

# Software design to calculate and simulate the mechanical response of electromechanical lifts

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**Abstract.** Lift engineers and lift companies which are involved in the design process of new products or in the research and development of improved components demand a predictive tool of the lift slender system response before testing expensive prototypes. A method for solving the movement of any specified lift system by means of a computer program is presented. The mechanical response of the lift operating in a user defined installation and configuration, for a given excitation and other configuration parameters of real electric motors and its control system, is derived. A mechanical model with 6 degrees of freedom is used. The governing equations are integrated step by step through the Meden-Kutta algorithm in the MATLAB platform. Input data consists on the set point speed for a standard trip and the control parameters of a number of controllers and lift drive machines. The computer program computes and plots very accurately the vertical displacement, velocity, instantaneous acceleration and jerk time histories of the car, counterweight, frame, passengers/loads and lift drive in a standard trip between any two floors of the desired installation. The resulting torque, rope tension and deviation of the velocity plot with respect to the setpoint speed are shown. The software design is implemented in a demo release of the computer program called ElevaCAD. Further on, the program offers the possibility to select the configuration of the lift system and the performance parameters of each component. In addition to the overall system response, detailed information of transients, vibrations of the lift components, ride quality levels, modal analysis and frequency spectrum (FFT) are plotted.

## 1. Introduction

Design engineers are interested in the response of the elevator system in a number of circumstances. The most typical for that is the normal functioning in a standard trip like quantifying car position, acceleration, speed or jerk, assess ride quality, roller guides efficiency or levelling problems though other exceptional circumstances like impacts provided by guide misalignment, buffer actuation or safety gear actuation by means of uncontrolled car movements are of interest too.

A mechanical model with 6 degrees of freedom of the elevator system connected by generalized joints was developed and the governing equations for residential or freight lifts with any roping arrangement were derived [1]. This paper presents the analysis, design and programming of software that enables the calculation of the mechanical response of a real electromechanical lift installation.

Several solutions strategies of the governing equations are analyzed in order to get the mechanical response of the system in every scenario. The main, more versatile, strategy consists on a step by step integration using the Kutta-Meden algorithm which can be implemented in the MATLAB platform.



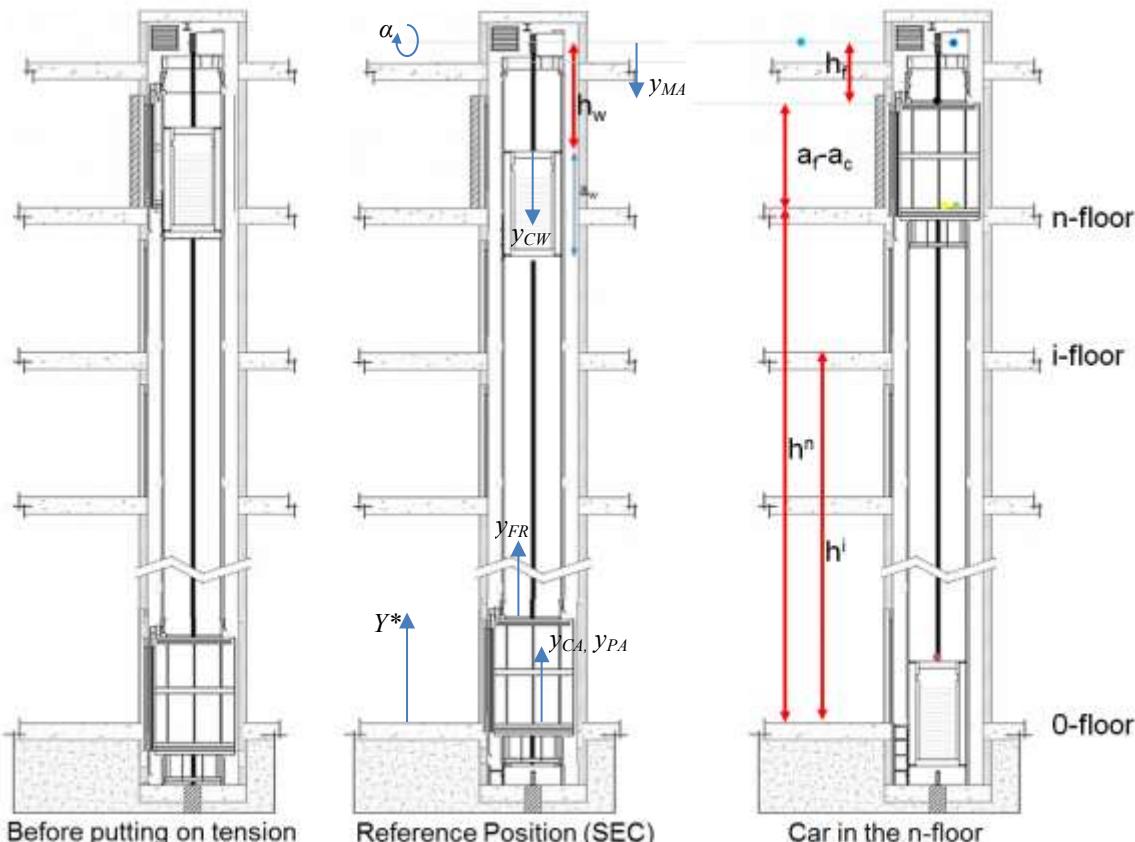
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The expected response consists of various variables. Firstly, the response includes the vertical displacement, velocity and acceleration of the car, counterweight, frame, passengers/loads and lift drive as a function of time. Secondly, the variables related to the electric and electronic drives, mainly the deviation of the lift drive speed from the setpoint speed which is used for the electronic controller to determine the machine supply frequency according to a predefined control law, and the torque applied to the machine.

Vertical transport control involves the conjunction of a number of components of different nature. Each of these components has its own dynamic behaviour that could be modelled by differential equations. In this paper, it is analysed the mechanical dynamics considering that both, electronic and electric components, have an immediate response in comparison with mechanical component dynamics, and are modelled by static equilibrium equations instead of using the corresponding differential equations. The main objective of including these static models within the general mechanical differential model is to highlight the differences when using different electronic controllers or electrical machines.

## 2. Input parameters and initial conditions

The characteristic of a certain installation are specified by means of the parameters shown in the right of figure 1 where  $h^i$  represents the distance from the 0-floor landing to the  $i$ -th floor landing and the colored circles represents the local origin of the displacements masses: red for the counterweight ( $y_{CW}$ ), black for the car frame ( $y_{FR}$ ), yellow for the car ( $y_{CA}$ ), green for the passengers/loads ( $y_{PA}$ ), blue for the machine drive ( $y_{MA}$ ) and cyan for the machine rotation ( $\alpha R$ ). The minimum distance from the car frame rope-ends to the stator machine axis is named  $h_f$ . In a similar way, the minimum in-service distance from the counterweight rope ends to the stator machine axis is named  $h_w$ .



**Figure 1.** Installation parameters of the lift system

The center diagram in figure 1 shows the reference configuration of the elevator system (*SEC*) which is made coincident with the typical configuration after putting on tension the elevator in the installation process and rotate the drive, at a very low speed, to make the car floor level coincident with the 0-floor level of the building.

From figure 1 is also possible to derive the steady-state configuration of the stopped lift with the machine brakes pressed at any car floor height  $y_{CA}$ . Then, the static deformations can be easily computed from the theory of elasticity, the sling properties (number of ropes  $n_R$ , net cross-sectional area of one rope  $A$ , rope density  $\rho$ , and instantaneous lengths  $l_f$  and  $l_w$  and equivalent stiffness  $k_{FR}$  and  $k_{CW}$  of both, frame rope-piece and counterweight rope-piece, respectively) the stiffness of the machine to vertical displacement  $k_{MA}$  and rotation with the brakes pressed  $k_{BR}$  and the car silent blocks stiffness  $k_{CA}$ . See table 1 in which  $m_{CW}$ ,  $m_{FR}$ ,  $m_{CA}$ ,  $m_{PA}$  and  $m_{MA}$  are the masses of the counterweight, car frame, car, passengers/loads and machine, respectively.  $g$  is the gravity constant.

**Table 1.** Steady-state deformations of lift components under full car pressed brakes.

Component	Global position [Y*]	Initial Elongation/Rope lengths
V. disp.Machine	$h + h_f + a_f - a_c$	$y_{0MA} = (m_{MA} + m_{FR} + m_{CA} + m_{PA} + m_{CW} + n_R \rho A (h + h_f + h_c)) g / k_{MA}$
Counterweight	$y_{CA} + h + h_f + a_f - a_c - h_w$	$y_{0CW} = y_{0MA} + (m_{CW} + n_R \rho A l_w / 2) g / k_{CW} + \alpha_0 R$
Car frame	$y_{CA} + a_f - a_c$	$y_{0FR} = -y_{0MA} - (m_{FR} + m_{CA} + m_{PA} + n_R \rho A l_f / 2) g / k_{FR} + \alpha_0 R$
Car	$y_{CA}$	$y_{0CA} = y_{0FR} - (m_{CA} + m_{PA}) g / k_{CA}$
Pasengers/Loads	$y_{CA}$	$y_{0PA} = y_{0CA} - m_{PA} / k_{PA}$
Machine Rot.	--	$\alpha_0 R = (m_{CW} + n_R \rho A (l_w - l_f) - m_{FR} - m_{CA} - m_{PA}) g / k_{BR}$
Car frame rope	$l_f = h + h_f - y_{CA}$	$l'_f = l_f + (m_{FR} + m_{CA} + m_{PA} + n_R \rho A l_f / 2) g / k_{FR}$
Counterw. rope	$l_w = h_c + y_{CA}$	$l'_w = l_w + (m_{CW} + n_R \rho A l_w / 2) g / k_{CW}$

The derived mechanical model with 6 degrees of freedom [1] is shown in figure 2. The initial conditions of the system are made coincident with the equilibrium configuration of all the components during a stop after putting on tension the lift, thus with the brakes pressed, no gap between the car floor and the level of the landing floor and a critical time interval passed for assuming that the damping of the system makes the speed of all the components to be zero. Formulas in table 1 are based on the assumption that the machine brakes of a stopped lift behave like an elastic spring of constant  $k_{BR}$ . It is considered that the brakes take some time to grip and release, then the spring/damper constants are linearly variant with time from full intensity to zero during transition from stopped lift to moving lift and vice versa. Alternatively, it is possible to model the brake action using a brake torque  $T_{BR}$  which is applied to the traction pulley according to controlling parameters (On/off switch, rotation direction) and the active torque  $T_{ACT}$  exerted by the gravity forces and motor drive on the traction pulley. Then, the initial rotation  $\alpha_0 R$  shown in table 1 is set to 0, and constants  $k_{BR}$  and  $c_{BR}$  are inexistent in the model.

A standard trip consists basically of three stages. In the first stage the passengers/loads get on the lift car. This makes the system to vibrate together with the passengers/load and after a certain time lapse  $\Delta t_1$  the car-frame sling will elongate

$$\Delta l_{1f} = \frac{m_{PA} g}{n_R E A} l_f \quad (1)$$

where  $n_R$  is the number of ropes,  $EA$  is the longitudinal rigidity of one rope. The gap between car-floor and landing will be:

$$y_{CA}^1(\Delta t_1) = y_{CA}^1(0) - \Delta l_{1f} - \frac{m_{PA}g}{k_{CA}} - \frac{m_{PA}g}{k_{MA}} \quad (2)$$

where  $k_{CA}$  and  $k_{MA}$  are the stiffness of the car and machine silent-blocks, respectively.

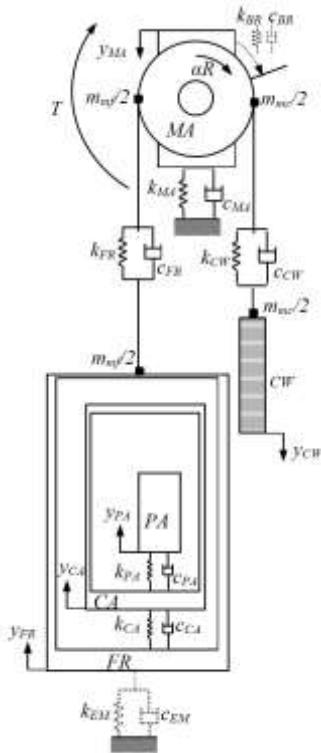
The second stage starts at time lapse  $t_i$  when the button of the desired floor is pressed. Then, a series of electric and electronic components called the controller are responsible to release the brakes and compute the necessary torque  $T$  for the car to follow the setpoint speed for the desired trip. This stage finishes at time lapse  $t_f$  once the setpoint car speed gets zero, the brakes are pressed and motor switch is set to off.

In the third stage the passengers/loads get off the car and the system vibrates during some time  $\Delta t_3$ . The gap between car-floor and landing will be:

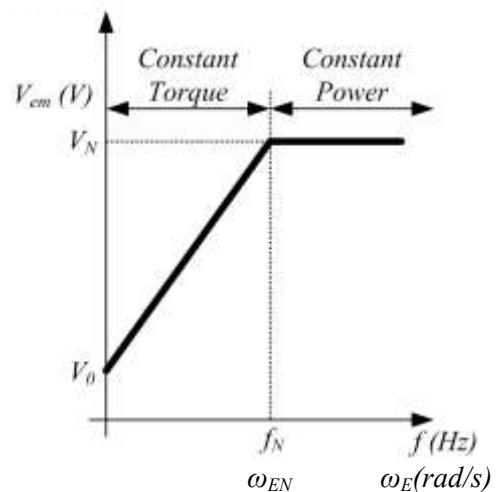
$$y_{CA}^3(t_f + \Delta t_3) = y_{CA}^3(t_f) + \Delta l_{1f} + \frac{m_{PA}g}{k_{CA}} + \frac{m_{PA}g}{k_{MA}} \quad (3)$$

The vibrations and response of the system in the three stages is obtained by the integration of the governing equations following the step by step process indicated in the next paragraph.

The traction forces on the rope have to be transmitted through friction grip between the traction sheave and the rope, resulting inevitably in slip. Following Berner [2] the slip is considered vanishing,



**Figure 2.** The 6 d.o.f mechanical model of the proposed elevator car system [1].



**Figure 3.** Scalar or V/f law control.

that's consistent with rope force ratios close to 1 and no gilding slip occurs.

### 3. Step by step process

The system of equations that govern the elevator system [1] are:

$$[M][\ddot{y}] + [C][\dot{y}] + [K][y] = [F] \quad (4)$$

where  $[M]$  is the mass matrix,  $[C]$  is the damping matrix,  $[K]$  is the stiffness matrix,  $[F]$  is the time dependent vector of applied forces and  $[y]$  is the displacement vector:

$$[y] = [\alpha R \quad y_{FR} \quad y_{CA} \quad y_{PA} \quad y_{CW} \quad y_{MA}]^T \quad (5)$$

The system of equations can be solved using the iterative method of Runge-Kutta[3]. This method can be applied easily in MATLAB after some preparation. First, we obtain the acceleration vector multiplying the members of the system of equations (4) by the inverse of the matrix  $[M]$ :

$$[\ddot{y}] = [M]^{-1} \{ [F] - [C][\dot{y}] - [K][y] \} \quad (6)$$

We define a vector of unknowns  $[Y(t)]$  including the displacement vector  $[y]$  and its derivate:

$$[Y(t)] = \begin{bmatrix} [y] \\ [\dot{y}] \end{bmatrix} \quad (7)$$

Then the derivate of  $[Y(t)]$  can be expressed in the form:

$$[\dot{Y}(t)] = \begin{bmatrix} [\dot{y}] \\ [\ddot{y}] \end{bmatrix} = \begin{bmatrix} [\dot{y}] \\ [M]^{-1} \{ [F] - [C][\dot{y}] - [K][y] \} \end{bmatrix} \quad (8)$$

whose terms can be rearranged:

$$[\dot{Y}(t)] = \begin{bmatrix} [0]_{6 \times 6} & [I]_{6 \times 6} \\ -[M]^{-1}[K] & -[M]^{-1}[C] \end{bmatrix} \begin{bmatrix} [y] \\ [\dot{y}] \end{bmatrix} + \begin{bmatrix} [0] \\ [M]^{-1}[F] \end{bmatrix} \quad (9)$$

If we call  $[A]$  to the first square matrix in (9), which is nearly a time-constant matrix, and  $[B(t)]$  to the vector added to the right, the expression (9) gives:

$$[\dot{Y}(t)] = [A][Y(t)] + [B(t)] \quad (10)$$

The vector  $[B(t)]$ , is a function of the excitation forces  $[F(t)]$  which has the form [1]:

$$[F] = \begin{bmatrix} \frac{T}{R} + (n_R \rho A l_w / 2 - n_R \rho A l_f / 2) g \\ -(m_{FR} + n_R \rho A l_f / 2) g \\ -m_{CA} g \\ -m_{PA} g \\ (m_{CW} + n_R \rho A l_w / 2) g \\ (m_{MA} + n_R \rho A l_f / 2 + n_R \rho A l_w / 2) g \end{bmatrix} \quad (11)$$

where  $T$  represents the applied torque (which is the sum of that produced by the electric machine  $T_{EL}$  and that of the brakes  $T_{BR}$ ) and will depend on the setpoint speed  $v_{ref}$  according to the control law

established by the electronic controller and on the electrical machine parameters, and  $R$  is the traction sheave radius. It is clear that the gravity torque and equivalent gravity forces on the car-frame and counterweight are not time-constant because the assumed rotation with no-slip of the traction sheave makes the rope mass to be transferred from one suspension side to the other.

Once the lift controller fixes the torque  $T$  to be exerted at a certain step  $i$  ( $t=t_i$ ) according to next paragraph, the set of governing equations (10) can be solved using the MATLAB function *ode45*. Then the position and velocities of all the components (7) are determined in the next step ( $t=t_i+\Delta t$ ) where  $\Delta t$  is the time increment of the integration process.

#### 4. The lift controller

As a first simulation step, an ideal controller which computes the motor torque according to kinetic momentum equation of the rotor machine drive as a function of the preset acceleration of the setpoint speed  $v_{ref}$  profile.

A second simulation step consists on the modelling of an electronic converter which is assumed to have a basic scalar (or V/f law) control. Its main function is to reconstruct a three-phase sinusoidal voltage controlling the RMS and frequency of the sinusoidal waveforms according to figure 3. This control operates the electrical motor in two zones, one named “Constant torque” and the other named “Constant power” where the control tries to maintain the torque (or control) constant in the electric machine. It has also to be highlighted that the applied voltage at zero speed is not zero, but a minimum value,  $V_0$ , in order to maintain the torque at low speeds [4-8].

The electronic control analyses the starting and ending point of the vertical trip, and establish the car speed profile for the setpoint speed  $v_{ref}$ , reacting to limit switches and actuating to press or release the mechanical brakes. Once the speed profile for the trip  $v_{ref}=v_{ref}(t)$  is established, the electronic control generates the sinusoidal voltage to be applied to the electrical motor at time  $t_i$ , according to next equations:

$$\omega_E = n_p \cdot v_{ref} \cdot k_L \cdot k_R \quad (12)$$

$$V_{em} = V_0 + (V_N - V_0) \frac{|\omega_E|}{\omega_{EN}} \quad (13)$$

where  $\omega_E$  is the angular speed of the voltage applied to the electric motor (adopted in absolute value as far as negative frequencies require positive voltages, and the interchange of the supply three-phase sequence),  $n_p \cdot k_L \cdot k_R$  is the factor to translate the vertical speed of the car  $v_{ref}$  into electrical motor angular speed  $\omega_E$  taking into account the number of pair of the asynchronous or induction electric motor  $n_p$ , the index ratio of the machine gear box  $k_R$  and the conversion factor from the vertical speed of the car and the mechanical angular speed of the traction sheave  $k_L$  ( $k_L=n/R$  for an  $n:1$  roping elevator),  $V_{em}$  is the RMS value of the voltage applied to the electric motor;  $V_N$ ,  $f_N$  and  $\omega_{EN}$  are the nominal RMS voltage, frequency and angular speed of the electric motor, and  $V_0$  is the minimum voltage applied by the electronic converter at zero speed. The electric motor is modelled as an instantaneous torque source  $T_{EL}$ , with next expression:

$$T_{EL} = k_R \frac{m_1 \frac{R'_2}{s}}{\omega_E + \omega_e} \frac{V_{em}^2}{\left( \left( R_1 + \frac{R'_2}{s} \right)^2 + (L_{cc} \cdot \omega_E)^2 \right)} \quad (14)$$

where  $m_1$  is the number of phases (3 in most cases),  $R_1$  is the stator resistance,  $R'_2$  is the rotor resistance passed to stator,  $L_{cc}$  is the electric machine series inductance and  $s$  is the electric machine slip given by:

$$s = \frac{\omega_E - n_p \cdot v_{mec} \cdot k_L \cdot k_R}{\omega_E} \quad (15)$$

where  $v_{mec}$  is the mechanical speed of the vertical movement. In (14) a value of  $\omega_e=0.001\text{rad/s}$  is included to prevent dividing by zero for zero angular speed  $\omega_E$  case. That is different from the set point speed  $v_{ref}$ , as the induction machine operates with the previously indicated slip, which is dependent on the torque the machine is producing. The mechanical speed  $v_{mec}$  is frequently made coincident with the tangent speed of the traction sheave  $\dot{\alpha}R$  which can be logged out easily by means of an encoder attached to the machine stator in engineering practice. We take  $\dot{\alpha}R$  from the seventh component of the solution vector  $[Y(t_i)]$  at time  $t_i$ . Another option is to make the mechanical speed coincident with that of the car speed  $\dot{y}_{CA}(t_i)$ .

## 5. Computing a trip

The step by step process starts in the stopped configuration. The spring constant  $k_{BR}$  is fixed to its nominal value typically  $k_{BR}=10^8\text{N/m}$  in the option of modelling the brake actuation by a spring/damper constant. That can be estimated by  $T_{BR}/(\varphi R^2)$  where  $T_{BR}$  is the torque exerted by the brakes on the motor and  $\varphi$  is the angle during the braking action). Then, preset time variant passengers mass  $m_{PA}(t)$  and spring constant  $k_{BR}(t)$  are introduced to simulate the getting on process. In the other option,  $k_{BR}$  is set to 0. Then the brake torque  $T_{BR}$  is computed from the active torque  $T_{ACT}$  and the instant maximum braking torque at every step.

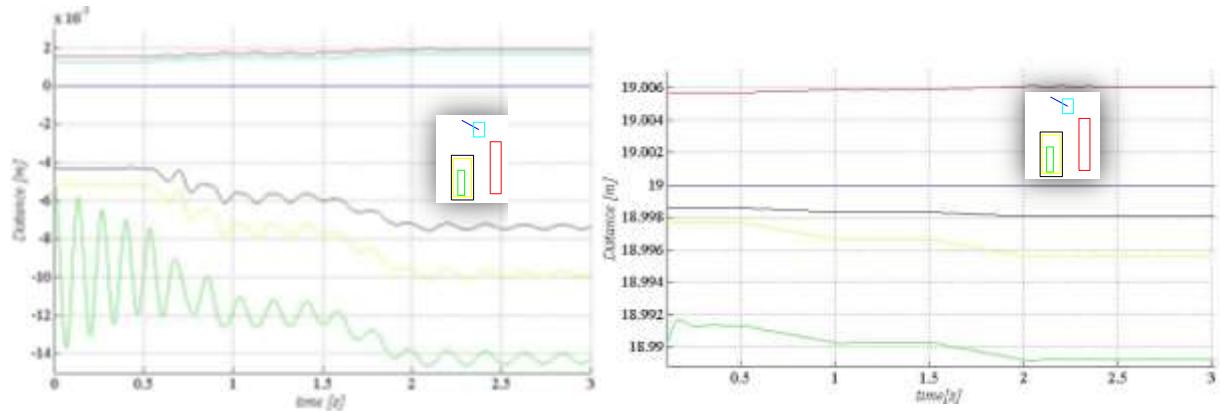
“Get on” and “Get off” routines are used before and after any standard trip. Initial conditions previous to getting on or getting off the car can be established by table 1 for the empty car ( $m_{PA}=0$ ) and stopped system (brakes pressed).

The simulated standard trip stage is the main routine in the program. The trip is simulated from  $i$ -th floor to  $j$ -th floor of an installation using MATLAB [9]. The program follows the step by step process previously described then  $\alpha R$ ,  $y_{CA}$ ,  $y_{MA}$ ,  $y_{FR}$ ,  $y_{PA}$ ,  $y_{CW}$  and its first, second and third derivates are estimated during the time interval  $[t_i; t_f]$  in time increments  $\Delta t$ . A general integration time lapse  $\Delta t$  of 0.005s is adopted.

Now we are ready to compare the performance of the various roping systems at the functional differences in the elevator installation.

## 6. Results

Two typical load/unload processes were analyzed. In the passengers’ load process the passenger mass  $m_{PA}$  is varying from 0 to its nominal value at the certain rate (typically 0.8-1 passengers per second [10]). Goods’ loading process is considered too. Handling trucks or freight cars loading is exerted in



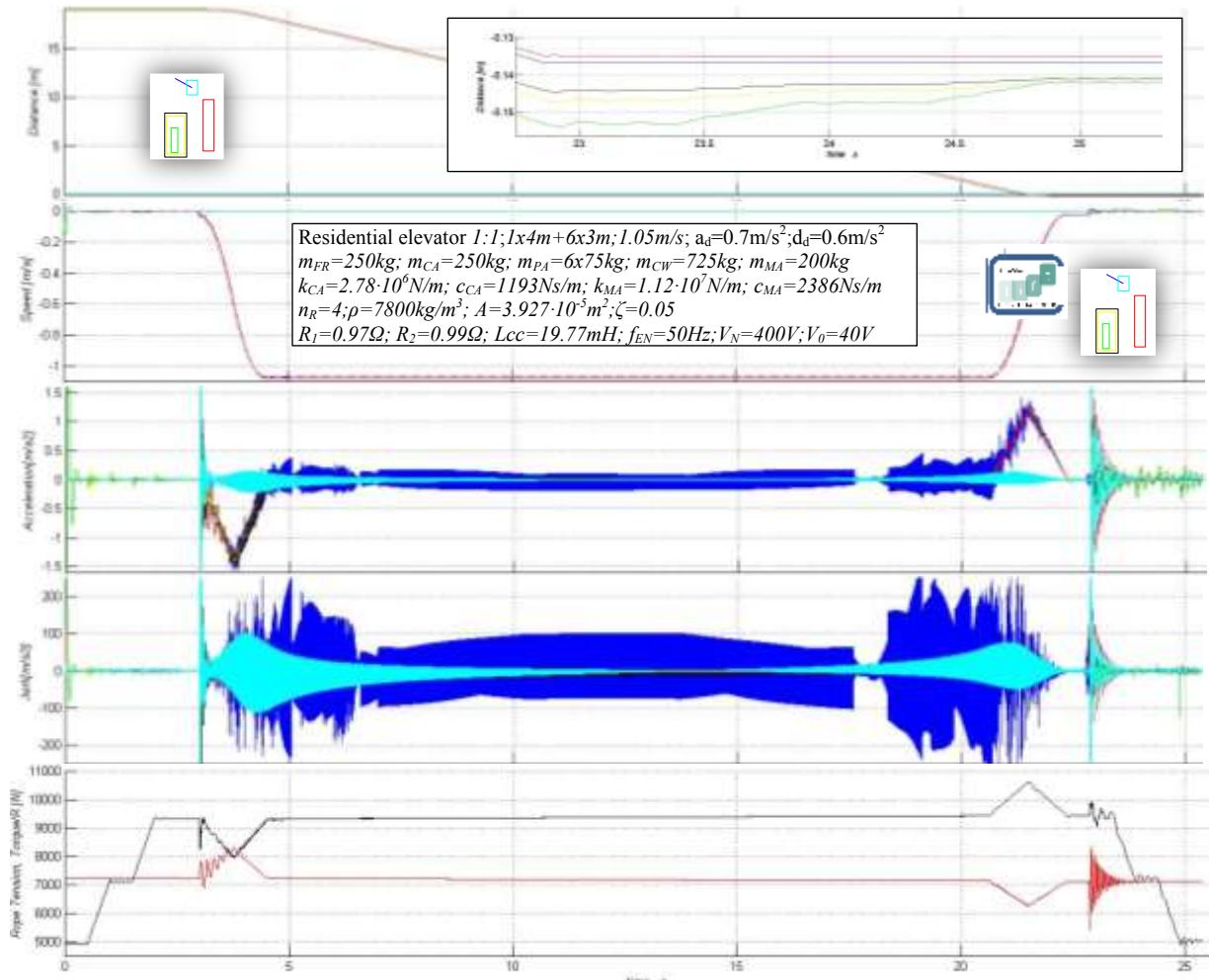
**Figure 4.** Vertical displacements during a residential elevator getting on process. (right) 8 passengers get on floor 6 (19m height) at 8 passengers per second rate: [0.5s-1s] 4 passengers are getting on [1s-1.5s] stay, [1.5s-2s] Additional 4 passengers getting on. (left) 450 kg palette truck get on 0-floor (0m height): [0.5-1s] front wheels are getting on, [1.5s-2s] rear wheels are getting on.

two steps: First, the front wheels are lifted up and then, after some time interval, the rear wheels are lifted up too.

Then, the plot of the instantaneous loaded mass  $m_{PA}(t)$  vs time  $t$  adopts a bilinear form interlaced with a constant mass period. Impacts during getting on/off the car is simulated using a passenger mass  $m_{PA}$  weighted by an equivalent dynamic/static impact coefficient during a short time interval.

During the load/unload stages all the lift components influences its mechanical response, the car and machine silent-blocks are strained and all the components vibrate vertically to some extent. The change in gap between landing floor and car-floor can be easily computed from the displacement plots comparing the initial ( $t=0$ ) and end values ( $t=\Delta t_1$ ) of the car displacement  $y_{CA}$ .

The step by step process indicated in previous paragraph starts once the stopped configuration, the load/unload type and the time variant passengers mass  $m_{PA}(t)$  are adopted. A general integration time

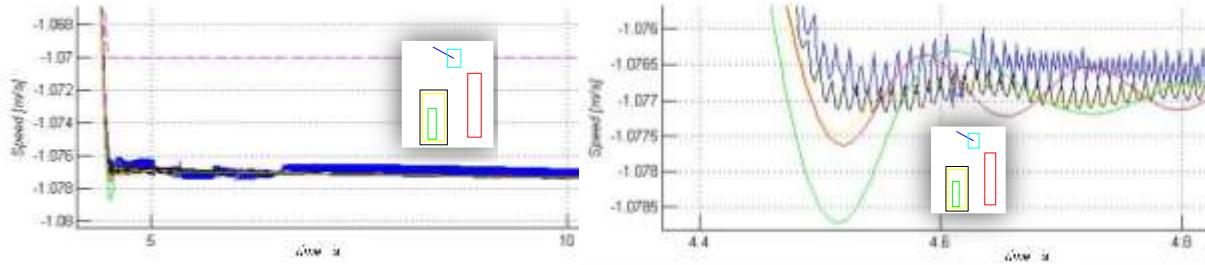


**Figure 5.** Full passengers' trip of an 8 passengers residential elevator at a rated speed of 1.05m/s from 6th to 0-floor inspired in the performance of Ziehl-Abegg's asynchronous machine drive (VFD132.M-4) and using scalar control routine. (Extra key colors, magenta: prescribed speed)

lapse  $\Delta t$  of 0.005s is initially set. After a period of computing time, the display will show the response of the system (figure 4), where we can see the system oscillations to reach balance or the steady state response during the load process.

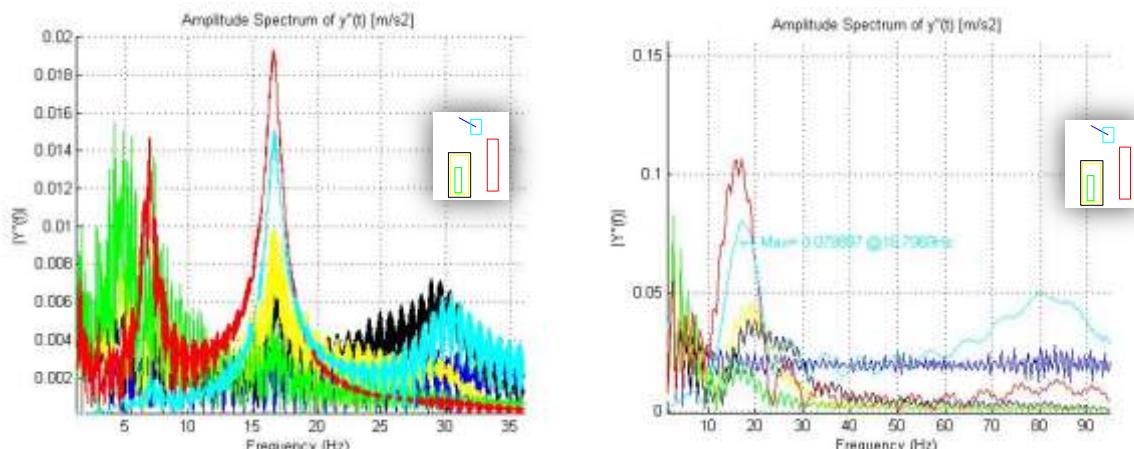
The results are shown in the computer display (Figure 5), where the position, velocity, acceleration and jerk of each component and the slings tension during the full trip are plotted. Using MATLAB zoom capabilities of the plotted figures a more detailed description of velocity, for example, can be

observed (figure 6). The frequency spectrum of all mechanical variables and components during a selectable time gap can be plotted (figure 7). It is observed a different frequency spectrum whether data includes the overall trip or only those from the constant speed portion.



**Figure 6.** Left: Zoom of the speed time history in figure 5 showing discrepancies to preset speed plot (– magenta) with increasing time. Right: Extra zoom of the plot in the left showing detailed component speeds.

Qualitative validation of the simulations for a number of characteristics scenarios has been made. A series of elementary validation tests were done by assigning an infinite stiffness value to all the stiffness parameters of the system except the ropes. These elements also had been assigned a minimum damping coefficient. Attending to the results, the program is fully validated with an overall error of about 0.3%.



**Figure 7.** Acceleration frequency spectrum of (left) the overall trip shown in figure 5 (right) constant speed travel [5-20s] portion.

## 7. Conclusions

A method for solving interactively the equations of motion of any specified lift system by means of a computer has been derived and implemented in the MATLAB platform. A number of dynamics phenomena concerning car position, acceleration, speed or jerk that occurs during the normal functioning in a standard trip and other related exceptional circumstances like impacts provided by guide misalignment or uncontrolled car movements can be reproduced with enough accuracy.

Scalar control and static model have been analysed for electronic controller an electric machine. In future work, more precise controls can be analysed, as field oriented control (FOC), and more detailed model including the dynamics of the electric machine can be applied to analyse the behaviour of the whole system deeply. Moreover, by using FOC control, advanced control techniques, as active damping of mechanic vibration, can be modelled to study how these techniques can mitigate undesired

effect in the whole system and how a more precise and smooth control during a standard trip can be achieved.

In future work more detailed and demanding validation tests can be performed to make test tower experiments compatible to simulated results.

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