

ON CONTROLLING THE MAGNETIC SUSPENSION OF A FLYWHEEL SYSTEM

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1. INTRODUCTION

The flywheel systems or “electromechanical batteries” were especially developed for energy storage systems in Hybrid Electric Vehicles (HEV) purposes. Flywheels seem to be highly appreciated in the design of HEV, because of they outperform conventional chemical batteries in many important areas, such as: shorter recharge time, longer driving range, higher reliability and practically absence of the maintenance. Moreover, in the last decade, in the power quality market the flywheel has regained consideration as a viable means of supporting a critical load during mains power interruption.

The main drawback of the flywheel system is its relatively higher cost, but the technical development should significantly reduce the costs of such systems over time.

In [1] and [2] the numeric control of a magnetic bearing destined to be included in a flywheel storage system was synthesized. The paper purpose is to develop the synthesized controller taking into account the disturbance influences of the other flywheel system components.

2. THE FLYWHEEL STORAGE SYSTEM

The electromechanical battery we are studied is composed by a magnetically suspended flywheel, a synchronous motor/alternator and an inductive position transducer (see fig.1). The magnetic suspension has only one active axis and is composed by two hybrid magnetic bearings that act as two electromagnets in opposition. The radial stiffness is assured by the minimum reluctance effect. In the case of the absence of any current through the coils, the radial stability is assured “passive” through the presence of the permanent magnets. The chosen drive machine is a disk-type permanent-magnet synchronous machine [4].

3. SYSTEM MODELING

The equations that describe the working state of a hybrid axial magnetic bearing are:

- the electric equilibrium equation of the circuitry constituted by the series connection of the two coils placed each one on the superior and, respectively, the inferior stator of the bearing

$$u(t) = R_t \cdot i + \frac{d}{dt}(\Psi_1 + \Psi_2) \quad (1)$$

Where R_t - the sum of the coil resistances, Ψ_1, Ψ_2 – the total fluxes in the two bearing components
 - the mechanical equilibrium equation:

$$m \cdot \ddot{\delta}(t) = -F_{rez} + (F_{ext} + m \cdot g) \quad (2)$$

where $F_{rez} = F_1 - F_2$ is the resultant force of the suspension electromagnets, F_{ext} is the disturbing external force and $m \cdot g$ is the mobile part weight (including the entire shaft with the rotors of the motor and transducer).

The disturbing force F_{ext} is composed by an aleatory part and also by the residual forces from motor/alternator and transducer. For an accurate synthesis of the magnetic-suspension control system, the residual forces must be evaluated.

As concern the permanent magnet disk-rotor machine, their influence is low because of the permanent magnets that are much thicker (5 mm) as the maximum of the air-gap (1 mm). A simulation program based on the finite element method showed us that the disturbance forces created by the motor can not exceed 1 N.

In opposition, the position transducer is a magnetic-type sensor (see fig. 2.a) and produces disturbing axial forces as we can see in figure 2.b.

Considering that the air-gap is the same in the magnetic bearing and in the magnetic position sensor, the disturbing force due to the position sensor can be calculated with:

$$F_{PT} = \frac{S_{pt} \cdot h_{MPt}^2 \cdot J_t^2}{\mu_0 [h_{MPt} + 2 \cdot \delta_1(t)]^2} \quad (3)$$

where J_t and h_{MPt} – the magnetization and the thickness of the permanent magnet, S_{pt} - the pole surface, δ_1 – the minimum of the two air-gaps from the transducer magnetic circuit.

We consider a power supply source characterized by the gain factor k_s and a position sensor that have an one-order transfer function. Considering low

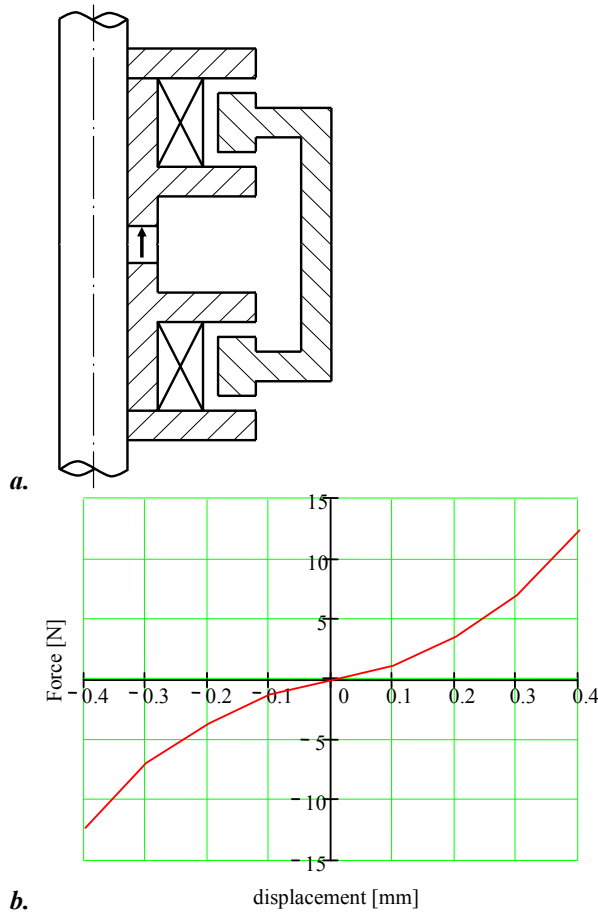


Figure 2. The position sensor geometry and the produced disturbing force.

variations of the system variables a linear input-state-output model is achieved [1]:

$$\dot{x}(t) = A \cdot x(t) + b \cdot u(t) + e \cdot p(t) \quad (4)$$

$$y(t) = c^T x(t) \quad x = \begin{bmatrix} \Delta e & \Delta \delta & \Delta \dot{\delta} & \Delta i \end{bmatrix}^T$$

the state-variables being:

$$x_1 = \Delta e; \quad x_2 = \Delta \delta; \quad x_3 = \Delta \dot{\delta}; \quad x_4 = \Delta i \quad (5)$$

where $\Delta \delta$ is the air-gap variation, Δi is the coil feeding-current variation and Δe is the variation of the transducer output voltage.

4. EXPERIMENTAL RESULTS

4.1. Controller synthesis

The experimental plant presented in figure 1 is characterised with the following parameters:

- the mobile mass: $m_f = 1.3 \text{ Kg}$ (whole system)
- bearing coil resistance: $R_b = 1.5 \Omega$
- turn number of the coil: $N = 180$
- remanent magnetisation in bearing: $J = 0.5 \text{ T}$
- remanent magnetis. in transducer: $J_t = 1.1 \text{ T}$

- natural frequency $\omega_{0n} = 100 \text{ rad} \cdot \text{sec}^{-1}$
- the air-gap sum: $\delta_1 + \delta_2 = 1.5 \text{ mm}$
- the pos. sensor time constant: $T_l = 0.002 \text{ s}$.
- the initial current: $i_0 = 0.2 \text{ A}$

In [1] a controller synthesis considering only the axial magnetic bearing is presented. In this paper we expand the previous controller for the whole flywheel system and analysed their characteristics through numeric simulations in Matlab environment.

To insure the flywheel system stability we chose a polynomial-type control system that is presented in figure 3, where $H(z)$ is the plant (the flywheel system including the two converters, numeric-analogue CNA and analogue-numeric CAN) transfer function, $H_R(z)$ is the transfer function of the RST-type controller with two degree-of-freedom and $H_{mr}(z)$ is the transfer function of the model that defines the tracking-rating behaviour between the

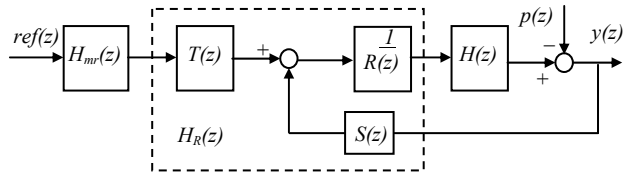


Figure 3. The closed-loop system structure.

reference $ref(z)$ and the plant output $y(z)$ and $p(z)$ is the external perturbation.

Considering a sampling period $T_s = 0.005 \text{ sec}$, the plant discrete transfer-function yields:

$$H(z) = \frac{B(z)}{A(z)} = \frac{-0.075z^3 - 0.455z^2 - 0.212z - 0.008}{z^4 - 3.728z^3 + 1.086z^2 - 0.384z + 0.026} \quad (6)$$

In the polynomial $B(z)$ there can be separate a stable part $B_s(z)$ and an unstable part $B_i(z)$, so that

$$B(z) = B_s(z) \cdot B_i(z) \quad (7)$$

where $B_i(z) = k(z - z_1)$ and $B_s(z) = (z - z_2)(z - z_3)$.

Taking into account the structure presented in figure 3, the close-loop transfer function $H_0(z)$ can be expressed through

$$H_0(z) = \frac{B_0(z)}{A_0(z)} = \frac{B(z)T(z)}{A(z)R(z) + B(z)T(z)} \approx \frac{B_i(z)B_{md}(z)}{A_{md}(z) \cdot (z - p_{1f})^{n_f}} \quad (8)$$

In the last relation, $A_{md}(z)$ is a second-order polynomial built with two dominant poles in order to have a desired dynamic behaviour. Considering a second-order element characterised by the natural

frequency $\omega_{nd} = 0.9\omega_n = 90$ rad/sec and a damping factor $\xi_d = 0.8$, its expression yields:

$$A_{md}(z) = z^2 - 1.344z + 0.487 \quad (9)$$

The supplementary pole $p_{1f} = 0.6$ having the order $n_f = 2$ was introduced to improve the dynamic performances of the close-loop system over the high frequency range.

The transfer function $H_{mr}(z)$ of the reference-trajectory model correspond also to a second-order element, which considering the natural frequency $\omega_n = 100$ rad/sec and a damping factor $\xi_d = 0.8$ results:

$$H_{mr}(z) = \frac{-0.08z + 1.53}{z^2 - 1.28z + 0.449} \quad (10)$$

The controller synthesis consists in finding out the four polynoms $R(z)$, $S(z)$, $T(z)$ and $K_0(z)$, which must satisfy a specific condition (that is called Bezout equation) presented in [2].

The expressions we used for these four polynoms are [2]:

$$\begin{aligned} R(z) &= (z-1)(z+r_0)B_s(z) \\ S(z) &= z^\alpha + s_{\alpha-1}z^{\alpha-1} + \dots + s_1z + s_0 \\ T(z) &= B_{mc}'(z)A_0(z) \\ K_0(z) &= z^2 \end{aligned} \quad (11)$$

where

$$B_{md}(z) = \frac{A_{md}(1) \cdot (1-p_{1f})^2}{k(1-z_1)}$$

The above mentioned condition can be written in this particular case as follows:

$$A(z) \cdot (z-1) \cdot (z+r_0) + B_i(z) \cdot S(z) = A_{mc}(z) \cdot K_0(z) \quad (12)$$

This equation put in a matrix form is useful to determine the unknown values for r_0 , $s_{\alpha-1}, \dots, s_1, s_0$. For the system parameters and chosen form of the polynoms we found the following numeric expressions:

$$\begin{aligned} R(z) &= z^4 + 0.81z^3 - 1.119z^2 - 0.663z - 0.028 \\ S(z) &= -11.958z^4 + 15.053z^3 - 4.31z^2 + 1.251z - 0.083 \\ T(z) &= -0.046z^2 \end{aligned}$$

4.2. Simulation results

The following figures present few simulation results, for the experimental plant-controller ensemble. For two values of the flotor mass, m_1 and $m_2 = 3m_1$, and for two variants of the control structure (with and without reference-trajectory-model), the

system behaviour is studied considering a step-jump of the reference variable.

The variables having index 1 corresponds to the mass m_1 and that having index 2 corresponds to the mass m_2 . A supplementary index $*_m$ refers to the case when the reference-trajectory-model is considered. The output voltages of the controller are showed in figures 4...7.

First time, we can see that the presence of the reference-trajectory-model lead just to a low delay in the controller response, both for the m_1 (figure 4) and m_2 (figure 5) flotor masses.

A comparison between the controller responses for the two masses is done in figure 6 (controller without reference-trajectory-model) and figure 7 (controller with reference-trajectory-model).

For the chosen cases, in the figures 8, 9 and 10 the output voltages of the position transducer are presented. One notes that the system displaces with a low delay when the reference-trajectory-model is used and, more, this delay increases corresponding to the flotor mass.

In the figures 11 is showed a magnitude-frequency characteristic corresponding to the perturbation - output variable transfer function, which is called also perturbation-output variable sensitivity function.

The obtained characteristics show a good stability and high dynamic performances for the synthesised numeric controller.

5. CONCLUSIONS

In the paper the influences of the elements composing a flywheel system over the magnetic bearing stability is analysed. In the proposed structure only the use of a magnetic-type position sensor lead to supplementary disturbance forces.

A polynomial-type controller for the whole flywheel system is synthesised. One notice that the presence of the magnetic-type position sensor lead only to some small modifications in the controller structure. This is owed to the identical behaviour of the bearing active forces and the sensor disturbing forces.

Considering variations of the constructive parameters and perturbation, by means of simulation programs the system behaviour analyses is performed. The simulation results show us that the synthesised controller assures, for the proposed flywheel system structure, a good stability and higher dynamic performances even for modifications in the control structure (with and without the reference-trajectory-model).

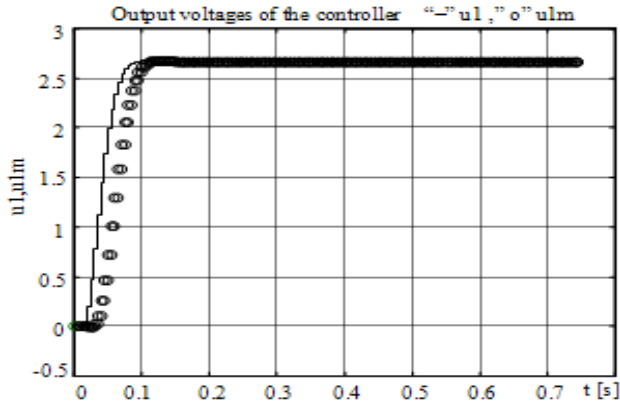


Figure 4.

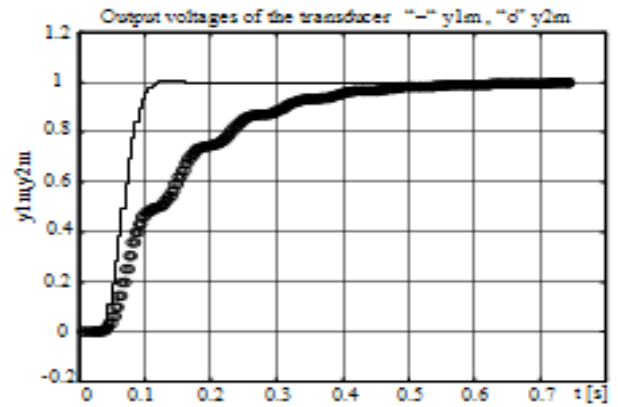


Figure 8.

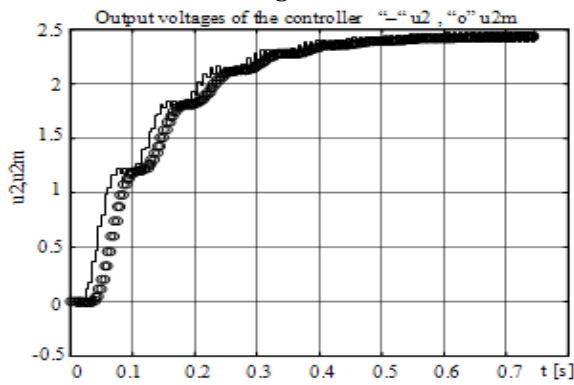


Figure 5.

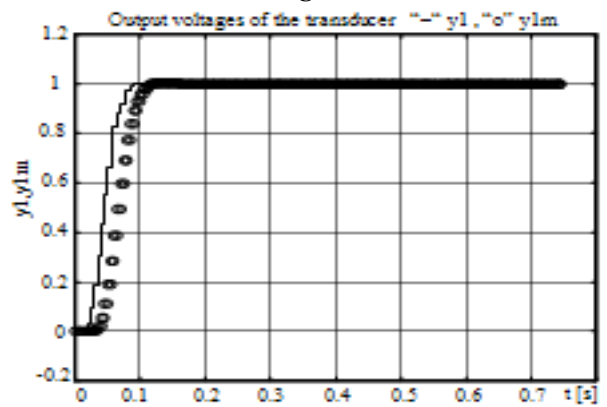


Figure 9.

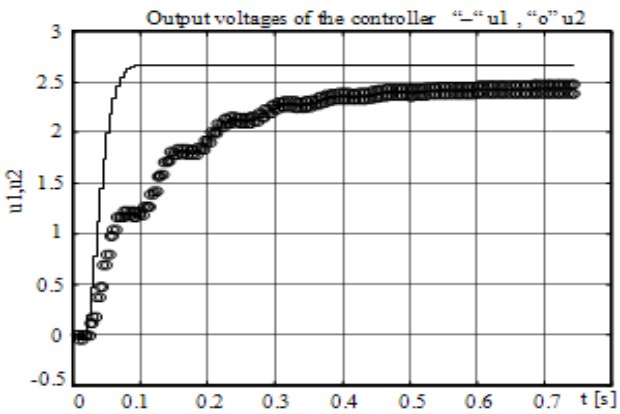


Figure 6.

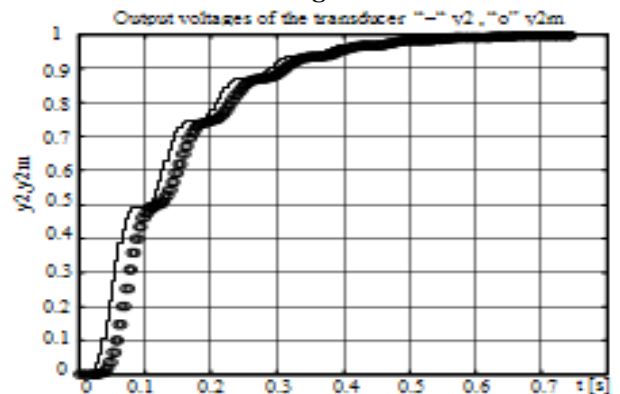


Figure 10.

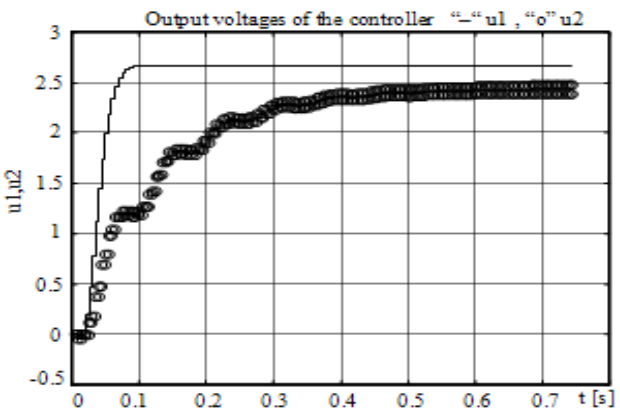


Figure 7.

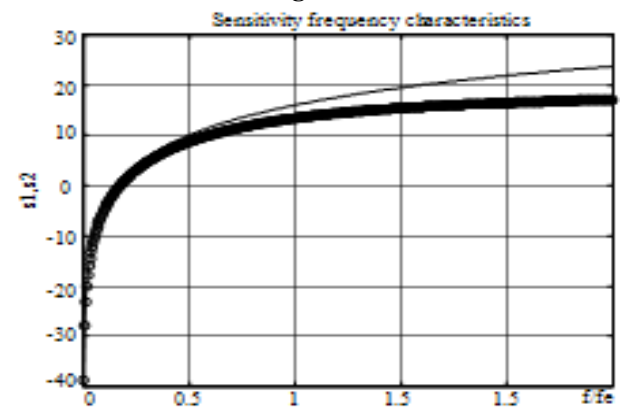


Figure 11.

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