Comparison study regarding bearing performance on 3 types of air bearings using Dyrobes software

Sigismund Becze^{1,*} and *Gheorghe Ioan Vuscan*¹

¹ Technical University of Cluj-Napoca, B-dul Muncii no. 103-105, 400614 Cluj-Napoca, Romania

Abstract. The gas bearings have low load capacity and high stiffness requirements beside the high precision which all are a concern in their development. The 2 large categories of gas bearings have each of them a drawback: the static gas bearing have the performances depending by the system which maintain the pressure, while the dynamic one's performances are increasing with speed. The article creates a comparison between 3 different type of dynamic gas bearing, using air as lubricant. While all 3 types of bearing having similar performances at max speed in term of eccentricity ratio vs. rpm, minimum film thickness and equilibrium locus, the cross-coupled damping and stiffness shown a rotor destabilization tendency. The 3 lobe rotor presented the highest pressure profiles at every speed. The pressure profiles present same shapes after 80000 rpm. The results shown potentially a multilobe rotor can over perform the plain bearing also in low speed.

1 Introduction

Initially the gas bearing found application where are required high speed, low load, oil and dust free environment like medical application but recently they also being applied successfully for hydrogen fuel cells in passenger cars [1, 2]. The main concerns regarding gas bearings are high precision, very limited load, low performance at low speed for dynamic gas bearings, stiffness [1, 3]. Despite those drawbacks, this type of bearing has some major advantages: oil or grease free, low viscosity, self-centering, low unbalance and they perform better where oil and ball contact bearings failing: at speeds over 80krpm [1,4].

Abbreviations

CAD - computer aided design

FEA - finite element analysis

* Corresponding author: bsigismund@gmail.com

2 Simulation conditions

The length of bearing was considered $L=80$ mm, diameter $D=10$ mm, bearing clearance $c=0.009$ mm, ambient pressure $P=1.01325$ bar, load force $W=11.7678$ N and the gas dynamic viscosity $\mu=0.01837$ centiPoise. The initial speed was considered at 5000 rpm and max speed 305000 rpm with an increment of 50000 rpm.

The bearing was considered a non-pressurized one with ambient side pressures $P=1.01325$ bar. The bearing characteristics were calculated using the following parameters:

- Bearing characteristics number $=\mu N/P$
- Where μ -absolute viscosity of lubricant (kg/m.s)
- N-speed of the journal (rpm)
- P-bearing pressure (N/mm²)
- In our case, for initial phase, considering the rotor speed:
- $N=5000$ rpm
- For plain journal bearing $P=0.0156871$ N/mm², for 3 lobe bearing $P=0.0214135$ N/mm², while for 4 lobe $P=0.0192296$ N/mm²
- $\mu=0.00001837$ kg/m.s

In this case, for each type the bearing characteristics will be

- a. 5.85512 for plain journal bearing
- b. 4.28935 for 3 lobe bearing
- c. 4.77694 for 4 lobe bearing

2.1 Sommerfeld Number

The bearing characteristic number or Sommerfeld number (S, equation 1) is a dimensionless value, generally containing all the variables specified in bearing design and is used as a reference in hydrodynamic fluid film bearings analysis. [5]

$$S = \left(\frac{r}{c}\right)^2 \cdot \frac{\mu \cdot \omega \cdot L \cdot D}{W} \quad (1)$$

Where:

S - Sommerfeld number (dimensionless)

r - shaft radius (m)

c - radial clearance (m)

μ - absolute viscosity of the lubricant (pascal*s)

ω - angular velocity of the shaft (rad/s)

L - bearing length (m)

D - bearing diameter (m)

W - applied load (N)

For all 3 cases Sommerfeld number is the same $S=2018.168746$ for 5000 rpm and $S=100908.437281$ for 250000 rpm.

2.2 Petroff's Law - shear stress in the lubricant

Bearing friction was first explained by Petroff's method (equation 2), who consider the shaft being concentric with bearing, and this method generates the equation called Petroff's Law from which results the friction coefficient. [6]

$$\tau = \frac{2 \cdot \pi \cdot r \cdot \mu \cdot N}{c} \quad (2)$$

Where:

τ - shear stress in the lubricant assuming a constant rate of shear. (pascal)

π - pi

r - shaft radius (m)

μ - absolute viscosity of the lubricant (pascal*s)

N - rotational speed of the shaft (revs/s) (1/s)

c - radial clearance (m)

The shear stress in all 3 air bearings: $\tau= 53435.9501872$ at 5000 rpm; while at 250000 rpm $\tau= 2670453.660976444$ Pascal.

2.3 Petroff's Law - Bearing coefficient of friction

Applicable to small load, Petroff's Law offers a reasonable estimation of friction coefficient [7]:

$$f = 2 \cdot \pi^2 \cdot \frac{\mu \cdot N}{P} \cdot \frac{r}{c}$$

(3)

Where:

f - bearing coefficient of friction (dimensionless)

π - pi

μ - absolute viscosity of the lubricant (pascal*s)

N - rotational speed of the shaft (revs/s) (1/s)

P - radial load per unit of project bearing area (pressure) (Pascal) (fig. 1)

r - shaft radius (m)

c - radial clearance (m)

Fig. 1. Surface area was simulated and measured using Catia V5 CAD software.

The area was used to calculate the pressure of radial load. In all 3 bearings the coefficient to friction is: $f= 0.0628634804849$ for 5000 rpm and $f=3.1431865467$ for 250krpm. [8]

Bearing Performance module of Dyrobes v2020 was used for the analyze of the plain air bearing (figure 2), 3 lobe (figure 3) and 4 lobe (figure 4).

Fig. 2. Plain bearing setup.

Fig. 3. 3 lobe air bearing setup.

Fig. 4. 4 lobe air bearing setup.

3 Calculation and data evaluation

Calculation was performed using Dyrobes Bearing Performance module with Metric units. Eccentricity is decreasing with speed increase, due to self-centering property of air bearing.

Fig. 5. Eccentricity ratio vs. rpm top left: a) plain bearing; b) 3 lobe, c) 4 lobe

The dynamic air bearing is stabilizing when speed is increased (figure 5).

Fig. 6. Minimum film thickness. a) plain, b) 3 lobe, c) 4 lobe.

Having a more constant eccentricity results in also a more stabilized film thickness at higher speed. All three bearings performing quite similar regarding film thickness (figure 6).

Fig. 7. Journal equilibrium locus. a) plain, b) 3 lobe and c) 4 lobe.

In term of journal equilibrium locus, the lowest performance at 5000 rpm are the 3 lobe bearing, but all performing quite similar near the max speed (figure 7).

At low rpm the 4 lobe and plain bearing pressures are close, only the 3 lobe bearing having a higher pressure (figure 8).

At 105000 rpm the pressure distribution changing, the plain bearing having the most distributed pressure and the lowest, while 3 lobe bearing having the highest pressure (figure 9).

At 305000 rpm the pressure distribution slightly changing (became closer to an equal distribution) for 3 and 4 lobe bearings, while for plain bearing remain the same as at low speed one (figure 10).

Highest pressure is registered at 3 lobe, then 4 lobe and lowest at plain bearing (figure 11). The pressure levels distribution is same at 105krpm and 305krpm: highest at 3 lobe bearing, lowest at plain bearing (figure 12 and figure 13).

Fig. 8. Pressure profile at 5000 rpm a) plain, b) 3 lobe and c) 4 lobe.

Fig. 9. Pressure profile at 105000 rpm a. plain, b. 3 lobe and c. 4 lobe.

Fig. 10. Pressure profile at 305000 rpm: a) plain, b) 3 lobe and c) 4 lobe.

Fig. 11. Pressure profiles at 5000 rpm: a) plain, b) 3 lobe and c) 4 lobe.

Fig. 12. Pressure profile at 105000 rpm: a) plain, b) 3 lobe and c) 4 lobe.

Fig. 13. Pressure profile at 305000 rpm: a) plain, b) 3 lobe and c) 4 lobe.

Fig. 14. Stiffness and damping vs. rpm: a) plain, b) 3 lobe and c) 4 lobe (red-damping; magenta-stiffness).

Fig. 15. Stiffness and damping vs. rpm cross-coupled coefficients (red-damping; magenta-stiffness): a) plain, b) 3 lobe and c) 4 lobe.

4 Conclusions

Cross-coupled coefficients negative value present a tendency to destabilize the rotor (figure 14 and figure 15).

Around 80000 rpm, once the film thickness and eccentricity ratio variation getting close to zero, the pressure distribution stabilizing, keeping the shape until max speed.

While for low speed (rpm) the pressure distribution look very similar between all 3 bearings, at above 100 krpm the 3 and 4 lobe bearings changing the distribution under the geometry influence (lobes). While for 3 lobe bearing the pressure is highest, for 4 lobe the pressure decreasing. For 4 lobe bearing the pressure distribution is more uniform, basically being distributed between lobes.

Between all 3 bearings, the lowest pressure was obtained using cylindrical plain bearing, but a multilobe bearing could match that with a better pressure distribution. All 3 bearings present isotropic behaviour above 40000 rpm, the both damping and both stiffness coefficients being almost equal.

The cross-coupling stiffness coefficients (linearized bearing damping and stiffness coefficients from figure 14) being negative, this shown a tendency to destabilize the rotor system.

References

1. J. Chen Wen, E.J. Gunter (Trafford Publishing, Canada, 2007)
2. E.J. Gunter, J. Chen Wen, Vibration Institute 34th Annual Meeting, Oak Brook, Illinois (2010)
3. E.J. Gunter, R.R. Humphris, H. Springer, **55** (1983)
4. E.J. Gunter, 7th IFToMM-Conf. on Rotor Dynamics, Vienna, Austria (2006)
5. <https://www.fxsolver.com/browse/formulas/Sommerfeld+Number+-+alternative+using+angular+velocity>, accessed on 30.04.2019
6. <https://www.fxsolver.com/browse/formulas/Petroff%27s+Law+-+shear+stress+in+the+lubricant>, accessed on 30.04.2019
7. <https://www.fxsolver.com/browse/formulas/Petroff%27s+Law+-+Bearing+coefficient+of+friction>, accessed on 29.04.2019
8. <https://www.fxsolver.com/browse/formulas/Sommerfeld+Number+-+alternative+using+angular+velocity>, accessed on 30.04.2019