Air Bearings for Heavy-Duty Industrial Applications - Effect of Bearing Type and Operating Conditions on Energy Efficiency

D.Sc. (Tech.) Olof Calonius

D.Sc. (Tech.) Panu Kiviluoma

Professor D.Sc. (Tech.) Petri Kuosmanen

Department of Engineering Design and Production, Aalto University School of Engineering, Otakaari 4, FI-02150 Espoo, Finland, E-Mail: olof.calonius@aalto.fi

Abstract

In the process industry, air bearing technology could provide a competitive alternative to the oil lubricated sliding bearing technology which has high power consumption due to the high viscosity of oil. Typically, air bearings are used in applications where frictionless and precise motion is needed. There are also air-cushion bearings for moving heavy loads in along the fairly rough factory floor in the production of, e.g., trains and large diesel engines. The purpose of this study is to explore the possibility for using air-cushion bearings in industrial machinery in cases with moderate counter-surface quality, fairly large tolerances and dynamic loading. The operating characteristics of an air-cushion type of bearing are put in contrast with those of an air bearing of the porous material type. It was found that the latter type is a good choice for machinery where adequate sliding surface quality can be achieved. High stiffness and fairly low air consumption was found. The air-cushion bearing lacks stiffness but it could function in machinery as additional load carrying unit. Good energy efficiency appears to be possible in the low-leakage mode of operation that was found. However, further testing is needed to determine if the low leakage is associated with contact between the air-cushion membrane and the counter surface.

KEYWORDS: air bearings, industrial

1. Introduction

Typically, air bearings are used in applications involving precise and nearly frictionless motion, for example in 3D coordinate measuring machines. In these applications, the counter surface of the bearing is manufactured with strict tolerances and good surface finish which increases the manufacturing cost. There are also air bearings for moving heavy loads in robust industrial systems, such as, in the production of trains and large diesel engines. These bearings can operate on the fairly rough factory floor. These air-cushion type of bearings are, however, not optimized for continuous operation with minimal air consumption. Instead, as the length and time of load transportation is limited, higher air consumption is not an issue.

In the process industry, air bearing technology could provide a competitive alternative to the oil lubricated sliding bearing technology which has high power consumption due to the high viscosity of the lubricating medium. Thanks to the low friction of air bearings, less heat is generated in the production facilities and much smaller drive units are needed for the machinery. Naturally, the benefits must be considered bearing in mind the energy consumption of the compressor used for producing compressed air. Therefore, air flow through the bearings should be kept at the lowest possible level.

Among air bearings, porous material bearings have gained increasing success because of their tolerance for contacts during the operation and low air consumption /1/. In addition, they feature a more uniform bearing pressure distribution than conventional, orifice compensated bearings /2/. These bearings can also have high stiffness provided that the permeability of the porous material is low /3/. Regarding the air-cushion type of bearing, Dvorianinov et al. /4/ and Sibgatullin et al. /5/ have studied the air flow between a flat wall and an elastic ring diaphragm and they present models to analyse its operation. Dvorianinov et al. /6/ studied the friction coefficient between the elastic diaphragm and the supporting surface.

In this study, operating characteristics of the air-cushion type of bearing are reported together with the characteristics of an air bearing of the porous material type. The purpose is to explore the possibility for using these bearings in industrial applications involving moderate sliding speeds, moderate counter surface quality, fairly large tolerances and dynamic loading.

2. Methods

2.1. Test rig

The test rig has been implemented in a large-size lathe, model 1M65. It was chosen as a test rig because it offered a rigid structure and precise guide ways and because it is easy to adjust the relative movement between the bearing and counter surface as well as the air gap height.

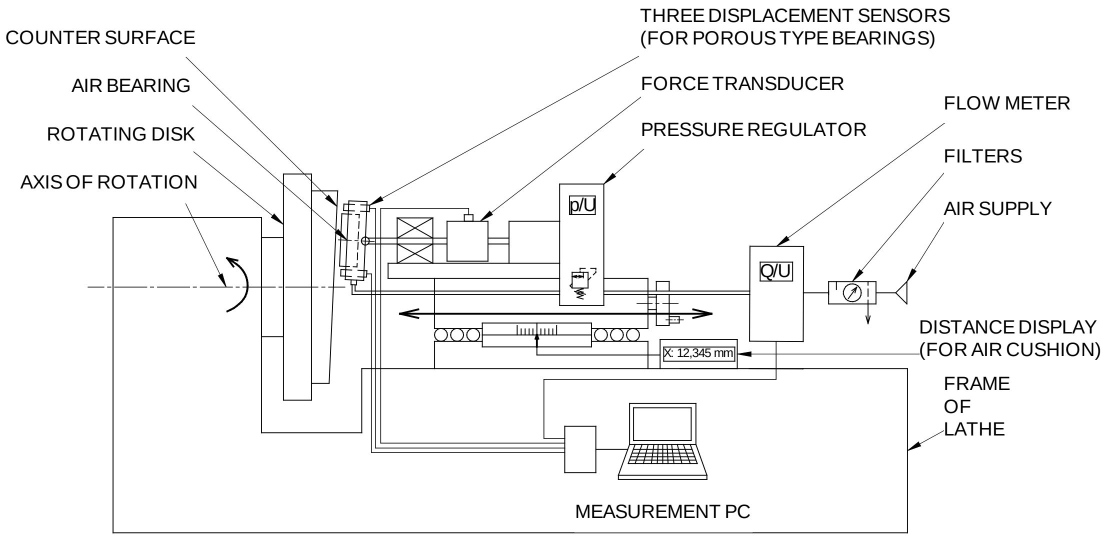


Figure 1: Schematic of the test rig.

The speed of rotation of the counter surface can be varied in a stepwise manner from 5 rpm to 500 rpm. The chuck limits the maximum diameter of the counter-face to 1000 mm. The axial run-out can be adjusted by misaligning the counter surface relative to the axis of rotation. The principle of the test rig is shown in **Figure 1**.

2.2. Loading and alignment of bearing

The loading mechanism is shown in **Figure 2**. The compression is adjusted by the hand screw of the tool slide in the lathe. A force transducer, attached to the bearing support, measures the bearing load. The test bearing is mounted to the end of a shaft by a threaded stud and a ball joint. The ball joint makes it possible to position the bearing in parallel with the counter surface. The aligning was made with the help of 10 mm gauge blocks put between the base plate of the bearing and the counter surface. The 20 mm diameter shaft is guided by aerostatic bearings (New Way S302001) which allow frictionless axial movement. The force transducer (HBM U2B 5 kN and 10 kN depending on load level) is mounted in series with the shaft by a threaded joint and the transducer rests against a fixed end.

During the dynamic tests when the test surface is rotating, a series of cup springs can be mounted between the force sensor and the fixed end to allow movement of the shaft caused by axial run-out of the counter face.

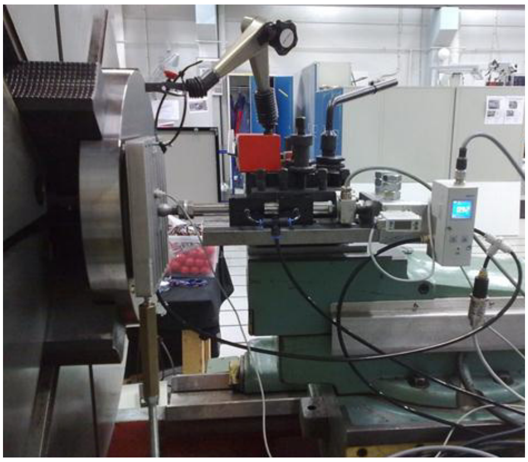


Figure 2: Loading mechanism.

2.3. Air gap height and bearing distance from counter surface

Gap height and bearing alignment for the porous bearing are measured using three eddy current displacement sensors directly attached to the bearing (MicroEpsilon, 1 mm measurement range). Zero adjustment of the sensors was done by loading the bearing without air infeed against the counter surface with constant load of 100 N. Because force and displacement between bearing and counter surface are measured directly, the flexibility of the structure does not influence the load and gap height measurements.

In the air-cushion bearing the air film is generated between the flexible membrane and the counter surface. This makes the measurement of the gap height difficult. Instead of gap height, the distance between the bearing base plate and counter surface is measured for this bearing type (Heidenhain tool slide displacement display unit).

2.4. Air supply

Aerostatic bearings set high requirements on the quality of the air supply. A service unit consisting of several filters (SMC air filter AF20, mist separator AFM20, micro mist separator AFD20), a pressure regulator (SMC AR20) and a gauge was placed between the pressure source and the system. For tests with the porous type bearing, two air flow meters (SMC PFM710, measuring range 0.2 – 10 L/min and FESTO SFE3, measuring range 5 – 50 L/min) were connected to the supply line to measure the air consumption. Because higher air leakage was anticipated with the air-cushion bearing, the two air flow meters were replaced by one FESTO SFAB-200 meter with a measuring range of 2 – 200 L/min. With the porous type bearing, a regulator with manual pressure setting was used (SMC IR1020), and with the air cushion type bearing a proportional regulator was used (Festo VPPM). The bearing pressure was set to values relevant for the bearing type and the purpose of the test. For porous bearings, the bearing supply pressure was 0.52 MPa (5.2 bar) and for air-cushion bearings it was varied from 0.1 – 0.3 MPa (1 – 3 bar).

2.5. Data acquisition

The measurements were recorded on a PC using a 14-bit analogue-to-digital board with a sampling rate of 10 kHz. An analogue 2.5 kHz low pass filter was used before sampling to avoid aliasing. The duration of one measurement was typically 10 seconds. After each measurement the data were saved to a file. The analysis of the measurements with a stationary counter surface was based on the averaged values of each channel. With the porous type bearing, dynamic tests with a larger rotating counter surface could be used. In these tests, the measurements were triggered using a laser-type photoelectric sensor (Omron E3C-LD11) with reflective tape glued on the rotating four-jaw chuck. The analysis of the measured data was based on synchronized averaging of load, displacement and air consumption.

2.6. Test bearings and counter surfaces

Two types of air bearings were used in the study (**Figure 3**): a porous type bearing (New Way Air Bearings, S1010001, recommended pressure range 0.41 – 0.55 MPa) and an air-cushion type bearing (Solving ML 8, max. pressure 0.3 MPa). The diameter of the surface of the porous bearing was 100 mm. The diameter of the flat air-cushion bearing was 200 mm. The circular contact area of the rubber membrane varies during the operation, but based on the wear mark seen after testing, the actual diameter was approximately 18 cm. The membrane was attached to a thin aluminium base plate which, in turn, was attached to a cast aluminium frame. During operation, the membrane assumes a ring-shape having a tear-like cross-section with the rounded part outwards and the pointed part towards the pressure chamber. The bearings were connected to the shaft by a 20 mm diameter, round end, ball mounting screw.

Two types of counter surfaces were used. The first was a used paper machine **roll end**, 1000 mm in diameter. The axial run-out of the roll end counter surface was 0.25 mm. The surface roughness (Ra) was 0.54 µm in the radial direction and 0.29 µm in the circumferential direction. The sliding surface of the roll end, however, proved to be too narrow for the air-cushion bearing that was acquired later. The second surface was a **test disk**, 400 mm in diameter and 50 mm in thickness, finished by surface grinding.

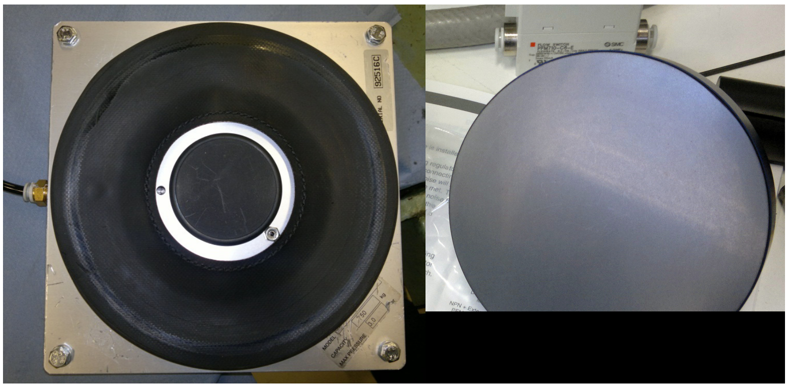


Figure 3: Air cushion bearing (ϕ200 mm) and porous material bearing (ϕ100 mm).

The axial run-out of the test disk was 0.10 mm. The surface roughness (Ra) was 0.89 µm in the radial direction and 0.76 µm in the circumferential direction. Both counter surfaces were made of steel.

2.7. Types of tests

Both a stationary (non-rotating) and a moving (rotating) counter surface was used. Quantities measured were air flow, load, bearing pressure and/or supply pressure.

2.7.1. Tests with porous material bearing

- Stationary counter surface
  - Load vs. gap height tests
- Rotating counter surface (roll end) with axial run-out, radius of rotation 0.4 m
  - Speeds of rotation 11 rpm, 32 rpm and 63 rpm, corresponding to sliding speeds 0.46 m/s, 1.34 m/s, 2.68 m/s

The bearing supply pressure was 0.52 MPa. During the measurement with the stationary counter surface, the load was increased or decreased in steps of 100 – 200 N. After the adjustment of the load, the bearing was let to stabilize for a period of 60 s before recording values. With the rotating surface, the maximum load was set by compressing the cup springs while the surface was at its maximum run-out value.

2.7.2. Tests with air-cushion bearing

- Stationary counter surface
  - Distance from bearing base plate to counter surface increased in steps (0.5 – 1 mm) from 10 to 21 mm; air consumption and load is measured
  - Three pressure levels: 0.1, 0.2 and 0.3 MPa

- Rotating counter surface (test disk) with axial run-out, radius of rotation 0.1 m

- o Same variation of the distance between bearing base plate and counter surface as above, but with speeds of rotation of 32 and 63 rpm, corresponding to center point sliding speeds of 0.34 m/s and 0.66 m/s
- o Three pressure levels: 0.1, 0.2 and 0.3 MPa

3. Results

3.1. Measurements against a non-rotating counter surface

3.1.1. Porous material bearing

The stiffness of the bearing was measured against the stationary test disk surface. The load was increased until one of the displacement sensors measuring the air gap indicated contact with the surface. The measurement was done both in the direction of increasing and decreasing values of the load. During the stiffness measurement, also the air consumption in the bearing was measured. Measured points and polynomial fits are depicted in **Figure 4**. The bearing stiffness in the gap height range 2.5 – 5 µm was (2855 – 2057) N / 2.5 µm = 319 N/µm (at 0.52 MPa bearing supply pressure).

3.1.2. Air-cushion bearing

The load carrying capacity and air leakage of the air-cushion bearing was measured against the test disk surface. Measurements were done with three different levels of input pressure. The measurement was started with a distance of 10 mm between the bearing base plate and counter surface. The distance was increased until either the pressure (and load) was lost or the air flow rate exceeded the flow meter range (200 L/min). Measured points and a polynomial fits are depicted in **Figure 5a-c**.

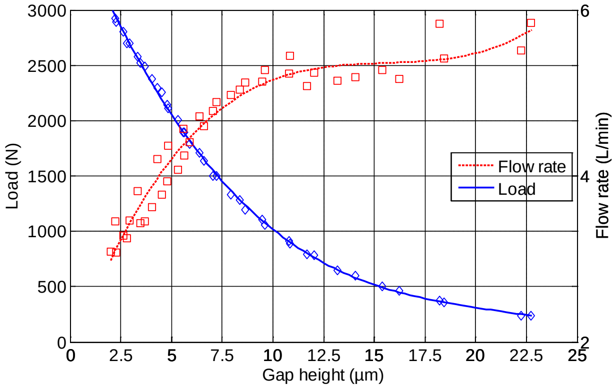


Figure 4: Porous type bearing, load and flow rate vs. gap height.

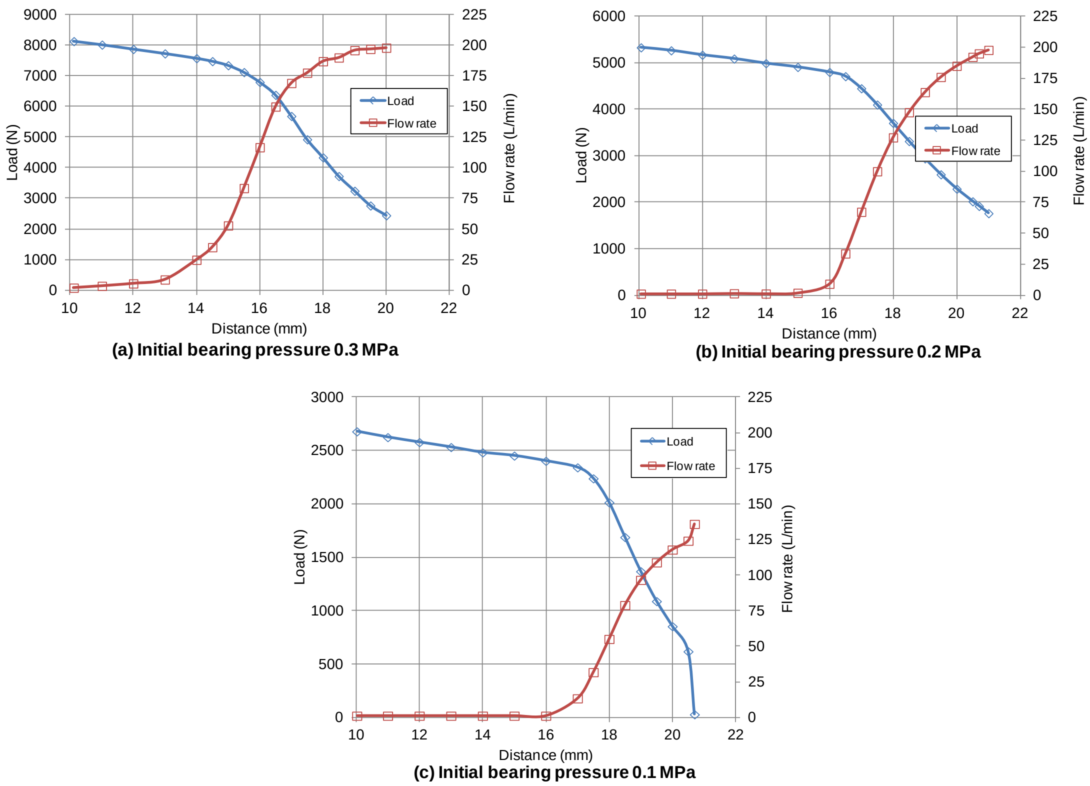


Figure 5: Air cushion, load and flow rate vs. base plate to counter surface distance. Note different load axes scales.

3.2. Measurements against a rotating counter surface

3.2.1. Porous material bearing

The load and flow rate were measured against the roll end surface. The averaged values for one rotation of the counter surface (obtained by synchronized averaging for a period of 60 s) are depicted in **Figure 6** and **Figure 7**.

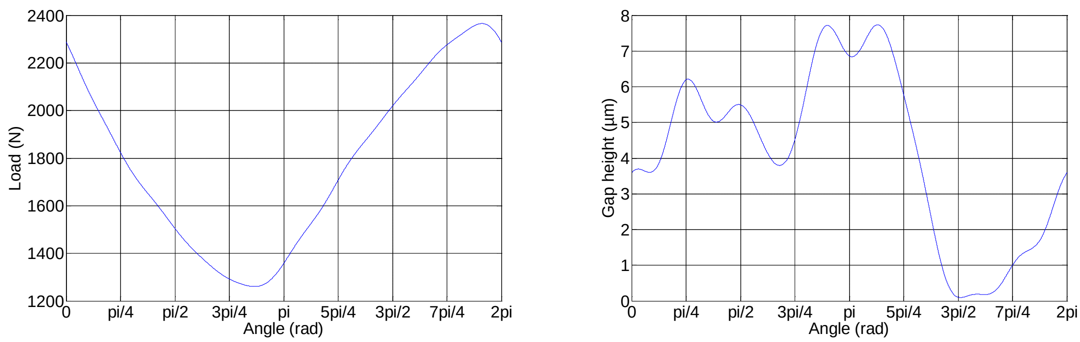


Figure 6: Porous material bearing, load and gap height vs. rotation angle.

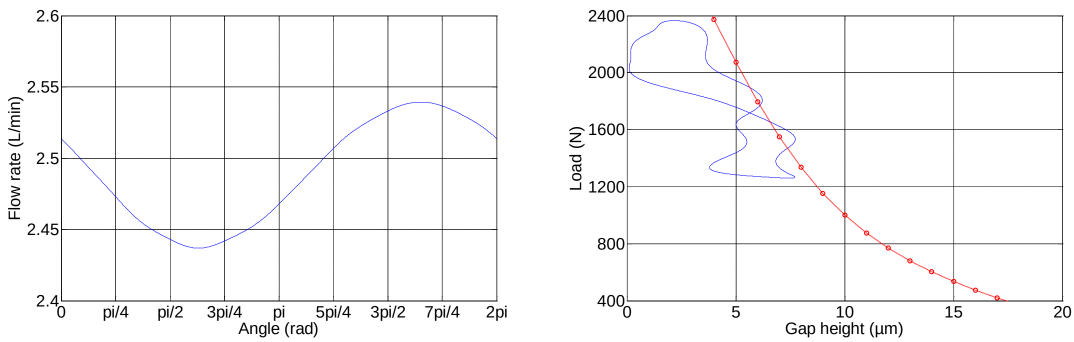


Figure 7: Porous material bearing, flow rate vs. rotation angle and load vs. gap height. Eight-shape curve (at right, continuous), shown together with sloped (dotted) curve for stationary counter surface.

3.2.2. Air-cushion bearing

The load and flow rate was measured against the test disk. The load and flow rate vs. distance between bearing base plate and the counter surface is shown in **Figure 8**. The values for both load and flow rate varied during the measurement, as can be seen in **Figure 9**. Therefore the data points in Figure 8 are average values for 10 s measurements.

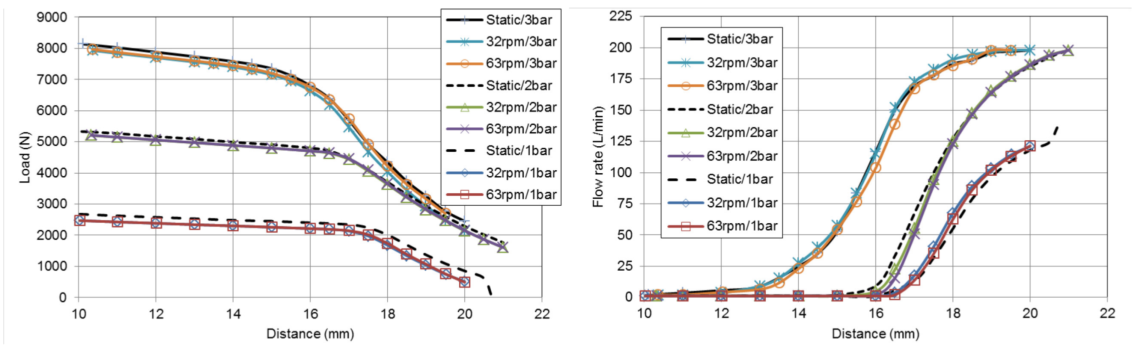


Figure 8: Air cushion bearing, load and flow rate vs. distance.

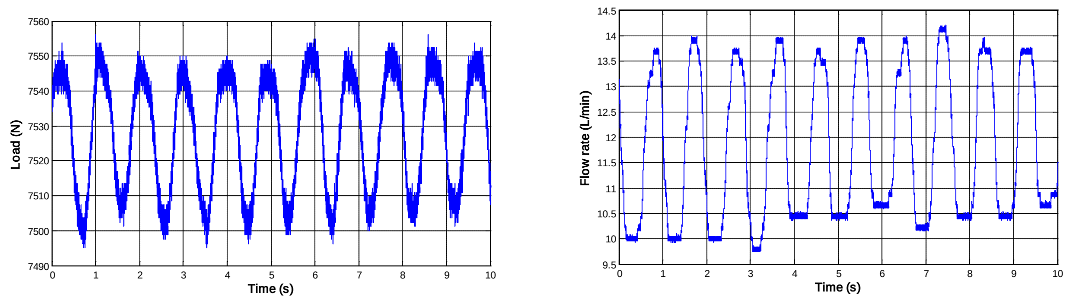


Figure 9: Dynamic load and flow rate (0.3 MPa, 63 rpm, 13.5 mm).

The reason for clipping of the flow rate curve peaks in Figure 9 is not known. One explanation could be that there are a certain number of shape configurations (small undulations) of the rubber membrane in the contact zone. At the time when the axial movement changes direction, only a certain shape configuration gets selected and this configuration can change slightly from cycle to cycle. A certain amount of leakage flow is associated with each configuration.

3.3. Power consumption

To assess the energy efficiency, the power consumption in the bearing and the test system was calculated as the product of the air supply flow rate and the measured pressure. For example, the power consumption of the air cushion bearing was calculated according to (1).

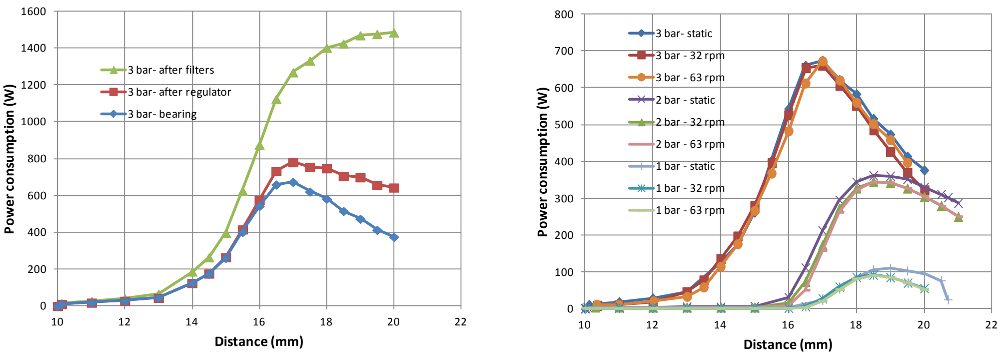
$$P_{\text{bearing}} = p_{\text{bearing}} \times Q_{\text{supply}} \tag{1}$$

In the system with the air cushion, pressure was measured at three points: after the filters in the service unit (entrance point to the pneumatic subsystem of the test rig), after the pressure regulator, and in the pressure chamber of the bearing. The variation of the power consumption as measured at these points is shown in **Figure 10a** (test series with 0.3 MPa nominal bearing pressure and a non-rotating counter surface). The power consumption based on measurements of the pressure in the air cushion pressure chamber is shown in **Figure 10b**.

For the porous type bearing, only the pressure at the pressure regulator (0.52 MPa), not the pressure in the bearing gap, could be measured. With the non-rotating counter surface the flow rate was always less than 6 L/min (Figure 4). Therefore the power consumption was less than 52 W. In the tests with the rotating counter surface, the averaged air consumption was approximately 2.49 L/min (Figure 7). With 0.52 MPa pressure this corresponds to 22 W. Lower air consumption with rotating surface is due to low dynamic gap height and lower Ra value compared to that of stationary surface.

4. Discussion

The characteristics of the bearings in this study are quite different from each other. The air-cushion type bearing was chosen to explore its performance as possible additional bearing unit in industrial applications involving moderate counter surface quality and run-out between the bearing and the counter surface. The porous type bearing was chosen as representative for state-of-the-art air bearing technology. This is nowadays a proven bearing type for precision machinery, but requires a good counter surface. Both bearing types perform well for the purpose that they were designed for.



(a) Measured at three points - non-rotating **(b) Power consumption in bearing**

Figure 10: Power consumption. (a) Pressure measured at three points in the system with non-rotating counter surface. (b) Pressure measured in the bearing, both non-rotating and rotating counter surface. Different scales on Y-axes.

Because of the large differences, a straightforward comparison of these bearing types would not be worthwhile. Instead the study should be seen as a search for a robust industrial air bearing solution initiated from two different starting points.

The porous material bearing operates with a very small air gap and its stiffness rises rapidly as the air gap gets smaller (Figure 4). These results agree well with the common knowledge about air bearings. The air gap of the cushion type bearing could not be measured and it is unclear whether the membrane is in contact with the counter surface or not. Apparently, when the air consumption was at its lowest (especially with 0.1 – 0.2 MPa bearing pressure), there has been surface contact as evidenced by the wear mark on the membrane, Figure 3. Also, when the counter surface was rotated by hand, a small breakaway friction was apparent. Measurement of the friction force (transverse force applied to the bearing from the contact) would indicate when there is contact and when an air bearing mode is reached. The air-cushion type clearly lacks stiffness (Figures 5 and 8), but it can easily accommodate large changes in bearing-to-counter-surface distance without significant changes in load carrying capacity or air consumption. It takes an outward displacement of several millimeters to cause significant loss of pressure in the bearing.

For the air-cushion bearing, there is a low-leakage mode of operation which is active at the initial bearing distance of 10 mm. This low-leakage mode continues even if the distance is increased by several millimeters, Figure 5. The extent of the low-leakage range depends on the bearing pressure as indicated by the extent of the flat initial part of the flow rate curves in Figure 5. Beyond this range, the load carrying capacity can be maintained for some additional distance if the air source is powerful enough.

In the tests against the rotating counter-face the effect of the axial run-out can be seen for both bearings as a sine variation of the measured quantities. In the tests with the porous bearing, the load variation (amplitude ≈ 1 kN, Figure 6) was much higher than for the air-cushion (amplitude ≈ 50 N, Figure 9) due to the higher run-out (0.25 mm vs. 0.10 mm) and especially due to the much higher load line stiffness in the former case. Regarding the measurements against the roll end (porous material bearing, Figure 6, right), the basic shape of the gap height curve should be a sine curve, due to the axial run-out. The curve, however, gets modified due to the surface imperfections which superimpose a signal of smaller amplitude and higher frequency on the run-out displacement signal. The peak load (2.3 kN) appears to be close to the maximum load value, as the minimum gap height is close to zero. By combining the data from the graphs in Figure 6, the dynamic load-gap height curve of Figure 7 was produced. The figure shows the dynamic operating range of the porous type bearing compared to the load-gap height data measured with the stationary counter surface.

For the air-cushion bearing, the load and flow rate curves for rotating and stationary counter surfaces are nearly the same, Figure 8. In this figure, it can be seen that the low-leakage range is clearly shorter with 0.3 MPa bearing pressure.

With the porous material bearing the power consumption remained low under all circumstances; below 52 W for the stationary counter surface and 22 W for the rotating roll end surface (which had lower surface roughness). For the air-cushion bearing, the power consumption varied according to the leakage flow rate, Figure 10. In the low-leakage range, the power consumption was of the same order of magnitude as for the porous material bearing, some tens of watts at 0.3 MPa and even less at 0.1 – 0.2 MPa. Beyond this range the power consumption increases rapidly. The power loss curve of the bearing reaches a peak value when the air supply cannot compensate anymore for the pressure loss caused by the leakage flow. The losses in the total pneumatic subsystem will still keep on increasing, Figure 10a. The speed of rotation (at least in the range used in the tests) does not have an influence on power consumption.

The low-leakage range of the air-cushion bearing appears promising from the point of view of energy efficiency. If the air-cushion can be used at less than the rated 0.3 MPa the savings in air consumption could be considerable; at 0.2 MPa the air consumption was less than 2 L/min up to a distance of 15 mm (Figure 8), corresponding to a power consumption of less than 6.7 W. Of course the load carrying capacity will decrease accordingly, so another alternative would be to redesign the membrane to seal for longer distances even at 0.3 MPa. More research is needed to determine when there is actually contact between the membrane and the counter surface and when there is low-leakage air lubrication. The wear properties of the membrane should also be tested if the bearing is aimed for continuous operation.

A number of improvements should be made to the test rig for future studies with air-cushion type bearings. A low-range flow meter should be added to study the low-leakage characteristics. In the present study, the nominal lower limit of the flow meter was 2 L/min (lowest recorded value 1.3 L/min). A glass counter surface could be used to study the deformation of the rubber membrane and to determine the actual contact diameter and sealing width. Addition of accurate measurement of friction force between the counter surface and the air-cushion membrane would be important to obtain information about the contact situation. With regular and porous material air bearings this has not been necessary, as there has not been surface contact when the bearings have been in operation. A larger counter surface should be manufactured; now only modest sliding speeds could be achieved. The aim is to be able to study situations where the relative velocity between bearing and counter-surface exceeds 10 m/s.

5. Conclusions

The porous material bearing appears to be a good precision machinery bearing providing both high stiffness (despite the porosity of the material) and low air consumption. However, the counter surface must be of good quality (surface finish typical for sliding bearings and seals).

The air-cushion type bearing can accept large axial displacements between the bearing and the counter surface without damage and with only minor changes in the load carrying capacity. This is due to the low stiffness of this type of bearing. For the air-cushion, a low-leakage operating range was found especially at the lower pressure levels. At 0.1 – 0.2 MPa bearing pressure, the leakage was less than 2 L/min when the bearing-to-counter surface distance was less than approximately 16 mm. This could be very useful when aiming for good energy efficiency. However, it is unclear whether the rubber part acts as an air bearing or a contact seal in this low-leakage mode; further studies are needed to determine if non-contact conditions can be obtained without excessive air consumption. If low leakage indicates contact, then the membrane will wear and frictional heat will be generated when running the bearing at higher sliding speeds. With 0.3 MPa pressure the low-leakage region is much narrower than for 0.1 – 0.2 MPa. After the low-leakage region the leakage increases rapidly with the distance. Therefore, the distance should be controlled to limit the air consumption. The effect of speed of rotation on air consumption appears weak; the air cushion pressure and the distance to the counter surface have a much stronger effect. The air-cushion bearing could be used as additional load carrying unit in machinery and it can operate on low-quality/low-cost surfaces. However, due to its low stiffness, the air-cushion cannot directly (without additional measures) replace traditional bearing solutions.

Both tested bearing types can operate with low air consumption and thus provide energy efficient air bearing solutions. However, the air-cushion bearing needs further testing to clarify if low leakage is associated with membrane contact, which would indicate the need for condition control in continuous use.

6. Acknowledgements

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7. References

/1/ Calonius, O., Kiviluoma, P., Kuosmanen, P. 2011. Large-scale test rig for assessment of characteristics of flat air bearings running against a rotating counter-face. 56th International Scientific Colloquium, Imenau, Germany, Sept. 12-16. 11 p.

/2/ v Beek, A. 2009. Advanced engineering design. Lifetime performance and reliability. Chapter 12 Design of aerostatic bearings. ISBN-10: 90-810406-1-8.

/3/ Fourka, M., Bonis, M. 1997. Comparison between externally pressurized gas thrust bearings with different orifice and porous feeding systems. Wear, 210, pp. 311-317.

/4/ Dvorianinov, V.G., Sibgatullin, N.R., Slezkin, N.A. 1977. Motion of a viscous gas in a layer with an elastic boundary. Journal of Applied Mathematics and Mechanics, 41 (2), Jan, pp. 289–297.

/5/ Sibgatullin, N.R., Slezkin, N.A., Sorokin, E.A. 1978. On the stability of an aerostatic bearing with an elastic skirt, Journal of Applied Mathematics and Mechanics, 42 (5), Jan, pp.947-957.

/6/ Dvorianinov, V.G., Sveshnikova, E.I., Sibgatullin, N.R. 1981. On friction resistance to motion of an aerostatic carrier with elastic diaphragm, Journal of Appl. Mathematics and Mechanics, 45 (6), Jan, pp.809-813.