DESIGN AND TESTING OF AERODYNAMIC THRUST BEARINGS FOR MICRO TURBOMACHINERY APPLICATIONS

Tobias Waumans, Joris Mattheijssens, David Meuws,
Peter Vleugels, Jan Peirs, Farid Al-Bender and Dominiek Reynaerts
Katholieke Universiteit Leuven, Department of Mechanical Engineering, Leuven, Belgium

Abstract: This paper presents the theoretical and experimental performance comparison of different grooved thrust bearing geometries for micro turbomachinery applications. A parameter study and optimisation are carried out to identify and quantify the effect of different geometrical parameters on bearing characteristics as load carrying capacity and viscous frictional losses. The obtained performance curves are experimentally validated up to speeds of 240,000 rpm. A reasonable agreement can be observed with the simulated data.

Key Words: aerodynamic thrust bearing, micro turbine, parameter study

SYMBOL LIST

d_{groove}	[m]	groove depth
h	[m]	clearance height
p_a	[Pa]	atmospheric pressure
p_{load}	[Pa]	loading pressure
r_{in}	[m]	inner bearing radius
r_{groove}	[m]	groove end radius
r_{out}	[m]	outer bearing radius
r_{tip}	[m]	herringbone tip radius
w_{groove}	[m]	groove width
w_{land}	[m]	land width
N		number of grooves
R	[m]	characteristic radius
$W = W/(\pi R^2 p_a)$		dimensionless load
α	[deg]	groove angle
ω	[rpm]	rotational speed
$\Lambda = 6\mu\omega R^2/(p_a h^2)$		bearing number

1. INTRODUCTION

Recently, a growing interest can be observed in power generation with micro- and mesoscopic scaled gas turbines. These compact and portable units can replace batteries as they claim to provide a higher energy density [1]. However, a lot of technical challenges are encountered while developing and integrating all different components. The combined requirement of high rotational speed, elevated working temperature and small overall scale turns the bearing design into a challenging task.

The work is performed within the PowerMEMS-project of the KULeuven. The final project goal is the development of a fuel-based micro gas turbine which can serve as a compact, mobile and autonomous energy source with an expected power output in the range of 1 kW. The system as a whole should fit in 1 dm^3. The operational rotational speed is set to 500,000 rpm for a compressor and turbine diameter of 20 mm. Further details can be found in [2], while other aspects such as production and material research are discussed in [3, 4].

Air bearings are seen as the only feasible bearing solution for this application. Different air bearing types are currently envisaged. For prototype applications, aerostatic bearings are the first choice. Aerodynamic bearings increase the systems autonomy as no external supply of pressurised air is required during operation. Current research focuses towards aerodynamic foil bearings for the journal bearing function, while aerodynamic grooved thrust bearings are a good alternative if they are able to meet the systems requirements. Axial loading can be as high as 15 N due to a pressure difference between the compressor and turbine side. The viscous frictional losses must be kept low while only a minimal detrimental effect on the rotordynamic stability can be tolerated.

The topic under investigation has only been briefly studied in recent literature. Some publications investigate the numerical aspect of the problem [3], but relevant experimental reference data at this scale is limited.

2. THEORETICAL PARAMETER STUDY

The pressure distribution of an air film between two narrowly spaced surfaces in relative motion is modelled by the Reynolds equation. A finite difference discretisation together with suitable boundary conditions allows the calculation of the static behaviour of any bearing geometry and working condition.

In order to use the same core programming code, a grid mapping technique is applied to transform the spiral shaped grid into a rectangular calculation grid.

A parameter study is carried out to identify the most influencing geometrical parameters on bearing characteristics as load carrying capacity and viscous frictional losses. Fig. 1 shows optimal performance with respect to load for geometries with a groove angle of ca. 20 degrees, a groove depth ratio of 3 and a groove width ratio of 1. The number of grooves has only a minor effect. These results correspond with the design rules found in literature [4, 5]. However, in the total design and optimisation process other criteria have to be taken into account, such as dynamic bearing properties as stiffness and damping.
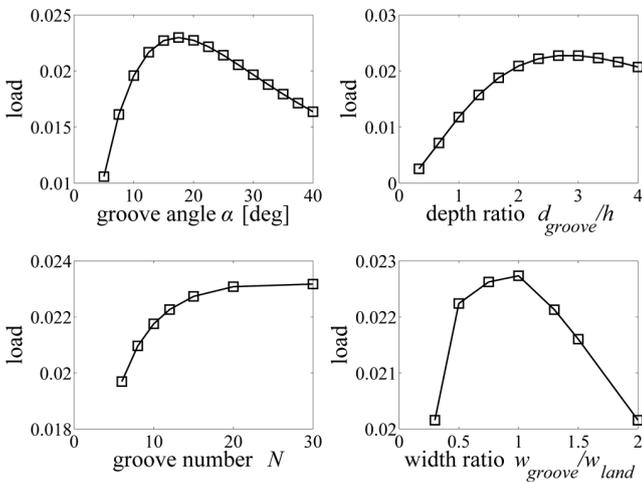

Fig. 1: Effect of the groove angle, depth ratio, number of grooves and width ratio on the dimensionless load carrying capacity (inward spiral geometry with bearing number Λ of 11.3).

3. EXPERIMENTAL VALIDATION

In order to validate the described modelling process, an experimental setup has been developed to test various grooved thrust bearing geometries over a wide range of working conditions. The setup allows a fine adjustment of the axial loading up to 30 N and has a design speed range up to 300,000 rpm. Measurement of the actual clearance height makes it possible to compare simulated performance curves relating the applied load W to the actual clearance h for different values of the rotational speed ω.

3.1 Experimental setup

The main part of the experimental setup (Fig. 2) consists of a shaft of dia. 6 mm with a shrink-fitted titanium rotor disc of dia. 20 mm which is supported by aerostatic journal bearings (radial clearance ca. 7.5 μm, length of 6 mm and with 6 feedholes of dia. 150 μm). The aerodynamic thrust bearing under testing acts on the inner side of the rotor disc and consists of an individual and interchangeable bronze part. The grooved geometry is machined by micro-milling.

A Pelton turbine drives the shaft up to the required speed range. A pressure chamber provides a controlled axial load by applying a superambient pressure at the outer side of the rotor disc. The air expands across the cylindrical rim of the rotor disc and escapes further on through the housing.
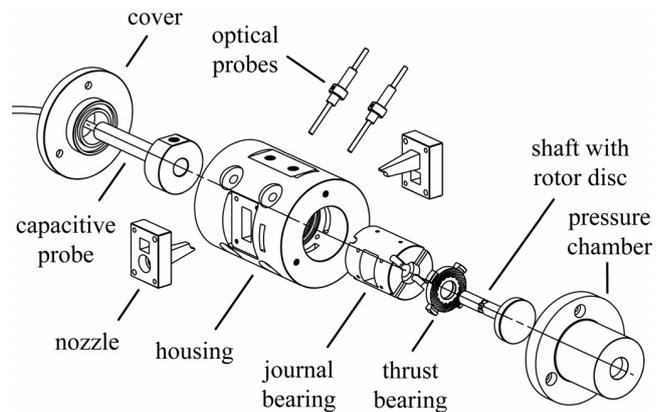

Fig. 2: Exploded view of the test setup.

Careful alignment of all components has proven to be crucial in achieving good test data. This is assured by specifying stringent fabrication tolerances on components which determine the uniformity of the air gap height (both static and dynamic).

3.2 Instrumentation

The actual bearing clearance h is measured with a capacitive distance probe (Lion Precision DMT12-C7) at the shaft side without rotor disc (Fig. 3). The applied pressure in the loading chamber p_{load} is recorded with an electronic pressure sensor (Druck PMP 1400). Two fibre optical transducers provide information concerning rotor speed and rotor vibration data. The latter is important to prevent operating at working conditions characterised by excessive (sub)synchronous whirling.
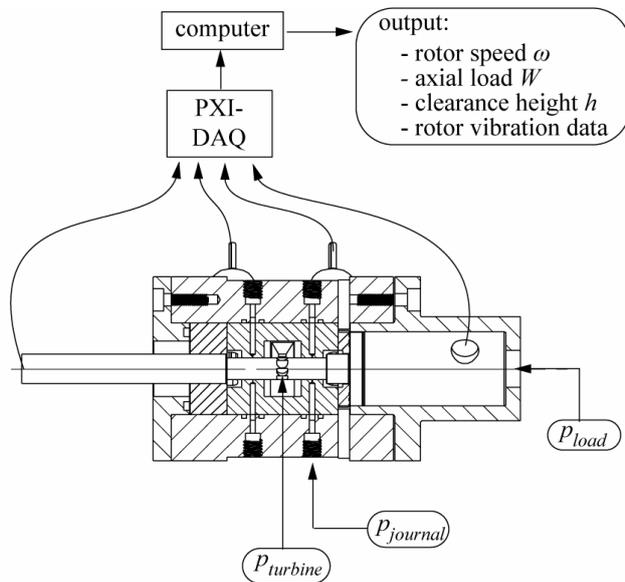

Fig. 3: Instrumentation overview.

All analog data is acquired by a National Instruments PXI-DAQ system. A MATLAB routine calculates and displays all necessary information concerning rotor speed ω, applied axial load W, actual clearance height h and rotor whirling.

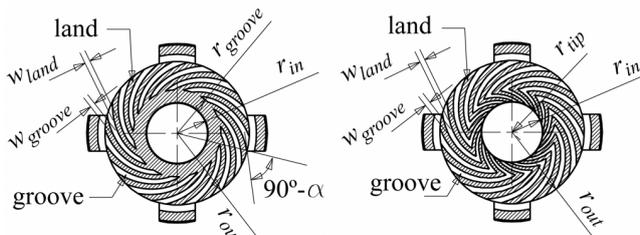## 3.3 Test bearing geometries

Fig. 4: Bearing geometry types and nomenclature.

Fig. 4 shows the bearing geometries which are experimentally analysed. The nomenclature and the actual values of all geometrical parameters are summarised in Table 1.

geometry parameter	inward spiral geometry	herringbone geometry
r_{in}	4.25 mm	4.25 mm
r_{out}	10 mm	10 mm
r_{groove}	6 mm	-
r_{tip}	-	6 mm
α	25º	25º
d_{groove}	45 μm	45 μm
w_{land}/w_{groove}	1	1
N	15	15

Table 1: Geometrical parameter values of both tested bearing types.

3.4 Test data

The performance of the different bearing geometries is experimentally obtained by recording the clearance value for different values of the applied load while keeping the rotational speed constant. Fig. 5 compares simulated and experimental performance data for the inward spiral geometry.
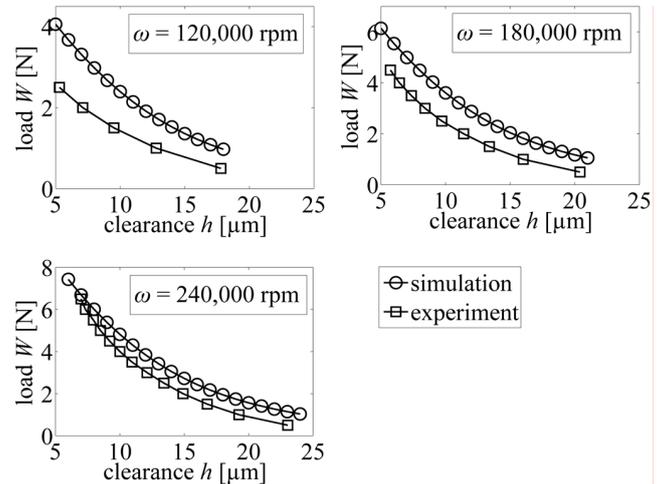

Fig. 5: Comparison of simulated and experimentally obtained load carrying capacity vs. clearance height for different rotational speeds (for inward spiral geometry).

4. DISCUSSION

The discrepancy between the predicted and measured load carrying capacity is most likely due to (i) surface and alignment imperfections of the bearing components (ii) deformation of the rotor disc due to centrifugal forces (iii) variation of the shaft length due to the Poisson effect. Some of these effects can be taken into account in a refined modelling process.

5. CONCLUSION

This paper analyses the effect of different geometrical parameters on the load carrying capacity by means of a parameter study. The performance of an inward spiral thrust bearing is experimentally verified. The agreement with predicted data is reasonable. Further work will focus on the model refinements to explain the existing discrepancy with experimental data.

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