DEVELOPMENT OF A CONTROLLER FOR AN ACTIVE SUSPENSION SYSTEM FOR HIGH PERFORMANCE ELEVATORS

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Abstract. *This work deals with the development of a feedback controller for an active suspension system to be applied in a high performance elevator. The elevator has a double-deck cabin and will be used in a 200 [m] course guide in a skyscraper of almost 500 [m]. The elevator's steady state velocity is 40 km/h. The development of the controller is based on the use of a linear state space model that represents the system's dynamics in two orthogonal planes. The synthesis of a state space feedback controller was done using the poles placement method with a full state observer. The parameters of the real system were measured and identified and a computational model was constructed. Results of passive and active system simulations are presented and discussed.*

Keywords*: active suspension, elevator dynamics control, high performance elevators, state feedback control, pole placement.*

1. INTRODUCTION

This work introduces the development of an active suspension system for high performance elevators that aims to reduce the lateral vibrations in the base of the elevator's cabin. The oscillations are caused mainly by disturbances that come from the contact between the rollers parts of the suspensions with the alignment guides fixed to the building structure during the elevator's movement. These guides, although modern processes of manufacture and installation, present inherently small misaligns that are caused, normally, by disturbances in the installation processes, thermal dilatations and long period effects of the materials behavior, among others. The movements caused for these misalignments, mainly for the high speed elevators of high building (skyscrapers), can compromise the requirements of security and comfort. Consequently, high technology companies are spending resources in the development of modern systems, as active control, to improve the performance of the elevator's suspensions without increasing its size and weight, as occurs with the traditional passive systems. This situation can be proven by the elevated number of patents in the area since the middle of the last decade. For example, Oh *et al.* (2006) uses a suspension system based on the control of repulsive forces of electromagnets to cancel lateral disturbances. Utsunomiya *et al.* (2004 and 2006) patented an active suspension system that uses the measured accelerations (using accelerometers located on the center of the elevator) to compare the cabin vibrations with the desired values and thus applying lateral forces by means of electromagnetic actuators. Husmann (2005) describes an active vibration damping system for the structural frame based on the measurement of the deformations of the frame of the elevator. The acceleration is measured with electro-resistive sensors fixed to the main frame in the perpendicular direction to the movement. Linear motors apply the forces requested for the control system.

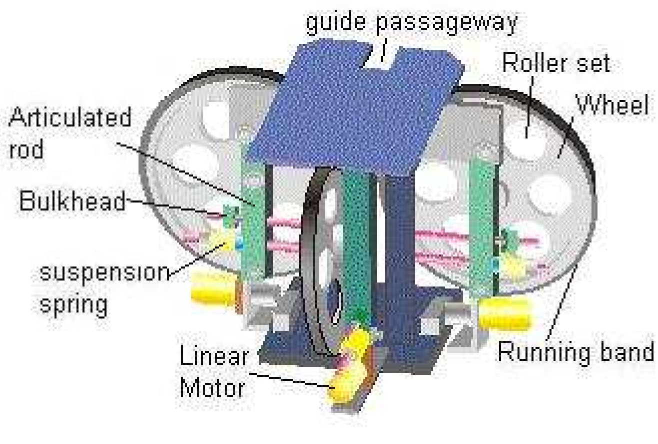
It was found in the bibliography few specific works about active suspension control of commercial elevators. For example, Istif *et al.* (2002) uses *Bond Graphs* to construct a states space model of a active suspension system. The control project is carried through a proportional-differential (PD) scheme to command a hydraulical system in the vertical direction of the elevator. The active system is used to control the acceleration and deceleration of the elevator aiming the reaching of comfortable behavior of the elevator. Nai *et al.* (1994) takes in account 20 differential second order equations to obtain a model for an elevator. The dynamics is invariant in time and depends on the position, amount of passengers and the axis being considered. They use two different strategies for the control, based on the variation of the speed to control the frequencies of vibration of the cables and of the insulators of the elevator (low frequencies), and employ another one based in the use of actuators in the suspension system to control the resonance in the cables and insulators through the uses of pole placement method with the project and implementation of a states observer. Schneider *et al.* (2001) models the system's dynamics of an elevator. The control strategy is based on neural networks and is used to control acceleration and deceleration of the elevator's cabin. Sha *et al.* (2002) presents a dynamic model of an elevator with hydraulical system using the classic equations of Newton and the continue flow laws. The different frictions involved in the system also are taken in account. The model is used to perform simulations via Matlab/Simulink. The solution proposed for the control is an Adaptive Sliding Mode Controller (ASMC) for discrete SISO systems. This control strategy combines nonlinear feedback control based on the direct method of Lyapunov and an adaptive sliding mode scheme. The reference dynamics for the system is specified with the use of and the pole placement method. The proposed control strategy is compared with a classic PID and the results show an improvement due to the proposed control technique. Skalski (1984) presents and compares two methods used at this time to carry through the control of the speed, acceleration and deceleration of elevators. The methods compared are the Silicon-Controlled Rectifier (SCR) Velocity Control and the Motor Generator (MG) Velocity Control. Both involve basically a PI control acting directly on the motor of the elevator. Experimental and simulation results show that both systems achieved good results, but with SCR-Drive is shown a better performance.

In the present work, aiming the development of an algorithm to control an active suspension system based on the use of linear electromagnetic actuators, a simplified model for the dynamics of the elevator suspension was taken into account using the classical methods of the rigid body dynamics. The controller was developed based on the method of pole placement (Ogata (2003), Friedland (2005), Franklin *et al.* (1994) and Preumont (2002)). The concepts and methods used in this work are of ample application in the project of systems to control active suspensions of automotive vehicles (the available bibliography on this application is far more extensive). Through the development presented in Section 2, it can be observed that the model considered for the elevator is similar to the classic model of half car, widely studied. Thus, the theoretical considerations found in the bibliography for these systems were extended for the case of the active suspension of elevators. In the field of development of automotives suspensions, Giua *et al.* (2004) presents a suspension system that uses a magneto-rheological actuator that increase or decrease the damping constant, in accordance with the disturbances of the road. Chen *et al.* (1999) developed an electronic controller based on a micro controller unit for a hydro-pneumatic active suspension with 2 degrees of freedom. The control strategy consists of a combination of a variable structure control (VSC) with a PID scheme. For the control of the vertical vibrations it was used a Linear Quadratic Regulator (LQR) project. The results presented were considered good. Ikenaga *et al.* (2000) and Campos *et al.* (1999) use a control strategy based on the Stability Augmentation System, classically used in the control of airplanes. In this method, the problem is divided in two parts: one part isolates the body of the car from the vibrations of the road and the other part controls the maneuvers of the vehicle. Moreover, in a different approach to the previous one, Ben Gaid *et al.* (2004) present a linear quadratic regulator (LQR) that isolates the mass of the vehicle from the external disturbances and optimizes the car maneuverability. The car's model is based on the full 7 degrees of freedom introduced by Ikenaga *et al.* (2000). The results are obtained through simulation with Matlab/Simulink and compared with a passive suspension model results.

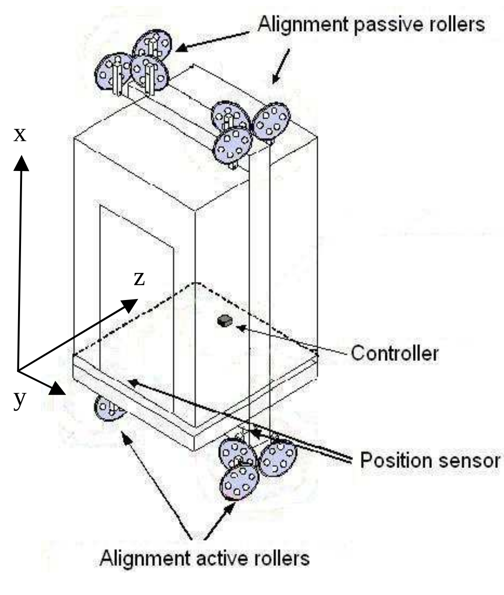
2. DYNAMIC MODELING

2.1. Introduction

In this section the development of the dynamic model of the elevator is presented.



a) Active suspension



b) Suspension systems installed in the elevator

Figure 1: Suspension system

The elevator's cabin is modeled as a solid parallelepiped. According with the objectives of the active suspension (to reduce the oscillatory movements of the base of the cabin, where would be located the passengers), it is necessary to take in account the displacements in two plans (XY and YZ, defined in accord with Fig. 1). In this work, only the XY plan problem will be approached. In Fig. 1a the scheme of the *suspension system* is presented. Four of these sets are used to align the elevator, as it's shown in the Fig.1-b. Each *suspension system* is composed of three arms, each one made up of an independent suspension (articulated connecting rod, spring, roller, linear motor, etc.). The vertical guides, attached to the civil structure, are placed vertically, aligned with the X axis. The rollers (joint + wheel + wheels roller band) are aligned to the XY plan (four) and hinder the movement in the direction of the Y and the rotation around the axis parallel to Z. The rollers are aligned to XZ plan (eight) and tend to hinder the movement in the direction of Z and the rotation about an axis parallel to X.

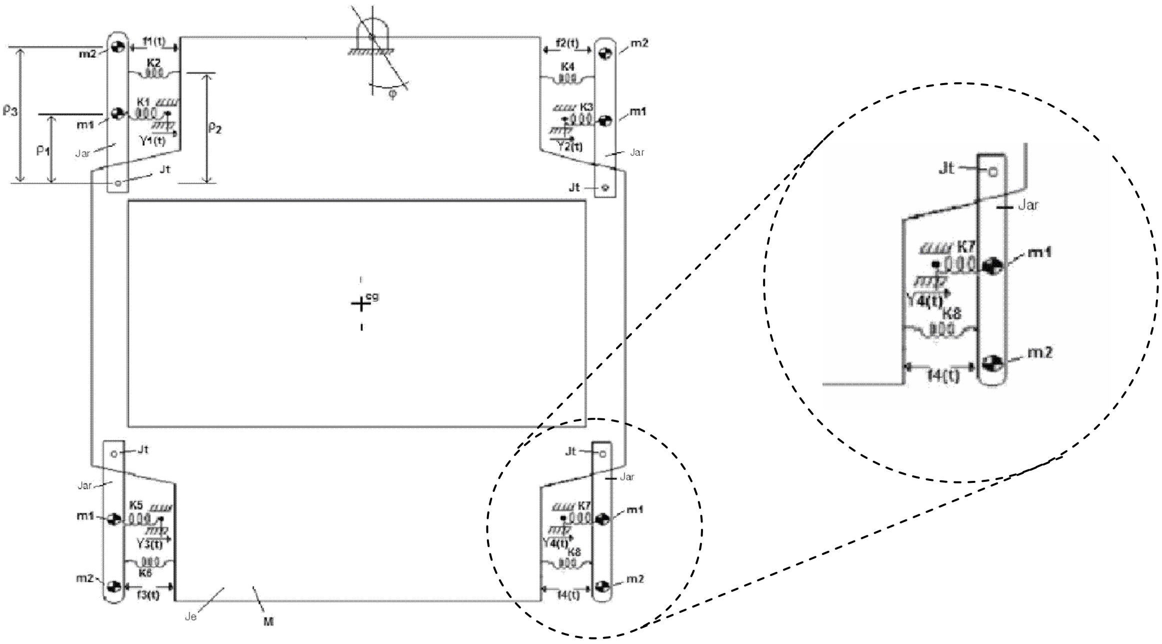


Figure 2: Scheme of the suspension system and elevator

Figure 2 shows the suspension system, its location on the elevator and a schematically drawing of the system. In Fig. 2, the variable and parameters are defined as follows: $f_i(t)$ (i=1,2,3,4) are the forces applied by the actuators, $Y_i(t)$ are the displacements of the rollers which contact the guides. The suspensions sets are considered equal, thus m_1 are the equivalents masses of each roller and m_2 are the mobile mass of each linear actuator. M is the total mass of the elevator (car weight + capacity), ρ_1, ρ_2 and ρ_3 are, respectively, the length of the arms of the rollers, the gyration radius of the connection point of the springs related to the fixed point of the arm of the suspension, and the rotation radius of the mobile mass of the actuator to the pivoting point. J_t is the total inertia moment of each suspension set related to the fixed pivoting point ($J_t = m_1\rho_1^2 + m_2\rho_3^2 + J_{ar}$, where J_{ar} is the moment of inertia of the arm's structure of the suspension referred to the fixed pivoting point). $K_1 = K_3 = K_5 = K_7$ are, respectively, the spring constants of the wheels of the roller's tires and $K_2 = K_4 = K_6 = K_8$ are, respectively, the stiffness constants of the helical springs of the suspensions. Finally, J_e is the elevator's moment of inertia referred to the top rotation center pivoting, where the cables are connected to the structure. In Fig. 2, φ is the rotational angle of the elevator referred to the connection pivoting point of the cables in the top of the cabin $tg\varphi = (Y_{lo} - Y_{up})/L \cong \varphi \Rightarrow \varphi = (Y_{lo} - Y_{up})/L$ where L is the length of the cabin.

2.2. Definition of the basic variables

The present work consists of an initial approach to the project of the active suspension. Thus, only the movement in plan XY is considered in this study. It is still considered, aiming at the simplification of the model for this problem, that the comfort and security of the passengers are associated mainly with the pendulum rigid body movement of the structure of the elevator, as shown in Fig. 2. Hence, using the hypothesis of small displacements, the movement in the base of the cabin, considering the interactions between the components and the gravitational effects, were approached by the system represented in Fig. 3. This simplified configuration models the system as a concentrated translational mass where two passive suspension sets are connected and two actuators are symmetrically located in the base of the cabin, as shown in Fig. 3. Therefore, a simplified representation of the pendulum system is necessary. The torsional spring effect caused by gravitational effect relative to the movement of rotation around the fixed pivoting point of the cabin (where the cables are connected) was considered by mean of the incorporation of a torsional spring to the model ($K_{\mathrm{tg}0} = MgL_0$), where L_0 is the length of the cable of the elevator that generates the pendulum movement referred to the top rotational pivoting center, M is the total mass of the elevator and g is the gravitational constant.

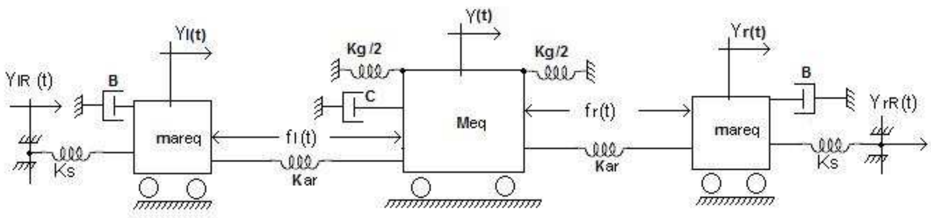


Figure 3: Translational equivalent model

In Fig. 3, K_s are spring translational constant of the roller's wheels bands, $K_{\mathrm{ar}} = K_2(\rho_2/\rho_1)^2$ are the elastic stiffness constants of the suspension's helical springs, so that the spring effects were considered as applied directly in the contact point of the roller's wheels bands with the guides, $m_{\mathrm{areq}} = m_2\rho_3{}^2 + m_1\rho_1{}^2 + J_{\mathrm{ar}}/\rho_3{}^2 = J_t/\rho_3{}^2$ are the equivalent mass of the arms referred to its rotational centers, $K_{\mathrm{tg}} = M_{\mathrm{eq}}gL$ is the torsional spring due to the rotational movement of the elevator around the connection of the cables in the top of the cabin. M_{eq} is the equivalent mass of the car ($M_{\mathrm{eq}} = J_{\mathrm{e}}/L^2$). $K_{\mathrm{g}} = K_{\mathrm{tg}}/L^2$ is the equivalent translational spring constant referred to the gravitational effects. The forces of the actuators were transferred to the position of the contact point between the band of the rollers wheels with the guides and were defined as $f_1(t) = f_3(t)(\rho_2/\rho_1)$ and $f_r(t) = f_4(t)(\rho_2/\rho_1)$. Finally, the displacement of the left arm was defined as $Y_{\mathrm{lR}}(t)$ while $Y_{\mathrm{rR}}(t)$ was for the right one. It was also considered the losses by friction effects through the introduction of equivalent viscous damping terms (dampers C and B).

2.3. Formulation of the motion equations

For the formularization of the motion equations for the system presented in Fig. 3, is still necessary to stipulate the direction of the forces applied for the actuators. For the situations of application of positive signal of control, the closing movement of the actuators was specified (that is, they apply attractive force). On the other hand, for a negative control signal, it was stipulated the opening movement of the actuators (that is, they apply a compression force to the elements in its edges). Through the application of the Second Law of Newton, the resultant equation system is the following one:

$$m_{\mathrm{areq}}\ddot{Y}_{\mathrm{l}} + B\dot{Y}_{\mathrm{l}} + K_{\mathrm{s}}Y_{\mathrm{l}} + K_{\mathrm{ar}}Y_{\mathrm{l}} - K_{\mathrm{ar}}Y = K_{\mathrm{s}}Y_{\mathrm{lR}} + f_{\mathrm{l}} \tag{1}$$

$$-K_{\mathrm{ar}}Y_{\mathrm{l}} + K_{\mathrm{ar}}Y + M_{\mathrm{eq}}\ddot{Y} + K_{\mathrm{g}}Y + C\dot{Y} + K_{\mathrm{ar}}Y - K_{\mathrm{ar}}Y_{\mathrm{r}} = f_{\mathrm{l}} + f_{\mathrm{r}} \tag{2}$$

$$-K_{\mathrm{ar}}Y + K_{\mathrm{ar}}Y_{\mathrm{r}} + m_{\mathrm{areq}}\ddot{Y}_{\mathrm{r}} + B\dot{Y}_{\mathrm{r}} + K_{\mathrm{s}}Y_{\mathrm{r}} = K_{\mathrm{s}}Y_{\mathrm{rR}} + f_{\mathrm{r}} \tag{3}$$

The real system uses potentiometers sensors to measure the relative displacement between the arms and the cabin (usually, in the literature, it is presented the use of accelerometers - the use of potentiometers aims the development of an alternative system). The main objective of the control will be the regulation of the dynamic behavior of the states in a way that the responses of the degrees of freedom of the system, mainly the trajectory of the cabin, converge fast and smoothly for the central equilibrium position, providing a comfortable and safe travel to the passengers.

$$\Delta y_l = Y_l - Y \Rightarrow Y_l = \Delta y_l + Y \tag{4}$$

$$\Delta y_r = Y_r - Y \Rightarrow Y_r = \Delta y_r + Y \tag{5}$$

Thus, deriving once and twice the Eqs. (4) and (5) and substituting them in to Eqs. (1) up to (3), it is obtained,

$$\ddot{Y} = \frac{1}{M_{eq}}\left[f_l + f_r + K_{ar}\Delta y_l - K_g Y - C\dot{Y} + K_{ar}\Delta y_r\right] \tag{6}$$

Substituting the derivatives of Eqs. (4), (5) and (6) in the expressions of $\Delta\ddot{y}_l$ and $\Delta\ddot{y}_r$, results

$$\begin{aligned}\Delta\ddot{y}_l = & \frac{1}{m_{areq}}\left[K_s Y_{lR} + f_l - B\Delta\dot{y}_l - B\dot{Y} - K_s\Delta y_l - K_s Y - K_{ar}\Delta y_l\right] - \\ & -\frac{1}{M_{eq}}\left[f_l + f_r + K_{ar}\Delta y_l - K_g Y - C\dot{Y} + K_{ar}\Delta y_r\right]\end{aligned} \tag{7}$$

$$\begin{aligned}\Delta\ddot{y}_r = & \frac{1}{m_{areq}}\left[K_s Y_{rR} + f_r - K_{ar}\Delta y_r - B\Delta\dot{y}_r - B\dot{Y} - K_s\Delta y_r - K_s Y\right] - \\ & -\frac{1}{M_{eq}}\left[f_l + f_r + K_{ar}\Delta y_l - K_g Y - C\dot{Y} + K_{ar}\Delta y_r\right]\end{aligned} \tag{8}$$

2.4. State variables

Equations (6), (7) and (8) represent the system in the state space format, where the state variable set is defined as $\mathbf{x} = \left[\Delta y_l \quad \Delta\dot{y}_l \quad Y \quad \dot{Y} \quad \Delta y_r \quad \Delta\dot{y}_r\right] = \left[x_1 \quad x_2 \quad x_3 \quad x_4 \quad x_5 \quad x_6\right]$. The representation in states-space matrix format $\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u}$ is:

$$\begin{aligned}\begin{pmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\\ \dot{x}_4\\ \dot{x}_5\\ \dot{x}_6\end{pmatrix} = & \begin{pmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\frac{(K_s+K_{ar})}{m_{areq}} - \frac{K_{ar}}{M_{eq}} & -\frac{B}{m_{areq}} & \left(\frac{K_g}{M_{eq}} - \frac{K_s}{m_{areq}}\right) & \left(\frac{C}{M_{eq}} - \frac{B}{m_{areq}}\right) & -\frac{K_{ar}}{M_{eq}} & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{K_{ar}}{M_{eq}} & 0 & -\frac{K_g}{M_{eq}} & -\frac{C}{M_{eq}} & \frac{K_{ar}}{M_{eq}} & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ -\frac{K_{ar}}{M_{eq}} & 0 & \left(\frac{K_g}{M_{eq}} - \frac{K_s}{m_{areq}}\right) & \left(\frac{C}{M_{eq}} - \frac{B}{m_{areq}}\right) & -\left(\frac{K_{ar}}{M_{eq}} + \frac{K_{ar}+K_s}{m_{areq}}\right) & -\frac{B}{m_{areq}} \end{pmatrix} \begin{pmatrix}x_1\\ x_2\\ x_3\\ x_4\\ x_5\\ x_6\end{pmatrix} \\ & + \begin{pmatrix} 0 & 0 & 0 & 0 \\ \left(\frac{1}{m_{areq}} - \frac{1}{M_{eq}}\right) & -\frac{1}{M_{eq}} & \frac{K_s}{m_{areq}} & 0 \\ 0 & 0 & 0 & 0 \\ \frac{1}{M_{eq}} & \frac{1}{M_{eq}} & 0 & 0 \\ 0 & 0 & 0 & 0 \\ -\frac{1}{M_{eq}} & \left(\frac{1}{m_{areq}} - \frac{1}{M_{eq}}\right) & 0 & \frac{K_s}{m_{areq}} \end{pmatrix} \begin{pmatrix} f_l \\ f_r \\ Y_{lR} \\ Y_{rR}\end{pmatrix}\end{aligned} \tag{9}$$

3. PROJECT OF THE SUSPENSION CONTROL

In this section it will be applied the pole placement method, following the steps of Ogata (1996) and Ogata (2003) for the project of the feedback control system. The pole placement method is analogous to the root place method, because, as in that method, the poles of the closed loop are placed in the desired positions. This method is also known as modern method because of the contrast with the classic method (based on frequency domain analysis). It is important to remark that the main advantage of this method is, for fully controllable systems, the possibility of choosing the location of the dominant poles as the not dominant ones. It makes this method to be very powerful, because it allows the designer usually to obtain a closed response to the desired one (Ogata (1996)). The desired closed loop poles can be determined based on the requirements of the response of the transient regimen and/or on the frequency response. Through the appropriate choice of a gain matrix for the feedback states, it is possible to force the system to have closed loop poles in the desired locations. However, there is a cost associated with the allocation of all the closed loop poles. It is due to the fact that the placement of all poles requires the measurement of all the states variables or the inclusion of a state observer in the system. Also, the system must be completely controllable so that the closed loop poles could be situated in arbitrarily chosen localizations.

3.1. Considerations for the application of state feedback control method

Considering the control system as $\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}u$ and $y = \mathbf{C}\mathbf{x} + \mathbf{D}u$ where $\mathbf{x}$ = state vector (vector of n order), y =output signal (scalar), u =control signal (scalar), $\mathbf{A}$ = constant matrix nxn, $\mathbf{B}$ = constant matrix nx1, $\mathbf{C}$ = constant matrix 1xn, $\mathbf{D}$ = constant (scalar). The matrices $\mathbf{A}$ and $\mathbf{B}$ are obtained from the Eq. (9). The methodology applied in this work follows the steps given in Ogata (2003). The control signal is chosen as being $u = -\mathbf{K}\mathbf{x}$, where matrix $\mathbf{K}$ (1xn) is called gain matrix of state feedback. Note that the control signal u is chosen in a way that the control signal is determined through the instantaneous value of the state vector $\mathbf{x}$. In the following analysis, it is admitted the inexistence of restrictions for the value of u. The methodology will be applied for a regulation problem.

3.2. Determination of the characteristic equation of the system

Inserting the parameters of the system, experimentally identified, in the Eq. (9), the characteristic equation of the original system (passive one) (det(s**I**-**A**)) results: $s^6 + 73.46398s^5 + 32125.1999s^4 + 1160007.7012s^3 + 238436053.2879s^2 + 474717373.73685s + 8824542640.72369$. Thus, the poles of the passive system are: P_1=-17.904 + 122.545i, P_2=-17.904 - 122.545i, P_3=-17.907 + 122.535, P_4=-17.907 – 122.535, P_5=-0.922 + 6.055i, P_6=-0.922 - 6.055i. All of them have its real part negative (so the system is stable) and the system is verified as controllable (due to the fact that the rank of controllability matrix, given in Section 3.5, is 6).

3.3. Choice of the desired parameters for the controlled system (Mp, tp e ts)

To simulate the original system (passive) with initial conditions of displacement for the elevator, it was defined a left guide displacement of $\Delta y_l = -0.0046463692m$ which results in the cabin initial position of $Y = 0.005m$ and right arm position of $\Delta y_r = -0.0046463692m$. The chosen parameters for the controlled system (desired ones) were taken considering the following transient parameters: overshoot $M_p \leq 5\%$ and settling time of $t_s(1\%) \cong 1.5s$.

3.4. Determination of the parameters ζ and ω

Based on Franklin *et al.* (1994), the parameters in the time domain with ζ and ω, were obtained by:

$$\begin{cases} M_p \leq 5\% \dfrac{1}{100} = e^{-\pi\zeta / \sqrt{1-\zeta^2}} \\ t_s(criterio1\%) \cong 1.5s = \dfrac{4.6}{\zeta\omega_n} \end{cases} \Rightarrow \begin{cases} \zeta = 0.7 \\ \omega_n = 4.38 \\ p = s^2 + 2\zeta\omega_n s + \omega_n^2 \end{cases} \Rightarrow p = s^2 + 6.132s + 19.18$$

3.5. Determination of the dominant poles of the desired system and the desired characteristic equation

The roots of the equation $p = s^2 + 6.132s + 19.18$ are the dominant poles ($p_1 = -3.07 + \sqrt{9.78}i$, $p_2 = -3.07 - \sqrt{9.78}i$). The others four poles must be arbitrary chosen in a way that they do not influence the desired dynamics. Due to the numeric conditioning of the matrix of the system (see Leonard (1992)), the positioning of the not-dominant poles was not trivial. In a first attempt to allocate the poles, whose position were defined from a desired dynamics, it had resulted in huge gains that have produced a very large power demand from the electrical actuators, exceeding the maximum force that they can provide (120N). This can be demonstrated through the curve of control force in this situation (Fig. 6-a). To reduce the power demand it was adopted a strategy based on the allocation of the poles through the use of LQR methodology (see Rivas and Perondi (2007). The LQR approach gave adequate results, because the power demand relative to the new poles positions could be supplied by the actuators, (therefore, with the LQR approach, the problem of control signal saturation was solved). The hypothesis that the force saturation were originated by faster poles position was then investigated through the following test: the original position of the dominant poles were maintained (aiming to verify if the project conditions could be attended), and the not dominant poles were placed in the positions obtained with the use of the LQR. This solution showed adequate results in terms of the project requirements as well as of power level of the actuators. It must be remarked that the solution using the LQR also presents good results, however, with distinct transient parameters of those defined originally by the project made with the time domain parameters. Several papers present strategies for the use of LQR with full pole placement constraints (Misra (1996), Sehitoglu (1993), Shieh *et al.* (1988) e Shieh *et al.* (1990)) and partial pole placement (Fujinaka e Omatu (2001) e Sugimoto (1998)).

The calculated not dominant poles were determined as being -33.65±119.39i and -17.91±122.54i (see Rivas and Perondi (2007)). According to the methodology given in Ogata (1996) and Ogata (2003), the matrices of controllability (**M)**, the auxiliary matrix (**W)** and the transformation matrix (**T)** are determined through the following steps. As the method demands that matrix **B** must be n x 1 that u be a scalar (actually, $\mathbf{u}^T = [f_l \quad f_r \quad Y_{lR} \quad Y_{rR}]$), considering that the problem is taken in the linear operation conditions, so, superposition principle is valid, the design was developed using just one of the controlled forces, that is, $u=f_l$. Thus, $\mathbf{B}^T = \begin{bmatrix} 0 & \frac{1}{m_{areq}} - \frac{1}{M_{eq}} & 0 & \frac{1}{M_{eq}} & 0 & -\frac{1}{M_{eq}} \end{bmatrix}$.

The controllability matrix $\mathbf{M} = [\mathbf{B} \quad \mathbf{AB} \quad \mathbf{A^2B} \quad \mathbf{A^3B} \quad \mathbf{A^4B} \quad \mathbf{A^5B}]$ has rank 6, then the system is controllable.

The auxiliary matrix **W** is given by $\mathbf{W} = \begin{bmatrix} a_5 & a_4 & a_3 & a_2 & a_1 & 1 \\ a_4 & a_3 & a_2 & a_1 & 1 & 0 \\ a_3 & a_2 & a_1 & 1 & 0 & 0 \\ a_2 & a_1 & 1 & 0 & 0 & 0 \\ a_1 & 1 & 0 & 0 & 0 & 0 \\ 1 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}$, where, $\underline{a}_i$ are the coefficients of the original characteristic equation. The matrix of transformation is **T** = **MW**. The following step is to obtain the desired characteristic equation using the matrix **J**, which is a diagonal matrix formed by the poles of the open loop system.

3.6. Determination of the state feedback gains

Also, based on the pole placement theory presented by Ogata (1996), it is obtained the matrix $\mathbf{K} = \hat{\mathbf{K}}\mathbf{T}^{-1} = [\alpha_6 - a_6 \quad \alpha_5 - a_5 \quad \alpha_4 - a_4 \quad \alpha_3 - a_3 \quad \alpha_2 - a_2 \quad \alpha_1 - a_1]\mathbf{T}^{-1}$, where $\underline{\alpha}_i$ are the coefficients of the desired characteristic polynomial and the $\underline{a_i}$ are the coefficients of the characteristic polynomial of the original system. The gains are:

Table 1: State feedback gains

K_1	3.462×10^3	K_4	4.974×10^3
K_2	0.559×10^3	K_5	0.787×10^3
K_3	-15.648×10^3	K_6	0.005×10^3

The new system is: $\dot{\mathbf{x}}(t) = (\mathbf{A} - \mathbf{BK})\mathbf{x}(t)$, thus, the poles of the controlled system are P_1=-33.654 + 119.392i, P_2=-33.654 – 119.392i, P_3=-17.905 + 122.540i, P_4=-17.905 - 122.540i, P_5=-3.07 + 3.127, P_6=-3.07 – 3.127i. All of them have its real part negative so the system is verified to be stable. Note that in the process of designing the pole placement control method it was assumed that the state **x**(t) is completely known and available for the feedback of the system, but in the reality this does not happen. So, it was designed a full order state observer to observe the states and allows the actual application of the methodology presented in this section.

4. FULL STATES OBSERVER DESIGN

The system model given by the Eqs. (6), (7) and (8) considers the state set given by the position and velocities of the system's masses ($\mathbf{x} = [\Delta y_l \quad \Delta\dot{y}_l \quad Y \quad \dot{Y} \quad \Delta y_r \quad \Delta\dot{y}_r]$). These variables are not accessible for measurement, that is, only the relative positions of the arms of the suspension (Δy_l, Δy_r) can be measured. Then, it was projected a full order states observer to allow the application of the full pole placement method for the control of the active suspension. The methodology adopted follows the steps of Ogata (1996), Ogata (2003) and Friedland (2005). The method is analogous to the pole placement design; hence the followed steps are similar. The former step is to determine the observability matrix for verifying if the system is completely observable. The observability matrix is $\mathbf{N} = [\mathbf{C'} \quad \mathbf{A'C'} \quad (\mathbf{A'})^2\mathbf{C'} \quad (\mathbf{A'})^3\mathbf{C'} \quad (\mathbf{A'})^4\mathbf{C'} \quad (\mathbf{A'})^5\mathbf{C'}]$, with **C**= [1 0 0 0 0 0]. So, as the matrix **N** has rank 6, the system is completely observable and using the matrices **N** and **W,** it is obtained the matrix **Q**= $(\mathbf{W}.\mathbf{N'})^{-1}$. The next step is to determine the desired characteristic polynomial that has to response to the desired conditions of the project. The observer must have a response between two and five times (Ogata (2003) recommendation) or between two and eight times (Friedland (2005) recommendation) faster than the observed system. Therefore, the error of the observer will tend quickly to zero. The chosen dominant poles are $p_{\text{obs1}} = -65.7$, $p_{\text{obs2}} = -65.7$, with 15 times faster response than the poles of the controlled system. The chosen not-dominant poles are the same ones that had been used for the states feedback project presented in Section 3. The desired characteristic polynomial is $p_{obs} = (s^2 + 91.98s + 4316.49)(s + 33.65 - 119.39i)(s + 33.65 + 119.39i)(s + 17.91 - 122.54i)(s + 17.91 + 122.54i)$. Then the observer gains K_{obs} are shown in Tab. 2:

Table 2: Gains of the observer

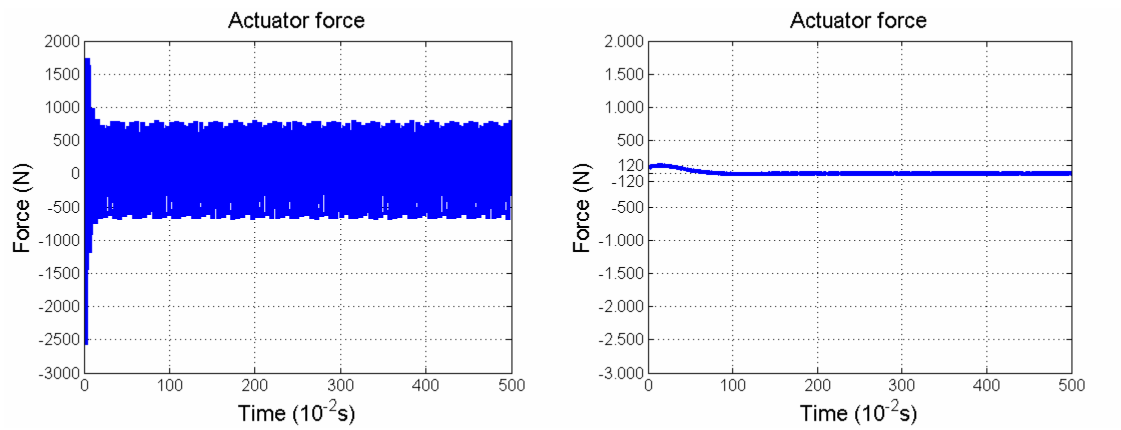
K_{obs1}	161.054	K_{obs4}	-4342.735
K_{obs2}	7043.211	K_{obs5}	138.043
K_{obs3}	-148.157	K_{obs6}	4053.939

The estimated states will be used in the place of the original state set. The control law is now $u = -\mathbf{K}\tilde{\mathbf{x}}$, remarking that $\tilde{\mathbf{x}}(t) = \mathbf{K}_{obs}\mathbf{x}(t)$ and $\dot{\tilde{\mathbf{x}}} = (\mathbf{A} - \mathbf{K}_{obs}\mathbf{C} - \mathbf{BK})\tilde{\mathbf{x}} + \mathbf{K}_{obs}y$. The complete system is of twelfth order and its characteristic polynomial is $|s\mathbf{I} - \mathbf{A} + \mathbf{BK}||s\mathbf{I} - \mathbf{A} + \mathbf{K}_{obs}\mathbf{C}|$.

5. SIMULATION'S RESULTS

The simulations results are not compared with others papers because lateral active suspension of elevators were found only in patents (Husmann (2005), Oh *et al.* (2006), Utsunomiya *et al.* (2004) e (2006)) while elevators velocity controls were found in some papers (Istif *et al.* (2002), Nai *et al.* (1994), Sha *et al.* (2002) e Skalski (1984)) and vehicle suspension were also found in articles (Ben Gaid *et al.* (2004), Campos *et al.* (1999), Giua *et al.* (2004) e Ikenaga *et al.* (2000)).

In Fig. 6 it is presented typical control force curves of both cases: original projected gains (with the poles allocated arbitrarily) and for the re-projected case (with the non dominant poles allocated with LQR).



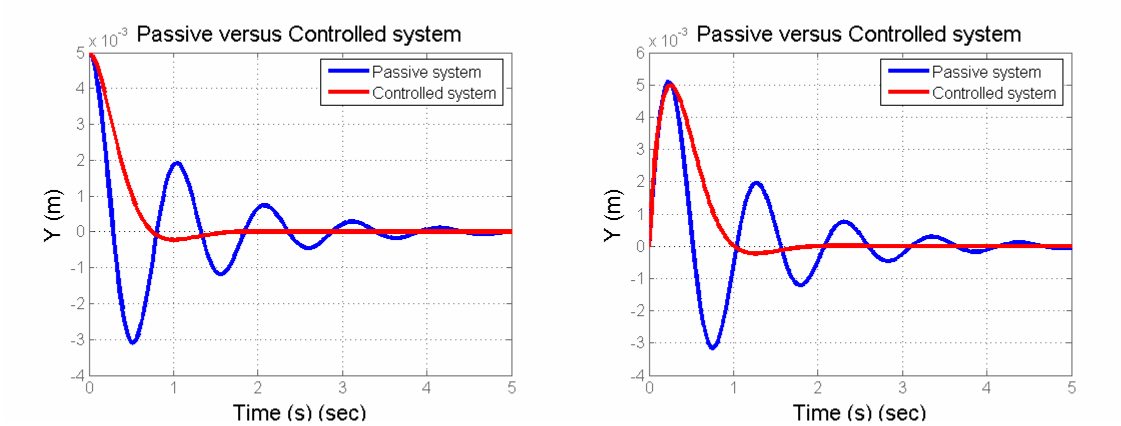
a) Control force for original projected gains

b) Control force for the re-projected gains

Figure 6 – Control force

The control force for the re-projected case is limited to 120N (at the same level as the real actuators) while the original case has the upper limit about 1750N (larger than the actuator capacity of 120N).

The time response to the initial conditions $\Delta y_l = -0.0046463692m$, $Y = 0.005m$ and $\Delta y_r = -0.0046463692m$, and for the impulse function obtained through the use of Matlab/Simulink are presented in Fig. 7.



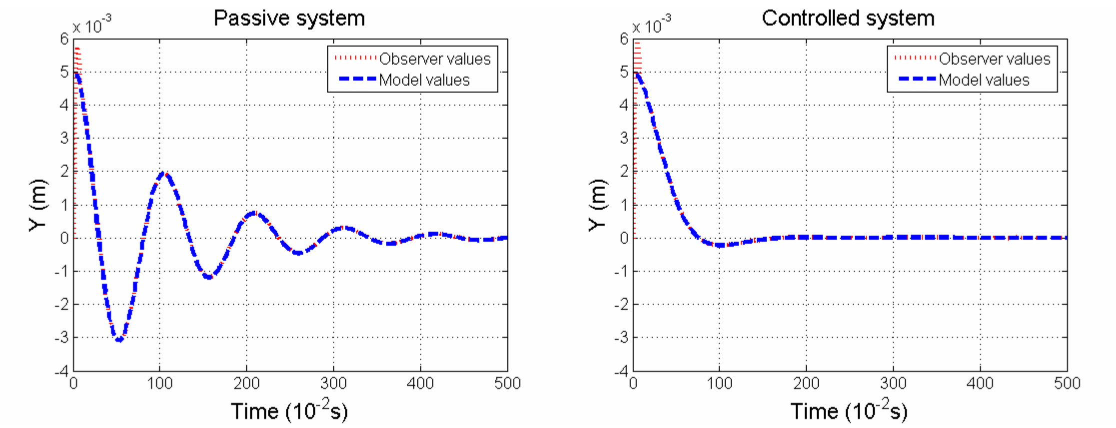
a) Passive an active response to initial conditions

b) Passive an active response to the impulse function

Figure 7 – Cabin position time response for initial conditions and impulse function

From the Fig. 7-a) it could be observed that the controlled response reaches the desired design parameters. The response overshoot is about 4.58% and the settling time (ts) was reached within the range of 0.92%. The impulse response of the controlled system presented in Fig. 7-b) shows an overshoot of about 4.56% and the settling time was reached in the range of 0.88%.

In Fig. 8 the observer to estimate the model cabin position is presented and compared with the cabin position for the initial condition case obtained from simulations with a Simulink model.



a) Cabin position response – passive case

b) Cabin position response – controlled case

Figure 8 – Observer results

Figure 8 presents the response to initial conditions, for passive and active cases, (a and b, respectively). It could be verified that although the observer response converge to model response, there is an initial deviation caused by the fact that the initial conditions of the observer are different of the model initial conditions. Based on the response presented in figure 8-b), it is possible to conclude that, in spite of the observer response presents some very little phase lag in relation to the model response, the closed loop response using the observer is within the design specifications. In this work, the values of the parameters used in the observer project are the same used in the model. The effects of parametric errors are now being studied.

6. CONCLUSION

The pole placement method applied with state feedback control and full state observer has demonstrated its capacity to control the elevator's active suspension system, as demonstrate the results obtained by simulations. It is shown in Section 4.1 that the design parameters established in the Section 3.4 were reached at all for both initial condition and impulse function cases. Also, the estimated states obtained thorough the use of a full state observer reached the model response. But to obtain better results, improvements in the observer design are being accomplished. It is important to remark that in the cases, as the present one, where matrices are bad conditioned (huge and small values present), the arbitrary (try and error) positioning of the desired closed loop poles could be a hard task, mainly because the gains obtained in the state feedback control design could be too elevated, resulting in control efforts that could overcome the power system's capacity. Therefore, methods to find acceptable locations for the not dominant closed loop poles, like LQR method, are very useful (see, for example, Rivas and Perondi, 2007).

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8. RESPONSIBILITY NOTICE

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