Spatially Sampled Pressure Profile Measurements in Externally Pressurized Gas Journal Bearings

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ABSTRACT

Externally pressurized gas journal bearings (EPGJB) are widely adopted to support high speed rotors. This paper presents novel experimental benchmark test data which enables EPGJB model validation. Axial and circumferential pressure profile measurements are presented for a 40mm diameter annular restrictors EPGJB operating at speeds up to 25 krpm. The corresponding rotor position, air mass flow rate, and load capacity are also measured. The effects of the supply pressure, rotational speed, and load on the measured variables are presented. The measurements are in good agreement with numerical model computations. The cases in which the model deviates from the measured data are discussed in terms of compressible flow theory.

INTRODUCTION

Externally pressurized gas journal bearings (EPGJB) are a widely-adopted technology to support high-speed rotors (1), and precision machine tools (2). The technology is attractive due to its high stiffness and contact-free operation even at zero rotational speed. Such qualities are at the kernel of high accuracy applications like manufacturing spindles and inertial instruments (3,4). The main disadvantages of the technology are: 1) its dependence on a continuous supply of a compressed fluid, 2) its high sensitivity to geometrical accuracy, hence, demanding tight manufacturing tolerances, 3) its susceptibility to dynamic instabilities, and 4) its vulnerability to pollution and contamination.

The load capacity in fluid film bearings is provided by the pressure field within the bearing clearance. Although knowledge of the pressure field as a function of the rotor position is imperative for the understanding of the behavior of rotor-bearing systems, as well as for the validation of bearing models, measurements of pressure profiles inside the gas film of EPGJB are very scarce in the literature.

Background

Most of the experimental effort to measure the flow field inside fluid film journal bearings were done on oil lubricated bearings. Roberts and Hinton (5) measured the radial and axial pressure profile inside a short ($L/D = 1/3$) circumferentially grooved bearing at different Reynolds numbers (40-50000). The bearing had a nominal diameter of 100 mm, and a clearance of 0.55 mm. Six equidistant piezo resistive transducers placed along a helix were used to measure the fluid film pressure. The sensors were connected to the bearing via 0.5 mm pressure taps. Results were in good agreement with theory for low eccentricity cases.

Brito et al. (6) performed similar measurements for a hydrodynamic journal bearing with two axial grooves. The measurements investigated the influence of lubricant supply conditions on the pressure and temperature profiles inside the bearing, as well as the rotor eccentricity. Pressure taps of Ø1 mm were drilled along the mid-plane of the bearing, and connected to high precision Bourdon pressure gauges. The rotor nominal diameter was 100 mm with a diametrical clearance of $171 \pm 4 \mu\text{m}$. Cristea et al. (7) measured the pressure and temperature inside the oil film of a circumferential grooved journal bearing. The pressure was measured using Ø1.5 mm pressure taps, the rotor nominal diameter was 100 mm, the diametrical clearance was $122 \pm 4 \mu\text{m}$.

The literature reports several studies presenting the pressure profile measurement in axial thrust bearings using pressure taps (8–13). However, few publications reported pressure measurements in gas journal bearings.

In 1961, Laub (14) reported the earliest zero rotational speed pressure profile measurement inside an EPGJB. The author used 6 pressure taps in the axial direction and 10 in the circumferential direction. The bearing was described to have 192 nozzles divided on 4 rows. No details on the manufacturing tolerances were given.

In 1962, Lemon (15) presented pressure profile measurements and compared it to analytical solutions. The bearing had pocketed (recessed) supply restrictors, yet the full dimensions of the bearing under investigation were not presented and experimental details were not provided.

The report by Burt in 1969 (16) is the most detailed experimental pressure measurement in EPGJBs that is available in the open literature. The tests considered a rotor supported on two journal bearings and one axial bearing. The journal bearings had two rows of 4 equidistant supply nozzles, with a relatively large diameter ($\approx \text{Ø}1.12 \text{ mm}$ for a 46 mm bearing diameter). The author reported large deviations in the measured bearing clearance relative to the nominal clearance (not quantified in the report). Only the axial pressure profile was measured using 7 equidistant pressure taps along the bearing length and at the same circumferential position. The supply flow rate was also measured to be 25% above the design value. Laminar flow theory was used to compute the axial pressure profile, however, the model underestimated the pressure compared to the measurement. The load capacity was overestimated by roughly 50% at $\epsilon = 0.5$. Better agreement in load capacity was achieved by changing the bearing design to a lower Reynolds number flow regime, suggesting an influence of turbulence on the performance of EPGJBs.

In 1979, Pink and Stout (17) presented pressure profile measurement in an EPGJB with feeding pockets. The results were presented at zero rotational speed and solely for the sake of numerical result validation. The paper is the most recent in the literature concerned by the gas film pressure in EPGJBs, but unfortunately, details of the measurements were not provided.

Motivation and Objectives

Reviewing the literature reveals a gap in experimental flow field benchmark data for gas lubricated EPGJBs. The few available articles are either incomplete, or missing crucial details, hence, making the experimental literature insufficient for a thorough model

development and validation. This paper provides point measurements of the axial and circumferential pressure profiles inside a 40 mm diameter ($L/D = 1$) EPGJB at different supply pressures and static loads. The paper also presents the bearing pressure profile measurement at different rotational speeds and up to 25 krpm. The corresponding mass flow rates are also measured, allowing the deduction of the bearing discharge coefficients. Quasi-static load displacement measurements were executed to investigate the bearing load capacity and stiffness as a function of supply pressure. Deviations between the computations of a hydrodynamic EPGJB model and the measured data are discussed in the framework of compressible flow theory providing insight into the origin of the deviation.

NUMERICAL MODELLING

The steady state isothermal compressible Reynolds equation for EPGJBs is adopted as presented in Lo et al. (18), and discretized and solved using the finite difference method implemented in Guenat and Schiffmann (19). The working fluid (air) is considered an ideal gas.

The supply restrictors are modelled as a source term present in the Reynolds equation that is active only at the nodes corresponding to the positions of the restrictors in the fluid film domain.

An isentropic and adiabatic expansion through the nozzles from stagnation is assumed, with the mass flow rate expressed as follows:

$$\dot{m} = Ap_s \varphi \left(\frac{2\rho_s}{p_s} \right)^{\frac{1}{2}} \quad \text{Eq. (1)}$$

where, depending on whether the flow is choked or not:

$$\varphi = \begin{cases} \left(\frac{\gamma}{2} \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{\gamma-1}} \right)^{\frac{1}{2}} & \text{if } \frac{p}{p_s} \leq \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} \\ \left(\frac{\gamma}{\gamma-1} \left[\left(\frac{p}{p_s} \right)^{\frac{2}{\gamma}} - \left(\frac{p}{p_s} \right)^{\frac{\gamma+1}{\gamma}} \right] \right)^{\frac{1}{2}} & \text{if } \frac{p}{p_s} > \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} \end{cases} \quad \text{Eq. (2)}$$

A discharge coefficient should be defined to model the actual mass flow rate through the restrictor. The sensitivity of the discharge coefficient to the fluid flow condition, and the restrictor's geometry, makes the development of a generic model very challenging. To the knowledge of the authors there are no models covering the specific geometry and flow conditions of the restrictors under investigation. The empirical correlation proposed by Belforte et al. (12) is developed using orifice restrictors, which are different than the restrictors under investigation. However, they are considered the closest possible to the latter. Hence, the correlation is used to estimate the discharge coefficient in the present study.

A grid dependency analysis has been executed starting with 90 nodes as an initial seed. The normalized load $\bar{W}$ (pressure field integral) was used as a control variable with a residual error defined as follows:

$$r_{\varepsilon} = \frac{\bar{W}^n - \bar{W}^m}{\bar{W}^m} \quad \text{Eq. (3)}$$

where n is a denser grid than m . In total 5 grids have been compared, the selected grid has 9360 nodes (65 axial and 144 circumferential) and yields a residual error r_{ε} in the order of 10^{-3} (0.1%) which is considered satisfactory.

EXPERIMENTAL SETUP

Test Rig Description

A test rig has been developed including two model EPJGBs constructed to measure the pressure at a series of points within the gas film. The apparatus is comprised of a rotor supported on two test journal bearings, which is driven by an 18kW electric motor through a quill-shaft - Figure 1. Each test bearing is equipped with two capacitive proximity probes (x-y) to measure the time resolved rotor position. The proximity probes have an uncertainty of $\pm 1.3\%$ and a bandwidth of 14 kHz.

The fluid film pressure is measured through taps connected to pressure scanners. The connection between the tap and the pressure scanner is made through internal passages inside the bearing followed by a flexible tube - Figure 2. The scanner measures the time averaged gauge pressure up to 0.7 MPa with an accuracy of $\pm 0.05\%$ of the full scale. Each scanner contains 16 thermally compensated piezoresistive pressure transducers. The sampling rate of the scanner is 5 Hz, hence only steady state measurements are considered. The supply

pressure is controlled with a pressure regulator, and measured using a different pressure scanner with a range up to 1.7 MPa.

The supply air mass flow rate of each bearing is measured individually using two Coriolis flowmeters. The measurement accuracy is between 0.67% (at 3 kg/hr) and 2.67% (at 0.75 kg/hr) depending on the flowrate. The pressure drop across the flowmeter ranges between 53 Pa (at 0.75 kg/hr) and 465 Pa (at 3 kg/hr) and is therefore negligible compared to the supply pressure.

The output signal of the proximity probes is communicated to the data acquisition controller through a high-speed acquisition card with simultaneously sampled channels. The pressure scanner and the mass flowmeter output signals were acquired using a data acquisition system to a personal computer.

The test rotor has a nominal diameter of 40 mm, is made of 100Cr6 Steel, weighs 1.3kg and has a rotational and transverse inertia of $3.912 \times 10^{-4} \text{ kgm}^2$ and $8 \times 10^{-3} \text{ kgm}^2$ respectively. The layout showing the relative distances between the motor and the two test bearings is shown in Figure 1, a photograph of the test rig is shown in Figure 2.

Test Bearing Description

The test bearings are made out of DIN EN 1982 Bronze. Each bearing has two feeding rows of 18 restrictor nozzles (100 μ m diameter and 1.1mm long). The two rows are located at $\bar{z} = \pm 0.5$ where $\bar{z} = \frac{z}{R}$. The two journal bearings are identical in terms of nominal dimensions and air supply, however one of them was designed to measure the axial pressure profile (bearing A), and the other to measure the circumferential pressure profile (bearing B). The supply and measurements have two physically separated internal passages inside the bearing structure. The tested bearings have a nominal radial assembly clearance of 30 μ m.

The pressure measurements are made through $\varnothing 0.3$ mm pressure taps at various locations inside the test bearings. A symmetric axial pressure profile around the bearing mid plane is assumed. Hence, bearing A is designed to have 6 measurement points capturing half of the axial pressure profile at $\bar{z} = -0.85, -0.75, -0.65, -0.5, -0.25$, and 0. The taps are located on 3 equidistant axial planes between the supply nozzles - Figure 3. Bearing B has 25 pressure taps along the bearing circumference (packaging maximum) yielding the circumferential pressure profile inside the bearing - Figure 4.

Dimensional Metrology of the Rotor and the Test Bearings

The effect of manufacturing tolerances on the performance of EPGJBs was the focus of several investigations. Stout and Rowe (20) studied the effect of manufacturing errors on orifice and slot entry bearings, and concluded that a double entry bearing is less sensitive to the effects of manufacturing errors than single entry bearings. Pink and Stout (21) extended the work by introducing a tolerancing procedure for compensated EPGJBs, and showed that

minor dimensional variations in the restrictor or the clearance can produce significant discrepancies in load capacity. Following these efforts, Stout (22) dedicated an article discussing in details possible manufacturing errors in EPGJBs and their corresponding effects on performance. He included the effects of the bearing roundness, form, and alignment, along with the restrictor geometry, and bearing assembly clearance. The author highlighted major deviations in load capacity due minor deviations in the bearing geometry. Stout and Pink (23) showed that knowledge of the geometrical deviations in orifice compensated EPGJBs can be used to estimate the deviation in performance of such bearings.

Hence, special attention is given to ensure the accurate geometrical control of the rotor-bearing system before embarking in the test program. The test rotors and bearings are measured using a commercial optoelectronic reader that converts the linear movement of the measurement probe into an electronic signal. The rotor has a diameter of $39.369 \text{ mm} \pm 1.16 \text{ }\mu\text{m}$. Bearing A has a diameter of $39.442 \text{ mm} \pm 1.48 \text{ }\mu\text{m}$. Bearing B has a diameter of $39.437 \text{ mm} \pm 1.87 \text{ }\mu\text{m}$. As a consequence the measurements yield a clearance of $36.5 \text{ }\mu\text{m} \pm 1.88 \text{ }\mu\text{m}$ in bearing A, and $33.86 \text{ }\mu\text{m} \pm 2.2 \text{ }\mu\text{m}$ in bearing B.

For the bearing A, the supply nozzles and the measurement taps are laser drilled from the outside of the bearing using 4 mm counterbores for access, which are sealed after the manufacturing - Figure 5a. The supply nozzles and the measurement taps were inspected and measured with a digital microscope – Figure 5b. The supply nozzles measurement indicates a mean diameter of $111 \text{ }\mu\text{m}$, with deviations within 6.6%. Hence, indicating an overall oversizing of the supply nozzle by 11%. The measurement taps are within 3% of the design

value (Ø0.3mm). For bearing B, the supply nozzles and the measurement taps were drilled from the inner diameter of the bearing using a special EDM (Electric Discharge Machining) electrode setup. The drilling accuracy lies within 5%.

TEST PROGRAM

Mass Flow Measurement and Discharge Coefficient Deduction

The mass flow rate was measured for each of the two bearings simultaneously at different supply pressures. Figure 6a shows that the mass flow increases linearly with the supply pressure, which is a clear indication of choked restrictors. The difference between the two bearings is due to the larger supply nozzle diameter and clearance of bearing A as indicated above. It was also observed that the measured mass flow rate was not affected whether or not the rotor was assembled inside the bearings. This can be explained by considering the effective (smallest) throttling discharge area, which is governed by the nozzle cross-sectional area $A = \frac{\pi d^2}{4}$, and not the curtain area $A = \pi dC$.

The measured mass flow rate is then compared to theoretical mass flow rates based on isentropic expansion through the restrictor. Hence, deducing the discharge coefficient as follows:

$$C_d = \frac{\dot{m}_{actual}}{\dot{m}_{isentropic}} \quad \text{Eq. (4)}$$

Although the flow is choked, the discharge coefficient evolves inversely proportional to the supply pressure - Figure 6b. This is due to the increase in the bearing pressure drop with the increase in the supply pressure - given that the bearing back pressure is atmospheric and independent of the supply pressure. The estimated discharge coefficients in the model of Belforte et al. (12) overestimates the measured experimental values by up to 15% – Figure 6b. This is attributed to the shorter restrictors ($l/d = 0.75$ to 1.5), and larger diameters (0.2 , 0.3 , and 0.4mm) used in the development of the empirical model. The manufactured restrictors in bearings A and B are long, with a length equal to 11 diameters. It is also worth mentioning that the manufactured restrictors in bearing A are slightly convergent due to the nature of the laser drilling technique. Moreover, and contrary to the model of Belforte et al. (12), the restrictors under investigation are not of the inherent type since the film thickness and the restrictor diameter are of the same order of magnitude. The restrictor's Reynolds number ($Re = \frac{md}{A\mu}$) ranged between 7777 and 4626 depending on the supply pressure level.

The uncertainty in the deduced discharge coefficient due to the uncertainty in the nozzle diameters is 12% and 9.3% for bearings A and B respectively. Hence, the measured mass flow rates are used as input to the bearing model. Making the assumption that all the nozzles have the same geometries in a first approximation allows the deduction of the mass flow rate for a given nozzle.

Effect of Supply Pressure on Circumferential and Axial Pressure Profiles

The axial and circumferential pressure profiles are measured at zero rotational speed for different supply pressures. Figure 7b, d, and f compares the measured and the model computed axial profiles inside bearing A supporting only the rotor (no external load and imposing the measured mass flow rate from Figure 6a). The maximum pressure is at the length corresponding to the location of the supply nozzles row ($\bar{z} = -0.5$). The pressure drop between $-0.5 < \bar{z} < 0$ is attributed to dispersion and to non-axial flow effects (24), which are the result of manufacturing variations in the supply nozzles that causes the air to flow around the bearing from regions of high pressure to low pressure. It should be stressed that the measurement points are located between the feeding restrictors, and not at the same circumferential angle—Figure 3. It is observed that doubling the non-dimensional supply pressure from 5 to 10, increases the pressure inside the bearing only by an average of 22%, which indicates that the restrictor flow resistance is high compared to the fluid film resistance.

Numerically, the pressure at the position of measurement taps is obtained from the model at the corresponding positions according to Figure 3 and Figure 4 and by imposing the measured mass-flow and supply pressure as operating conditions. The influence of the variance in the bearing geometry as well as deviations in the supply nozzles diameters are used to obtain a band of prediction. Mean geometrical values are used for a nominal simulation. It is found that the pressure field is strongly influenced by slight deviations in the bearing clearance and in the supply nozzle diameter. The bounds of the manufacturing uncertainty at the point $\bar{z} = 0$ yield pressure deviations between -7.7% and 8.9% relative to nominal. The model is in good agreement with experimental measurements between $-0.5 <$

$\bar{z} < -1.0$, highlighting the effectiveness of the measurements for model validation.

Overestimation in pressure is observed between the supply nozzle and the middle of the bearing. Possible explanatory hypotheses are:

1. The effect of dispersion and circumferential flow are significant in the test bearing.

Dudgeon and Lowe (25) studied the two effects on the performance of EPGJBs and attributed large model overestimation to these factors.

2. Certain supply nozzles (potentially close to measurement taps) are beyond the standard deviation of the measured nozzles, hence, creating circumferential flows.

In support of the circumferential flow hypothesis, a simple numerical optimization is performed using the bearing model, where the individual diameter of each nozzle is multiplied by a factor to perturb the mass flow rate of each nozzle. In a first case, the 36 nozzles adopted factors varying between 0.4 and 2 (wide range). In a second case, the factors varied between 0.94 and 1.06 (tight range). For each case, the optimizer finds the set of factors minimizing the absolute error between the computed and the measured axial pressure profiles. Both cases, seems to capture some pressure drop in the center of the bearing.

However, the first case is in better agreement with the experiment as shown in Figure 8.

Hence, implying that the deviation in the nozzle diameters is creating a circumferential flow, which in turn is the root cause of the pressure depression in the center of the bearing.

Figure 7a, c, and e presents the same exercise for the circumferential pressure profile in bearing B – without external load. Both the experiment and the simulation show a clear pressure gradient between the supply restrictors, with the pressure building up near the nozzle locations and dropping slightly in between them. Qualitatively, the model is considered in good agreement with the experiment. A slight over estimation of the pressure, however, persists. Yet, the error is lower in comparison to the axial profile validation. This can be attributed to the specific location of the measurement at $\bar{z} = -0.5$ which is away from the pressure depression region in the center of the bearing. It should be noted that the circumferential and axial profiles are measured inside two different bearings (i.e. different clearances and supply nozzle diameters).

In order to quantify the recurring error between the simulation and the experiment, the pressure values of the model at the measurement points are compared against the corresponding experimental values. The error is defined as follows:

$$Error = \frac{\bar{p}_{model} - \bar{p}_{exp}}{\bar{p}_{exp}} \quad \text{Eq. (5)}$$

For the design case of 0.7 MPa, the error along the bearing circumference at $\bar{z} = -0.5$ had a mean of 5.52%, a maximum and a minimum of 10.11% and 2.68% respectively. The error is reaching a maximum at 330° and a minimum at approximately 120°, phasing roughly 180°. Such observation suggests a slight eccentricity towards 120°, which is also observed during the measurements of the rotor position.

Effect of Rotational Speed on Circumferential and Axial Pressure Profiles

Pressure profiles at the design supply pressure of 0.7 MPa were measured at different rotational speeds up to 25 krpm. It should be stressed that since the axial and circumferential profiles are measured inside two different bearings with slightly different clearances, the bearing compressibility numbers are different for the same rotational speed (maximum compressibility 0.86 and 1.0 for bearing A and B respectively). Increasing the rotational speed, increases the pressure inside the bearings, however, the effect of rotor speed is limited for this range of compressibility numbers. Qualitatively, the shape of the pressure profile was unaffected by the rotational speed.

The limited pressure buildup is attributed to the centrifugal rotor growth, which is calculated according to Dubbel (26):

$$\Delta h = \frac{\rho_{Rot} \Omega^2 R}{E} [2c_1 r^2 + (c_1 - c_2) R^2] \quad \text{Eq. (6)}$$

where

$$c_1 = \frac{3 + \nu}{8} \quad \text{Eq. (7)}$$

$$c_2 = \frac{1 + 3\nu}{8} \quad \text{Eq. (8)}$$

where Δh is the reduction in radial clearance, ρ_{Rot} is the rotor material density, Ω is the rotor speed, R and r are the rotor's outer and inner radii respectively, E is the Young's modulus, and ν is the Poisson ratio. For the Steel rotor used in the tests, the centrifugal growth at 25 krpm results in a 1.3 μm decrease in radial clearance. In order to test the hypotheses of the pressure buildup due to the rotor centrifugal growth, two cases are simulated, one with nominal clearance, and second reduced by 1.3 μm - Figure 9. The results of the simulations plotted in Figure 9 are in good agreement with the measurement, hence supporting the initial hypotheses.

The mass flow rate is measured at the different test speeds, showing no evident effects at the nominal supply pressure of 0.7 MPa. This is because the throttling area is unaffected by the centrifugal growth of the rotor (1.3 μm in radius), also the pressure buildup downstream of the nozzles maintains the choked condition.

Effect of Load on Displacement

Fleming et al. (27) confirmed the assumptions of Powell (24) and showed that the stiffness of EPGJBs is not significantly affected by the rotational speed. Cunningham and Gunter (28) suggested that measured stiffness values at zero rotational speed of such bearings is sufficient

to estimate the rigid body modes of the rotor. Hence, it was found essential to quantify the stiffness of the bearings under investigation.

A quasi-static load-displacement auxiliary setup was added to the test rig in order to measure the resulting displacement for a given load at zero rotational speed. The setup consists of a loading mechanism, which is comprised of a load cell attached to the test rotor side and connected to a torque arm via wire. A vertical load is gradually exerted to the rotor using the torque arm, hence loading the two test bearings simultaneously. The corresponding displacement is measured through the proximity probes attached to the bearing bushings - details on the load-displacement rig can be found in (29). The accuracy of the proximity probe is $\pm 1.3\%$, the accuracy of the load cell is $\pm 1\%$. The uncertainty propagation for the eccentricity ratio and stiffness deduction is estimated to be $\pm 6.57\%$ and $\pm 1.64\%$ respectively (including the uncertainty in the measured bearing clearance).

Figure 10a shows the measured load versus eccentricity ratio at different supply pressures in bearing B. The load capacity increases both with eccentricity and with the supply pressure. It is observed that the effect of supply pressure on load capacity is more predominant at high eccentricities ($\epsilon > 0.5$).

At very large eccentricities ($\epsilon > 0.8$) and for low supply pressures (0.3 and 0.4 MPa), an area of zero stiffness (instead of the expected single inflection point) is observed. This phenomenon can be explained by considering a single nozzle under load, which will build up

pressure as the eccentricity increases. At a certain threshold eccentricity, the pressure will drop drastically due to blockage (reduction in curtain area). Considering the rotor-bearing system under investigation, and given their circular nature, at a given eccentricity, the supply nozzles are not loaded equally. Hence, some nozzles are heavily blocked and stop contributing to load capacity, while others are at peak pressure. Such behavior would propagate along the loaded bearing circumference as the eccentricity continues to increase. Thus, the net reaction force will be nearly constant at this range of eccentricities. Finally, the aggressive change in slope near $\epsilon=1.0$ is due to the mechanical contact of the rotor with the bearing surface.

Figure 10b presents a deduced normalized stiffness ($\frac{d\bar{W}}{d\epsilon}$) versus eccentricity ratio for the different supply pressures. The observed stiffness is highly nonlinear as a function of eccentricity. Thus, implying the necessity of deducing the stiffness at a given eccentricity ratio, within the design process of EPGJBs.

In order to assess the bearing model, a comparison of the measured and computed load capacities is presented. For a given eccentricity ratio, the bearing load is evaluated by integrating the pressure over the fluid film domain. The pressure field is simulated with two different approaches. The first is assuming a constant discharge coefficient for the nozzles (independent of the local film thickness), and accounting only for the reduction in the minimum discharge area (curtain area vs. restrictor area), in addition to the local thermodynamic conditions (heat capacity ratio, pressure and density at supply conditions). The initial unloaded case uses the measured mass flow rate as a starting point - Figure 6a.

The second uses the model of Belforte et al. (12) for estimating the discharge coefficient as a function of the local film thickness and Reynolds number.

Figure 11 compares the model predictions and the experimental measurement at 0.7 MPa supply pressure and at zero rotational speed. The two approaches similarly overestimate the bearing load capacity up to half the clearance, with a maximum of 50% overestimation (at $\epsilon = 0.25$). Up to that eccentricity, both the measurement and the simulations adopt a relatively linear trend, beyond which the two approaches behave differently. The constant C_d approach evolves into a non-linear trend similar to the measurement, while gradually decreasing the overestimation as a function of eccentricity. The approach using the Belforte et al. (12) model is maintaining its linear trend and intersecting with the measured values at $\epsilon = 0.655$. Both simulation approaches show a slight change in slope at $\epsilon = 0.26$, signaling the transition onset of the throttling area, making the mass flow rate dependent on the rotor position. Similar trends are observed at different supply pressures. Higher levels of overestimations were reported at large eccentricities by Burt (16), and Dudgeon and Lowe (25) -100% and 130% overestimation respectively.

Effect of Static Load on Pressure Profiles and Mass Flow Rate

Given the observed disparity between the computed and the measured load capacity, it was deemed necessary to further investigate the root cause of this discrepancy. Therefore, the circumferential pressure profile is measured at zero rotational speed for a supply pressure of 0.7 MPa at four different loading levels exerted along the 180° direction ($\bar{W} =$

0, 0.314, 0.628, 0.942) and represented in Figure 12. The pressure profile at no load is relatively axisymmetric around the bearing center. As the load increases, the pressure is observed to increase on the loaded side of the bearing and decrease on the opposite side.

The same modelling approaches described in the previous section are used for comparison with the measurements. The models are capable of describing the qualitative evolution of the pressure profile with a high level of details. However, they are slightly overestimating the pressure values at the loaded side of the bearing. Such observation corroborates with the initial observation of the load capacity computation. At an eccentricity ratio of 0.618, the two pressure profiles (constant C_d and C_d by (12)) are practically overlapping at the loaded side, thus yielding similar computed load capacity as suggested in Figure 11. At high eccentricity ($\epsilon = 0.768$), the results of the two modelling approaches deviate significantly, with the constant C_d approach in good agreement with the experiment, again verifying the conclusions from the load capacity computation.

It is hypothesized that the pressure buildup inside the loaded bearing is highly sensitive to the mass flow rate through the individual supply nozzles, which are influenced by the rotor position. At a given choked pressure ratio across the nozzle, the two variables governing the mass flow rate through the nozzle are the throttling area (cross-section or curtain), as well as the discharge coefficient (12).

Figure 13 shows the reduction in the measured total mass flow rate, as well as the two modelling approaches, as a function of eccentricity. Comparing the measurement to the constant C_d modelling approach - which is also identifying the area reduction as a function of eccentricity, it is observed that the mass flow rate is not reduced as much as the throttling area, evidencing that the discharge coefficient is variable at different eccentricities (loads). Hence, in order to compensate for the relatively higher mass flow rate, it is then deduced that the discharge coefficient is inversely proportional to the gas film thickness once the curtain area is engaged – within the measured range, as at very large eccentricities ($\epsilon \rightarrow 1.0$), the discharge coefficient will tend to diminish. The discrepancy between the total nozzle area and the measured mass flow rate can be also explained by the potential dissimilarity among the nozzles. Hence, shifting the transition of the nozzle area towards higher eccentricity ratios.

The second approach (Discharge coefficient model of Belforte et al.(12)) shows a clear mismatch in mass flow rate even at concentric position, hence, confirms the inadequacy of the model in the estimation of the discharge coefficient of the bearings under investigation. It is suggested that this discrepancy is a consequence of the larger L/D ratio of the restrictors in compared to the ones investigated by Belforte et al., thus underlining the need for new correlations for predicting discharge coefficients of more capillary-like restrictors. Further, the evolution of the effective throttling area in Figure 13 and the evolution of the measured discharge coefficient as a function of the supply pressure suggest that the discharge coefficient for more capillary-like restrictors varies less with eccentricity and more with the supply pressure compared to classical restrictors according to Belforte et al.

CONCLUSIONS

The paper presents a comprehensive set of benchmark measurement data for externally pressurized journal bearings. A novel test rig dedicated mainly for pressure measurement within the micro gas film of the bearing is presented. The following conclusions are made:

- a. The axial and circumferential pressure measurement successfully validated the bearing model by Lo et al. (18).
- b. Discrepancies in the supply nozzle shapes create circumferential flows at zero rotational speed inside the bearing, which are the main cause of the pressure drop in the bearing center between the two rows of nozzles.
- c. The simulated bounds of manufacturing errors result in pressure deviations between - 7.7% and 8.9% relative to the nominal (based on mean geometrical values).
- d. The empirical model by Belforte et al. (12), which has been validated for small L/D restrictors is not adequate for the estimation of the correct mass flow rates through the nozzles. This is likely due to the different range of the nozzle geometries used in the development of the empirical correlation. The restrictors implemented the bearings of this investigation are clearly outside of the range of available discharge correlations in literature.

- e. Knowledge of the exact mass flow rate of each individual nozzle as a function of the film thickness is required for the accurate estimation of the load capacity. This is highlighted in the presented pressure profiles at different loads, which were strongly influenced by the discharge coefficient of each individual nozzle. As a consequence of the lack of this knowledge, the simulated loaded pressure profiles, as well as the load capacity, are overestimated by up to 50%.
- f. The measured load capacity, and the deduced stiffness, show a non-linear stiffening behavior, in particular above 20% eccentricity. Hence, highlighting the importance of stiffness estimation at the specific design eccentricity.
- g. The load capacity can be overestimated by the model by up to 50%, this shall be taken into consideration while defining the design's factor of safety.

Finally, it should be stressed that further research should be directed towards developing more universal discharge coefficient models – especially for micro-holes ($\varnothing < 0.1$ mm), with large l/d restrictors. It is believed that the presented pressure measurements will serve as a comprehensive validation platform for EPGJB model development.

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NOMENCLATURE

l	supply nozzle length	[m]
d	supply nozzle diameter	[m]
ϵ	eccentricity ratio	[-]
Q	mass flow rate	[kg/s]
$\bar{W}$	Load capacity	[-]
e	eccentricity	[m]
C	Bearing clearance	[m]
P_a, P_{amb}	Ambient pressure	[Pa]
L	Bearing length	[m]
D	Bearing diameter	[m]
P_{abs}	Absolute pressure	[Pa]
C_d	Discharge coefficient	[-]

Z	Axial bearing position	[m]
Δh	Reduction in bearing clearance	[m]
ρ_{Rot}	Rotor material density	[kg/m ³]
Ω	Shaft rotational speed	[1/s]
R	Shaft outer radius	[m]
r	Shaft inner radius	[m]
E	Young's Modulus	[MPa]
$\bar{p}_{model}$	Estimated pressure	[Pa]
$\bar{p}_{exp}$	Measured pressure	[Pa]
$\dot{m}_{actual}$	Measured mass flow rate	[kg/s]
$\dot{m}_{isentropic}$	Isentropic mass flow rate	[kg/s]
A	Throttling area	[m ²]
θ°	Circumferential angle	[°]
$\bar{P}$	Normalized Pressure $\frac{P_{abs}}{P_{amb}}$	[-]
$\dot{m}$	Mass flow rate	[kg/s]
p_s	Supply Pressure	[Pa]
ρ_s	Supply Density	[kg/ m ³]
γ	Heat Capacity Ratio	[-]

μ	Dynamic Viscosity	[Pa.s]
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Figure 2 – Photograph of the test rig highlighting the 31 pressure measurement points, the driving spindle (left), the test rotor (right), and the quill shaft coupling (center)

Figure 3 – Pressure tap locations in bearing A for axial pressure profile measurement

Figure 4 – Pressure tap locations in bearing B for circumferential pressure profile measurement

(b)

Figure 5 – (a) Ø0.1mm bearing supply nozzle geometry (dimensions in mm), (b) nozzle and tap inspection and measurement using digital microscopy

Figure 6 – (a) Mass flow rate measurement for the two test bearings as a function of supply pressure ($\bar{w} = 0.04$), and (b) corresponding deduced discharge coefficients

Figure 7 – Measured and predicted circumferential and axial pressure profiles at different supply pressures ($\bar{W} = 0.04$)

Figure 8 - Numerical axial pressure profile with optimized nozzle diameters compared to measurement

Figure 9 - Effect of centrifugal growth on the fluid film pressure profile ($\bar{w} = 0.04$)

Figure 10 – (a) Measured normalized load capacity as a function of the eccentricity ratio at different supply pressures and at 0 rotational speed, and (b) corresponding deduced normalized stiffness

Figure 11 – Normalized measured load as a function of the eccentricity ratio compared to model data obtained (i) by imposing a constant measured C_d and (ii) by applying the C_d correlation from reference (12).

Figure 12 – Measured circumferential pressure profiles under different loads for a supply pressure of 0.7 MPa and zero rotational speed compared to model data obtained (i) by imposing a constant measured C_d and (ii) by applying the C_d correlation from reference (12).

Figure 13 – Effect of load on mass flow rate and effective throttling area at 0.7 MPa supply pressure