Nonlinear rotordynamic-thermal analysis of micro gas turbines

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Summary. A compliant tilting pad air bearing concept has been proposed as a solution for low power micro gas turbines. In this paper, the nonlinear dynamic models for this bearing concept are extended with a thermal model. With simulations and experiments it is demonstrated that the thermal expansion of the bearing journals may negatively affect the power loss, and the nonlinear rotordynamic response of the rotor and these bearings. Therefore, designing these bearings for a micro gas turbine requires both a nonlinear rotordynamic model and a thermal model of the micro turbine.

Introduction

The smallest commercial micro gas turbines (MGTs) deliver about 30 kWe. However, applications with even smaller MGTs will reach the market soon, for instance in the fields of combined heat & power systems [2] (2017) and small hybrid power plants [1] (2019).
The objective of this research is to develop an aerodynamic bearing solution to support the rotor of MGTs in the 3-5 kWe output power range. Although aerodynamic bearings are already commonly used in larger MGTs, it is not simply a case of scaling down because: 1) A smaller rotor has to run at higher rotation speeds to achieve optimal cycle performance. 2) The operational temperatures remain approximately the same, requiring a system that can handle a higher heat flux.
To comply with the rotordynamic and thermal requirements the tilting pad bearing concept shown in Figure 1 has been proposed. It has been made compliant to thermal expansion by allowing for radial displacement of the pivot points of the pads. Nonlinear 2D and 3D rotordynamic models of this concept have been described in [3]. The non-linearity stems from the aerodynamic force in the air gap between journal and pad. Depending on the bearing design, this force may cause nonlinear behaviour like: self-excited oscillations (pad flutter), and (quasi-)periodic or even chaotic responses to imbalance loads.
In this paper, these nonlinear rotordynamic models are extended with a thermal model. Thermal expansion of the journal diameter may negatively affect performance (power loss), and the rotordynamic response of the bearing.

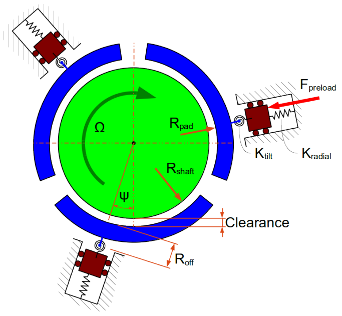

Figure 1: Compliant Tilting Pad Air Bearing (CTPAB) concept.

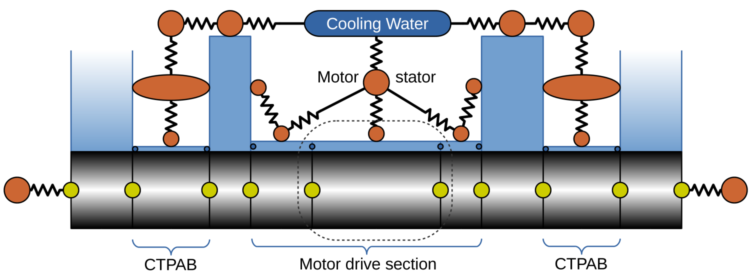

Figure 2: Thermal model of air bearing test rig with two 15 mm CTPAB bearings and a 120 mm long rotor. The dashed line encloses a single rotor section with 5 thermal nodes.

Thermal model

From a thermal point of view a micro gas turbine is a collection of interconnected heat sources and sinks. The temperature that occurs at some location, say the bearings, therefore depends on the temperature or power of the sources and sinks and the resistance to heat flow of the intermediate structure. As a consequence a thermal model of the whole micro gas turbine is necessary to predict the temperature distribution in the micro gas turbine that has to be designed.
The temperatures of the main components in an air bearing test setup have been estimated with a thermal network model, schematically shown in Figure 2. A proper placement of the thermal nodes and links is needed, together with an accurate estimation of, respectively, thermal capacity and conductivity. For the rotor segments this is achieved assuming that the perceived conductivity of the air gap is low compared to the metal parts, and that the heat conduction through the air gap can be modelled with convection coefficients at the air contact surfaces. This leads to the following set of coupled differential equations for the temperature in a single rotor section with constant radius R_r surrounded by a turbulent air gap of constant thickness:

$$c_r\rho_r A_r \frac{\partial T_r}{\partial t} = k_r A_r \frac{\partial^2 T_r}{\partial x^2} - 2\pi R_r \alpha_r \left(T_r - T_a\right) + \dot{q}_r \tag{1}$$

$$c_a\rho_a A_a \frac{\partial T_a}{\partial t} = k_a A_a \frac{\partial^2 T_a}{\partial x^2} - c_a \dot{m} \frac{\partial T_a}{\partial x} + 2\pi \left(R_r \alpha_r \left(T_r - T_a\right) + R_s \alpha_s \left(T_s - T_a\right)\right) + \dot{q}_a \tag{2}$$

where x is the rotor axial coordinate, $T\left(x,t\right)$ is the temperature, c is the specific heat, ρ is the density, k is the (effective) axial heat conduction coefficient, α is the convection coefficient, A is a cross section area, $\dot{m}$ is a constant axial mass flow

through the air gap, and $\dot{q}$ is internal heat generation per unit length. The subscripts r, a, and s indicate properties of rotor, air gap, and stator, respectively.
Applying the Finite Volume method to equations (1) and (2) converts them into a matrix equation that can be connected to the thermal network model using the nodes at the boundary of the rotor section and its air gap. Since the time scales of the thermal model (seconds, minutes) and the rotordynamic model (milliseconds) are very different, as a first approach the initial cold start condition and the long term steady-state temperature distribution can be used as limiting cases to evaluate the effect of temperature changes on the rotordynamic performance of the system. For the long term steady-state the left hand sides of the equations become zero. The internal temperature distribution of a single rotor section is then completely determined by its boundary temperatures and internal heat generation.
The advantage of this approach is that the equivalent 5-node thermal network of a single rotor section (e.g. the section enclosed by the dotted line in Figure 2) can now be derived directly from its geometry, material properties c, ρ, k, and engineering quantities like α and $\dot{q}$ for which (semi)empirical relations often exist [4].

Thermal effects on rotordynamic performance and stability

Thermal expansion of the rotor may have important effects on the rotordynamic performance of CT-PAB bearings. Figure 3 shows the static component of the radial pad displacement that has been obtained from simulations and experiments at several rotation speeds. Omitting the thermal expansion, as predicted by the thermal model of Figure 2, leads to large differences between simulation and experiment, especially at higher speeds. Furthermore, there is a large difference in predicted friction power loss. At 170 krpm the simulation without thermal expansion predicts 19 W, whereas the simulation with thermal expansion predicts 35 W per bearing.

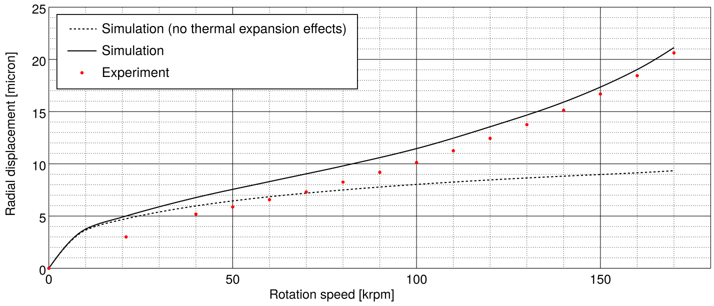

Figure 3: Simulated and measured radial displacement of tilting pad at different running speeds.

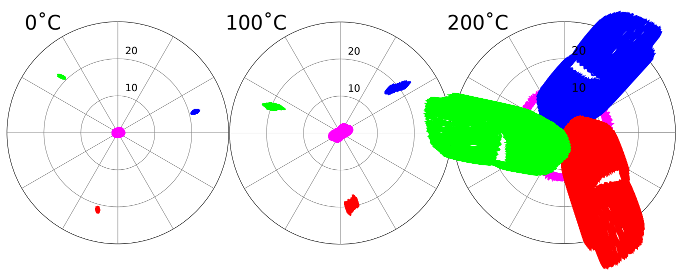

Figure 4: Effect of thermal expansion on nonlinear dynamic response. Steady-state trajectories at 240 krpm of the journal (magenta), and the centre points of the tilting pads (red, green and blue).

Figure 4 shows how the thermal expansion of the journal affects the nonlinear dynamic response of a 2D bearing model. The figure shows the steady-state trajectories of the centre point of the journal (middle) and the centre points of the three tilting pads for three different operating temperatures. In all three cases the rotor spins at 240 krpm and is loaded by a rotating imbalance of 0.1 gmm. In terms of the parameters of Figure 1 the bearing has the following properties: $R_{shaft} = 7.52$ mm, $C = 30$ µm, $R_{pad} = R_{shaft} + C$, $R_{off} = 4$ mm, Ψ= 20°, $K_{radial} = 0.3$ N/µm, $K_{tilt} = 0$ Nm/rad, $F_{preload} = 5$ N, and the length of the bearing is 15 mm.

The response of this particular bearing design clearly deteriorates with increasing temperature. This, however, is not a general rule, because there are also designs that hardly react to a temperature change, and even cases that show an improved dynamic response with increasing temperature.

Conclusions

With simulations and experiments it has been demonstrated that the friction power loss and nonlinear dynamic response of a compliant tilting pad air bearing can be strongly affected by a high operational temperature. Therefore, designing these bearings for a micro gas turbine requires both a nonlinear rotordynamic model and a thermal model of the micro turbine.

References

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