Design of a Highly Reliable Fan with Magnetic Bearings

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***Abstract*—Long term analysis at ebm-papst St. Georgen GmbH & Co. KG showed that bearing failures are responsible for 90% of compact fan breakdowns. A magnetically levitated impeller has no contact with the stator and therefore the life expectancy can be essentially increased. This paper presents the design of a magnetically levitated fan. To meet the low-cost requirement, passive magnetic bearings (PMBs) are used to stabilize the radial and tilt deflections of the rotor. By applying an optimized viscoelastic support to the stator, sufficient damping is introduced to the passively stabilized degrees of freedom. The optimization of the stiffness and damping is discussed and the design of the key components, namely the PMB, the active magnetic bearing (AMB) and the passive damping device, is described. Finally the built prototype and measurements are presented.**

I. Introduction

A. Reliability

The reliability of electronic systems essentially depends on the operating temperature. Fans are the common thermal management solution for cooling of electronic systems. However, wear in the bearings reduces the life time of conventional fans. A study by Tian [1] identified bearing defects as major failure cause of fans.

To estimate the rating life of roller bearings, ISO 281 specifies methods for calculation of the basic dynamic load rating [2]. As fans usually have a very light load, the use of the ISO 281 standard is not suitable for life estimation [3]. The reliability of the roller bearings instead depends mainly on the retention of its lubricant (grease or oil). Grease life equations can be found in [4].

Fan manufacturers usually specify the lifetime when ten percent of the population fails under the specified test conditions (e.g L_{10} at 40°C). However, care must be taken when the lifetimes L_{10} of different manufacturers are compared. The differences are briefly discussed below.

To cut down the test time, accelerated life tests (ALT) are used to get the life expectancy. As the bearing temperature is the most important acceleration factor for the roller bearing life, ALT are done at the highest possible operation temperature. The industry standard IPC-9591 was developed in 2006 in order to standardize fan's performance parameters, including reliability [5]. The norm suggests an acceleration factor of 1.5 for 10 Kelvin temperature rise ($AF|_{IPC9591} = 1.5/10\,\mathrm{K}$). Nevertheless, often unacceptably high acceleration factors are used by fan manufacturers [6]. The acceleration factor used at ebm-papst St. Georgen GmbH & Co. KG is even below the value suggested by the norm ($AF|_{ebmpapst} = 0.7/10\,\mathrm{K}$), leading to more conservative life time predictions [7].

Also the test strategy of the reliability test should be considered if the fan life is predicted. The *zero-failure test strategy* determines the test time for each fan dependent on the sample size n. The test time where the n samples must not fail depends on the estimated parameters of the Weibull distribution. The *test with failure strategy* leads to longer test times, as it stops when a predefined number of defects has been reached. As it is not necessary to assume the Weibull distribution parameters this strategy can be considered more reliable.

B. Measures for Improved Reliability

1) Nonrotating Air Moving Devices: One approach to improve the reliability of air moving devices is to use an oscillating beam instead a rotating fan blades. The vibration can be realized by a cantilever beam, which is actuated by a piezoelectric element [8] or by the magnetic forces of coils [9]. The beam is excited at the first or second eigenfrequency. However, conventional fans with comparable installation space achieve considerably higher pressures and flow rates. Nevertheless, a vibrating beam might be an alternative for applications demanding rather low flow rates.

2) Magnetic bearings: A magnetically levitated fan could be the solution to reach a durable air cooling device with high cooling power. This approach will be investigated in this paper.

II. Magnetic Bearing Concept

As magnetic bearings are employed primarily in high-tech applications, their realization in a compact fan poses a great challenge. Commercial success requires concepts that are only marginally more expensive to manufacture than traditional fans. The magnetically levitated fan uses two simple passive magnetic bearings (PMBs) to meet this low-cost requirement, see Fig. 1. The upper and lower PMB stabilize four degrees of freedom: the radial and tilt motions of the rotor. Due to Earnshaw's Theorem [10], the axial direction is unstable and therefore controlled by an active magnetic bearing (AMB). The remaining degree of freedom (rotation of the impeller) is actuated by a permanent magnet synchronous motor (PMSM). The cross section of a magnetically levitated fan that was jointly developed by ebm-papst St. Georgen, the Institute of Electrical Drives and Power Electronics at the Johannes Kepler University Linz and the Linz Center of Mechatronics (LCM) is shown in Fig. 1.

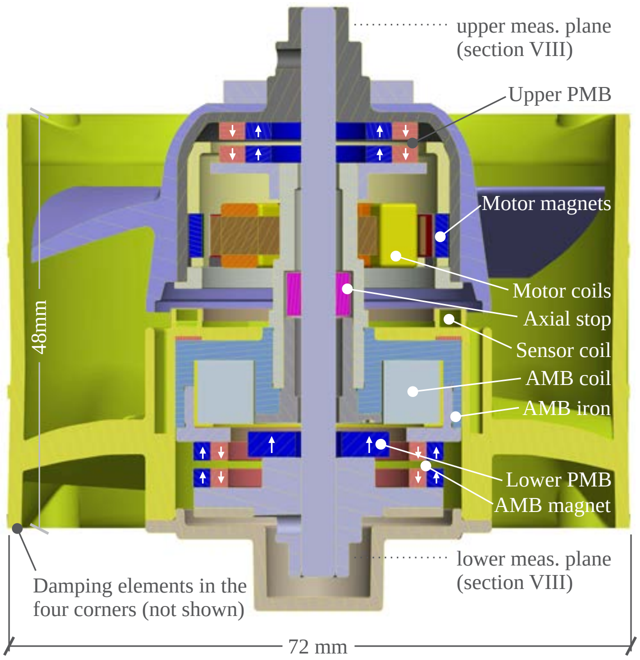


Figure 1. Cross-section of magnetically levitated fan. The radial and tilt directions are passively stabilized by two permanent magnetic bearings (PMB). Stabilization of the axial degree of freedom is achieved by an active magnetic bearing (AMB).

Each PMB consists of two axially magnetized permanent magnets on the stator and the rotor. The shown alignment of the magnets causes radially stable magnetic forces. All PMB magnets are mounted on a back iron which increases the radial stiffness. However, the value of the destabilizing axial stiffness increases disproportionately [11], [12]. The back iron ensures that the magnetic field of the PMB is kept inside the fan. The stacking of the magnets allows to achieve a higher radial stiffness per volume ratio [13], [14].

In Fig. 1 the rotor is shown in the levitating, axially force free position. Here, the sum of the axial forces of the upper and lower PMB and the axial force of the currentless active magnetic bearing (AMB) is zero. Only small positive and negative currents are therefore necessary to keep the rotor levitating. The axial position of the impeller is measured by an eddy current sensor.

Fig. 2 depicts the tool which was built to assemble the magnetically levitated fan. It is equipped with a load cell and two position sensors. This device enables to mount the rotor PMB parts so that the lower and the upper air gap of the axial stop are of same size in the force free position. This has proven to be a good method to compensate for the force tolerances due to variations in the PMB parts (magnet material, magnetization, magnet size, magnet position, magnet shape, etc.).

Figure 2. Assembly tool for magnetically levitated fan.

III. Passive Magnetic Bearing (PMB)

Although the permanent magnets of the upper PMB are mounted on a back iron, the axial force and the axial and radial stiffnesses can be calculated analytically. A novel calculation method based on the method of images for the stiffnesses is described in [12].

IV. Active Magnetic Bearing (AMB)

AMBs with one degree of freedom using permanent magnetic bias flux can be distinguished in two main groups.

A linear force per current relationship is achieved by placing the coil in the magnetic field of the permanent magnet. This arrangement corresponds to a voice coil except that the moving part in the magnetic bearing is the magnet (and the attached iron) and not the coil. These types of actuators have no negative stiffness, which makes them well suited for rotors with passive permanent magnetic radial bearings [15]. An actuator providing both axial forces and torque is shown in [16].

In the second AMB concept, which is used in the presented magnetically levitated fan, the coil is embedded in iron. The magnetically levitated fan uses a space-saving configuration where the magnetic circuits of the AMB and the lower PMB are combined. Fig. 3(a) shows the general geometry of this active/passive magnetic bearing. A drawback of the design is the rather high reluctance of the AMB coil. However, the design was chosen due to the simple construction and the easy assembly of rotor and stator.

In the available space this arrangement achieves a higher axial force per ampere winding ratio k_Θ compared to the voice coil type actuator. The voice coil actuator causes no magnetic forces if the AMB coil is not energized. Contrarily, the AMB concept of Fig. 3(a) has a negative axial stiffness, a negative tilt stiffness and a possibly negative radial stiffness[1]. Furthermore, all these stiffnesses change with the current of the AMB coil. As the AMB iron of the prototype fan was not laminated or built from soft magnetic composite (SMC), there is additionally a strong frequency dependence of the force per ampere winding ratio k_Θ due to eddy currents. The effect on the closed loop stability was investigated by Panholzer et al. [17].

[1]The radial stiffness with the lower PMB is always positive. However, the AMB alone can be radially stable or unstable depending on the actual geometrical dimensions of the AMB.

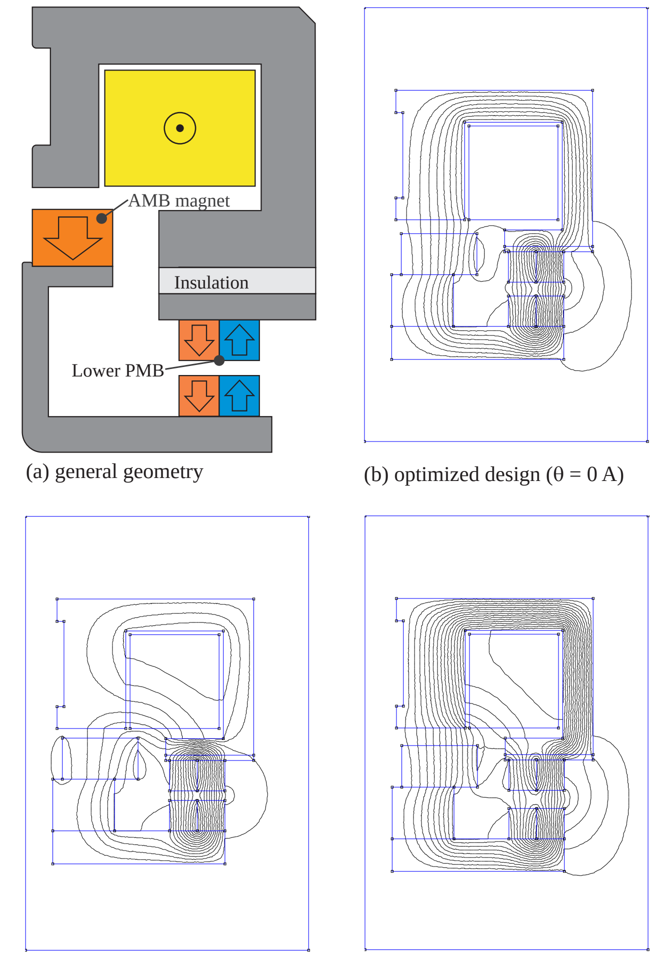
A proper design of the AMB involves the fulfillment of several criterias:

- low copper losses at rotor start
- high stability or at least low instability in radial direction
- low decrease of the radial stiffness when the coil is energized
- low overall height of the AMB
- low magnet volume

A multi-objective optimization using genetic algorithms was carried out to find a good compromise between these conflicting objectives using the software MagOpt [18]. All geometric parameters of the AMB have been used as design variables. For the optimization process a short calculation time of all stiffness values is crucial. Since the original rotational symmetry is lost in case of a radially deflected or tilted rotor, the radial stiffness and the tilt stiffness are usually calculated with 3D-FE software. But these time-consuming calculations would prevent fast optimization results. Therefore, fast approximate calculations of the radial and tilt stiffnesses which need only 2D finite element analyses were used. This method is described in [19]. The 2D finite element calculations were done in FEMM [20].

Fig. 3(b) shows the selected Pareto-optimal design which was realized in the magnetically levitated fan. The field lines are drawn for zero magnetomotive force and the rotor is shown in the force free rotor position, i.e. the sum of the axial forces of the AMB and the two PMBs is zero. The insulation thickness between the AMB and PMB is zero in this design.

To start from the upper mechanical stop, the axial force of the AMB has to be weakened by the current, shown in Fig. 3(c). Contrarily, to start from the lower mechanical stop the flux density in the air gap above the AMB magnet is increased by the coil, shown in Fig. 3(d).



(a) general geometry

(b) optimized design (θ = 0 A)

1. (c) start from upper stop

(d) start from lower stop

Figure 3. Cross section of the active magnetic bearing (AMB) and the lower passive magnetic bearing (PMB) of the magnetically levitated fan.

V. Electrical Drive

The cross section of the electrical drive was optimized with MagOpt [18] and is shown in Fig. 4. The desired speed was 16300 rpm, where the load torque of the impeller is circa 13 mNm. The motor is operated with block voltage commutation without an angle sensor by detecting the zero-crossing of the back-emf. This operation mode was implemented in a motor simulation within the MagOpt project using the modeling strategy as suggested by Weidenholzer et al. [21]. A multi-objective optimization with the following objectives was carried out:

- low instability in radial direction (k_{drive})
- minimum losses at load point
- low motor height (including the end windings)
- low torque ripple (at load point, gained from block voltage commutation simulation)
- low magnet volume

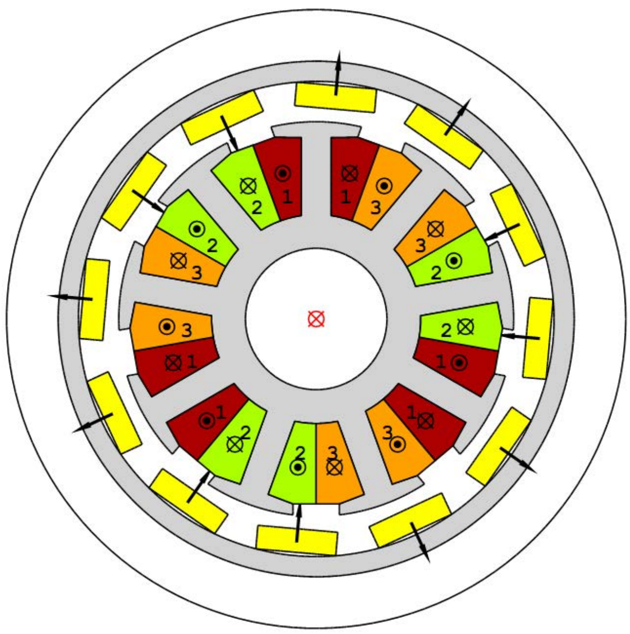


Figure 4. Cross section of the electrical drive of the magnetically levitated fan (9 slots, 6 poles, 2 magnets per pole)

VI. Optimization of PMBs and Damping Elements

A. Mathematical Model

For magnetic bearing systems using passively stabilized degrees of freedom, the optimization of the stiffnesses of the

PMBs and viscoelastic damping elements (VDE) is essential in the design process. Marth et al. showed that an analytical model is well suited to calculate the rotordynamic behavior of a passive magnetic bearing system including viscoelastic damping elements [22]. All stiffness and damping coefficients of the PMBs, the AMB, the motor and the viscoelastic dampers have to be considered.

The equations of motion were derived for the system illustrated in Fig. 5, where k_{B1} and $k_{\varphi B1}$ denote the radial and tilt stiffness of the upper PMB, k_{B2} and $k_{\varphi B2}$ denote the radial and tilt stiffness of the lower PMB and the AMB when it is powered off. Each PMB by itself is unstable against tilting ($k_{\varphi B1} < 0$, $k_{\varphi B2} < 0$). The positive tilt stiffness of the rotor is achieved by the positive radial stiffnesses ($k_{B1} > 0$, $k_{B2} > 0$) and the distance between the PMBs. The destabilizing effect caused by the radial reluctance force between motor magnets and the motor lamination stack is considered using k_{drive} ($k_{drive} < 0$). The rotor and the stator can move in the radial directions and incline about both diametral axes. The axial movement of the stator and the rotor as well as the rotation of the stator were neglected. Thus, eight degrees of freedom remain to describe the position of the rigid rotor and the rigid stator. The nonlinear equations of motion were linearized around the rest position for a fixed angular speed Ω.

B. Viscoelastic Damping Elements (VDE)

In order to damp the vibrations of the rotor, the stator was mounted on four VDEs placed in the corners of the fan housing. The dynamic behavior of the VDEs can be calculated as product of the VDE static stiffness with the dynamics of the viscoelastic material [23]. The oscillation frequency and the temperature have the biggest influence on the material properties[2]. Frequency dependent stiffness and damping values $[k_{VDE}(\omega), d_{VDE}(\omega), k_{\varphi VDE}(\omega), d_{\varphi VDE}(\omega)]$ can be used to describe the dynamics of the VDEs at one temperature, where k_{VDE} and d_{VDE} denote the radial stiffness and radial damping, $k_{\varphi VDE}$ and $d_{\varphi VDE}$ denote the tilt stiffness and tilt damping and ω is the excitation frequency. However, no transient simulation is possible with this model. Therefore, the frequency-dependence was implemented by using the generalized Maxwell or Wiechert model [24], which introduces additional state variables to the linear system in order to model the frequency-dependence.

C. Optimization with MagOpt

The mathematical model was implemented and optimized in MagOpt [18]. For a constant angular speed Ω the system is linear and has around 70 states[3].

The deflection of the rotor relative to the stator is the main criteria for the stability of a magnetically levitated fan. Table I lists the considered disturbance forces and the corresponding objectives. Each objective was calculated at $-10\,°C$ and $+60\,°C$ damping element temperature. The objectives of the optimization were set to the worst case value. In this paper the deflections due to unbalance will be discussed in more detail. A comparison with measurements is carried out in section VIII.

Since the axial force of the upper PMB has to compensate the axial force of the lower PMB and the axial force of the currentless AMB, the PMBs and the AMB have to be adjusted appropriately. Therefore feasible upper and lower bearing combinations were identified before starting the optimization. The upper PMB was calculated analytically [12], while the AMB and the lower PMB were calculated with FEMM (as described in section IV). The stiffnesses of every suitable design were saved to a table. Every table row represented

[2]Static pre-load or nonlinear material behavior due to large strain amplitudes were not considered in the model.

[3]The exact number depends on the number of state variables used to describe the viscoelastic material

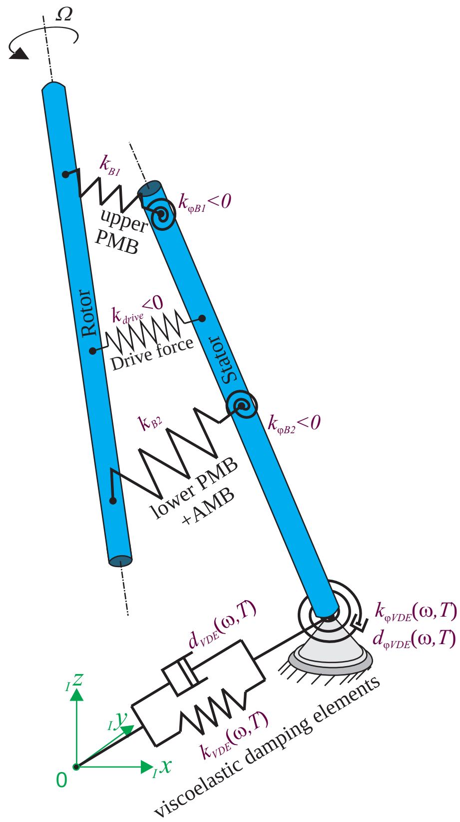


Figure 5. Schematic figure of the dynamic model. The rotor and stator can move in radial direction and incline about both diametral axes of inertia.

Table I
CONSIDERED DISTURBANCE FORCES AND OBJECTIVES FOR THE OPTIMIZATION OF THE PASSIVE MAGNETIC BEARING AND DAMPING ELEMENT STIFFNESSES

	Force application	Force type	Objective
1	stator	external acceleration	fulfillment of the vibration norm IEC 60721-3-4 Class 4M5
2	rotor	force step	high transient damping ratio
3	rotor	unbalance force	low deflection in the planes of the PMBs even if the rotor speed is equal to the frequency of a rigid body mode

a design with different radial stiffnesses but where the axial forces add up to zero. In the optimization the bearings were varied by using these precalculated table values. Furthermore, the geometry and material of the VDEs were altered during the optimization process.

VII. Prototype

Fig. 6 depicts the built magnetically levitated fan prototype and its basic data. The power electronics of the prototype were assembled in an external housing, but integration within the fan housing is possible. For levitation of the rotor (speed 0 rpm), the power consumption of the whole system is 1.7W.

Nominal data	
Operating voltage	48 V
Nominal speed	16.300 rpm
Power consumption	
...for levitation (standstill)	1.7 W
...at nominal speed	30 W

Figure 6. Prototype of magnetically levitated fan.

VIII. Measurements

In this section the measured rotor deflections due to unbalance and magnetic tolerances are compared to calculated values. Magnetic tolerance means that the magnetic axis, which is given by the position where the sum of the magnetic forces is zero, does not coincide with the symmetry axis of the rotor. The magnetic axis further depends on the rotor angle. Magnetic tolerances as well as unbalance are the major internal excitations in radial and tilt direction [25].

In a first step the rotor was balanced as good as possible. This was done by measuring the forces on the stator corresponding to the method described in [26, pages 24-28]. The radial rotor deflections caused by the remaining unbalance and by the magnetic tolerances were measured by laser distance sensors, see Fig. 7. The two measuring planes are shown in Fig. 1. Since the deflections at low speeds ($\Omega \approx 0\,rpm$) are caused by magnetic tolerances they cannot be compensated by balancing weights. As shown in Fig. 7, the balancing worked well since there are no distinct resonances.

The rotor deflections were also calculated by the mathematical model described in section VI-A. However, the remaining unbalance and magnetic tolerance of the rotor are initially unknown. Therefore these excitations were identified by using an optimization with genetic algorithm. The calculated rotor

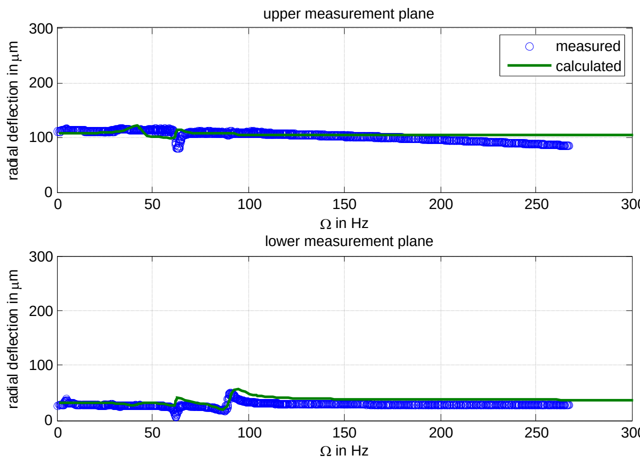


Figure 7. Measured and calculated rotor deflections of the balanced rotor.

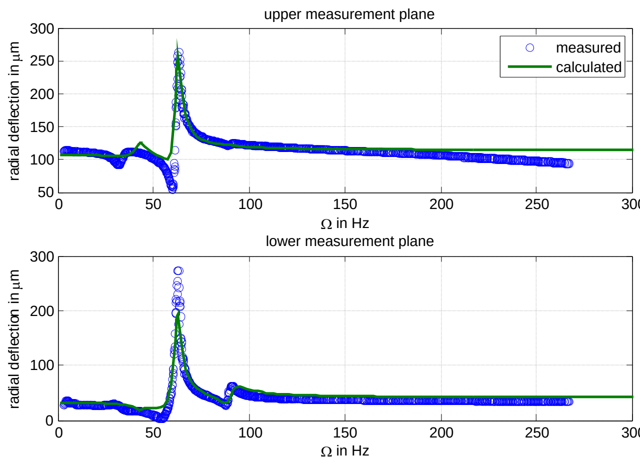


Figure 8. Measured and calculated rotor deflections. An additional weight causing an unbalance of $0.5\,$gmm was added to the rotor top.

deflections are also shown in Fig. 7. It can be seen that the absolute value and phase of the unbalance and magnetic tolerance could be found and that the mathematical model is able to describe the behavior of the rotor.

In a second step a weight was added to the rotor top which caused an unbalance of $0.5\,$gmm. This unbalance mass was also considered in the mathematical model. In Fig. 8 the measured rotor deflections were compared to the calculated ones. Both Fig. 7 and Fig. 8 show that the measured deflections decline at higher rotor speeds, especially in the upper measurement plane. This effect, which might be caused by aerodynamic forces, was not considered in the mathematical model.

IX. Further Work

The focus of the development is to enhance the life expectancy of compact high speed fans under adverse environmental conditions. The goal is to achieve less than 2.5 percent failures in 10 years. Experiments must be made to prove if this ambitious goal can be met. In particular, the longtime stability of the viscoelastic damping elements and the electronics have to be tested. Before the tests can be started the damping elements must be integrated in the housing. To keep dirt and dust away from the air gaps of the passive magnetic bearings and the electrical drive further design changes might be necessary.

X. Conclusion

This paper describes the concept and design process of a magnetically levitated fan and its components. It was shown that two radial passive magnetic bearings and an axial active magnetic bearing can be successfully implemented in the available space of a 72 mm x 72 mm x 48 mm compact fan.

The necessary damping injection for the passively stabilized degrees of freedom was obtained by a viscoelastic support of the stator. In order to achieve the required damping, an optimization of the stiffnesses of the passive magnetic bearings and damping elements was absolutely necessary.

The calculated rotor deflections due to unbalance and magnetic tolerances were compared to measurements and it was shown that the mathematical model is sufficiently precise to predict the deflections of the rotor and the viscoelastically mounted stator.

Acknowledgment

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