HIGH-SPEED ELECTRICAL MOTOR/GENERATOR SUPPORTED BY AIR BEARINGS

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Abstract: This paper reports on the development of a high-speed electrical machine which can be used as motor or generator for application in micro heat engines. The machine has a rated power of 200 W at a rotational speed of 600 000 rpm and is supported by self-acting air bearings. The design process outlined in the paper focuses on aspects such as electromagnetic design, bearing design, rotordynamic behaviour and thermal modelling. The outcome of this global design process is the result of a close interaction between all these different aspects. In order to verify the obtained design results, an experimental setup is realised. Currently, the individual components of this setup have been manufactured and commissioning is scheduled in the near future.

Keywords: electrical motor/generator, high-speed, air bearings

INTRODUCTION

High-speed electrical machine technology comprises a key component in various energy-related application fields, such as micro gas turbines and in a broader context, micro compressors or micro expansion turbines. All these applications require a high-speed electrical machine to convert the mechanical power into electrical power or vice versa. The bearings fitted in these machines should be able to cope with the high rotational speeds in combination with a long lifetime and the possibility to operate at elevated temperatures. The work of this paper is therefore concerned with the development of such a motor/generator prototype including bearings in order to demonstrate the feasibility thereof.

The work is undertaken as a joint development between the company Celeroton AG and the Katholieke Universiteit Leuven. Both partners have demonstrated their expertise in respectively high-speed electrical drive systems and high-speed air bearing technology [1,2]. This machine prototype might therefore be regarded as a device that combines both technologies.

Based on the machine design specifications (200 W at 600 000 rpm), the global design process starts with the electromagnetic design and optimisation leading to the initial dimensions of the rotor (magnet and retaining sleeve) and stator (winding and back iron). The second aspect deals with the design of the self-acting air bearings. The main concern here is the dynamic stability problem of the bearings at high speed. Hereafter, the rotordynamic behaviour is dealt with. This design task predicts the critical speeds, stability threshold and bending modes of the rotor-bearing system and provides feedback to the motor and bearing design. Finally, the transient and steady-state thermal aspects are dealt with through the formulation of an equivalent thermal circuit.

Finally, an experimental setup is realised in order to verify the obtained design results.

MOTOR/GENERATOR SPECIFICATIONS

Based on the information and experience gathered from previous projects in the micro turbomachinery domain, the design specifications of the prototype machine were set to 200 W electrical output and a nominal rotational speed of 600 000 rpm. The outer diameter of the machine is kept at 19 mm.

The bearings of the prototype have to provide a durable long-life solution at the design speed of the machine and up to elevated operating temperatures. Air bearings are for this purpose an interesting candidate on condition that the dynamic stability issue is tackled. Given the application fields in mind, the choice for self-acting air bearings is evident as this will result in a more autonomous system.

In order to obtain a good overall efficiency of the machine, the various loss contributions (electromagnetic, bearing and windage losses) have to be kept minimal.

The machine prototype discussed in this paper concerns a free-running system without any functional components installed on the shaft ends. The main goal is to demonstrate the feasibility of a high-speed electrical machine on air bearings. The data obtained from experimental results will allow to validate the modelling tools used for the design and optimisation process.

DESIGN ASPECTS

Electromagnetic design

The electromagnetic part of the rotor of the motor/generator consists of a cylindrical permanent magnet and a retaining sleeve, the stator consists of a slotless winding and an amorphous iron core. The electromagnetic design is integrated into an analytic optimization routine according to [3]. Constraints for this optimisation are the mechanical strength of the rotor sleeve (titanium grade 5) and the rotor magnet (Sm_2Co_{17}), and the outer dimensions of the electromagnetic part of the motor/generator. The diameter of the electromagnetic part of the motor/generator is set to 11 mm, leaving 3 mm in radius for the cooling sleeve to reach the 19 mm outer diameter. The active length of the machine is set to 9 mm, which has been found in an iteration loop with the rotordynamic design.

The optimization routine minimizes the losses for the rated operating point of 200 W and 600 000 rpm. The resulting losses and the efficiency are specified in table 1. The rotor diameter of the motor/generator results in 5.5 mm.

Table 1: Electromagnetic design results.

Symbol	Quantity	Design value
P	Rated power	200 W
n	Rated speed	600 000 rpm
P_{Fe}	Iron losses	0.4 W
P_{Cu}	Copper losses	22.1 W
$P_{f,m}$	Frictional losses	7.1 W
η	Efficiency	87.1%

Bearing design

The rotor is supported at both ends by two identical self-acting journal bearings with a wave-shaped film height geometry, as described in [2]. The design parameters and characteristics are in table 2. The dynamic stability problem typically encountered at high speed is tackled by providing a flexible, damped support for the bearing bushes. A theoretical background and a more elaborate description of this stabilisation technique can be found in [2].

Table 2: Journal bearing design parameters and characteristics (per bearing) at a rotational speed of 600 000 rpm and perturbation frequency of 1 rad/s.

Symbol	Quantity	Design value
r	Bearing radius	2.5 mm
L	Bearing length	5 mm
c	Mean radial clearance	8 µm
k_{ii}	Direct stiffness	0.44 N/µm
$P_{f,j}$	Frictional loss	3.6 W

Since no functional axial loading occurs in this machine prototype, the thrust bearing configuration consists of two relatively small self-acting bearings placed at both sides of a thrust disc (figure 1). Pressure is generated by means of a classical Rayleigh-stepped geometry [4].

The total frictional loss contribution of both journal and thrust bearings amounts to 9.8 W at the design speed of 600 000 rpm.

Table 2: Thrust bearing design parameters and characteristics (per bearing) at a rotational speed of 600 000 rpm.

Symbol	Quantity	Design value
r_i	Inner radius	3 mm
r_o	Outer radius	4.5 mm
h	Clearance height	20 µm
d_{recess}	Recess depth	34 µm
l_{recess}	Recess length ratio	0.55
w_{recess}	Recess width ratio	0.5
N_{pad}	Number of pads	6
W_{max}	Max. load	0.8 N
$P_{f,t}$	Frictional loss	1.3 W

Rotordynamic behaviour

The model used for the prediction of the rotordynamic behaviour is shown in figure 1. The input for the model consists of: (i) rotor mass properties of table 3; (ii) stiffness and damping coefficients of the bearings at different combinations of rotational speed and perturbation frequency; and (iii) support parameters. In this case, the tilt effect of thrust bearings on the rotordynamic behaviour is neglected. The output consists of the critical speeds, stability performance and unbalance response.

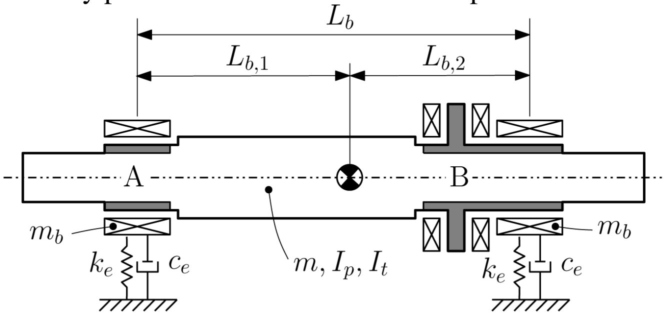

Fig. 1: Bearing configuration and rotordynamic model.

Table 3: Rotor mass properties.

Symbol	Quantity	Design value
m	Rotor mass	4.34 g
I_t	Transv. moment of inertia	468.8 gmm^2
I_p	Polar moment of inertia	18.1 gmm^2
$L_{b,1}/L_{b,2}$	Distance between bearing and mass centre	14.1/13.5 mm
m_b	Bush mass	0.28 g

As mentioned, a flexible, damped bearing support is introduced to stabilise the rotor-bearing system. In the bearing configuration of the described motor/generator, this support consists of a bearing bush with mass m_b, which is supported by means of elastomeric O-rings. These O-rings provide a support stiffness value k_e and damping value c_e.

The following design guidelines have to be fulfilled in order for the 'external' damping to be effective: (i) the bush mass must be considerably smaller than the rotor mass (m_b < m/10); (ii) the support stiffness may not exceed the bearing stiffness; and (iii) a proper amount of support damping has to be introduced [2].

For a support stiffness of 0.1 N/µm and support damping of 30 Ns/m, the damping ratio of all modes (eight in this case) is plotted as a function of the speed (see figure 2). Satisfactory performance in terms of stability is obtained over the entire operational speed range. Additional simulations show a negligible effect of the negative magnetic stiffness (-730 N/m) and of the transverse motor torque (max. 0.25 mNm) on the rotordynamic behaviour.

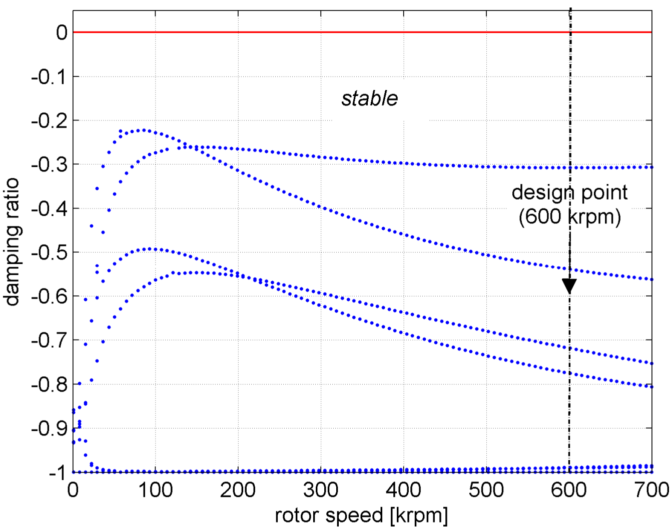

Fig. 2: Damping ratio as a function of the rotational speed.

The mechanical layout of the rotor is designed in such a way that the first bending mode lies above the maximum rotational speed (including a safety margin). Assuming a rigid connection between the rotor parts and by neglecting the bearing stiffness, the first bending mode is estimated at 13.95 kHz (see FEM-calculation in figure 3).

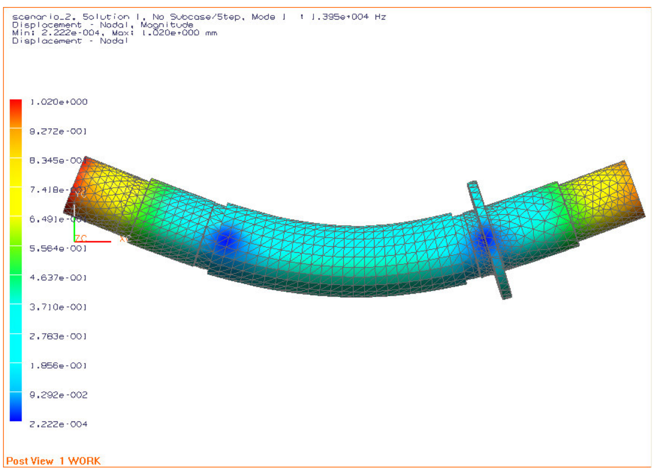

Fig. 3: First bending mode of the rotor at 13.95 kHz.

Thermal behaviour

In order to predict the temperature rise at various locations within the system, a thermal model is composed. This model starts from a lumped-parameter formulation consisting of different nodes interconnected with thermal resistors. Each node has a given temperature and thermal capacitance. The different loss contributions are treated as individual power loss sources.

To model the thermal effects within the gas film, the following assumptions are made: (i) the direct effects of the temperature distribution on the pressure distribution within the gap are insignificant [5]; (ii) in the normal direction of the gas film the heat is transferred solely by conduction; (iii) the heat generated in the bearing gap by the dissipative component of the friction force is modelled as one single heat source.

The thermal model is solved by using Matlab/Simulink when starting from ambient temperature conditions and for the situation of nominal speed and a torque of 1 mNm (one third of the rated torque) over a period of 180 s. Figure 4 shows the output of the model in terms of temperature rise at the various locations within the system. The increase in temperature is maximal at the air gap between rotor magnet and stator (ca. 180 ºC), but the rotor with ca. 160 °C remains still below the temperature limit of the magnet material (300 °C). At the bearing locations the temperature rises to respectively 100 and 120 ºC, which also is below the limitations of the bearing materials and supporting O-rings.

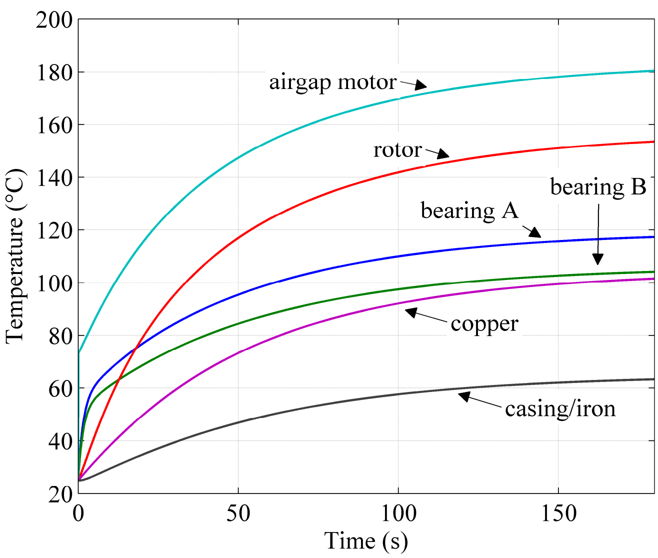

Fig. 4: Temperature increase at various locations within the system.

EXPERIMENTAL VERIFICATION

An experimental setup is designed and the parts described in the previous sections have been manufactured: a rotor including the electromagnetic part and the air bearing seats. The electromagnetic part of the stator is integrated into a cooling housing, where also the bearing flanges will be attached to. Finally, the cooling housing is attached to a holder, and pressurized air can be supplied to the cooling housing if forced cooling is required. A CAD-view of the entire experimental setup is depicted in figure 5, and in figure 6 the manufactured stator cooling housing and the rotor without bearing parts are shown.

After assembly and commissioning, the machine will be tested in motor mode in order to assess the dynamic stability, thermal behaviour and to characterise the various loss sources present in the system.

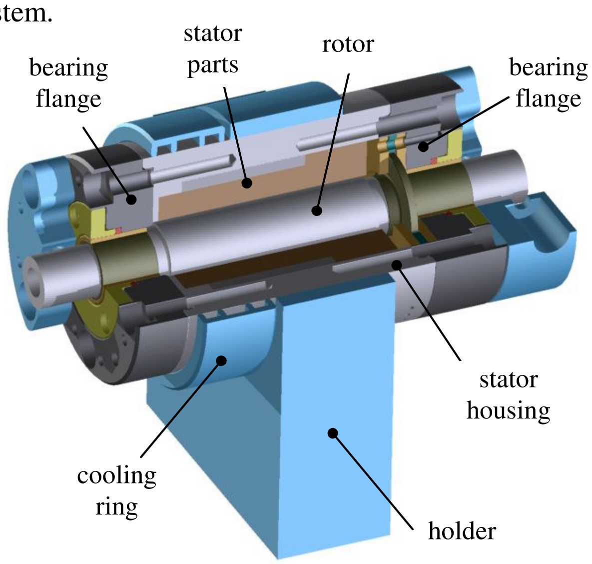

Fig. 5: CAD cut-away view of the experimental system.

Fig. 6: Stator cooling housing and rotor (without bearing parts) of the experimental setup.

CONCLUSION AND PERSPECTIVES

Integrated high-speed electrical machine and air bearing technology is a key component in various micro-turbomachinery applications. Electromagnetic, rotordynamic and thermal design analysis shows the feasibility of a 600 000 rpm, 200 W motor/generator supported by flexibly, damped air bearings. The design results will be verified with an experimental setup which is under construction.

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