REDUCED-ORDER MODELS FOR ELECTROHYDRAULICAL SYSTEMS CONTROL

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Abstract. In this paper some linear and nonlinear models of the electrohydraulical systems are analyzed and reduced-order techniques are applied in order to obtain useful models for control design. The exactly linearization feedback technique and singular perturbation method are widely analyzed and applied. Computer simulations are included to illustrate the behavior of the reduced-order models.

***Key words*:** Electrohydraulic systems, Modeling, Control, Simulation.

1. INTRODUCTION

Electrohydraulic servovalves used in automatic systems for position, speed and force tracking are equipment, which are the interface between the electrical signals of control and the hydraulic actuator. The electrohydraulic servovalve is the essential element of all the electrohydraulic servo-systems. In order to establish a control law that assures the desired performances, it is necessary to establish a mathematical model of the servovalve, model that must be as exact as possible.
We must also notice the fact that the very small frictions and the high hydraulic rigidity of this system (the very small coefficients of damping) raise us some problems in obtaining the needed performances. Another important characteristic of these high order systems is also the strong nonlinear character of the mathematical model.
Previous literature on electrohydraulic control systems has incorporated the servovalve dynamics to various extents. It is well known that these electrohydraulics systems of high order cannot be controlled by a simple law of type PID (the control based on tangent linearized model with a simple feedback or a PID controller cannot assure always the desired performances). In order to design other control laws some authors ignore the servovalve dynamics (Hayase et al. 2000).
Other authors who consider servovalve dynamics have used either an assumed second-order model or a third-order model. Nonlinear servovalve models have been presented in several papers (Bobasu 2002a), (Bobasu 2002b), (Hayase et al. 2000), (Kim et al. 2000).
In order to take into account the nonlinearities of these systems, some types of control laws have been elaborated (the control law based on tangent linearized model, the exactly linearizing control law, the robust and the adaptive control law).
The establishment of these control laws was made only for the models of reduced order and neglecting the Coulomb friction force and the force due to the hydraulic rigidity and the losses of the flow. The neglecting of these losses and of friction must be done only after a carefully analysis, because the system may become uncontrollable and/or unobservable (Bobasu 2001).
The paper is organized as follows: in Section 2, the models of the electrohydraulical systems are studied, Section 3 deals with the reduction-order techniques for these models and Section 4 presents the simulation results and comparisons. Finally, Section 5 collects the conclusions.

2. DYNAMICS MODELS OF ELECTROHYDRAULIC SYSTEM

The electrohydraulic system shown in Fig. 1 consists of a two-stage flow control servovalve and a double-ended actuator. The servovalve has a symmetrical double-nozzle and a torque-motor driven flapper for the first stage, and a closed center four-way sliding spool for the second stage. Figure 1 displays two types of feedback spring commonly used: a cantilever spring connecting the flapper and spool, and a spring directly acting on the spool.

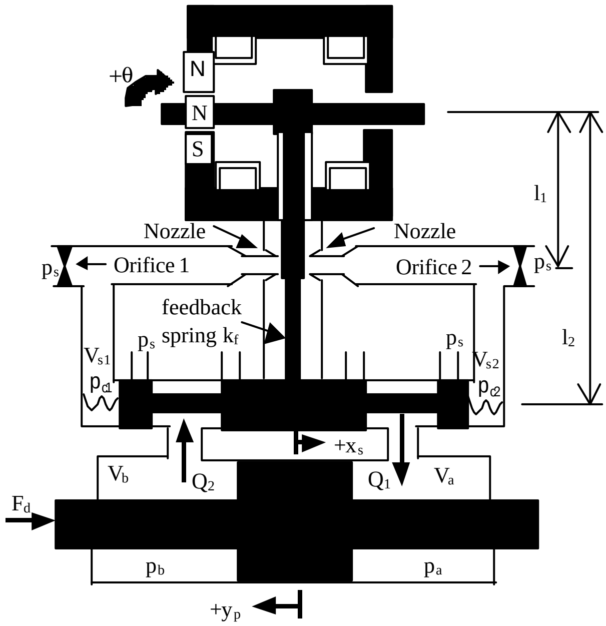


Fig.1 Schematic view of the electrohydraulic servovalve

The nonlinear dynamic model presented next is a compilation of results (Bobasu 2000b), (Hayase et al. 2000), (Kim et al 2000).

The torque-motor stage dynamics are given by

$$\frac{di}{dt}=\frac{Ga}{L}u-\frac{R}{L}i \tag{1}$$

$$\frac{J}{l_1}\ddot{x}_F=\alpha\cdot i-F_{Rj}l_1-k_f l_2\left(\frac{l_2}{l_1}x_F+x_s\right)-\frac{B_F l_2}{2}\dot{x}_F \tag{2}$$

where the flow force acting on the flapper is determined by the pressure difference between the two nozzles has the expression

$$F_{Rj}=\left\{p_{c1}-p_{c2}+\frac{16c_d^{\,2}}{d^2}\left[(u_0-x_F)^2 p_{c1}-\right.\right.$$

$$\left.\left.-(u_0+x_F)^2 p_{c2}\right]\right\}\frac{\pi d^2}{4} \tag{3}$$

The flapper-nozzle stage dynamics are given by

$$\dot{p}_{c1}=\frac{E}{V_{s0}+A_s x_s}\left[c_0 A_c\sqrt{\frac{2(p_s-p_{c1})}{\rho}}-\right.$$

$$\left.-c_d\pi d(u_0-x_F)\sqrt{\frac{2p_{c1}}{\rho}}-A_s\dot{x}_s\right] \tag{4}$$

$$\dot{p}_{c2}=\frac{E}{V_{s0}-A_s x_s}\left[c_0 A_c\sqrt{\frac{2(p_s-p_{c2})}{\rho}}-\right.$$

$$\left.-c_d\pi d(u_0+x_F)\sqrt{\frac{2p_{c2}}{\rho}}+A_s\dot{x}_s\right] \tag{5}$$

The force balance on spool is given by

$$\ddot{x}_s=\frac{1}{M_s}\left[A_s(p_{c1}-p_{c2})-B_s\dot{x}_s-k_f\left(x_s+\frac{l_2}{l_1}x_F\right)-\right.$$

$$\left.-2k_s x_s\right] \tag{6}$$

Note that k_s=0 for a servovalve with a cantilever feedback spring, and k_f=0 with a direct feedback spring.

The flow continuity through actuator is given by

$$\dot{p}_a=\frac{E}{V_{a0}+S_a y_p}\left[c_s w(\varepsilon+x_s)\sqrt{\frac{2(p_s-p_a)}{\rho}}-\right.$$

$$\left.-c_s w(\varepsilon-x_s)\sqrt{\frac{2p_a}{\rho}}-S_a\dot{y}_p\right] \tag{7}$$

$$\dot{p}_b=\frac{E}{V_{b0}-S_b y_p}\left[c_s w(\varepsilon-x_s)\sqrt{\frac{2(p_s-p_b)}{\rho}}-\right.$$

$$\left.-c_s w(\varepsilon+x_s)\sqrt{\frac{2p_b}{\rho}}+S_b\dot{y}_p\right] \tag{8}$$

The force balance on actuator is given by

$$\ddot{y}_p=\frac{1}{M_p}\left[S_a p_a-S_b p_b-F_r-F_c-F_d\right] \tag{9}$$

where

$$Fr=E\left(\frac{S_a^{\,2}}{V_{a0}+S_a y_p}+\frac{S_a^{\,2}}{V_{b0}-S_b y_p}\right)y_p \tag{10}$$

$$F_c=f_s\,sgn(\dot{y}_p) \tag{11}$$

where:

u is the input voltage to servovalve; G_a gain of servo amplifier; J moment of inertia of torque motor; R resistance; L inductance of torque motor; l_1 length of flapper; l_2 length of feedback spring; k_f feedback spring constant; k_s stiffness of each direct feedback spring at the spool; B_F drag coefficient of flapper; x_F displacement of flapper measured from the center position; x_s displacement of spool measured from the center position; α gain of torque motor; $A_c=A_{c1}=A_{c2}$ cross-sectional area of orifice; c_d nozzle flow coefficient; d diameter of nozzle; u_0 maximum flapper displacement; A_s spool area; B_s damping coefficient of spool; θ angular position of armature/flapper; p_{c1}, p_{c2} pressures on left and right side of spool, respectively; E bulk-modulus of fluid; c_0 orifice flow coefficient; p_s supply pressure; A_s area of spool valve; x_s spool position; ρ density of fluid; V_{s0} enclosed volume on each side of spool when $x_s=0$; M_s spool mass; w port width; ε underlap of spool; p_a, p_b pressure in left and right cylinder chambers, respectively; $V_{a0}=V_{b0}$ enclosed volume on each side of actuator where y=0; S effective area of double-ended piston; M piston mass; B_c damping coefficient of actuator; F_r the force introduced by the hydraulic rigidity: F_c frictional force; F_d disturbance force input on actuator; *sgn* denotes a sign function.

The order of nonlinear model for the electrohydraulic system described by the equations (1)-(11) is 11. It is difficult to apply the feedback linearization technique directly to this rather complicated system.

3. REDUCED-ORDER MATHEMATICAL MODELS

3.1. Nonlinear reduced-order models

The control laws designed for these systems are based on a mathematical model of reduced order. In order to obtain the needed performance, some nonlinear control laws have been designed (feedback linearization technique).

The reduction of these systems order is based on the singular perturbations method. The electrohydraulic servovalve is made by subsystems described by differential equations with small parameter, which multiplies the highest derivative. Due to the high value of the bulk-modulus of fluid, to the low value of the fluid occupied volume and also to the spool valve mass low, the equations describing the dynamical behavior of the spool valve can be considered singular perturbed equations.

We consider $\varepsilon_1 = \dfrac{V_{s0}}{E}$ and $\varepsilon_2 = \dfrac{V_{a0}}{E}$.

As the first step we consider the simplified nine-order. Pressure in the left and right side of spool p_{c1} and p_{c2} in the expression are given as

$$p_{c1} = \frac{\left(b_2^{\,2} - b_1^{\,2}\right)\dot{x}_s^{\,2} - 2b_1 b_2 \dot{x}_s \sqrt{\left(b_1^{\,2} + b_2^{\,2}\right)p_s - \dot{x}_s^{\,2}}}{\left(b_1^{\,2} + b_2^{\,2}\right)} + \frac{b_1^{\,2}\left(b_1^{\,2} + b_2^{\,2}\right)p_s}{\left(b_1^{\,2} + b_2^{\,2}\right)} \tag{12}$$

$$p_{c2} = \frac{\left(b_4^{\,2} - b_3^{\,2}\right)\dot{x}_s^{\,2} + 2b_3 b_4 \dot{x}_s \sqrt{\left(b_3^{\,2} + b_4^{\,2}\right)p_s - \dot{x}_s^{\,2}}}{\left(b_3^{\,2} + b_4^{\,2}\right)} + \frac{b_3^{\,2}\left(b_3^{\,2} + b_4^{\,2}\right)p_s}{\left(b_3^{\,2} + b_4^{\,2}\right)} \tag{13}$$

where

$$b_1 = \frac{c_0 A_c}{A_s}\sqrt{\frac{2}{\rho}};\; b_2 = \frac{c_d \pi d\left(u_0 - x_F\right)}{A_s}\sqrt{\frac{2}{\rho}};\; b_3 = \frac{c_0 A_c}{A_s}\sqrt{\frac{2}{\rho}} \tag{14}$$

$$b_4 = \frac{c_d \pi d\left(u_0 + x_F\right)}{A_s}\sqrt{\frac{2}{\rho}}$$

Introducing Eqs. (12), (13) and (14) into Eq. (6), we obtain the following relation

$$M_s \ddot{x}_s = f_s\left(\dot{x}_s, x_F\right) \tag{15}$$

In the expression (15) f_s represents the driving force of the spool as a nonlinear function of the spool velocity $\dot{x}_s$ and the displacement of flapper x_F.

In order to obtain reduced model of the order 7 we also consider that $\varepsilon_2 = 0$.

Pressure in the left and right cylinder chambers p_a and p_b in the expression are given as

$$p_a = \frac{\left(a_2^{\,2} - a_1^{\,2}\right)\dot{y}_p^{\,2} - 2a_1 a_2 \dot{y}_p \sqrt{\left(a_1^{\,2} + a_2^{\,2}\right)p_s - \dot{y}_p^{\,2}}}{\left(a_1^{\,2} + a_2^{\,2}\right)} + \frac{a_1^{\,2}\left(a_1^{\,2} + a_2^{\,2}\right)p_s}{\left(a_1^{\,2} + a_2^{\,2}\right)} \tag{16}$$

$$p_b = \frac{\left(a_4^{\,2} - a_3^{\,2}\right)\dot{y}_p^{\,2} - 2a_3 a_4 \dot{y}_p \sqrt{\left(a_3^{\,2} + a_4^{\,2}\right)p_s - \dot{y}_p^{\,2}}}{\left(a_3^{\,2} + a_4^{\,2}\right)} + \frac{a_3^{\,2}\left(a_3^{\,2} + a_4^{\,2}\right)p_s}{\left(a_3^{\,2} + a_4^{\,2}\right)} \tag{17}$$

where

$$a_1 = \frac{c_s w}{S}\sqrt{\frac{2}{\rho}}\left(\varepsilon + x_s\right);\; a_2 = \frac{c_s w}{S}\sqrt{\frac{2}{\rho}}\left(\varepsilon - x_s\right);$$

$$a_3 = \frac{c_s w}{S}\sqrt{\frac{2}{\rho}}\left(\varepsilon - x_s\right);\; a_4 = \frac{c_s w}{S}\sqrt{\frac{2}{\rho}}\left(\varepsilon + x_s\right) \tag{18}$$

Introducing Eqs. (16), (17) and (18) into Eq. (9), we obtain the following relation

$$M_p \ddot{y}_p = f_p\left(\dot{y}_p, x_s\right) \tag{19}$$

In the expression (19), f_p represents the driving force of the piston as a nonlinear function of the piston velocity $\dot{y}_p$ and the spool displacement x_s.

Because the nonlinear mathematical reduced-order models of the electrohydraulical system are not analytical linear, the exactly linearization technique through state feedback cannot be applied, and thus the nonlinear control law cannot be obtained.

For that reason, in order to apply the exactly linearization technique, we consider that the hydraulical cylinder is controlled with two three-ways electrohydraulical servovalves or with a single fifth-ways electrohydraulical servovalve. Also, the motor-torque dynamic and the dry friction are neglected.

Under these assumptions, the nonlinear mathematical model of the electrohydraulical system is described by the next differential equations:

$$\dot{p}_a = \frac{E}{V_{a0} + S_a y_p}\left[Q_{m1} - q_e p_a - q_i\left(p_a - p_b\right) - S_a \dot{y}_p\right] \tag{20}$$

$$\dot{p}_b = \frac{E}{V_{b0} - S_b y_p}\left[Q_{m2} - q_e p_b + q_i\left(p_a - p_b\right) + S_b \dot{y}_p\right] \tag{21}$$

$$\ddot{y}_p = \frac{1}{M_p}\left[S_a p_a - S_b p_b - F_r - F_d\right] \tag{22}$$

where

$$Q_{m1}\left(i_a, p_a\right) = A_{sa}\left(i_a\right)\sqrt{p_s - p_a} - A_{ae}\left(i_a\right)\sqrt{p_a} \tag{23}$$

$$Q_{m2}\left(i_b, p_b\right) = A_{sb}\left(i_b\right)\sqrt{p_s - p_b} - A_{be}\left(i_b\right)\sqrt{p_b} \tag{24}$$

q_i, q_e are flow losses factors.

We are using the currents i_a and i_b as control variable. In order to apply the nonlinear control law for the system described by Eqs. (20)-(22), we first have to adjust it to a special form. Therefore we consider that the flow factors for the two servovalves are not depending with the currents i_a and i_b and we will use two new control variables A_a^* and A_b^* witch are supposed to contribute linear in state equations.

Thus, (23) and (24) will be changed by

$$Q_{m1} = -Q_{pa}\left(p_a\right) + \varphi_a\left(p_a, sgn\left(A_a^*\right)\right)A_a^* \tag{25}$$

$$Q_{m2} = -Q_{pb}\left(p_b\right) - \varphi_b\left(p_b, sgn\left(A_b^*\right)\right)A_b^* \tag{26}$$

where

$$A_a^*\left(i_a\right) = \Psi_a\left(i_a\right) = \begin{cases} A_{sa}\left(i_a\right) - A_{pa} & if\ i_a \ge i_{a0} \\ -\left(A_{ae}\left(i_a\right) - A_{pa}\right) & if\ i_a < i_{a0} \end{cases} \tag{27}$$

$$A_b^*\left(i_b\right) = \Psi_b\left(i_b\right) = \begin{cases} A_{be}\left(i_b\right) - A_{pb} & if\ i_b \ge i_{b0} \\ -\left(A_{sb}\left(i_b\right) - A_{pb}\right) & if\ i_b < i_{b0} \end{cases} \tag{28}$$

$$\varphi_a\left(p_a, sgn\left(A_a^*\right)\right) = \begin{cases} \sqrt{p_s - p_a} & if\ A_a^* \geq 0 \\ \sqrt{p_a} & if\ A_a^* \langle 0 \end{cases} \quad (29)$$

$$\varphi_b\left(p_b, sgn\left(A_b^*\right)\right) = \begin{cases} \sqrt{p_s - p_b} & if\ A_b^* \langle 0 \\ \sqrt{p_b} & if\ A_a^* \geq 0 \end{cases} \quad (30)$$

$$Q_{pa} = A_{pa}\left(\sqrt{p_a} - \sqrt{p_s - p_a}\right) \quad (31)$$

$$Q_{pb} = A_{pb}\left(\sqrt{p_b} - \sqrt{p_s - p_b}\right) \quad (32)$$

Introducing Eqs. (25)-(32) into Eqs. (20) and (21), choosing as state variables x^T=[p_a, p_b, v_p, y_p] and the inputs vector $u^T = \left[A_a^*(i_a), A_b^*(i_b)\right]$, the state space representation (20)-(22) can be written as a linear-analytical form:

$$\dot{x}(t) = f(x) + g(x)u(t) \quad (33)$$

in which smooth vector fields f(x) and g(x) have the following expressions

$$f(x) = \begin{bmatrix} -\dfrac{E\left[Q_{pa} + q_e p_a + q_i(p_a - p_b) + S_a \dot{y}_p\right]}{V_{a0} + S_a y_p} \\ \dfrac{E\left[-Q_{pb} - q_e p_b + q_i(p_a - p_b) + S_b \dot{y}_p\right]}{V_{b0} - S_b y_p} \\ \dfrac{1}{M_p}\left[S_a p_a - S_b p_b - F_r\right] \\ v_p \end{bmatrix} \quad (34)$$

$$g(x) = \begin{bmatrix} \dfrac{E\varphi_a\left(p_a, sgnA_a^*\right)}{V_{a0} + S_a y_p} & 0 \\ 0 & -\dfrac{E\varphi_b\left(p_b, sgnA_b^*\right)}{V_{b0} - S_b y_p} \\ 0 & 0 \\ 0 & 0 \end{bmatrix} \quad (35)$$

For the system (33), a nonlinear control law can be designed, both in the monovariable case (the electrohydraulic system controlled by a fifth-ways servovalve) and in the multivariable case (the electrohydraulic system controlled by a three-ways servovalve) (Bobasu et al. 1996).

In the paper (Hayase et al., 2000) it is considered that the reduced model of order two which is obtained following hypothesis: the dynamics of the torque motor, the compressibility of the working fluid and the Coulomb friction on the piston are all ignored, only retaining the nonlinear pressure-flow characteristic of the spool valve. In the simplified model the spool displacement is proportional to the input signal $x_s = Ku$.

In order to apply the exactly linearization technique, Hayase has considered only the Eq. (19), and the nonlinear system (19) coupled with the linearizing control law has a double integrator behavior.

3.2. Linearized reduced-order models

The derivation of the linearized model with respect to an equilibrium state from the above nonlinear model is tedious but straightforward. The equilibrium states are derived for zero inputs, $u = 0,\ F_d = 0,\ x_F = 0,\ x_s = 0,\ y_p = 0$. We obtain $p_{c10} = p_{c20} = p_{c0} = p_s / 2$.

Linearizing the mathematic model of electrohydraulic system around the equilibrium position, we obtain the next linearized mathematic model:

$$\begin{bmatrix} \dot{x} \\ \dot{z} \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \cdot \begin{bmatrix} x \\ z \end{bmatrix} + \begin{bmatrix} B_1 \\ B_2 \end{bmatrix} \cdot u \quad (36)$$

where

$$A_{11} = \begin{bmatrix} -a2 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ a3 & -a4 & -a5 & -a7 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & -a14 & 0 & -a13 & -a12 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & 0 & 0 & -a19 \end{bmatrix}$$

$$A_{12} = \begin{pmatrix} 0 & 0 \\ 0 & 0 \\ -a6 & 0 \\ 0 & 0 \\ a11 & 0 \\ 0 & 0 \\ 0 & a18 \end{pmatrix} \quad (37)$$

$$A_{21} = \begin{bmatrix} 0 & a9 & 0 & 0 & -a10 & 0 & 0 \\ 0 & 0 & 0 & a16 & 0 & 0 & -a17 \end{bmatrix}$$

$$A_{22} = \begin{bmatrix} -a8 & 0 \\ 0 & -a15 \end{bmatrix};\ B_1^T = \begin{bmatrix} a1 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}$$

$$B_2^T = \begin{bmatrix} 0 & 0 \end{bmatrix}$$

$$x^T = \begin{bmatrix} i & x_F & v_F & x_s & v_s & y_p & v_p \end{bmatrix} \in R^7$$

$$z^T = \begin{bmatrix} \Delta p_c & \Delta p \end{bmatrix} \in R$$

in which

$$a1 = \frac{G_a}{L};\ a2 = \frac{R}{L};\ a3 = \frac{l_1 \alpha}{J}$$

$$a4 = \frac{k_f l_2^2 - 16 c_d^2 \pi u_0 p_{s0} l_1^2}{J};\ a5 = \frac{B_F l_1 l_2}{2J}$$

$$a6 = \frac{\pi l_1^2\left(d^2 + 16 c_d^2 u_0^2\right)}{4J};\ a7 = \frac{k_f l_1 l_2}{J} \quad (38)$$

$$a8 = \frac{1}{\sqrt{\rho p_s}}\left(c_0 A_c + c_d \pi d u_0\right);\ a9 = 2 c_d \pi d \sqrt{\frac{p_s}{\rho}}$$

$$a10 = 2A_s;\ a11 = \frac{A_s}{M_s};\ a12 = \frac{B_s}{M_s};\ a13 = \frac{k_f}{M_s}$$

$$a14 = \frac{k_f l_2}{M_s l_1};\ a15 = \frac{2 c_s w a}{\sqrt{\rho p_s}};\ a16 = 4 c_s w \sqrt{\frac{p_s}{\rho}};\ a17 = 2S$$

$$a18 = \frac{S}{M};\ a19 = \frac{B_c}{M};\ S_a = S_b = S$$

$$\varepsilon_1 = \frac{V_{s0}}{E};\ \varepsilon_2 = \frac{V_{a0}}{E};\ \varepsilon_3 = J$$

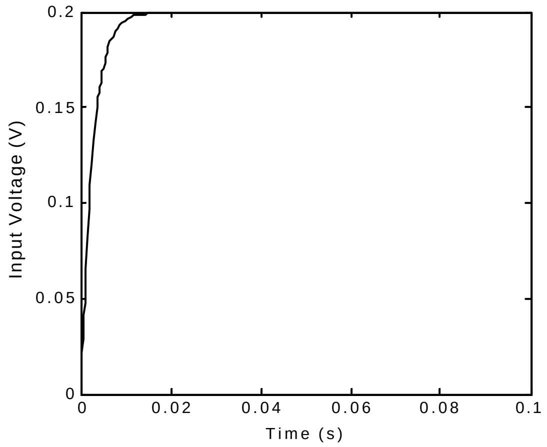
4 SIMULATION RESULTS

4.1. The mathematical complete model of the analyzed elctrohydraulic servovalve has the order 11 and it is strongly nonlinear. In order to obtain some mathematical models of reduced order, the singular perturbation method is applied. Two reduced models of order 9 respectively 7 are obtained by using the singular perturbation method. The coefficients values for these models were computed using numerical values from (Hayase et al., 2000).

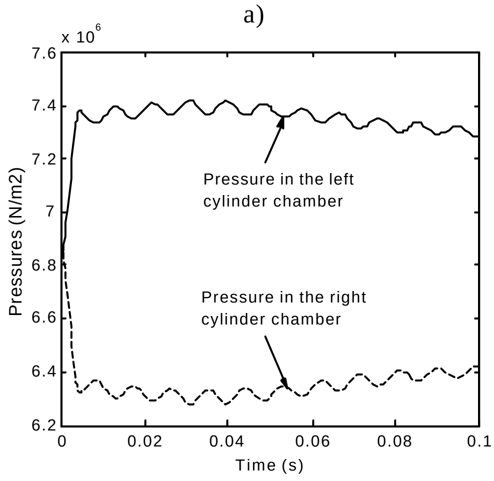
The dynamic behavior of these two models was compared with the dynamic behavior of the reduced model of order two (Hayase et. all., 2000).

In Fig. 2a the dynamic behavior for the initial model is presented. For an input voltage u=10V and the disturbance force F=1500 N, the current through the torque motor is quickly stabilized to 0.2 A. Also is presented the pressures evolution for the left and right cylinder chambers (Fig. 2b).

In Fig. 2c is presented a comparison for the actuator position between the full order model and reduced order model (11 and 2 degree respectively).



a)



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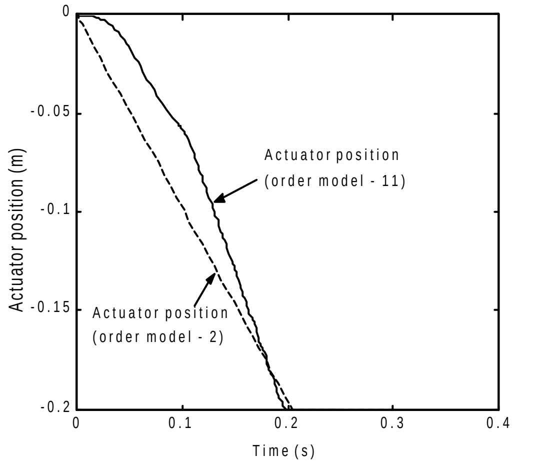

c)

Fig. 2. Dynamical behavior of servovalve

4.2. For the analyzed electrohydraulic system a linearized model of 9-order is considered. The coefficient values for this model were computed using numerical values from (Hayase et. al 2000). Then, the singular perturbation method was applied for this model considering initial that the parameters ε_1 and ε_2 are very small and a reduced model of 7-order was obtained.

In Fig. 3 the frequency characteristic for the initial model and the reduced model are presented. It can be seen a good behavior of the reduced order model until 10^3 rad/s (3 b).

If we consider that the parameter ε_3 is also very small a reduced model of order 6 is obtained.

In Fig. 4 the frequency characteristic for the initial model and the reduced model are presented. It can be seen a good behavior of the reduced order model until 10^3 rad/s (4 b).

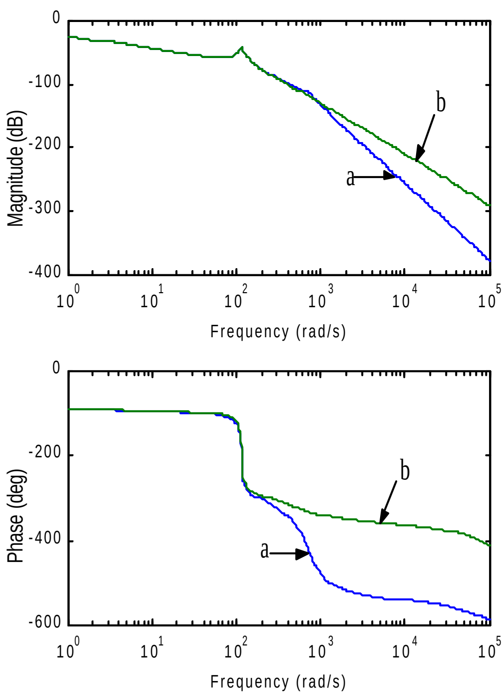

Fig. 3.

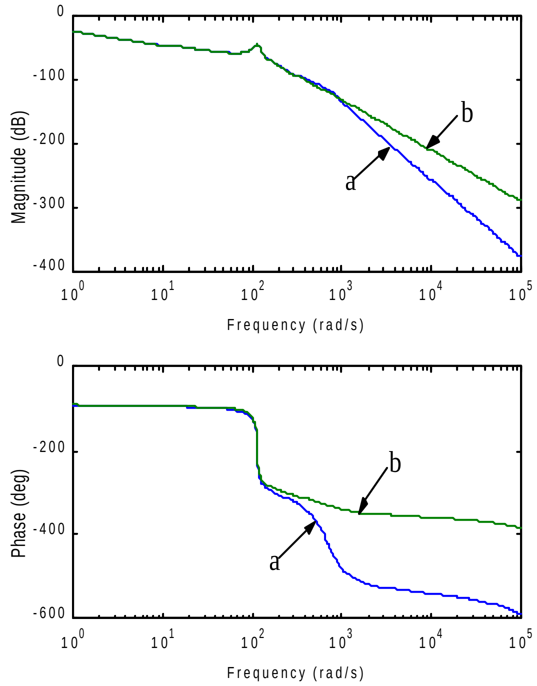

Fig. 4.

5. CONCLUDING REMARKS

The model coefficients are expressed in term of the system physical parameters and therefore reveal model structural properties. Using the singular perturbation method, the initial nonlinear model of 11-order has been reduced to 9, respectively 7-order system. The dynamic behavior of the reduced order models approximates quite well the complete order model dynamic. The neglected dynamic is not important in the real environment control.

A linearized servovalve model has been derived from the nonlinear model for an electrohydraulic system consisting of a linear actuator piston and a two-stage servovalve.

Using the singular perturbation method the initial linearized model of 9-order has been reduced to 7, respectively 6-order systems.

The frequency characteristics show that the approximations are good in the frequency range of the electrohydraulical servovalve.

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