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DESIGN AND HIGH-SPEED TESTING OF AIR BEARINGS FOR AN ULTRA-MINIATURE GAS TURBINE

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Abstract: This paper presents both the design aspect and the experimental testing of air bearings for an ultra-miniature gas turbine. The test-setup consists of a turbo-shaft rotor supported on aerostatic bearings with a target speed of 500,000 rpm. The bearing design process mainly focuses on rotordynamic stability at high rotational speed while the experimental work consists of in-situ balancing experiments of the rotor up to 75000 rpm.

Key words: air bearing, ultra-miniature gas turbine, stability

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 [1], while other aspects such as production research [2] and preformance map testing [3] are also presented at this conference.

2. TURBO-SHAFT SETUP

2.1 Overview

The test-setup of this paper is intended for low-temperature testing (without combustion) of compressor and turbine performance and therfore also allows to study the behaviour of the rotor-bearing system at high speed.

The exploded view of Fig. 1 shows the different components of the test-setup. A central part houses the bearing unit and compressor volute. The turbine stator vanes, turbine volute and diffuser are connected to this central part.

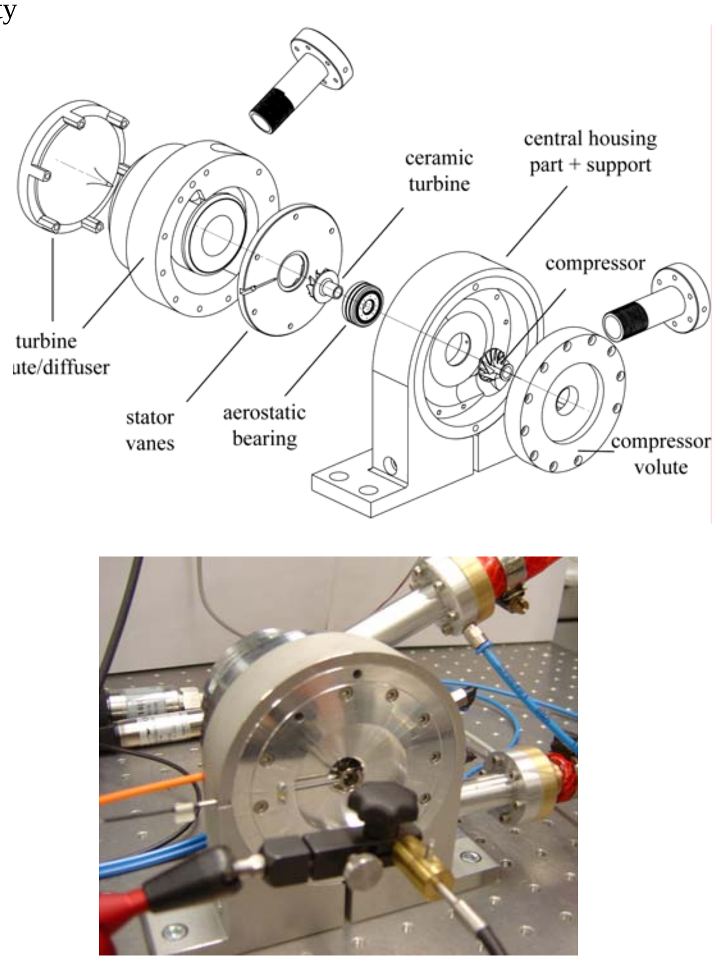

Fig. 1: Overview of the test-setup.

To overcome alignment and manufacturing problems of a split bearing design and to permit the use of a multi-material rotor, a split rotor concept was opted for (Fig. 2). A ceramic turbine (Si3N4/TiN ceramic composite) is connected to a titanium compressor by means of an internal coupling part. This rotor with a mass of 10.58 g is supported by an aerostatic bearing unit which acts on the ceramic turbine shaft and on the inner side of the compressor and turbine rotor disc.

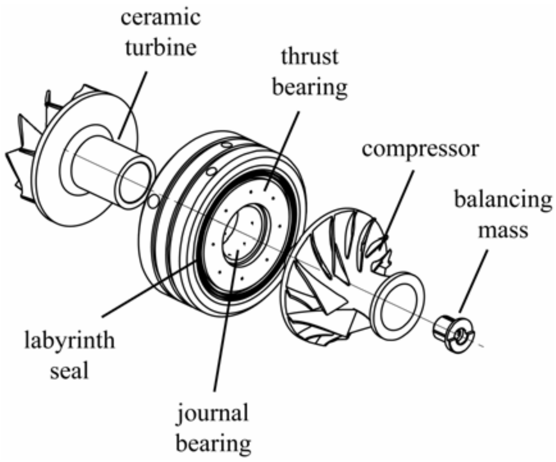

Fig. 2 Rotor-bearing system.

2.2 Instrumentation

The rotor whirling data is recorded by two fiber optical displacement probes located at the compressor inlet nose and turbine rim. An optical keyphasor signal (MDI K310) serves as triggering source. All data is acquired by a NI PXI-DAQ sytem (PXI-6123 with simultaneous sampling up to 500 kHz). A real-time bearing monitoring system provides information about the rotor speed and the actual rotor vibrations (synchronous and subsynchronous). This makes in-situ balancing possible and allows critical speed and stability analysis. More important, it prevents operating at working conditions characterised by excessive (sub)synchronous whirling and therefore reduces the risk of a fatal bearing crash.

2.3 Bearing design

The bearing design is regarded as an optimisation process in which different geometrical bearing parameters such as nominal clearance height and feeding arrangement lead to an optimal combination of film stiffness/damping coefficients, viscous frictional losses and gas flow rate. The dynamic bearing coefficients serve together with the rotor mass properties as input for a rotordynamic study which concludes on the critical speeds and stability of the rotor-bearing system. Guaranteeing of rotordynamic stability of both the cylindrical and conical whirling modes has proven to be the main concern, even when using aerostatic bearings. Aspects as fabrication tolerances, centrifugal and/or thermal effects are also taken into account in this design part. More information on this modelling process can be found in [4]. The entrance flow modelling aspects in inherently compensated bearing are explained in detail by [5].

Table 1: Journal bearing design values (evaluated at a rotational speed of 300 000 rpm and perturbation frequency of 15 500 rad/s).

diameter	8 mm
length-to-diameter ratio	1
radial clearance	5 µm
feeding details	two rows of 8 dia. 150 µm
mass flow rate at 8 bar (absolute)	0.037 g/s
frictional losses at 350,000 rpm	14.6 W
stiffness	4.40 N/µm (direct) -2.41 N/µm (cross-coupled)
damping	119.9 Ns/m (direct) 132.5 Ns/m (cross-coupled)

Table 2: Thrust bearing design values (evaluated at a rotational speed of 300 000 rpm and perturbation frequency of 17 000 rad/s).

inner diameter	11 mm
outer diameter	16 mm
nominal clearance	8 µm
feeding details	one rows of 8 dia. 150 µm
mass flow rate at 8 bar (absolute)	0.050 g/s
frictional losses at 350,000 rpm	10.20 W
tilt stiffness	34.35 Nm/rad (direct) -2.30 Nm/rad (cross-coupled)
tilt damping	0.118e-3 Nm s/rad (direct) 0.032e-3 Nm s/rad (cross-coupled)

2.4 Labyrinth seal

At both the compressor and turbine rim, air wih a pressure ratio of 3 (at target speed of 500 000 rpm) can deteriorate the stability of the conical whirling modes. To prevent this, a labyrinth seal is included into the bearing unit between the thrust bearing surface and rotor discs (Fig. 3).

A three-step labyrinth seal with a clearance of 10 µm results in a leakage flow of 2 x 0.176 g/s at target speed. The surface of the seal teeth is kept minimal (width of 0.15 mm) to prevent cross-coupling effects which can cause whirling instability.

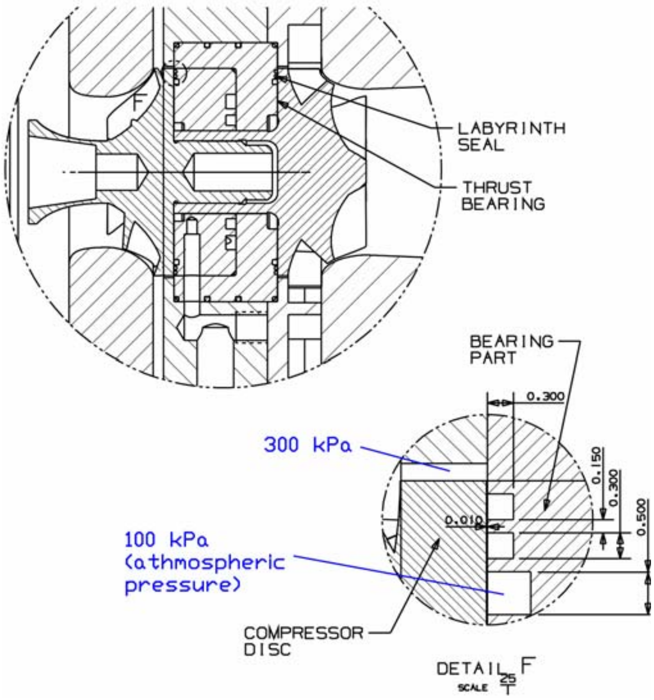

Fig. 3 Section view of air bearing unit and labyrinth seal detail view.

2.5 Fabrication technology

The limitations of the available production technology should always be kept in mind during the design process. This limits for instance the minimal air bearing clearance which can be realised.

The titanium compresor is produced by 5-axis micro-milling. More information on the production aspects of the ceramic turbine can be found in [2]. The bearing unit of this low-temperature setup is made out of bronze and is composed of two press-fitted parts to incorporate supply and venting channels. The bearings surfaces are fine-turned on a precision lathe while the labyrinth seal features are realised by micro-milling. µEDM technology enables the fabrication of the dia. 150 µm feedholes (Fig. 4).

Fig. 4 Production of the feedholes by µEDM technology.

3. TEST RESULTS

3.1 Balancing experiments

The systems target speed of 500,000 rpm lies above the estimated critical speed values of the rotor-bearing system. Therefore, the rotor should be sufficiently balanced to safely pass the critical speeds and to minimise any residual imbalance at supercritical operation.

The setup provides the possibilty of in-situ balancing by means of small balancing weights which can be mounted in the compressor inlet nose (Fig. 2) and at the turbine end. The two-plane influence coefficient method is used to determine the rotor imbalance. This two-plane method is adequate since the target speed stays far below the first bending mode of the rotor.

A first step in the balancing procedure is to compensate for large rotor imbalance by material removal at both compressor and turbine. After this step, in-situ fine balancing is required to compensate for imbalance caused by the assembly of the split rotor.

In a first test, the roughly balanced rotor was accelerated to a rotational speed of 75 000 rpm. Hereby, the amplitude of the synchronous rotor vibration at both compressor and turbine stayed below 2 µm. This first balancing step was however not sufficient to safely pass the critical speeds.

4. CONCLUSION AND FUTURE WORK

This paper decribes the design aspects of air bearings for an ultra-miniature gas turbine. The first part gives an overview of the turbo-shaft setup designed for low-temperature testing of the aerodynamic and rotordynamic performance. A next section elaborates on the air bearing design and the labyrinth seal. Finally, the balancing strategy and the results of a first test-run are explained.

Future work will focus on the balancing of the rotor. A well-balanced rotor will allow supercritical operation and can therefore give information about critical speeds and stability limits. A comparison of this test data with the predicted values can then serve as a validation of the rotordynamic modelling process.

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