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ADAPTIVE CONTROL FOR AN ACTIVE SUSPENSION OF AN ELEVATOR

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Abstract. *The main aim of this paper is to report the developing of a state feedback controller for an active suspension system to be employed in a high-performance elevator that is in a "skyscraper" of about 500 [m]. The proposed controller is of the pole-placement type, augmented with adaptive capabilities for obtaining improved robustness properties. The pole-placement part of the controller is dedicated to defining a desirable dynamic behavior of the suspension system of the elevator in the presence of vibrations, whereas its adaptive part aims at compensating the variations in such behavior as a function of the mass that is transported by the elevator. The employed mass estimation algorithm and its inclusion in the overall control scheme are described, and its convergence properties are discussed. Simulation results are used to illustrate the efficiency of the proposed control scheme when applied to different loading conditions of the elevator.*

Keywords*: active suspension, elevator dynamics control, high performance elevator, state feedback control, adaptive control*

1. INTRODUCTION

This work discusses an active suspension system for high performance elevators that aims at reducing the lateral vibrations in the base of the elevator's cabin. Such oscillations occur as the elevator moves, originated mainly from the contact between the rollers parts of the suspensions and their alignment guides that are fixed to the building structure. These guides present inherently small misalignments that are caused by many different reasons, such as disturbances in the installation processes, thermal dilatations and aging of their construction materials, among others. The vibration induced by such misalignments can compromise the requirements of security and comfort of the passengers. This problem is especially serious in the case of high speed elevators in very high buildings (skyscrapers).

Because of its importance, this problem has been attacked by means of many different control techniques. For example, Oh *et al.* (2006) uses a suspension system based on the repulsive forces of electromagnets to cancel lateral disturbances. Utsunomiya *et al.* (2004 and 2006) describes an active system that uses the signals from accelerometers located at the center of the elevator to compare the cabin vibrations with their acceptable values and applies adequate lateral forces through electromagnetic actuators. Husmann (2005) employs an active damping system for the structural frame of the elevator based on the measurement of its deformations. 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. Other approaches also include *Bond Graphs* associated with PD-control (Istif *et al,* 2002), adaptive sliding mode control (Sha *et al,* 2002) and neural networks (Schneider *et al.*, 2001).

In Rivas and Perondi (2007b), an active suspension system based on linear electromagnetic actuators was studied. In order to control such actuators, a linear control scheme based on the pole placement technique, associated with a full state observer (Friedland, 2005) was developed. The position of the desired poles for this controller was determined by means of the linear quadratic regulation (LQR) technique (see, for instance, Kirk, 1970), so that satisfactory results could be obtained within the energetic limits imposed by the available actuators.

This work proposes to add an online estimator of the mass transported by the elevator system to the control scheme originally proposed by Rivas and Perondi (2007b). With such mass estimate, the gains of the controller can be updated so as to reduce performance fluctuations due to the constant changes in the number of passengers during the operation process of the elevator. Thus, the robustness properties of the controlled system with respect to mass variations are enhanced. Furthermore, as the mass is real-time evaluated, this scheme presents the additional advantage of providing the bus of the logical operation system of the elevator with an on-line estimation of number of passengers transported. This information can be useful to other sectors of the elevator system, such as the power control of its engines. The mass estimation scheme is based on the self-tuning control (STC) approach (Slotine and Li, 1991). The resulting performance of the proposed controller is illustrated by means of simulation results.

This paper is organized as follows. Section 2 discusses the theoretical model of the suspension system, while, in Section 3, the control scheme is described. The algorithm for estimating the transported mass is presented in Section 4, including the theoretical analysis of its convergence properties. Simulation results are presented and compared in Section 5. Finally, the main conclusions are outlined in Section 6.

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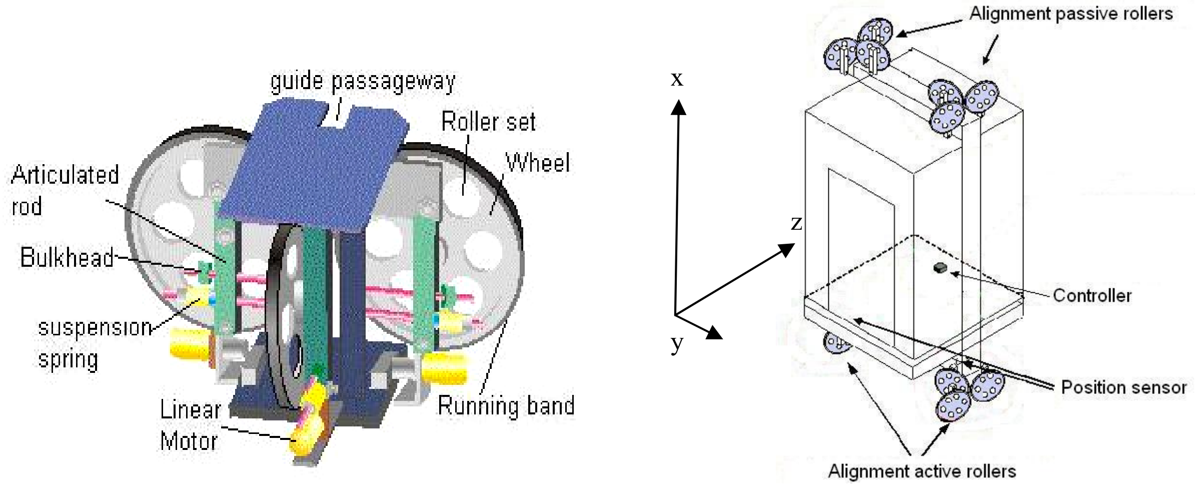
2. DYNAMIC MODEL

In this section, the dynamic model of the elevator is presented. The determination of such model is described in detail in Rivas and Perondi (2007b).



a) Active suspension b) Suspension systems installed in the elevator

Figure 1: Suspension system

In Fig. 1, the main elements of the studied system are depicted. The objective is to reduce the oscillatory movements of the base of the cabin where the passengers would be located. Therefore, it is necessary to take into account the displacements in two planes (XY and YZ). In this work, only the XY-plane problem will be approached. Also, the set cabin + passengers is treated as a solid parallelepiped. Each *suspension system* is composed of three arms, each one made of an independent suspension (articulated connecting rod, spring, roller, linear motor, etc.). The vertical guides are attached vertically to the civil structure. Four rollers (joint + wheel + roller band) are aligned to the XY-plane and hinder the movement in the direction of the Y-axis and the rotation around the Z-axis. Other eight rollers are aligned to XZ-plane, and are intended to hinder the movement in the direction of Z and the rotation about an axis parallel to X.

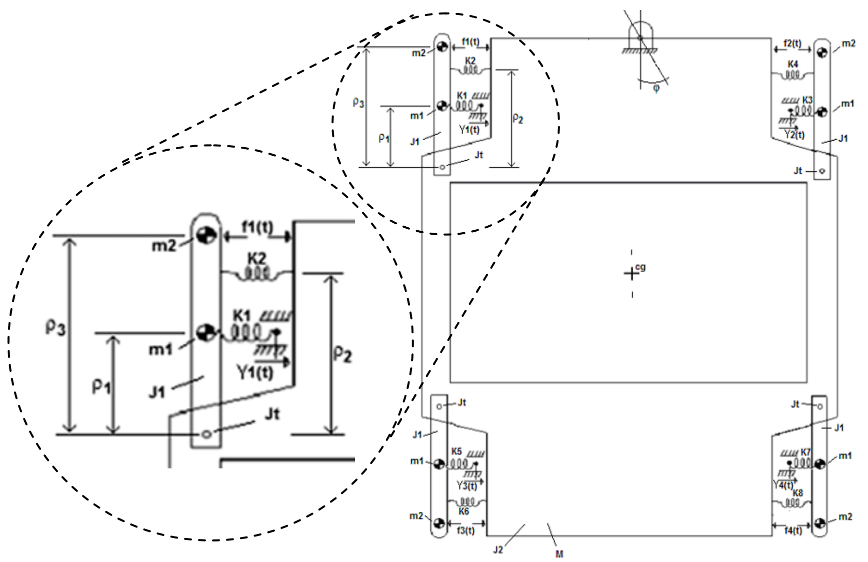


Figure 2: Scheme of the suspension system and elevator

Figure 2 shows the suspension system and its location on the elevator. In this figure, the variables 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 suspension 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 rotatio 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

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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 arms 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 tires of the rollers and $K_2 = K_4 = K_6 = K_8$ are, respectively, the stiffness constants of the helical springs of the suspensions. Also, J_e is the moment of inertia of the elevator referred to the top rotation center pivoting, where the cables are connected to the structure. Finally, φ is the rotational angle of the elevator referred to the connection pivoting point of the cables in the top of the cabin. It can be determined as $\varphi = (Y_{lo} - Y_{up})/L$ where L is the length of the cabin.

As previously mentioned, this study considers only the movement in the XY-plane. In addition, it is assumed 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, are modeled by means of the system represented in Fig. 3. This simplified configuration treats 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. Therefore, a simplified representation of the pendulum system is necessary. The gravitational effect due to the rotational movement of the elevator around the connection of the cables in the top of the cabin is given by an equivalent translational spring constant K_g defined as $K_g = M_{eq}g/L$, where g is the acceleration of the gravity.

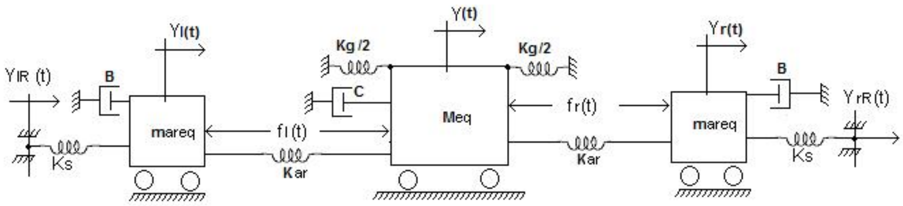

Figure 3: Translational equivalent model

In Fig. 3, K_s are the spring translational constants of the wheels bands, $K_{ar} = K_2(\rho_2/\rho_1)^2$ are the stiffness constants of the springs of the suspension, so that their effects are considered as applied directly in the contact point of the wheels bands with the guides; $m_{areq} = m_2\rho_3^2 + m_1\rho_1^2 + J_{ar}/\rho_3^2 = J_t/\rho_3^2$ is the equivalent mass of the arms referred to their rotational centers; M_{eq} is the equivalent mass of the car ($M_{eq} = J_e/L^2$). The forces of the actuators are transferred to the position of the contact point between the band of the rollers wheels with the guides, and they are defined as $f_l(t) = f_3(t)(\rho_2/\rho_1)$ and $f_r(t) = f_4(t)(\rho_2/\rho_1)$. These forces represent the control signal that is applied to the system, and are defined as positive when the actuators are retracted (that is, they apply a force that tends to pull the cabin towards its alignment guides), and negative when they are expanded (the applied force pushes the cabin away from the guides). The losses by friction are also considered through the introduction of equivalent viscous damping terms (dampers C and B). Finally, the displacements of the left and right arms are defined as Y_{lR}(t) and Y_{rR}(t), respectively.

By employing the Second Law of Newton, the system modeled in Fig. 3 can be described in the state variable form $\dot{\mathbf{x}} = \mathbf{Ax} + \mathbf{Bu}$, with $\mathbf{A}$, $\mathbf{x}$, $\mathbf{B}$ and u defined as (Rivas and Perondi, 2007b):

$$\mathbf{A} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\frac{(K_s + K_{ar})}{m_{areq}} - \frac{K_{ar}}{M_{eq}} & -\frac{B}{m_{areq}} & \left(\frac{K_g}{M_{eq}} - \frac{K_s}{m_{areq}}\right) & \left(\frac{C}{M_{eq}} - \frac{B}{m_{areq}}\right) & -\frac{K_{ar}}{M_{eq}} & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{K_{ar}}{M_{eq}} & 0 & -\frac{K_g}{M_{eq}} & -\frac{C}{M_{eq}} & \frac{K_{ar}}{M_{eq}} & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ -\frac{K_{ar}}{M_{eq}} & 0 & \left(\frac{K_g}{M_{eq}} - \frac{K_s}{m_{areq}}\right) & \left(\frac{C}{M_{eq}} - \frac{B}{m_{areq}}\right) & -\left(\frac{K_{ar}}{M_{eq}} + \frac{K_{ar} + K_s}{m_{areq}}\right) & -\frac{B}{m_{areq}} \end{bmatrix} \quad (1)$$

$$\mathbf{x} = \begin{bmatrix} \Delta y_l & \Delta \dot{y}_l & Y & \dot{Y} & \Delta y_r & \Delta \dot{y}_r \end{bmatrix}^T \quad (2)$$

$$\mathbf{B}^T = \begin{bmatrix} 0 & \frac{1}{m_{areq}} - \frac{1}{M_{eq}} & 0 & \frac{1}{M_{eq}} & 0 & -\frac{1}{M_{eq}} \end{bmatrix} \quad (3)$$

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$$u = f_l \tag{4}$$

where Δy_l and $\Delta \dot{y}_l$ are the relative position and the relative velocity of the left arm of the suspension set, respectively, Δy_r and $\Delta \dot{y}_r$ are the same quantities referred to the right arm, Y is the absolute position of the cabin, and $\dot{Y}$ is the absolute velocity of the cabin, and $f_l(t)$ is the force applied by the left actuator of the system. Hence, in this work, the influence of only one actuator is considered.

The real system uses potentiometric sensors to measure the relative displacement between the arms and the cabin (in most cases, other works employ accelerometers; therefore, 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.

3. PROJECT OF THE SUSPENSION CONTROL

The proposed controller to be applied to the suspension system is based on the pole placement method (Ogata, 2003) augmented with a self-tuning scheme (Slotine and Li, 1991) for compensating the variations in the mass of the system as the number of passengers changes with time. The pole placement method allows choosing the location of all poles of the controlled system (provided that the open loop system is fully controllable), so that its dynamical behavior matches very closely with the one that is ideally desired, whereas the self-tuning scheme makes it possible to reach such desired dynamics regardless of the number of transported passengers. The pole placement process is carried out by means of the LQR method (Kirk, 1970). This work is focused on the employment of the mass estimation part of the proposed control algorithm. The development of the pole placement part of the proposed controller is discussed in detail in Rivas and Perondi (2007b).

3.1. Open-loop poles and desired parameters of the system

As determined in Rivas and Perondi (2007b), inserting the experimentally identified parameters of the system in Eq. (1), the characteristic equation of the original system becomes:

$$\begin{gathered} s^6 + \left(71{,}63 + \frac{2058}{M_{eq}}\right)s^5 + \left(3{,}195 + \frac{19{,}21}{M_{eq}}\right)10^4 s^4 + \left(1{,}098 + \frac{68{,}97}{M_{eq}}\right)10^6 s^3 + \left(2351 + \frac{3{,}648}{M_{eq}}\right)10^9 s^2 + \\ \frac{5{,}316}{M_{eq}}10^{11} s + \frac{9{,}883}{M_{eq}}10^{12} = 0 \end{gathered} \tag{5}$$

Thus, for a nominal mass of 1120 kg, the poles of the open-loop system are given by $P_{1,2} = \text{-}17{,}904 \pm 122{,}545\mathrm{i}$, $P_{3,4} = \text{-}17{,}907 \pm 122{,}535\mathrm{i}$ and $P_{5,6} = \text{-}0{,}922 \pm 6{,}055\mathrm{i}$, where $P_{i,j}$ denotes two complex-conjugate poles P_i and P_j. As all these poles have negative real parts, the open-loop system is stable. Also, it can be verified that it is fully controllable (Rivas and Perondi, 2007b).

The desired transient parameters adopted in this for the controlled system are: maximum overshoot M_p of 5%; settling time at 2% of the final value of 1.5s. Therefore, the characteristic equation of the dominant dynamic behavior for the desired system is calculated as (Franklin *et al.,* 1994):

$$\begin{cases} M_p = e^{-\pi\zeta / \sqrt{1-\zeta^2}} \le \dfrac{5}{100} \\ t_s(2\%) = \dfrac{4}{\zeta\omega_n} = 1{,}5s \end{cases} \Rightarrow \begin{cases} \zeta = 0{,}6904 \\ \omega_n = 3{,}864 \\ s^2 + 2\zeta\omega_n s + \omega_n^2 = 0 \end{cases} \Rightarrow s^2 + 5{,}333s + 14{,}93 = 0 \tag{6}$$

The roots of this equation are the dominant poles $p_{1,2} = -2{,}667 \pm \sqrt{7{,}82}i$. The others four poles must be chosen so that they exert as small an influence over the desired dynamics as possible. In order to choose their positions in a way that they respect the maximum force that the actuators can provide (120N), the LQR method is employed (see, for instance, Fujinaka and Omatu, 2001). Thus, the non-dominant poles are determined as $p_{3,4} = \text{-}33{,}65 \pm 119{,}4\mathrm{i}$ and $p_{3,4} = \text{-}17{,}90 \pm 122{,}5\mathrm{i}$ (Rivas and Perondi, 2007a). Therefore, as the values of all poles are known, the desired characteristic equation $\Delta_d(s) = (s - p_1)(s - p_2)(s - p_3)(s - p_4)(s - p_5)(s - p_6)$ is given for the system.

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3.2. Determination of the state feedback gains

The vector $\mathbf{k} = [k_1 \quad k_2 \quad k_3 \quad k_4 \quad k_5 \quad k_6]$ of feedback gains is calculated by making the coefficients of $\Delta_d(s)$ equal to those of the characteristic equation of $\mathbf{A} - \mathbf{Bk}$ (the matrix that represents the dynamics of the controlled system). If the system is fully controllable and all of its parameters are known, $\mathbf{k}$ is unique and all of its components are constants. This case is described in Rivas and Perondi (2008b). In the present study, the elevator mass M_{eq} is unknown. For this reason, $\mathbf{k}$ is calculated in a time-variant form, depending on the value of the estimated mass $\hat{M}_{eq}$, whose estimation algorithm is discussed in the following section. For this reason, it is necessary to write $\mathbf{A} - \mathbf{Bk}$ in a parameterized form, in which the term $\hat{M}_{eq}$ is explicitly given:

$$\hat{\mathbf{A}} - \hat{\mathbf{B}}\mathbf{k} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ \left(-15{,}34 - \frac{19{,}03}{\hat{M}_{eq}}\right)10^3 & -35{,}81 & \left(\frac{6{,}662}{\hat{M}_{eq}} - 14{,}25\right)10^3 & \frac{20{,}58}{\hat{M}_{eq}}10^3 - 35.81 & -\frac{19{,}03}{\hat{M}_{eq}}10^3 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{19{,}03}{\hat{M}_{eq}}10^3 & 0 & -\frac{6{,}662}{\hat{M}_{eq}}10^3 & \frac{-20{,}58}{\hat{M}_{eq}}10^3 & \frac{19{,}03}{\hat{M}_{eq}}10^3 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ -\frac{19{,}03}{\hat{M}_{eq}}10^3 & 0 & \left(\frac{6{,}662}{\hat{M}_{eq}} - 14{,}25\right)10^3 & \frac{20{,}58}{\hat{M}_{eq}}10^3 - 35{,}81 & \left(-15{,}34 - \frac{19{,}03}{\hat{M}_{eq}}\right)10^3 & -35{,}81 \end{bmatrix} - \begin{bmatrix} 0 \\ 0{,}057 - \frac{1}{\hat{M}_{eq}} \\ 0 \\ \frac{1}{\hat{M}_{eq}} \\ 0 \\ \frac{-1}{\hat{M}_{eq}} \end{bmatrix} [k_1 \quad k_2 \quad k_3 \quad k_4 \quad k_5 \quad k_6] \tag{7}$$

Defining $\mathbf{I}$ as the identity matrix, the explicit calculation of $|\mathbf{A} - \mathbf{Bk} - \mathbf{I}s| = \Delta_d(s)$ results in a 6x6 linear system parameterized in terms of $\hat{M}_{eq}$ that must be solved for the vector of gains $\mathbf{k}$ at each time integration step. Thus, it is expected that the desired poles for the controlled system are closely tracked as the transported mass varies, resulting in a closed loop dynamics that does not vary significantly as the loading condition of the elevator changes with time.

It is important to notice that the control method described in this work assumes that the state $\mathbf{x}(t)$ is fully measurable. In practice, however, this is not the case, and only the relative positions of the arms of the suspension (Δy_l, Δy_r) are measured. In a previous work (Rivas and Perondi, 2007b), a fixed full order observer was developed by following the same methodology described in Ogata (2003), Ogata (1996) and Friedland (2005). For the currently proposed control approach, an adaptive version of such approach is currently under development, and it is expected to be also capable of taking into account the variations in the transported mass. As the development of such state observer is still under research, this work assumes that all necessary variables can be obtained by direct measurement.

4. MASS ESTIMATOR

In this work, the mass estimation algorithm presented in Slotine and Li (1991) is employed. In order to implement such estimator, it is assumed that both the force $F(t)$ applied to the system and the lateral cabin acceleration $\ddot{x}(t)$ are known. The basic idea is to use the knowledge of these two variables to calculate the difference between the estimated force applied to the system (that is, the product between the measured acceleration value and the current estimate of the mass $\hat{m}(t)$) and the actual value of such force. Then, such difference is used to update the mass estimate $\hat{m}(t)$. Therefore, the difference between estimated and actual force applied to the system is:

$$e = \hat{m}(t)\ddot{x}(t) - F(t) \tag{8}$$

Also, the acceleration signal is employed to calculate the auxiliary function $P(t)$, defined as:

$$P(t) = \left[\int_0^t \ddot{x}^2(\tau) d\tau \right]^{-1} \tag{9}$$

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With the definitions (8) and (9) and the acceleration signal, the mass estimation update law is given by:

$$\dot{\hat{m}} = -P\ddot{x}e \tag{10}$$

The algorithm defined by equations (8), (9), (10) presents an important convergence property. Let $\tilde{m} = \hat{m} - m$ be the estimation error in the mass calculated by means of the proposed algorithm. If the system is persistently excited, then $\|\tilde{m}\| \to 0$ as $t \to \infty$. This statement can be proved as follows:

Consider the definition for $P(t)$ given in Eq. (9). It follows from that expression that

$$\frac{d}{dt}P(t)^{-1} = \ddot{x}^2(t) \tag{11}$$

Using this result combined with Eq. (10), it can be shown that:

$$\frac{d}{dt}\left[P(t)^{-1}\tilde{m}(t)\right] = 0 \tag{12}$$

Separating variables and integrating Eq. (12) yields:

$$\tilde{m}(t) = \frac{\tilde{m}(0)}{P(0)}P(t) \tag{13}$$

From Eq. (9), it can be seen that if the system is persistently excited, all terms of $P(t)$ tend to zero as $t \to \infty$. Then, from Eq. (13), it follows that $\|\tilde{m}\| \to 0$ as $t \to \infty$, which completes the proof.

6. SIMULATION RESULTS

This section is dedicated to presenting the simulation results that illustrate the effectiveness of the proposed control scheme. First, it is presented the effect that the variation of the transported mass can cause on the performance of the closed-loop system when the controller is kept fixed with its original gains, calculated according to its nominal operation conditions. Afterwards, a series of simulation results are presented for illustrating the performance of the controlled system under different loading conditions. In all cases, it is simulated the dynamic behavior of the position of the elevator cabin in response to a initial condition problem, in which the cabin is initially displaced from its equilibrium position of 0,005 m (5 mm). In order to evaluate the performance of the proposed controller, simulation results obtained by means of other control schemes are also presented for comparison.

6.1. Effect of the transported mass

In order to verify the efficiency of the proposed mass estimator, it is important to observe how the variations of the transported mass can affect the performance of the closed loop system when the pole placement controller is employed with fixed gains. Fig. 3 depicts how the positions of the poles of the system vary as the transported mass is increased (the arrow indicates the direction in which such mass is augmented).

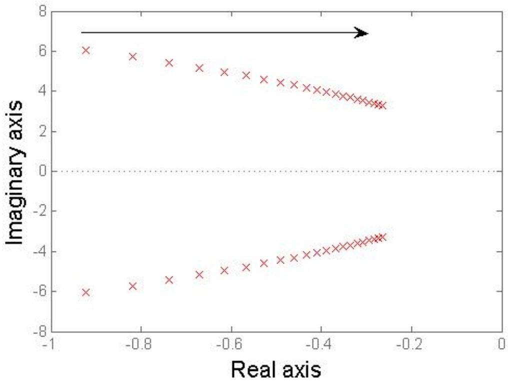

Figure 3: Poles variation according with the mass variation

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In Fig. 3, it is observed that the damping of the closed-loop system diminishes as the transported mass is increased. Thus, even though the system remains stable, its response becomes more oscillatory. This conclusion is reinforced by the results shown in Fig. 4: when the transported mass is at its nominal value of 1120 [kg], the dynamics of the system when controlled by means of the pole placement method assuming nominal conditions matches with small deviations the desired one; when the transported mass is 3920 [kg], however, the fixed gains of the controller lead to a more pronounced overshoot in the response of the system.

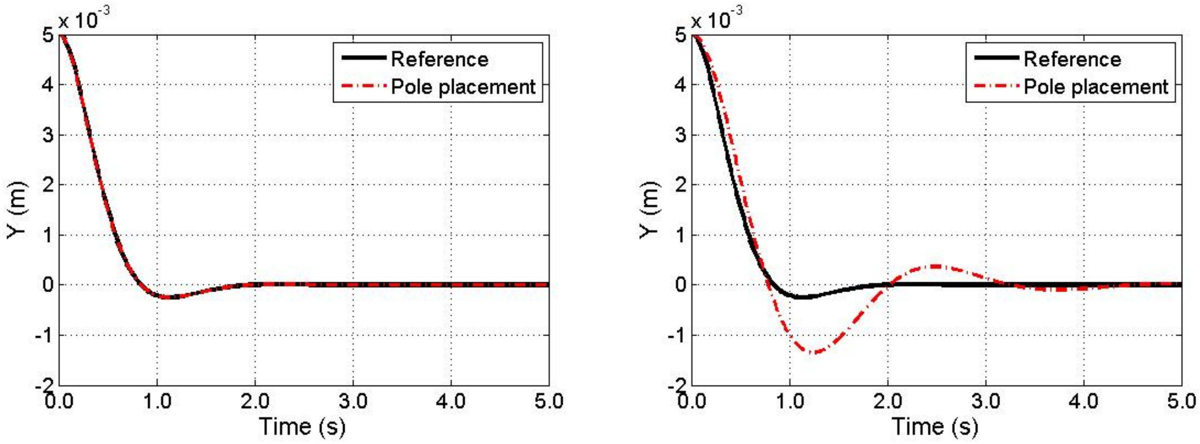

Figure 4: Performance of the system when controlled by means of the fixed pole-placement algorithm:
a) nominal mass b) 100% passengers transported

6.2. Comparison with other controllers

After observing the effect of the variations of the mass upon the performance of the closed-loop system when no action is taken to compensate such value changes, the results obtained by means of the proposed control approach are presented. The simulations were performed for three loading conditions of the elevator: nominal mass (zero passengers, transported mass equivalent to 1120 [kg]), half-capacity (20 passengers, transported mass equivalent to 2520 [kg]) and full capacity (40 passengers, transported mass equivalent to 3920 [kg]). For comparison, the performances obtained by means of three other controllers are also presented: pole placement controller, with fixed gains calculated for the nominal case, PID control with respect to position, and Sky Hook (see Reichert, 1997, or Ahmadian et al., 2004, for instance). The last two control algorithms are commonly used in the discussed application, whereas the two versions of the pole placement controller (with and without adaptation of the estimated mass) constitute the control approach that is proposed in this work. Besides the performance of the different controllers, the behaviour of the estimated mass that is transported by the elevator when calculated according to the algorithm discussed in Section 5 is also presented, so that the convergence properties of the employed estimator can be verified.

The PID scheme applied to the control of the position of the system is a classical approach, commonly used because of its relative simplicity in design and analysis (see Ogata, 2003, for instance), and it is one of the most widely employed approaches in the area of active suspension schemes. Examples of its application can be found in Chantranuwathana and Peng (1999), Chen et al. (1999), Istif et al. (2002), Feng et al. (2003), Senthil and Vijayarangan (2007), Çetin and Demir (2008), and Senthil (2008), among others. For this reason, the performance of the closed loop system obtained by means of such controller is an important parameter for evaluating the performance characteristics of the proposed algorithms. The gains of the employed PID controller were calculated by means of the PID Tuning with Actuator Constraints Matlab/Simulink Toolbox so that the desired dynamics for the closed loop system in nominal conditions was obtained. The values of such gains are: K_d= 3,8850x10^3, K_i=5,0554x10^3 and K_p=3,0449x10^3. The Sky Hook is another commonly used control approach in the area of active and semi-active suspension systems. Its concept was introduced by Karnopp et al. (1974). Basically, the Sky Hook approach consists of designing a controller based on a reference frame that is “attached to the sky”, so that it is kept fixed in spite of the movement of the base of the suspension system. Its main characteristics are discussed in Sá (2006). Examples of its application can be found in Reichert (1997), Ahmadian et al. (2004) and Zuo et al. (2004). In the present case, the Sky Hook approach was employed for implementing a PID controller with respect to velocity, with gains Kp=3563, Ki=1187 and Kd=-46.18. As in the case of the PID control of the position of the system, these values were obtained by choosing the desired dynamics of the system and employing the same Matlab/Simulink Toolbox.

The results obtained by means of each controller and for the three loading conditions of the system are presented in figures 5, 6 and 7.

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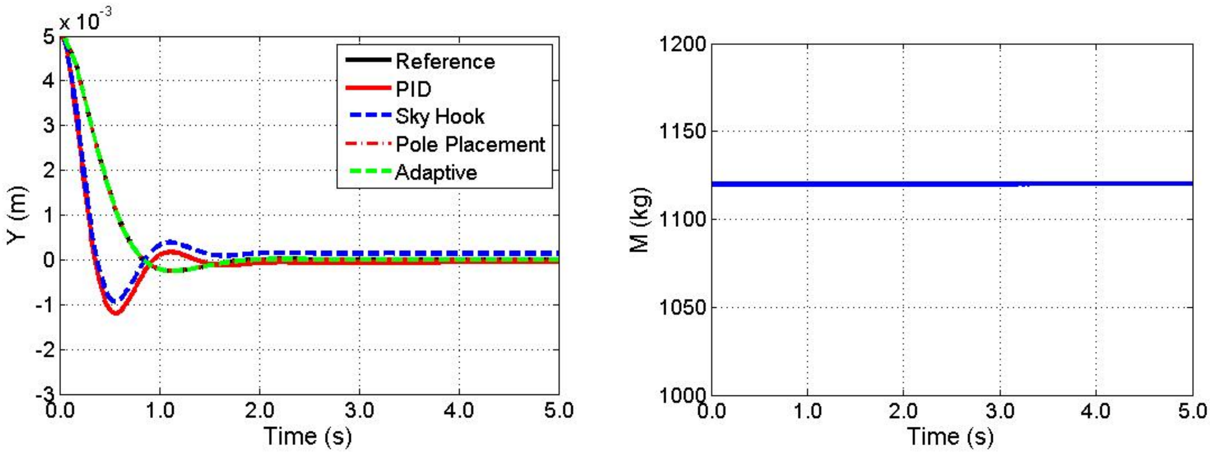

Figure 5: Cabin position for M_{eq}=1120kg:
a) Overall performance; b) Mass estimation results

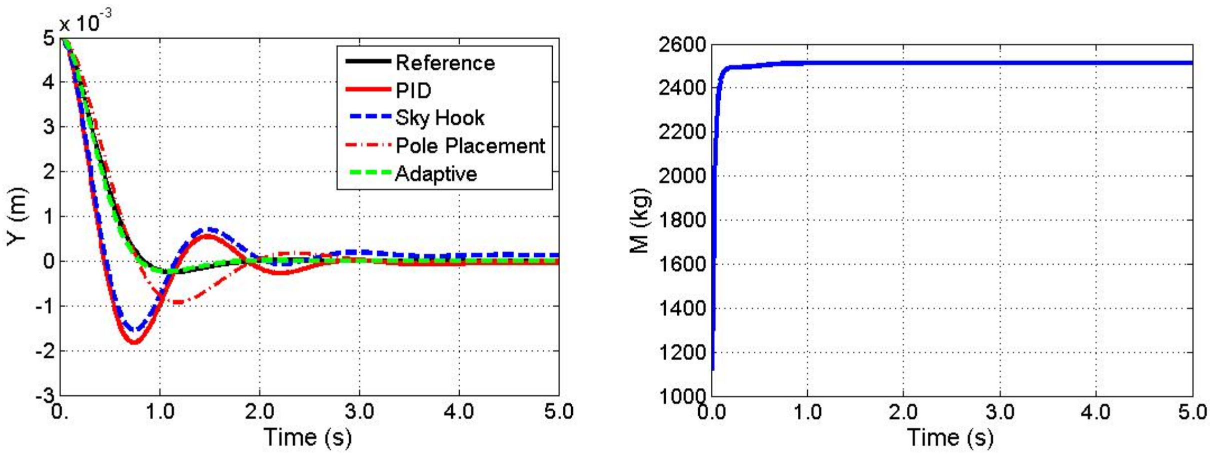

Figure 6: Cabin position for M_{eq}=2520kg:
a) Overall performance; b) Mass estimation results

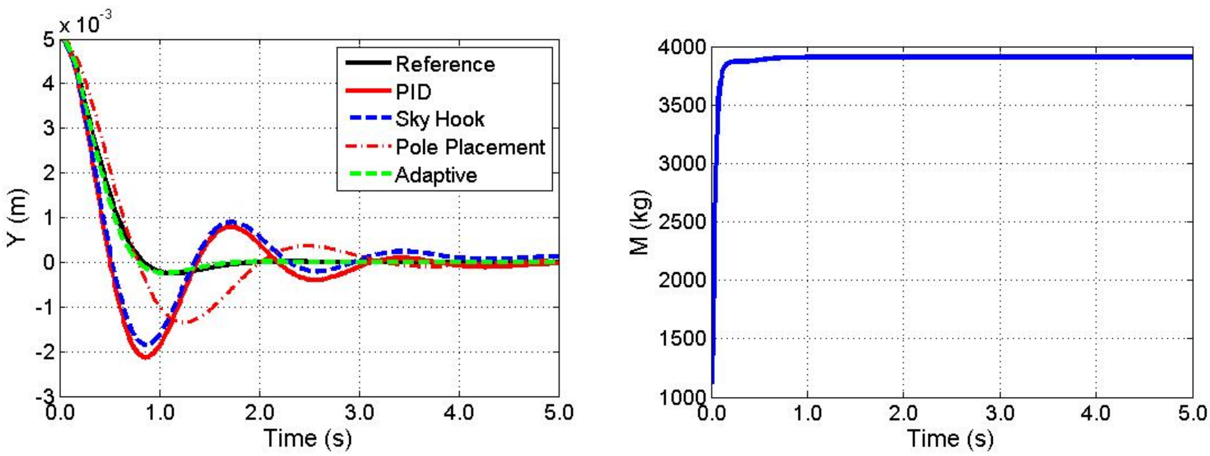

Figure 7: Cabin position for M_{eq}=3920kg:
a) Overall performance; b) Mass estimation results

From the results presented in the figures 5, 6 and 7, it can be observed that the controlled system becomes noticeably less sensitive to the variations in the transported mass when the proposed algorithm is employed, because the changes in its value are quickly compensated. It is also noticed that the estimates of the transported mass actually converge to their real values or to values that are very close to the real ones, and that such convergence occurs in relatively short periods (in Fig. 5(b), the mass estimate does not vary because it is already at its correct value). Thus, it is seen that the proposed control scheme can be a valuable tool in ensuring a reliable operation of the controlled system regardless of the number of passengers that are transported by the elevator.

Stability of adaptive systems is significantly affected by the value of the employed sampling time (Slotine and Li (1991), Aström and Wittenmark (1995)). In the present case, it was observed that the mass estimator is stable for sampling times smaller than 23 milliseconds. In the simulations whose results are presented in figures 5 through 7, the employed sampling time was 10 milliseconds.

7. CONCLUSION

In this work, an adaptive pole-placement control algorithm was proposed to be employed in the suspension system of a high-performance elevator. The main features of the controlled plant were presented, and the development of the pole-placement part of the controller was outlined. The development of the mass estimation algorithm and its inclusion as part of the employed control scheme were described, and the convergence properties of such estimator were proven analytically. Simulation results were used to illustrate the performance of the proposed controller when compared to other approaches. According to such results, it was observed that the proposed controller is able to guaranteeing a reliable and repetitive performance of the controlled system regardless of the mass transported by the elevator.

Further work will focus on developing an adaptive state observer to be employed with the proposed controller, so that its application to the real elevator system can be carried out and experimental validation of the algorithm can be obtained.

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