The 15th International Symposium on Magnetic Bearings, August 3-6, 2016, Mojiko Hotel, Kitakyushu, Japan

ISMB15

Turbomolecular pumps on conical active magnetic bearings

Nicola AMATI, Angelo BONFITTO, Lester D. SUAREZ and Andrea TONOLI

Department of Mechanical and Aerospace Engineering, Politecnico di Torino

Corso Duca degli Abruzzi, 24 – 10129 TORINO, Italy

E-mail: angelo.bonfitto@polito.it

Abstract

A field in which the magnetic suspensions technology is finding wide application with an increasing rate is the vacuum industry and, in particular, the market of turbomolecular pumps. Nowadays, most used solutions adopted in particular for small to medium pumping capacity are based on hybrid architectures where passive magnetic bearings are combined to ceramic ball bearings. By converse, fully active magnetic suspensions with cylindrical configuration represent the standard for medium to high pumping rate machines. Although simple, the cylindrical configuration is prone to drawbacks related in particular to the straines growing in the disc of the axial actuator that motivate the investigation of alternative architectures. As shown in previous literature, the use of conical configuration, besides compacting driving electronics, seems to be promising considering that the control of the radial and axial degree of freedom is performed simultaneously by the same devices in two actuation planes.

This paper describes the main design steps and experimental characterization of a turbomolecular pump supported by conical active magnetic bearings. The control design is based on a SISO decentralized technique with position and control embedded loops. A rotor centering technique based on the characterization of current loops is exposed while the external position loop is tuned measuring relevant transfer functions to refine the controller allowing critical speed crossing. The power actuation of the eight electromagnets is performed with a three-phase configuration drive technique instead of standard H-bridges to minimize the number of power switches. Experimental results along with numerical computations obtained with simulation models are reported in order to prove the validity of modeling approach and the effectiveness of the conical actuation system.

Keywords: Turbomolecular, Pump, Conical, AMB, Vacuum.

1. Introduction

Modern rotating machines require performance that have become more and more severe throughout the last decades. As a matter of fact, the demands of high accuracy and precision, high rotation speeds and low noise are of major interest in industrial operations. Although the support of rotors with ball bearings is considered a well-known, reliable and economic solution, it presents a set of limitations consisting mainly in the use of lubricants, the necessity of periodical and expensive maintenance, fatigue and tribology issues. Overcoming these drawbacks is indispensable in several industrial fields obliging to address research investigations towards alternative technologies. Among them, active magnetic bearings received an increasing attention due to their specific properties and are more and more widespread in particular industrial applications (Bleuler, et al., 2009), (Chiba, et al., 2005). Indeed this technology presents several advantages: a) the absence of mechanical friction constitutes the basis for high speed machines and low noise emissions, b) the absence of lubrication and contaminating wear allows the use in vacuum and ultra-vacuum systems, clean and sterile rooms and in high temperature applications, c) the active control allows high precision positioning of the rotor and permits to modify its dynamic behavior even during working operations, d) the maintenance costs are drastically reduced recommending the use of this technology in critical and hostile environments. By converse, magnetic bearings technology still needs to be improved in terms of costs and compactness in order to face large scale productions and to establish itself as a standard solution in applications like special vacuum pumps, ultra-high speed machining spindles and micro/nano linear stages where the size reduction is one of the main design challenges. To this end, last years' research efforts have been focused also on the investigation of configurations different from the conventional one realized with five couples of cylindrical shape electromagnets to perform five degrees of freedom active control. In particular, exploiting a conical profile of the electromagnets as alternative actuators allows to exert forces both in axial and radial direction simultaneously allowing to save one couple of electromagnets (Lahteenmaki et al, 1988), (Lee and Jeong, 1996), (Bonfitto et al). Furthermore, this geometry permits to reach higher spin speeds, limited in the cylindrical solution by the strains growing in axial bearing disc. Nevertheless, although the bearing results very compact, the design turns out to be more complex than the one for cylindrical actuators, in particular in control design phases (Huang and Lin, 2003), (Jing et al.). This limitation obstructed the spread of cone-shaped configuration which however keeps being very interesting and with a promising potential. This motivated the poor literature of conical magnetic bearings in particular regarding the cases of study conducted until experimental validation.

This paper aims to illustrate a methodology for the control design of a turbomolecular pump on conical active magnetic bearings. Nowadays the use of magnetic bearings in the field of vacuum and high vacuum, where this kind of pumps belongs, is very widespread. This is mostly due to the necessity of maintaining the environment of work non contaminated hence excluding the use of lubricated bearings. Actually hybrid solutions with a ceramic ball bearing in the high pressure side and a passive reluctance bearing in the low pressure side are still used but mainly for small to medium pumping capacity (less than 1500 l/s). On the other hand, the support of higher pumping rates machines is realized with fully active cylindrical active magnetic bearing and conical solutions for vacuum pumps have not been investigated yet. As well known, one of the main issues in magnetic bearings setup is to achieve the correct centering of the rotor. This problem is way more evident in conical configuration where axial and radial actions are inherently combined. The present work illustrates a control strategy to achieve rotor centering analyzing the behavior of the currents flowing in the eight electromagnets and to cross critical speeds with low displacement.

Experimental results at null speed and in rotation in terms of current and position transfer functions, waterfall, unbalance response, orbital view, orbital tube and deceleration view show the effectiveness of the proposed approach and urge on further research.

Nomenclature

e = voltage applied to an electromagnet
g = airgap
g_0 = nominal airgap
i = current
k_i = current-force factor
k_m = back-electromotive force factor
k_x = negative stiffness due to electromagnet
α = semi-conicalness angle
F = magnetic force
F_c = control force
F_u = unbalance force
L_0 = nominal inductance
N = number of turns of an electromagnet
R = resistance
S = cross-sectional area of the magnetic circuit
λ = magnetic flux linked by an electromagnet
$\boldsymbol{M}$ = mass matrix
$\boldsymbol{C}$ = damping matrix
$\boldsymbol{G}$ = gyroscopic matrix
$\boldsymbol{K}$ = stiffness matrix
$\boldsymbol{H}$ = circulatory matrix

2. System architecture and modelling

The proposed control design and the experimental results have been conducted on a turbomolecular pump with a high pumping speed (more than 2000 l/s) mostly used in coating and plasma processes industry. It is composed of a main shaft where the rotating parts of the electric motor and the conical actuators and a mock-up of the pumping impeller are fitted. The original bladed rotor is replaced with an equivalent component with the same inertial properties allowing to get rid of blades dynamics whose control is out of the purposes of this activity.

The power actuation of the eight electromagnets is performed with a three-phase configuration drive technique instead of standard H-bridges to minimize the number of power switches.

Due to confidentiality issues, detailed design and operating characteristics cannot be provided in this paper.

2.1 Mechanical subsystem

The rotor has been modelled by means of the Dynrot code, an FE code dedicated to rotordynamics. The rotating part has been modelled using Timoshenko beam elements. Fig. 1 shows the main nodes of the mesh along the shaft; a) represents the electric motor whose behaviour is considered as unstructured mass giving a contribute in terms of mass and not of stiffness, b) and c) are the rotating laminated parts of the conical actuators separated by a spacer (unstructured component) and d) is the mock-up of the pumping rotor provided with two discs to replicate the inertial behaviour of the original bladed impeller. Gyroscopic effects are taken into account in all elements and rotating masses.

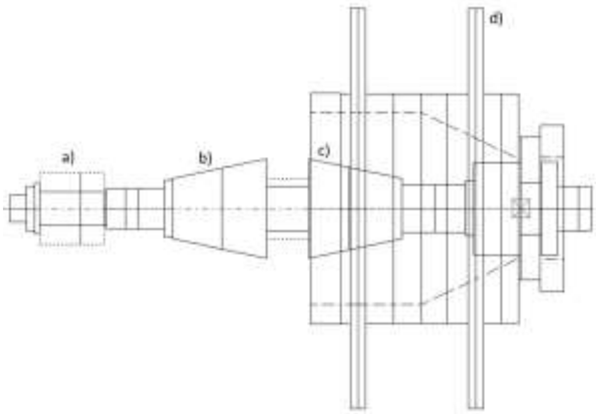

Fig. 1 - FEM Discretization of the pump. a) Electric motor; b) Lower bearing; c) higher bearing and d) Pumping rotor mock-up.

As explained in (Genta, 2005), the dynamic equation of the mechanical subsystem in the rotor reference frame is:

$$\mathbf{M}\ddot{q}(t)+(\mathbf{C}+\mathbf{G})\dot{q}(t)+(\mathbf{K}+\mathbf{H})q(t)=F_c+F_u \tag{1}$$

where $q(t)$ includes the generalized displacements, $\mathbf{M}$ is the symmetric mass matrix, $\mathbf{C}$ and $\mathbf{G}$ are the symmetric damping and the skew-symmetric gyroscopic matrices, $\mathbf{K}$ and $\mathbf{H}$ are the symmetric stiffness and the skew-symmetric circulatory matrices, F_c and F_u are the control and unbalance forces. Fig. 2 reports the Campbell diagram of the pump.

2.2 Electro-mechanical interaction

The four electromagnets installed on each actuator plane are assumed to be identical with no coupling between the two electromagnetic circuits so that the mutual inductance is neglected. Each electromagnet can be considered as a two-port element (electrical and mechanical). The relative force for each axis of actuation is expressed as:

$$F_j(i_j,g)=\frac{\partial E_j(i_j,g)}{\partial g}, \qquad j=1,2 \tag{2}$$

where E_j is the energy stored in the electromagnet j and g the airgap.

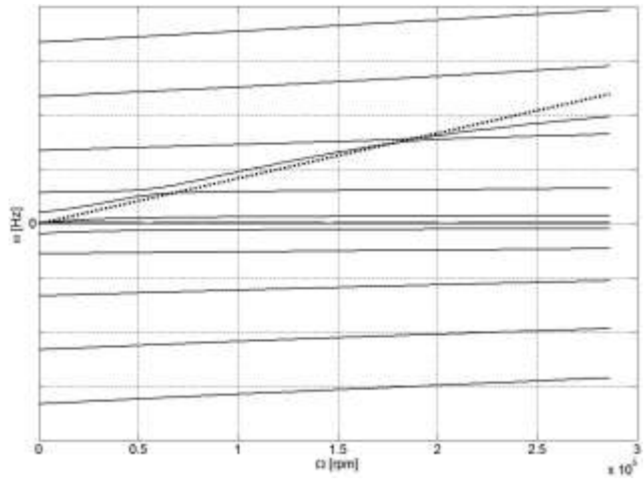

Fig. 2 - Campbell diagram. (Frequency values are omitted for confidentiality issue).

In the general case, the total magnetic flux λ_j and the coil current i_j are related by a nonlinear inductance which takes into account the flux leakage in the airgap and the saturation in the ferromagnetic material.

$$\lambda_j = \Phi_j(i_j, g_j) \tag{3}$$

Owing to Faraday and Kirchoff laws, the dynamic equations of the system in the electrical domain are:

$$\begin{aligned} \dot{\lambda}_1 + Ri_1 &= e_1 \\ \dot{\lambda}_2 + Ri_2 &= e_2 \end{aligned} \tag{4}$$

where R is the coils resistance and e_j is the voltage applied to electromagnet j. An in depth description of the nonlinear electromagnets behaviour with explicit expressions of inductance and force is reported in (Amati, 2010).

The system dynamics is linearized around a working point corresponding to a bias voltage e_0 imposed to both electromagnets:

$$\begin{aligned} &i_j = i_0 \pm i_c \\ &e_j = e_0 \pm e_c \qquad j = 1,2 \\ &F_j(i_j, g) = \pm F_0 \pm \Delta F_j \end{aligned} \tag{5}$$

where g_j is the radial airgap of electromagnet j defined as:

$$g_j = g_0 \mp q, \qquad j = 1,2 \tag{6}$$

where g_0 is the nominal airgap, assumed to be identical for the two opposite electromagnets controlling one actuation direction. $\mp$ stands for upper and lower magnets, F_0 is the initial force generated by each electromagnet due to the bias current $i_0 = e_0/R$ and ΔF_j is the variation of the electromagnets' forces due to the control.

The non-linear functions defined in equations Eqs. (2) and (3) can be linearized as follows:

$$F_j(i_c, q) = k_x q + k_i i_c, \qquad \lambda = L_0 i_c + k_m q \tag{7}$$

where L_0, k_i, k_m and k_x are the inductance, the current-force factor, the back-electromotive force factor, and the so-called negative stiffness of one electromagnet, respectively, given by:

$$L_0 = \left.\frac{\partial \Phi_j}{\partial i}\right|_{\substack{e_0\\ i_0}}, k_m = \left.\frac{\partial \Phi_j}{\partial g}\right|_{\substack{e_0\\ i_0}}, k_i = \left.\frac{\partial F_j}{\partial i}\right|_{\substack{e_0\\ i_0}} k_x = \left.\frac{\partial F_j}{\partial g}\right|_{\substack{e_0\\ i_0}} < 0 \tag{8}$$

Due to the conservative nature of the electromagnetic interaction, $k_m = k_i$.

The single actuator exerts a force acting simultaneously axially and radially as illustrated in Fig. 3 (for the plane XZ) where α is the semi-conicalness angle, F_1, F_2, F_3 and F_4 are the forces perpendicular to the actuator surfaces, F_1', F_2', F_3' and F_4' are the radial forces and F_1'', F_2'', F_3'' and F_4'' are the axial forces.

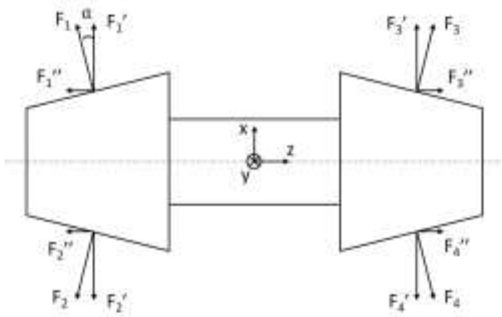

Fig. 3 Force generation scheme. Conical bearings exert simultaneously axial and radial force allowing to get rid of dedicated axial actuators. The resulting force of the single actuator is perpendicular to the conical surface (F_i) and can be expressed as the combination of radial (F_i') and axial (F_i'') components.

The forces are expressed as follows:

$$F_i = F_i' \cos\alpha + F_i'' \sin\alpha \qquad i = 1,2,3,4 \tag{9}$$

Table 1 reports the values of actuators parameters.

Table 1- Actuator parameters.

Symbol	Value	Unit
$\boldsymbol{N}$	112	-
$\boldsymbol{S}$	3.93e-4	m^2
$\boldsymbol{g_0}$	0.4e-3	m
$\boldsymbol{\alpha}$	12	°
$\boldsymbol{R}$	0.8	Ω
$\boldsymbol{L_0}$	10e-3	H
$\boldsymbol{k_i}$	25	N/A
$\boldsymbol{k_m}$	25	Vm/s
$\boldsymbol{k_x}$	-125	kN/m

3. Control design and experimental results

The aim of the control strategy is to allow the rotor reaching its nominal spin speed with radial displacement not exceeding 0.1 mm peak. Since the nominal velocity is lower than the first flexural critical speed, the main role of the position control is to reduce displacement during rigid body critical speeds crossing. The adopted control architecture is a decentralized strategy exploiting five PID position loops along with eight embedded PI current loops.

A major issue in the control of active magnetic bearings is the identification of the rotor equilibrium point. For conical configurations this problem is even more severe due to the inherent interaction between axial and radial forces. Overcoming this drawback is possible by identifying accurately the geometric center of the rotor. To this end, the first step of the proposed strategy is to perform preliminary calibrations replacing the landing bearings with two collets mounted with no clearance between the rotor and the stator: the rotor is mechanically centered and the sensor offsets can be calibrated with good accuracy. Furthermore, this procedure allows to tune current control parameters in order to make the eight actuators work with the same dynamics. It is well known that actuator response is strongly dependent on the precision of sensors acquisition chain and on the value of the inductance that changes considerably even with small airgap variations. Fig. 4 illustrates the transfer functions between the reference and measured current, obtained on the lower actuation stage of the pump with the rotor centered by means of the collets, before (a) and after (b) current control tuning. Although the nominal values of the four actuators electrical parameters are the same and the rotor is forced on its geometric center, relevant differences on the bandwidth of the current loops are measured resulting in bad actuation behavior in operating conditions and making tuning necessary.

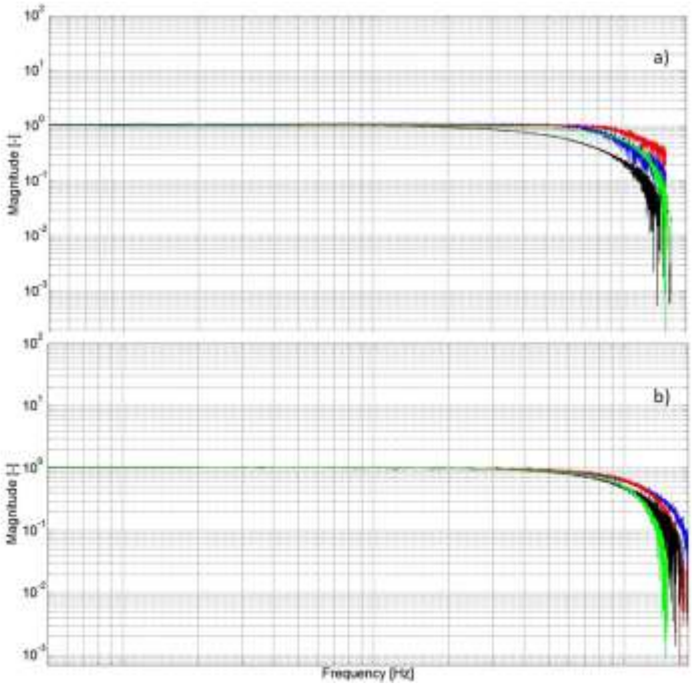

Fig. 4 - Transfer function between current reference and current measurement on lower actuation stage before (a) and after current control parameters tuning. The responses are obtained with the rotor centered by means of two collets. The tuning of current control is necessary to obtain a homogeneous electrical behavior of the four actuators. The same results are obtained on the higher stage. (Frequency values are omitted for confidentiality issues).

Once defined sensor offsets and current control, external position control loops have been designed with the aim of damping rigid body modes. Experimental results have been conducted on the pump in order to validate the effectiveness of the modelling and control design procedure. Figure 5 reports the transfer function between the current reference and the measured position on X-axis of lower actuation stage at null speed. The rigid body modes are damped to allow critical speed crossing with low displacement. The good correspondence between experimental (solid line) and numerical (dashed line) results is a proof of the correctness of the model.

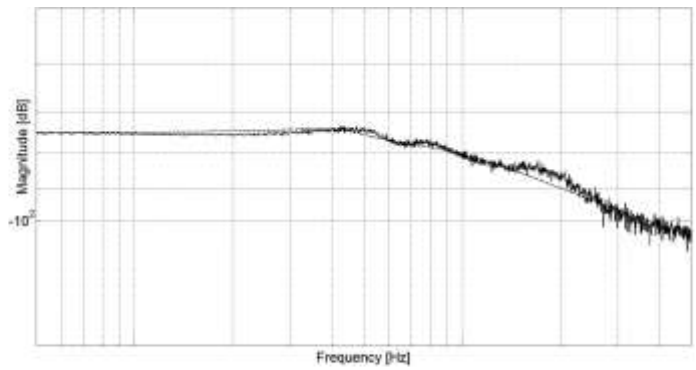

Fig. 5 - Current reference to measured position transfer function on X-axis of lower actuation stage at null speed. Solid line:

Experimental results. Dashed line: Numerical results. Rigid body modes are damped in order to allow critical speed crossing. Frequency values are omitted for confidentiality issues.

The control has been validated also with the pump in rotation measuring on lower actuation stage the waterfall diagram (Fig. 6), the unbalance response (Fig. 7, solid line: X-axis, dashed line: Y-axis), the orbital tube, the orbital view and the deceleration views (Fig. 8). The pump is able to reach its nominal speed with a displacement that does not exceed the design requirement of 0.1 mm.

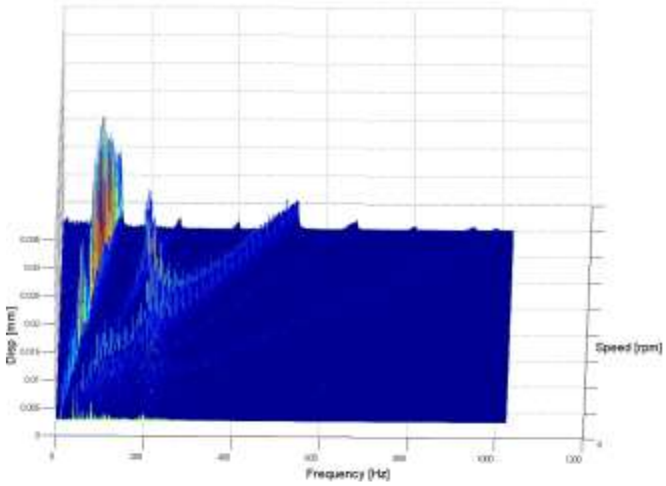

Fig. 6 – Waterfall diagram. Speed values are omitted for confidentiality issues.

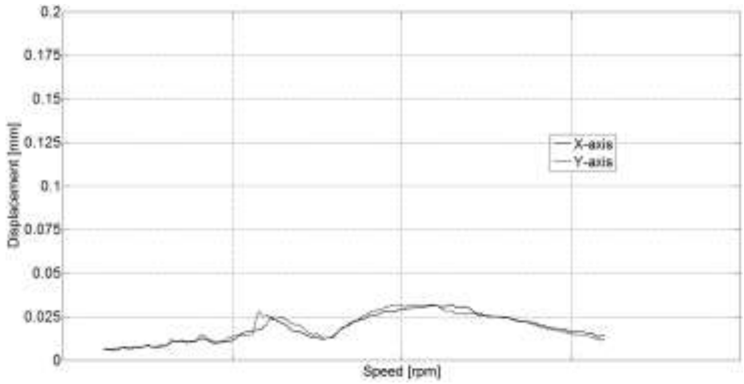

Fig. 7 - Unbalance response computed on X and Y axis of lower actuation stage. Displacement is lower than 0.1 mm as per control specification. Speed values are omitted for confidentiality issues.

4. Conclusions

The modelling, the control design and the experimental results of conical active magnetic bearings applied on a turbomolecular pump has been presented. The goal of the paper was to illustrate a procedure of calibration of position sensors offsets along with the tuning of current and position control loops allowing the pump to cross critical speeds with measured displacements not higher than 0.1 mm. Experimental tests at null speed (reference current to measured current and reference current to measured position transfer functions) and with the pump in rotation (waterfall diagram, unbalance response, orbital tube, orbital view and deceleration view) have been performed to validate the correctness of the modelling approach and to proof the effectiveness of the control strategy.

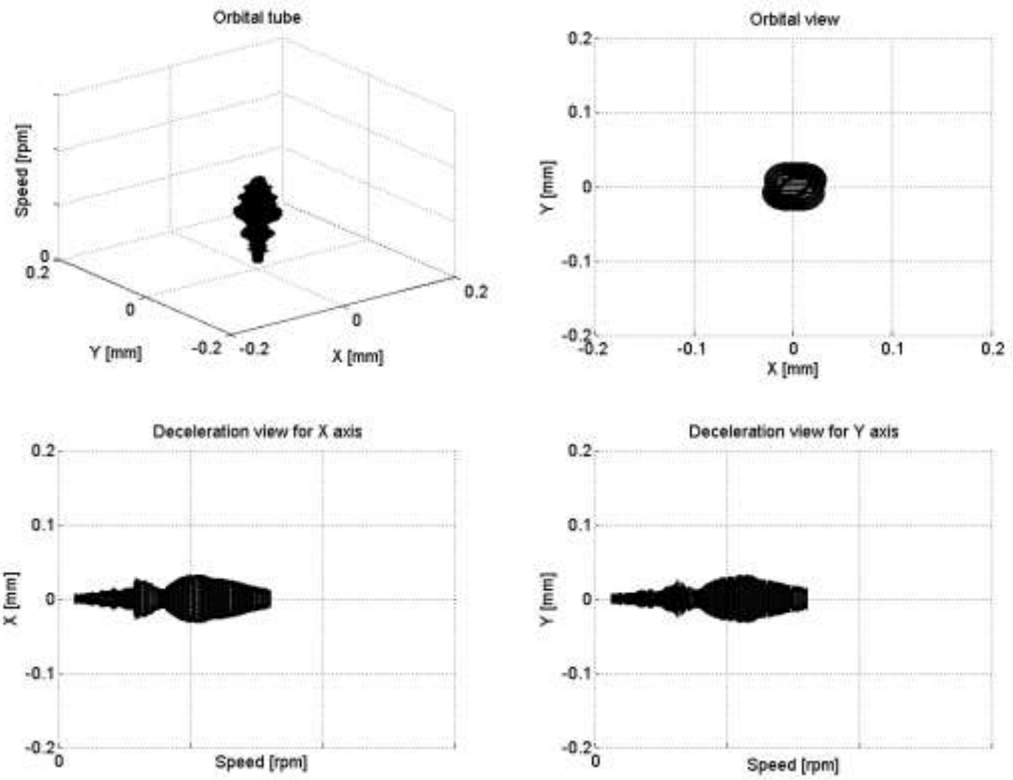

Fig. 8 - Orbital tube, orbital view and deceleration views from X and Y axis on lower actuation stage. Speed values are omitted for confidentiality issues.

References

Amati N., Silvagni M., Tonoli A., Electromechanical Dampers for Vibration Control of Structures and Rotors, Vibration Control, Mickael Lallart (HRV), pp. 32, Vol. 1, ISBN: 9789533071176, (2010), pp. 1-32.

Bleuler, H., Cole, M., Keogh, P., Larsonneur, R., Maslen, E., Nordmann, R., Okada, Y., Schweitzer, G. and Traxler, A., Magnetic Bearings Theory, Design, and Application to Rotating Machinery, Springer, Berlin Heidelberg, (2009).

Bonfitto, A., Tonoli, A., Botto, G., Silvagni, M., Suarez, L.D., Rotor on cone-shaped active magnetic bearings with three-phases power drivers, 13th International symposium on magnetic bearings, (2012).

Chiba, A., Fukao, T., Ichikawa, O., Oshima, M., Takemoto, M. and Dorrell, D.G., Magnetic Bearings and Bearingless drives, Elsevier, Oxford, (2005).

Genta, G., Dynamics of rotating systems, Springer Verlag, New York (2005).

Huang, S. G., Lin, C. L., Fuzzy modeling and control for conical magnetic bearings using linear matrix inequality, Journal of Intelligent and Robotic systems 37, 2003, pp. 209-232.

Lahteenmaki, J. K. / Lantto, E. J., Optimization of the Conical Angle of Cone-Shaped Active Magnetic Bearings, 6th Proceedings of the international symposium on magnetic bearings, (1988), 224-233.

Lee, C. W., Jeong, H. S., Dynamic modeling and optimal control of cone-shaped active magnetic bearing systems, Control Practice Engineering, Vol. 4, No.10, 1996, pp 1393-1403.

Jing, Y., Lie, Y., Jing, X., Coupled dynamics and control of a rotor conical magnetic bearing system, Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, vol. 220 no. 7, 2006, 581-586.