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PARAMETER IDENTIFICATION OF A FEED DRIVE FOR HIGH SPEED MACHINE TOOLS

Agustín Casquero, agustincasquero@gmail.com
Facultad de Ingeniería, Universidad Nacional de La Pampa. *Calle 110 esq. 9, General Pico (6360), La Pampa, Argentina*

Rogelio Hecker, hecker@ing.unlpam.edu.ar
Facultad de Ingeniería, Universidad Nacional de La Pampa - *CONICET. Calle 110 esq. 9, General Pico (6360), La Pampa, Argentina.*

Diego Vicente, vicente@ing.unlpam.edu.ar
Facultad de Ingeniería, Universidad Nacional de La Pampa. *Calle 110 esq. 9, General Pico (6360), La Pampa, Argentina*

Marcelo Flores, gmflores@ing.unlpam.edu.ar
Facultad de Ingeniería, Universidad Nacional de La Pampa. *Calle 110 esq. 9, General Pico (6360), La Pampa, Argentina*

Abstract.
Appropriate control techniques can be tuned to reduce the vibrations induced in High Speed Machining (HSM) due to the excitation of the vibration modes. Therefore, the present work proposes an estimation procedure to obtain the parameters for the low frequency model and the frequency of the first resonant mode. A feed drive with ball screw, lineal ball guides, brushless motor and a rotational encoder was used to run the experiments. First, the motor was excited with two different input signals: a train of frequency variable steps and a train of steps with modulated amplitude and constant frequency. The measured rotational position was then used to estimate the parameters for the low frequency model and a coulomb friction value using the least square technique. A set of estimates was performed with different torque amplitude for both signals, showing the best results for the modulated amplitude signal at the maximum torque. Afterwards, the motor was excited with a chirp signal running from 50 Hz to 280 Hz. The Fast Fourier Transform from the measured position shows a resonant frequency at 155 Hz, identified as the first vibration mode of this system. Therefore, the procedure was successful for the estimation of the low frequency model complemented with the coulomb friction and the location of the first resonant mode. This model can now be used to tune controllers with an adequate closed-loop bandwidth for HSM.

Keywords*: feed drive, parameter estimation, vibration modes*

1. INTRODUCTION

Servo-control loops manage the axes of machine tools based on requirements of velocity, position, and acceleration. The success of modeling and control techniques applied to this level sets the basis for production of parts with high quality as well as cycle time reduction, Hecker *et al.* (2008). Today, High Speed Machining (HSM) is possible due to the advances in cutting tools and spindle speed. Particularly, the axes of a machine for HSM require tracking of high acceleration and high jerk trajectories, which in turn demands high bandwidth closed-loops. However, this solution can excite the vibration modes of the system, inducing vibrations degrading the real trajectory and causing direct consequences on parts precision.

Therefore, it is convenient to have a representative model in order to apply adequate control schemes to reduce these negative effects. This model must represent the system with a certain degree of complexity, which could vary depending on the particular application. There are several methods to obtain a model, including parametric or non parametric models. In the case of parametric models, some parameters can be measured directly, but some others are difficult to measure or can vary with time, such as Coulomb or viscous friction, therefore must be identified.

In this case, the system is a linear positioning axis, which is composed by a brushless motor connected to a ball screw that drives a carriage, guided by lineal ball bearings. First, a rigid model plus a coulomb friction model are proposed and identified. Two type of signal were used, both at different levels of motor current. The estimated parameters where analyzed for each set of experiments and the optimal values were proposed, based on some priori parameter knowledge. The first vibration mode can be identified to augment the rigid model and then to tune an adequate controller, like notch filter, to avoid vibrations in high bandwidth closed-loops. Previous research shows that the first vibration mode of this kind of feed-drive systems can go from *65 Hz* (Smith, 1999) to *349 Hz* (Varanasi, 1999), where a bandwidth of at least *100 Hz* is recommended for HSM. This work proposed to excite the system with a chirp signal and analyze the response with a Fast Fourier transform to identify the frequency of this resonant mode.

2. SYSTEM DESCRIPTION

The system is a linear positioning axis, as it is shown in Fig. 1. It is basically composed by a brushless motor with a rotational encoder, a ball screw, a moving carriage and lineal ball bearings.

The brushless motor has a maximum speed of *3600 rpm*, a nominal torque of *3.53 Nm* and a maximum torque of *12,2 Nm*. This motor is attached to a lead-screw drive, with a *10 mm/rev* lead, which is fixed to the base by means of two bearings, a rigid one in one side and a floating one in the other side. The nut of the screw-drive is preloaded, in order to minimize backlash, and it is connected to the moving carriage by means of the nut holding. The moving carriage is guided by two linear bearings, which supports the top and side loads, and allows the carriage to move in a straight line.

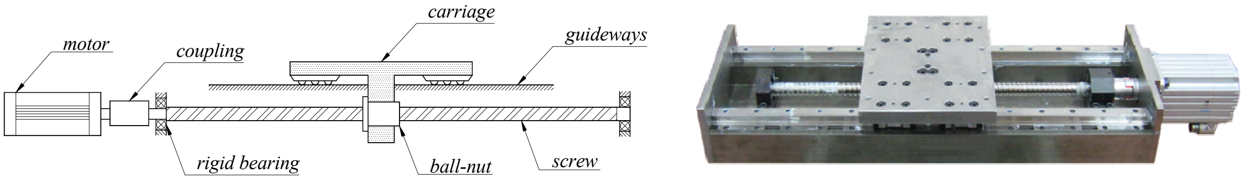

Figure 1: System outline

The motor has a rotational encoder that provides, in combination with the ball screw, 2.5 $m\mu$ of resolution in the linear position. The mechanical system is connected to a real time control system that is mounted in an industrial PC, and serves as the connection between the user and the system. The control system consists of a PCI expansion board with an onboard processor that runs a real time operating system. The control system is connected to the hardware using an analog output to command the motor torque (by means of its power amplifier) and a quadrature encoder input in order to read the angular position of the motor. The analog output has a *12 bits* resolution and the quadrature decoder has a *16 bits* counter, expandable to *32 bits* by software.

3. LOW FREQUENCY MODEL

The motor controller tries to set, in a closed-loop configuration, a motor current proportional to the input signal. In general, the bandwidth of this current closed-loop is much higher than the frequency range of interest of the mechanical system. Therefore, this dynamics is discarded and the input to the system is directly the motor current, or in this case, the motor torque.

Now, a low frequency model that includes the viscous friction is considered. First, the rotational behavior of the system can be represented by

$$J_c\,\ddot{\theta} = T_m - B_\theta\,\dot{\theta} - T_r \tag{1}$$

where J_c is the combined inertia of the motor, the screw and the coupling, θ is the angular coordinate, B_θ is the rotational viscous coefficient, T_m is the motor torque, and T_r is the reaction torque applied by the carriage. Similarly, the linear behavior of the system can be represented by

$$M\,\ddot{x} = F_r - B_x\,\dot{x} - F_p \tag{2}$$

where M is the carriage mass, B_x is the linear viscous coefficient, F_r is the reaction to the resistant torque applied by the carriage to the screw, F_p is a disturbing force, and x is the linear coordinate.

The rotational and the linear equations are coupled by the screw-nut interface considering that $T_r = F_r\,r$, where r is the screw lead, and also tacking into account that $r = x/\theta$. Using these relations, Eq. (1) and Eq. (2) can be combined to obtain

$$T_m = \frac{\ddot{x}}{r}\left(J_c + r^2 M\right) + \frac{\dot{x}}{r}\left(B_\theta + r^2 B_x\right) + r\,F_p \tag{3}$$

The transfer function from the motor torque to the linear position is found by applying the Laplace transform to Eq. (3) and considering that no disturbing forces are present ($F_p=0$)

$$\frac{X(s)}{T_m(s)} = \frac{r}{s\,(s\,J + B)} \tag{4}$$

where J is the system combined inertia and B represents the system combined viscous coefficient, which can be written in terms of the previously defined parameters as follows

$$J = J_c + r^2\, M \quad and \quad B = B_\theta + r^2\, B_x \tag{5}$$

The continuous model represented by Eq. (4) is transformed to discrete variables using a Zero Order Hold to obtain

$$\frac{X(z)}{T_m(z)} = \frac{k_1\, z + k_2}{(z-1)(z-p)} \tag{6}$$

where

$$k_1 = \frac{r\,T}{B\,\ln p}\left(1 - p + \ln p\right), \quad k_2 = \frac{r\,T}{B\,\ln p}\left(-1 + p - p\ln p\right), \text{ and } \; p = e^{-\frac{B}{J}T} \tag{7}$$

and where T is the sampling time. Now, the model can be used to know the order of the system in a parametric identification process.

3.1. Parameters estimation method

In order to obtain the parameters of the low frequency model, a similar procedure as the one proposed by Erkorkmaz and Altintas (2000) is followed. This approach uses the Least Squares Estimation (LES) method, but taking into account the Coulomb friction of the system. Therefore, the results from the estimation process are the estimated values for the combined inertia, combined viscous friction of the system, and dynamic Coulomb friction.

From Eq. (6), adding the Coulomb friction (T_f), the following equation in differences can be formulated

$$x(k) - (1+p)\; x(k-1) + p\; x(k-2) = k_1\left(T_m(k-1) - T_f(k-1)\right) + k_2\left(T_m(k-2) - T_f(k-2)\right) \tag{8}$$

and writing the equation in terms of $x(k)$

$$x(k) = (1+p)\; x(k-1) - p\; x(k-2) + k_1\; T_m(k-1) + k_2\; T_m(k-2) - k_1\; T_f(k-1) - k_2\; T_f(k-2) \tag{9}$$

Now, this equation can be simplified defining the following variable

$$\Delta x(k) = x(k) - x(k-1) \tag{10}$$

Additionally, the friction term is Eq (9) is replaced by a simplified model of the Coulomb friction, described by the following equation

$$T_f = T_{din}^{+} \cdot VP\left(w(k)\right) + T_{din}^{-} \cdot VN\left(w(k)\right) \tag{11}$$

where T_{din}^{+} and T_{din}^{-} represent the positive and negative dynamic torque, respectively, whereas VP and VN are two functions that depend on the velocity as follows

$$VP\left(\omega(k)\right) = \begin{cases} 1 & if\;\; \omega(k) > \omega_{\min} \\ 0 & if\;\; \omega(k) \le \omega_{\min} \end{cases} \qquad VN\left(\omega(k)\right) = \begin{cases} -1 & if\;\; \omega(k) < -\omega_{\min} \\ 0 & if\;\; \omega(k) \ge \omega_{\min} \end{cases} \tag{12}$$

where the parameter ω_{min} is a minimum non-cero velocity value, used to minimize the effects of error and noise in the velocity calculation, specially around zero velocity.

Using the friction model, Eq. (9) can be rewritten to obtain

$$\begin{aligned} \Delta x(k) = {} & p\; \Delta x(k-1) + k_1\; T_m(k-1) + k_2\; T_m(k-2) - k_1\; T_{din}^{+} \cdot VP\left(w(k-1)\right) - \\ & - k_1\; T_{din}^{-} \cdot VN\left(w(k-1)\right) - k_2\; T_{din}^{+} \cdot VP\left(w(k-2)\right) - k_2\; T_{din}^{-} \cdot VN\left(w(k-2)\right) \end{aligned} \tag{13}$$

Using Eq. (13) with N samples, the matrixes for the LSE are obtained, and these can be written as

$$\mathbf{Y} = \mathbf{\Phi} \cdot \mathbf{\theta} \tag{14}$$

where **Y**, **Φ** y **θ** are defined as

$$\mathbf{Y} = \begin{bmatrix} \Delta x(3) \\ \Delta x(4) \\ \cdots \\ \Delta x(N) \end{bmatrix} \quad \mathbf{\theta} = \begin{bmatrix} p \\ k_1 \\ k_2 \\ -k_1 T_{din}^{+} \\ -k_1 T_{din}^{-} \\ -k_2 T_{din}^{+} \\ -k_2 T_{din}^{-} \end{bmatrix} \quad \mathbf{\Phi} = \begin{bmatrix} \Delta x(2) \; \Delta x(3) \cdots \Delta x(N-1) \\ T_m(2) \; T_m(3) \cdots T_m(N-1) \\ T_m(1) \; T_m(2) \cdots T_m(N-2) \\ VP(2) \; VP(3) \cdots VP(N-1) \\ VN(2) \; VN(3) \cdots VN(N-1) \\ VP(1) \; VP(2) \cdots VP(N-2) \\ VN(1) \; VN(2) \cdots VN(N-2) \end{bmatrix}^T \tag{15}$$

And the vector that minimizes the estimation error is found by (Franklin *et al*, 1998)

$$\mathbf{\theta} = \left(\mathbf{\Phi}^T \mathbf{\Phi}\right)^{-1} \mathbf{\Phi}^T \mathbf{Y} \tag{16}$$

Once that **θ** ($p \; k_1 \; k_2$) is calculated, the transfer function in Eq. (6) can be obtained by replacing the identified coefficients. With this equation the continuous representation represented by Eq. (4) can be found, by using the *Matlab* function D2C, from where the values of J and B can be calculated, where the screw lead (r) is a known parameter.

The values of the dynamic Coulomb friction (both positive and negative) can also be obtained; they can be calculated by the following equations, based on the identified data

$$T_{din}^{+} = -\left(\frac{-k_1 \, T_{din}^{+} - k_2 \, T_{din}^{+}}{k_1 + k_2}\right) \qquad T_{din}^{-} = -\left(\frac{-k_1 \, T_{din}^{-} - k_2 \, T_{din}^{-}}{k_1 + k_2}\right) \tag{17}$$

3.2. Experiments and results

A set of experiments were conducted to get the needed data for the identification process. Two types of exciting signals were used: One consisting of a series of steps of varying sign and height (as proposed by Erkorkmaz and Altintas, 2000); and the other consisting on a series of steps of varying sing and frequency, but with constant height. Both types of signals can be seen in Fig. 2.

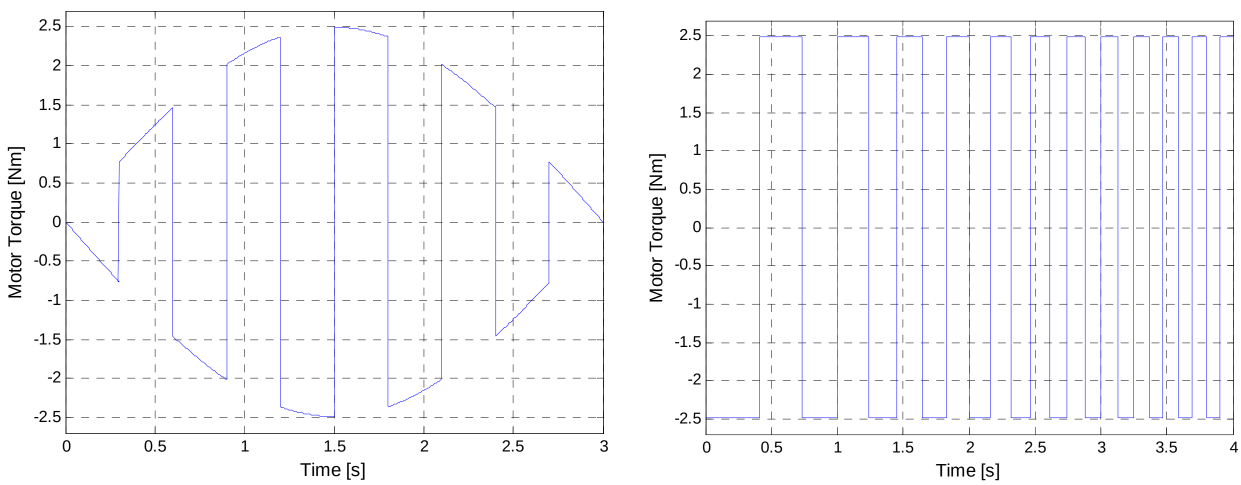

Figure 2: Varying Height and Varying Frequency Exciting Signal

These signals were applied to the system, with different values of maximum current, i.e.: *1 A, 1.5 A, 2 A, 2.5 A, 3 A, 3.5 A, 4 A* and *4.5 A*. For each signal the continuous parameters of the system were estimated (combined inertia J and combined viscous friction B). The comparison for the parameters obtained with each exciting signal is shown in Fig. 3.

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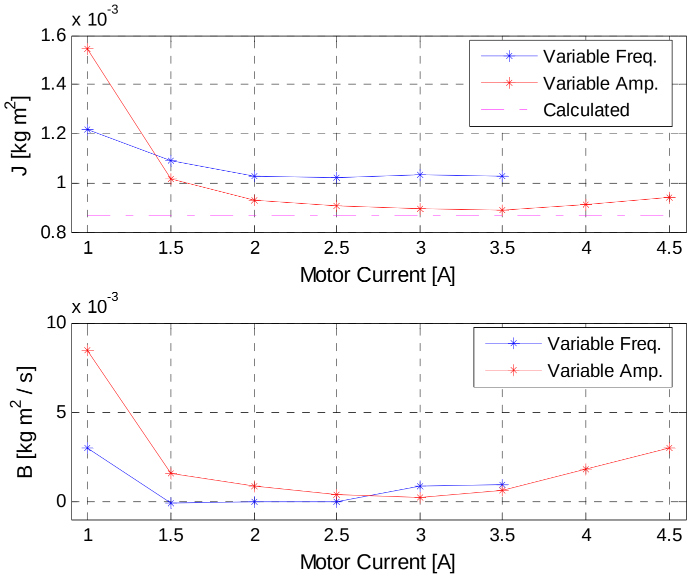

Figure 3: *J* and *B* model's parameters obtained with the exciting signasl, varying the maximum motor torque (or current)

It can be seen in Fig. 3 that, for both signals, the parameters identified (*J* and *B*) seem to stabilize at a given value as the maximum torque is increased. It can be also seen that for currents over *2.5 A* (for the frequency varying signal) and over *3.5 A* (for height varying signal) the identified values tend to increase, but this is due to a saturation of the motor speed, which introduce a non-linearity into the system.

From the graphic of the inertia, it can be seen that the signal of steps of varying height is the one that best approaches the theoretical inertia for this system (doted line), so it can be concluded that this type of signal is the best among the ones used.

Finally, the values of the estimated parameters were chosen from the experiment which has the best approximation to the theoretical value of the combined inertia, this is for the varying height signal with a maximum current of *3.5 A*. The identified values are resumed here as

$$\hat{J} = 8{,}885 \cdot 10^{-4}\, kg\, m^2 \quad \hat{B} = 6{,}061 \cdot 10^{-4}\, kg\, m^2 \quad \hat{T}_{din}^{+} = 0{,}605\ Nm \quad \hat{T}_{din}^{-} = 0{,}620\ Nm \tag{18}$$

1. FIRST VIBRATION MODE IDENTIFICATION

The system was excited with a high frequency signal to identify the first vibration mode. The signals were applied during a short period of time, while the position and velocity were measured. After that, a Fourier Transform analysis was applied to the position and velocity data, to determine the frequency of the vibration mode.

As an excitation signal, a chirp type signal was used, commanded directly to the motor current amplifier. This kind of chirp signal was suggested by various authors, such as Ljung (1999) and Franklin *et al.* (1998), and can be generated using the following equation

$$T_{ref} = T_{\max}\ K \sin\left(\omega_i t\right) \qquad \omega_i = \omega_s + \frac{\left(\omega_e - \omega_s\right)}{t_{\max}} \qquad K = \begin{cases} \dfrac{t}{a \cdot t_{\max}} & 0 \le t \le a \cdot t_{\max} \\ 1 & a \cdot t_{\max} < t < (1-a) \cdot t_{\max} \\ \dfrac{t_{\max} - t}{a \cdot t_{\max}} & (1-a) \cdot t_{\max} < t < t_{\max} \end{cases} \tag{19}$$

where T_{ref} is the reference motor torque function, T_{max} is the maximum motor torque, ω_i is the instant frequency, which stars at ω_e and ends at ω_e, K is the modulation function that is defined by the total signal time (t_{max}) and the percent of the total time that the signal is modulated (a). The instant frequency (ω_i) of the signal (T_{ref}) varies linearly with time, and the amplitude is modulated at the beginning and the end by the saturation function K.

This kind of signal generates a nice window of excitation frequencies in the range of interest, as Fig. 4 shows.

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4.1. Experiments and results

For the identification of the first resonant mode of the system, chirp type signals were used, as described before. To do this, it is necessary to define the chirp parameters like: Starting and ending frequency, maximum torque, total time, and modulation.

In order to identify the first vibration mode, the window of excitation frequency should be in the range of this expected resonant mode. To have some rough idea about the frequency of the vibration mode a detailed model of the system was simulated. This model, developed by Vicente *et al.* (2007), takes into account the first four axial vibration modes and the first four angular vibration modes of the system, and the coupling between them. Using this model it was found that the frequency of the first resonant mode was about *167 Hz*, for the carriage positioned at the center of the axis.

With the approximated information of the vibration frequency, the starting and ending frequency were chosen to be *50 Hz* and *280 Hz* respectively. The maximum torque of the signal was fixed at *3.55 Nm* (*5 A*). The total length of the experiment was limited to *2.4 s*, with a sampling time of *0.3 ms*, and the initial and final modulation was about *2 %* of the experiment time.

Once the position and velocity data were obtained using the defined exciting function, the FFT was applied to these signals, which are shown in Fig. 4. The presence of a peak at *155 Hz* can be seen in the figure, both in position and in velocity. The FFT of the exciting signal is also plotted.

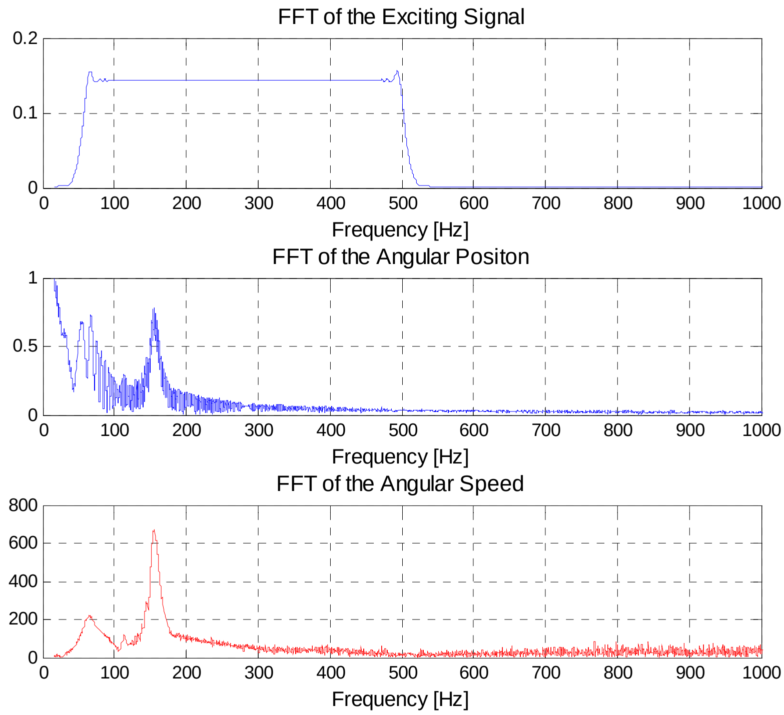

Figure 4: FFT of the excitation signal, the angular position and the angular speed, respectively.

The experiment was repeated varying some parameters, like starting and ending frequency and sampling time, and in all cases a peak at *155 Hz* was observed. It should be note that all the experiments were made with the carriage in the center of the positioning system, so the resonant frequency identified will be for that position.

5. CONCLUSIONS

A method to identify a low frequency model, including the Coulomb friction effect, of a linear feed drive system was presented. This method was tested experimentally using different excitation signal, where the best set of identified parameters was proposed.

Alternatively, the frequency of the first vibration mode was obtained to augment the rigid model. The first resonant mode was clearly observed in the FFT of the velocity and the position signals for a convenient excitation signal.

The augmented model of the system in this work can be used to select and tune control techniques for feed-drives requiring high bandwidth closed-loops.

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