Control and Commissioning of Active Magnetic Bearing (AMB) System for High-speed 1 MW Turbo Blower

Nikita UZHEGOV [a], Alexander SMIRNOV [a], Aleksi SALMI[b], Ville LAHDENSUO[b]

a SpinDrive, Laserkatu 6, 53850 Lappeenranta, Finland, nikita.uzhegov@spindrive.fi

b Runtech Systems, Kastarintie 27, 35990 Kolho, Finland

Abstract

This paper describes the design process, control and commissioning of active magnetic bearings (AMB) for a 1 MW, 11 500 rpm turbo blower used in the pulp and paper industry. Turbo blowers in pulp and paper applications work in a harsh environment. During the operation, the rotor is prone to high unbalance due to dirt buildup on the impeller. Implementing magnetic bearings helped to improve overall system efficiency and allowed stable operation with unbalance forces. AMB topology selection is described and justified from the overall turbo blower unit design perspective. Due to limited observability and controllability, only the robust model-based controller was selected to provide stable levitation throughout the operational range. The model based $\mathcal{H}_\infty$ approach scheduled over the speed with mixed-sensitivity formulation has been selected as a solution. The commissioning was performed by adjusting the plant model to the measured peak values and then by accelerating step by step and capturing identification data at each speed point. It allowed reaching the nominal speed despite working close to the first forward and on top of the first backward mode. Further system tests confirmed stable operation under all load conditions. Stable turbo blower operation has also been validated in situations of high unbalance force tested by installing a dummy impeller with added weights.

Keywords: *AMB, high-speed machine, turbo blower, rotor dynamics, flexible rotor*

1. Introduction

Energy efficiency at various operating points is an essential parameter for industrial equipment. With growing electricity prices, the rotating electrical equipment electricity costs may exceed 70% of the total lifetime equipment costs (De Almeida, Ferreira and Quintino, 2012). For example, a turbo blower is a rotating equipment typically used in industrial and manufacturing settings to provide large volumes of air or gas driven by high-speed electric motors. Turbo blowers can provide high energy efficiency and deliver a constant flow rate, making them a popular choice in many industrial applications, e.g. in pulp and paper. However, high-speed, high-power turbo blowers set challenging requirements for the bearing system needed for an efficient and reliable unit operation. This paper describes active magnetic bearing (AMB) control and commissioning of 1 MW, 11 500 rpm machine. The high output power and high rotational speed of the turbo blower unit resulted in the operation close the first forward mode, and simple control like PID was not able to provide stable operation.

AMBs are used in various industrial applications, where the combination of high rotational speed and high external loads set challenging requirements on the bearings (Uzhegov *et al.*, 2017). Magnetic bearings have several benefits, e.g., improved overall system efficiency due to higher possible rotational speeds and elimination of friction (Smirnov *et al.*, 2017). In addition, AMBs provide virtually maintenance-free and no wear bearing operation due to the absence of physical contact between the rotor and stator. Active magnetic bearings eliminate the risk of oil contamination due to the absence of lubricants. Finally, built-in position sensors and magnetic bearing controllers with significant computational power provide opportunities for condition monitoring of the rotating equipment.

The most common control approach for AMB systems is simple decentralized PID, where each axis is assumed to be independent. The reference for generating such a PID controller is available in fundamental textbooks on magnetic bearings (Chiba *et al.*, 2005; Larsonneur, 2009). The general properties and benefits of PID are summarized by (Knospe, 2006). However, decentralized control has its own limitations, which start to be evident in gyroscopic systems and with flexible rotors. In these cases, the coupling between axes and rotor ends cannot be neglected. To address the challenge, the model-based approaches are utilized such as $\mathcal{H}_\infty$ synthesis. This has also been addressed in the literature for AMB-supported machines (Jastrzebski, Hynynen and Smirnov, 2010; Balini, Scherer and Witte, 2011). Eventually, if a robust model-based approach is not sufficient, the solution is to utilize an adaptive controller. In the AMB system, the most straightforward is adapting to the rotational speed changes. This can be undertaken with the help of the gain-scheduling control theory (Balini, Witte and Scherer, 2012).

This paper describes how limited observability and controllability of the AMB system were solved during the 1 MW turbo blower commissioning. The robust model-based controller takes into account speed dependent model, foundation resonances and external disturbances to achieve the desired performance. During the commission, the model was adjusted, relying on identification results.

Changing turbo blower bearings from roller to active magnetic bearings allowed for optimizing impeller selection. The open impeller was possible to utilize due to the capability of AMBs to monitor and control axial pull. Turbo blower measurements demonstrated an efficiency increase from 62% to 70% by implementing an open impeller and active magnetic bearings. This tested variable speed and variable capacity turbo blower is able to save 30–70% in energy consumption compared with traditional vacuum systems like liquid-ring-pump, depending on the end-user.

2. AMBs for high-speed high power turbo blower

The decision to utilize magnetic bearing in vacuum blower is based on the challenging requirements. These machines are operated in the pulp and paper industry under harsh conditions. There are typically high axial loads with possibilities for surge effects occasionally. In addition, after some time, the dirt accumulates on the impellers resulting in a significant unbalance. The traditional ball bearings under such conditions require frequent maintenance resulting in production downtime, labour work and additional costs.

2.1 AMB Design

The vacuum blower has a standard 16-pole heteropolar radial bearing presented in Figure 1. The primary motivation for this topology is to reduce the journal thickness in order to keep the rotor dynamics acceptable and utilize unified manufacturing for the square-type coils. The system has an identical radial bearing at each end of the rotor.

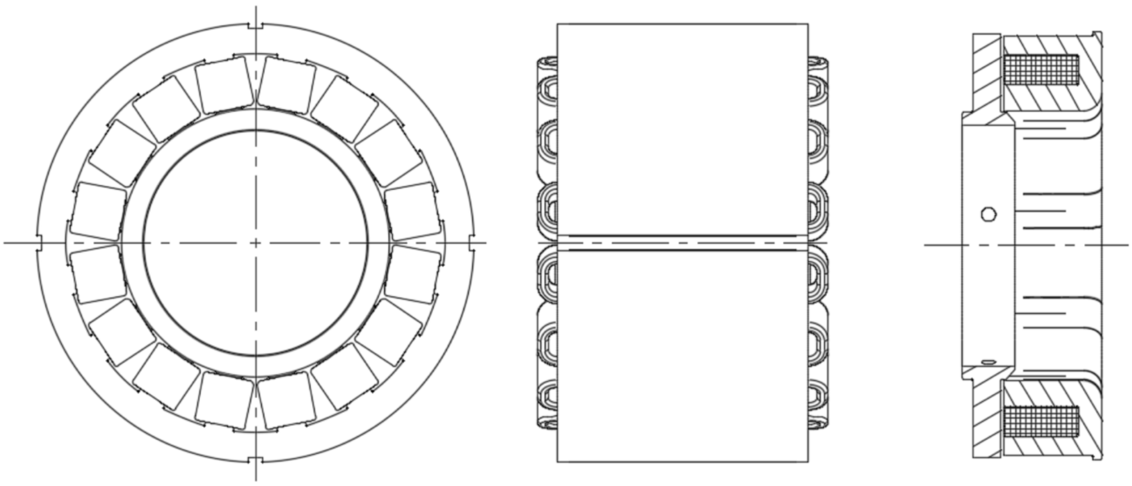

Figure 1 Overview of the radial and axial magnetic bearing actuators

The axial direction is controlled with a classical C-shaped axial bearing. Two axial stators are placed at each rotor end and facing the dedicated disc. The solution allows effectively utilizing the space under end windings of the electrical motor that otherwise will be empty.

The main parameters of the bearings are summarized in Table 1. The magnetic gap for the axial bearing has been increased to accommodate the possibility of uneven thermal expansion between the rotor and the stator.

Table 1 Main parameters of magnetic bearings

Parameter	*Value*
Radial	
Outer diameter	245 mm
Journal outer diameter	150 mm
Magnetic air gap	0.6 mm
Current stiffness	1.07 kN/A
Positions stiffness	30 N/µm
Inductance	40.9 mH

Parameter	Value
Axial	
Magnetic air gap	1.3 mm
Current stiffness	1.81 kN/A
Positions stiffness	9.39 N/µm
Inductance	174 mH

Overall, the bearings can provide the force with enough safety margin to levitate the system and reject disturbance from the process. The axial bearing stator has additional radial slits that break the path for the propagation of eddy currents and thus increase the bandwidth of the axial force.

2.2 Control

To create the linearized radial bearing model for the control purpose the general linearization of the system around its operating point is done. The current and position stiffnesses are given in Table 1. The actuator has been approximated as a first-order low-pass filter with a bandwidth of 350 Hz. The rotor model has been created as 1DOF FEM based on the Timoshenko beam theory. The overall view of that model is presented in **Error! Reference source not found.**, along with free-free modes. It can be seen that the nodes of the first mode go through the actuator locations.

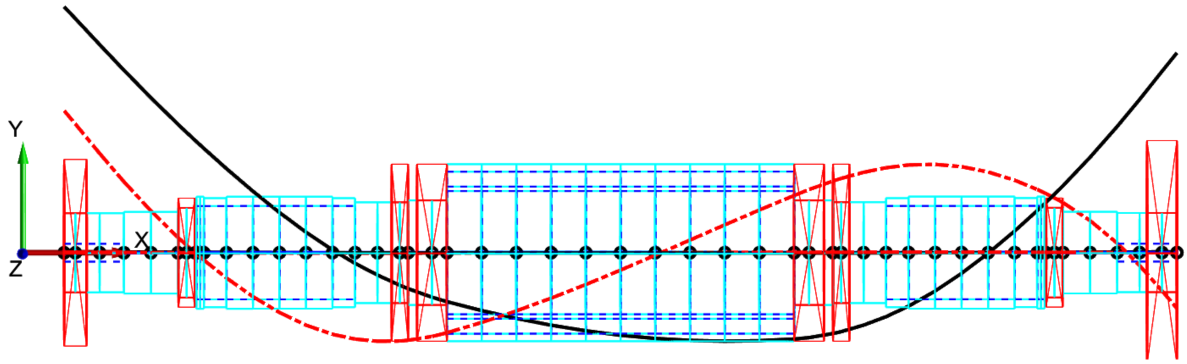

Figure 2 Free-free modes of the rotor model

The summary of the model is presented in

Table 2, where it is seen that 1st model has very limited controllability and observability on the right end of the rotor. Eventually, the resulting rotor model was reduced to keep only information on the two lowest frequency flexible modes to keep the controller order feasible.

Table 2 Main parameters and summary of the rotor model

Parameter	Value
Rotor mass	361 kg
Nominal speed	192 Hz
Polar moment of inertia	3.27 kg m^2
Diametral moment of inertia	62.4 kg m^2
Frequency of the first mode	217 Hz
Frequency of the second mode	412 Hz
Acutator locations from the center of mass (left/right)	490mm/423mm
Sensor locations from the center of mass (left/right)	596mm/529mm
Observability of the 1st mode (left/right)	44%/18.5%
Controllability of the 1st mode (left/right)	16.3%/8.42%

The final model was evaluated in the frequency domain, and the rotor speed was assumed as uncertainty. The corresponding singular value plot is presented in Figure 3. It is seen that the splitting of the first mode is quite considerable and results in roughly 57 Hz spread between forward and backward one at nominal speed. With limited controllability, it provides a challenge for the robust controller to handle. For this reason, the model-based $\mathcal{H}_\infty$ approach scheduled over the speed has been selected as a candidate solution. In particular, the mixed-sensitivity formulation of the problem (Zhou and Doyle, 1998).

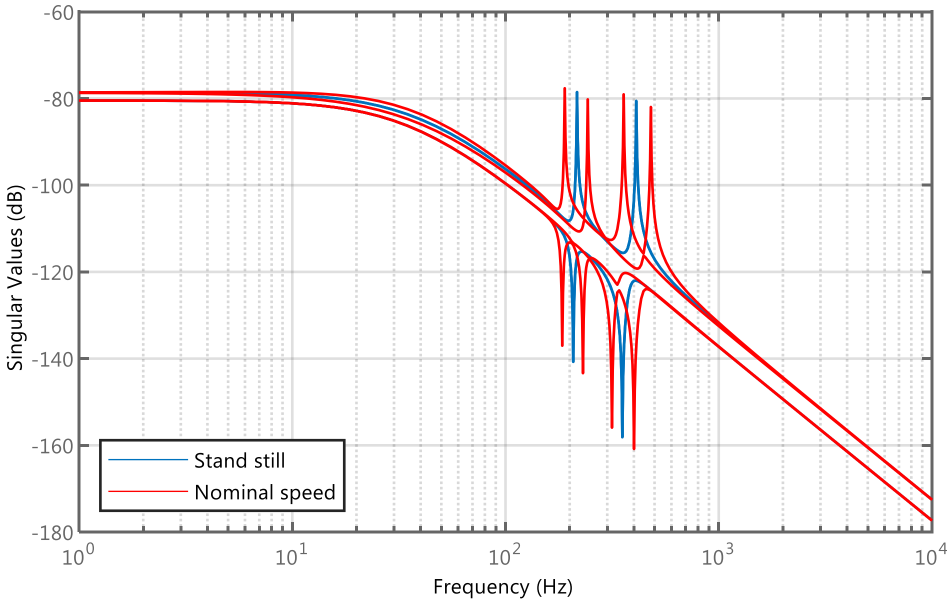

Figure 3 Singular value plot of the model at zero and nominal speeds

The weights for the control effort and sensitivity functions were defined as follows

$$W_{KS} = \frac{10s^3 + 6.87 \cdot 10^4 s^2 + 2.36 \cdot 10^8 s + 4.05 \cdot 10^{11}}{s^3 + 3.44 \cdot 10^4 s^2 + 5.93 \cdot 10^8 s + 5.1 \cdot 10^{12}}$$

$$W_S = 0.45 \frac{s + 150}{s + 1}$$

The control effort weight is rolled off with a 3rd order function at 500 Hz to remove the possibility for exciting higher frequency modes. The sensitivity weight defines the desired DC gain of the controller and limits the possible peak value of the closed loop function. This provides sufficient disturbance rejection.

2.3 Commissioning

The system was assembled in the test facility, where the identification data was obtained after initial levitation. The results are demonstrated in Figure 4, where the predicted model has a reasonable match to the measured results. The

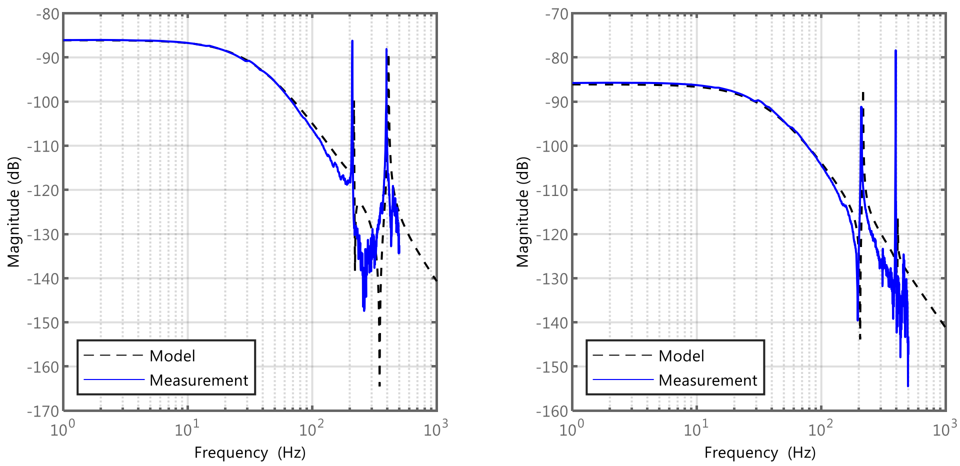

Figure 4 Identification results. Left on left end, right – right end

discrepancy appears in the exact frequencies of the flexible mode. This is expected as the beam model assumes some contact stiffness between the shaft and attached elements, which is difficult to predict. The other discrepancy is in the peak of the first mode for the left end and the second mode for the right end. This is related to the inaccuracy of the mode shape and its damping. These parameters were adjusted in the model with respect to the obtained measurements.

For control purposes, only the first two modes are considered, as their peaks are above the DC gain of the plant, and they require effort from the control system to provide additional dumping. The plant model was adjusted to the actually measured peak values so that the model-based controller could react adequately.

In the next step, the system was accelerated step by step, capturing identification data at each speed point. The resonances are demonstrated as red dots in **Error! Reference source not found.**. They do not perfectly correspond to the linear reduced order model. To overcome the discrepancy, the whole speed range is separated into five points between which the position controller is interpolated. At each point, the frequency of the flexible modes is tuned to the actual measurements to ensure the model match.

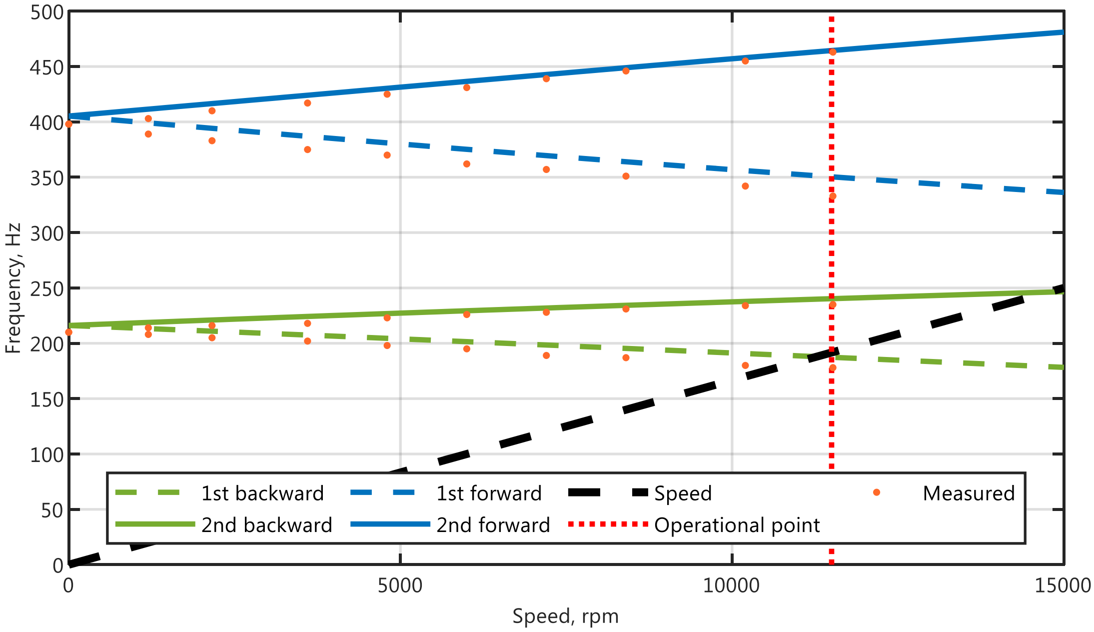

Figure 5 Estimated Campbell diagram and resonance frequencies measured from the setup

This approach allowed for reaching the nominal speed and operating the system under all load conditions. Furthermore, the stability has also been validated in situations of high unbalance force by installing a dummy impeller with specifically added weights. After that, the whole machine has been under test to ensure the overall aerodynamics performance and robustness.

3. Results

The system was thoroughly tested over a 1-year period in all operational conditions. The AMBs demonstrated the expected robustness and maintenance-free operation. In addition, the diagnostics capability of the bearings and, specifically, the possibility to estimate the forces allowed to optimize the impeller. During certain operational conditions, an unexpected axial pull was captured. After analysis of the data, the impeller was updated from closed to open design, which provided a process efficiency increase from 62% to 70%.

The other benefit was obtained by using the dummy impeller with a predefined unbalance level. Based on the experimental data and unbalance orbit estimated by the bearing controller, it was possible to predict the proper intervals for impeller cleaning. On top of that, the AMBs can tolerate high unbalance level without risk of damaging the system.

The high-speed turbo blower with AMBs inside the testing bunked is shown in Figure 6.

4. Conclusions

This paper describes how robust control was implemented for the AMB system used in a 1 MW, 11 500 rpm turbo blower. AMB topology, design parameters and rotor dynamics analysis affecting causing limited controllability and observability were described. The control synthesis procedure is presented, which has taken into account various factors like speed-dependent system behavior, foundation resonances and external disturbances. Finally, the control system was tuned to the actual model of the plant based on the experimental measurements. The gain-scheduling has been applied to provide the necessary performance in the entire operational range because of the flexible modes splitting.

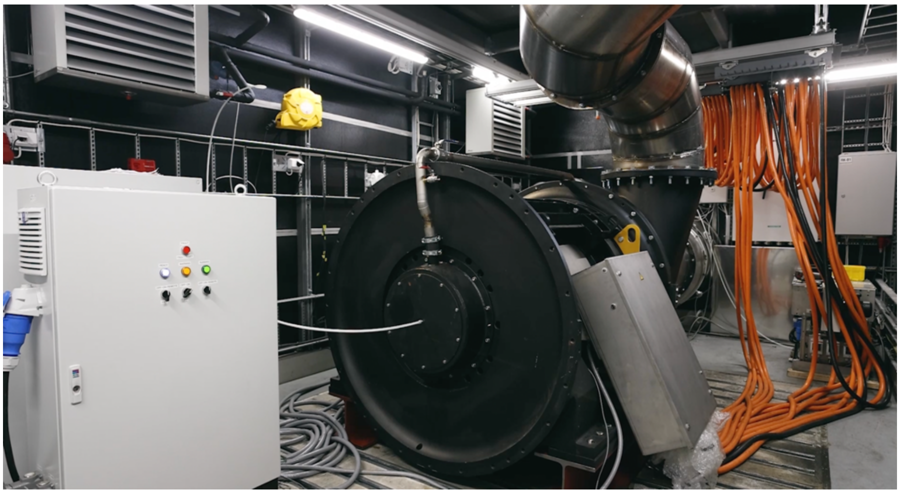

Figure 6 One MW turbo blower under tests

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