, LDV measurements are done in the model of a journal bearing with enlarged gap size [6, 7].

2. Experimental Setup

In this simplified axial thrust bearing, planar, fixed pads are used. During each measurement series, the gap width is constant and exhibits a higher film thickness compared with real size conditions (figure 1). The chosen pressure level ensures a flow without cavitation phenomena. The rotating parts of the bearing were driven by a DC-motor. The rotational speed is adjusted by the power supply of the motor. A constant width of the gap during the experiments is realized by pre-stressed ball-bearings on the shaft of the test rig. The parts of the axial thrust bearing such as the rotating disc, the pads and the housing were manufactured from acrylic glass in order to ensure the accessibility for laser-optical measurements of the velocities.

In figure 2 the scheme of the model-bearing is shown. The housing of the model bearing has a square bottom plate and four planar walls on the sides to minimize the optical distortion for the measurement. The minimal distance between the sidewall and the five pads of the bearing is 30 mm. In the presented investigation, the pads of the bearing were stiff and inelastic to insure a constant geometry during the measurements. The pads of the bearing have an outer diameter of 190 mm and an inner diameter of 70 mm; between the pads is a space with a width of 20 mm and a depth of 19 mm. The space between two pads is marked with a black box in the picture in figure 3 (bottom). A chamfer of 1 mm is built on the radial edge of the pad in the direction against the rotational direction of the rotating disk. Each pad is equipped with a lubricant supply hole with a diameter of 3 mm. Every supply hole is connected to two grooves in radial and two grooves in circumferential direction. The grooves have a depth of 3 mm and a width of 3 mm. In each pad the length of three grooves is 20 mm and the fourth groove in direction to the chamfer of the pad has a length of 19 mm. The lubricant supply hole and the grooves of one pad are depicted in the picture on bottom left in figure 3. The grooves are also visible in figure 2.

The gap between the bearing pad and the rotating disc has a nominal distance of 1 mm in the current model. The rotating disc has an outer diameter of 190 mm like the pads. At the diameter of 70 mm (inner diameter of the pads) the disk has a backward step of 2.2 mm. The rotating disc also
includes, at this inner diameter region, eight equally spaced, axial holes with a diameter of 10 mm to allow for the outflow of lubricant.

![Figure 3. Pictures from details of the bearing-model.](image)

![Figure 4. Left: location of the transducers; right: picture of the pressure transducers.](image)

The measurement of pressure fluctuations is obtained using five pressure transducers of the type 2MI from the manufacturer Keller. These pressure transducers are mounted in a hole on the top of the rotating disc and have a diameter of 6 mm and a height of 2 mm. The accuracy of the measured pressure signal is 1% related to the measured pressure. The connection to the gap is realized by a pressure tap with a diameter of 1 mm on the bottom side of the rotating disc. The five pressure transducers are located at the different radial positions, each of them with radial distance of 10 mm. So
the pressure is measured at the rotating disc in radial steps of 10 mm over five complete rotations. In figure 4 on the right four pressure transducers are depicted. It can be seen, that each pressure transducer is shifted 45° to the next transducers. These measured pressure signals presented in this report are located in the three radial locations detailed in left side of figure 4.

The transmission of the power supply and signals to and from the pressure transducers are realized by slip rings on the rotating shaft. In addition to the pressure measurements, the lubricating water discharge rate is measured with an electromagnetic flow meter, while the rotational speed is sampled by a light barrier.

Measurements of the velocity in the space between two pads with particle image velocimetry (PIV) are accomplished. For the velocity measurement a two-dimensional PIV-system is used. The PIV-system consists of a double-pulsed Nd:YAG-laser with 25 mJ per pulse and a maximum repetition rate of 20 Hz. The laser light is coupled via a light arm to the optics. The photos were taken by a CCD-camera with a resolution in space of 1280 x 1024 pixels and a 12 bit brightness resolution by a repetition rate of 8 Hz. The seeding is created using silver coated hollow glass spheres with an average diameter of 14 µm. An interrogation area with a size of 32 x 32 pixels was used in the measurement. For each time averaged velocity vector-field 1000 vector fields are averaged. The accuracy of the measured velocities is estimated to 2.5% related to the circumferential velocity of the rotating disc at the radius of the PIV-measurement (u = 0.34 m/s). For this estimation the common resolution of the 0.1 Pixel reached by the sub pixel interpolation is assumed. In figure 5 the test rig with PIV measurement equipment and the data acquisition system is shown.

![Figure 5](image.png)

**Figure 5.** Left: location of the transducers; right: picture of the pressure transducers.

### 3. Selected operating points

The variation of the investigated operating points is realized by the lubrication discharge of the water through the lubricant supply holes. Therefore, different flow configurations are investigated being independent of the gap width. The measurements for all presented operating points were done with constant rotational speed of about 50 rpm. Also the thickness of the lubricant film, meaning the width of the gap, is constant at 1mm. This leads to the following Reynolds number (Re), which is set constant in the measurements for all presented operating points:

\[
Re = \frac{U \cdot h_o}{v} \approx 500
\]
The Reynolds number is calculated with the width of the gap ($h_0 = 1 \text{ mm}$), the circumferential velocity ($U$) of the disc at the outer diameter and a rotational speed of $n = 50 \text{ rpm}$. The fluid in the bearing is water at ambient temperature with a kinematic viscosity of $10^{-6} \text{ m}^2/\text{s}$. The selected Reynolds number is similar to the Reynolds number of a prototype bearing with a diameter of 2 m and a width of the gap of 10 µm at a rotational speed of 500 rpm. It has to be mentioned that there is no geometric similitude of the diameter and the width of the gap between prototype and model. This means that the width of the gap in the model bearing is larger than the gap in a real bearing and the diameter of the model-bearing is smaller than the diameter of a real bearing.

4. Accomplishment of the measurement
The measurements were performed in each operating point at steady state condition. Before each measurement of the operating point the rotational speed of the disk and the discharge through all lubricant supply holes were adjusted. The accuracy of the adjusted rotational speed is 3% and the accuracy of the discharge is 0.5% related to the measured values. For the adjusted operating point the data of the operating point and the measured values of the pressure transducers are stored. Synchronously the velocity measurement with PIV is acquired.

The measured values of the pressure transducers are captured over five rotations of the disc. After passing a low pass filter with a filter frequency of 100 Hz the values were stored. For the presentation of the results the phase shift by the different circumferential positions of the transducer were eliminated. This is done by moving the timestamp of the measured signals with the delay-time of the transducer, respectively.

The velocity vector field was measured in the space between two pads. The investigated PIV-measurement-plane is located at a radial distance of 65 mm to the rotational axis of the bearing. This means that the measurement plane is in the radial middle position of the pad. In addition to the measurement of the complete space between two pads detailed PIV-measurements in the areas of the flow into and out of the gap were investigated, too.

5. Results of the Measurement
The variation of the operating points was realized by different discharge values through the lubricant supply holes. The discharge values given here are related to the total water flow through all five lubricant supply holes.

Figure 6 shows the velocities in the space between two pads for four operating points in the plane at the radius of 65 mm. The vector field at the top left shows the velocity without a discharge of water in the lubricant supply holes. The other pictures of figure 6 show the velocity vectors for a gradual increase of the water discharge through all five supply holes. The black line on the top of the velocity vector fields represents the rotating disc with a circumferential velocity of 0.34 m/s at this particular radius. The rotation of the rotating disc corresponds to a movement from right to left in all pictures presented in this paper.

The velocity in the space between the two pads is nearly zero at the operating conditions without a discharge of water through the lubricant supply holes. Only near the rotating disc are higher velocities created due to the friction within the flow. The results at the operating points with discharge through the lubricant supply holes show a cavity flow structure with one main vortex in the space between two pads. Also small vortices in the bottom corners are visible when the discharge through the lubricant supply holes is increased. The velocity of the vortex is higher with higher discharge through the lubricant supply holes. The operating points with discharge through the lubricant supply holes show higher velocity on the left surface of the pad than on the right surface of the pad.
Additional to the measurement of the velocity vector field in the complete space between two pads, detailed measurements at the area of the inlet and outlet of the gap were accomplished. In figure 7 the
flow at the area of the inlet to the gap is shown. In figures 8 and 9 the flow at the area of the outlet of the gap is shown. The red line on the top of the velocity vector fields again represents the rotating disc. Due to the strong reflections of the laser light on the rotating disc it was not possible to measure the velocity near the rotating disc. The velocity of the rotating disc is 0.34 m/s from right to left.

In figure 7 the flow at the area of the inlet of the gap at different operating conditions is depicted. The discharge of water through all lubricant supply holes is given in the respective velocity vector field. In the operating point without discharge to the lubricant supply holes a drag flow by the rotating disc at the inlet of the gap is shown. In the velocity vector fields with discharge through the lubricant supply holes a stagnation point at the chamfer of the inlet of the gap can be seen. The stagnation point is traveling downward with increasing discharge to the lubricant supply holes. Also, there are higher velocities through the gap in the operating point with a higher discharge of water through the lubricant supply holes than in the operating point with the lower discharge.

![Figure 8. Flow in the corner of the space between two pads at the outlet of the gap.](image)

In figure 8 the flow at the area of the outlet of the gap at operating conditions without discharge and with a discharge of 150 l/h through the lubricant supply holes is shown. The velocities in the gap have higher values in the operating point with a discharge through the lubricant supply holes. In both operating points the height of the area with higher velocities is increasing with the distance to the outlet of the gap in the space between the two pads. The increase leads to a small decrease in the velocity level.
In figure 9 the flow at the area of the outlet of the gap at operating conditions with a discharge of 750 l/h through all lubricant supply holes is shown. At the corner to the gap a flow against the rotational direction of the rotating disc is visible. In figure 10 the velocities from the result of a numerical simulation at the same operating condition is depicted. At the corner of the gap the velocity vectors of the simulation result are also oriented against the rotation direction of the rotating disc. This is caused by gap areas with lower pressure than the pressure level in the space between two pads. This is only the case for operating conditions with a high discharge through the lubricant supply holes. For the operating points without or low discharge through the lubricant supply holes there are no velocity vectors oriented against the rotational direction of the rotating disc (see figure 8).

Figure 9. Flow out of the gap by a discharge through the lubricant supply holes of 750 l/h.

Figure 10. Flow out of the gap (CFD-Results [4]).

Figure 11. Measured pressure for different operating points.
In figure 11 the measured pressure on the wall of the rotating disc is drawn against the rotation angle for different operating points. The yellow line represents the pressure at a radius of 45 mm; the black line shows the measured value of the pressure at a radius of 65 mm and the green line displays the pressure at a radius of 85 mm. A pressure peak on the rotating disc can be seen at the stagnation point on the top of the lubricant supply hole. For operating points with higher gap discharges the amplitude of the pressure peak is increasing. There are also areas with lower pressure levels around the stagnation point. The pressure level in those areas shows a decrease (meaning a higher negative value) by increasing the discharge through the lubricant supply holes.

6. Conclusion

For different operating points velocity and pressure measurements were accomplished in a simplified model of a hydrostatic axial thrust bearing. The variation of the operating points in this experimental investigation was realized by different discharges through the lubricant supply holes in the gap of the slide bearing. All other operating conditions were nearly constant in all shown operating points. With the results of these measurements a first database for validating the CFD-analysis of the flow in water lubricated axial thrust bearings is composed.

The results of the PIV-measurements show a cavity flow structure with one main vortex in the space between two pads. Also smaller vortices in the corners at the bottom of the space between two pads are investigated for higher discharge through the lubricant supply holes. The flow in the corner of the space between two pads at the outlet of the gap shows velocity vectors against the rotation direction of the rotating disc for 750 l/h lubricant flow. This flow against the rotation direction of the rotating disc at the corner to the gap is caused by the lower pressure in some areas of the gap than in the space between two pads. The velocity vector fields in the corner of the space between two pads at the inlet of the gap show a stagnation point on the chamfer when the bearing operates with discharge through the lubricant supply holes. This stagnation point travels downward on the chamfer when the discharge is increased.

In addition, a higher pressure variation in the gap can be measured at the wall of the rotating disc when the discharge in the gap is increased. A pressure peak at the stagnation point on top of the lubricant supply hole can be seen for operating points with discharge through the lubricant supply holes. The value of that pressure increases with increasing lubricant discharge. Around this stagnation point an area with lower pressure is located. The pressure is decreasing in this area with increasing lubricant discharge.

Nomenclature

D    outer diameter
h₀    width of the gap
ν    kinematic viscosity
Re    Reynolds number

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