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Identification of mechanical parameters at low velocities for a micropositioning stage using a velocity hysteresis model *

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Abstract

The paper presents a velocity hysteresis friction model. This model can be used for simulation or control of an accurate micropositioning servomechanism. The mechanical and friction parameters are identified from measured signals at low velocities, and with the use of a closed loop identification method. The measured signals are very noised. The number of measurement samples available is very low. The friction model depends on position, velocity and acceleration and includes the Coulomb, Stribeck and viscous frictions. A hysteresis effect integrated by an exponential term and the acceleration is added in the model. The identified parameters are validated by applying an efficient precision method based on the sensitivity analysis of nonlinear models and a normalized mean-square-error criterion.

1. INTRODUCTION

Nowadays the miniaturization become a very widespread subject demanded in industries of manufacturing electronic and optical devices, semiconductors and convenience goods. The development of manufacturing systems conduct to new smart solutions, to reduce the energy consumption, as well to improve the environmental conditions [17]. A major task in the manufacturing of miniaturized electronic devices is the necessity of servomechanisms that are characterized by a high precision of micropositioning. The servomechanisms with ball-screw transmission are the most popular driving systems recognized for high speed and long stroke precision positioning stage [17]. However, these type of systems could be affected by nonlinear effects,

thus the positioning accuracy of the servomechanism can be deteriorated by friction.

Friction is a natural phenomenon, a tangential reaction force between two surfaces in contact and a significant nonlinear effect that is hard to be precisely modeled. The friction is given by relations (static friction models) between the friction force and the relative velocity between the surfaces in contact. The static models contain Coulomb, viscous or static friction terms, and combination of these three effects, also adding Stribeck effect. More advanced friction models that cover better the friction compensation are the dynamic friction models based on the static friction models [9], an internal variable which provide the friction evolution during the sliding displacement, and added different phenomena as hysteresis or stick-slip effects. The most significant dynamic friction models are presented in [2]. The spring-like behavior during stiction is described by the Dahl model where the friction force depends on position and velocity [2]. The LuGre model combines the Dahl model with an arbitrary steady-state friction characteristics [12], and allows to describe the frictional lag in sliding regime, hysteresis behavior in presliding regime and varying break-away force depending on the rate of change of the applied force [13], [19].

The identification of mechanical systems, more precisely micropositioning servomechanisms can be realized by different methods. One of the most popular and basic identification method is the least square method applied frequently at linear systems, or linearized systems [10], [11], [20]. This method allows to identify a minimal set of parameters [20], which can be used then in direct or inverse dynamic identification methods. Unfortunately the method does not give good results if the measured signals are strongly perturbed and the friction models used in mechanical system modeling are non linear in the parameters [11]. The inverse dynamic identification method based on the output error minimizing the quadratic error between the measured and simulated output respectively, is a successfully method used to identify parameters of different mechanical systems and industrial robots [10]. Other

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In this work we propose to identify mechanical and friction parameters for a micropositioning system and calculate the precision of the identified parameters. The experimental plant designed for this study is presented in the second part of this paper, describing the dynamic equations, as well as the models used for friction. In the third part is presented the closed loop identification method where for the optimization criterion is used the control input of the mechanical system. Through further studies was shown that using the output of the mechanical system the identification goes wrong. Finally before presenting the results is described a method based on the sensitivity analysis applied for the computation of the identified parameters precision.

2. SERVOMECHANISM MODELING

The work was effectuated for the development of a winding machine which should realize complex steps during the RFID tags manufacturing. The difficulty of this work appears when the gold wire of micrometric size is wound around the magnetic material. It should be avoided the break of gold wire and keep a relative constant step between the windings. For this is needed a good precision of winding obtained from a good synchronization between the linear axis displacement and the magnetic material rotation.

In order to achieve the precision of micropositioning was conceived an experimental plant with two types of linear axes (one with ball-screw transmission, a second one with compliant nut-screw) and were studied individually. Each mechanical system consists of a brushless motors coupled with an incremental encoder, a positioning controller from Maxon Motor company, a linear stage, an incremental linear encoder and a PC. The resolutions of the angle encoders are 500 impulses/revolution and there is connected to the positioning controller. Finally the communication for programming and data transfer between the positioning controller and the PC is realized by a RS-232 serial link. In Fig.1 are represented the mechanical components involved in the mechanical sub-system (MSS) with the typical notions. The behavior of the MSS is expressed by the following equations obtained from the fundamental principle of dynamics:

$$\tau_m = (J_m + J_v)\ddot{\theta}_m + C_m + \tau_f \quad (1)$$

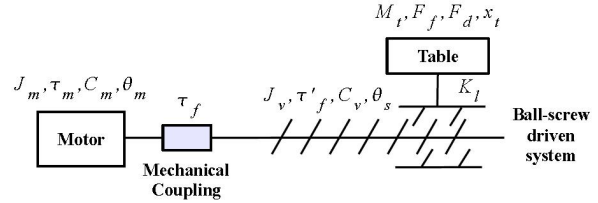


Figure 1. Schematic structure of the linear positioning system

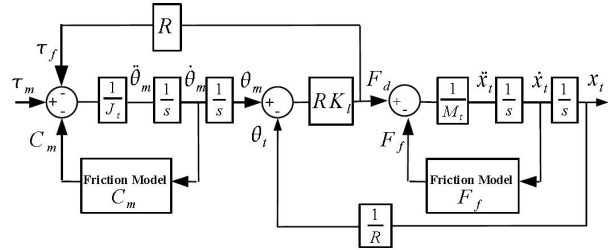


Figure 2. Model of the mechanical system

$$\tau_f = RF_d \quad (2)$$

$$F_d = RK_l(\theta_m - \frac{1}{R}x_t) \quad (3)$$

$$F_d = M_t \ddot{x}_t + F_f \quad (4)$$

where τ_m is the motor torque, τ_f is the torque on the ball-screw, F_d is the driving force, θ_m is the motor position measured by an encoder and x_t is the axial position of the table. The rotor speed was noted by $\dot{\theta}_m$ and the acceleration by $\ddot{\theta}_m$. J_m and J_v are the moments of inertia of the motor shaft and the screw respectively. C_m is the friction torque on the motor and screw side and F_f is the friction force between the table and the straight guide. M_t is the table mass, p is the thread pitch and K_t is the equivalent stiffness coefficient in the axial direction. The linear to rotational motion ratio is represented by a parameter R which is calculated in function of the thread pitch p :

$$R = \frac{p}{2\pi} \quad (5)$$

The MSS was developed using the modeled system found in [14]. The model have a complex form, and a hypothesis is taken into account that the mechanical coupling stiffness is infinite, and also the motor position θ_m is supposed to be equal to the screw position $\theta_s = \theta_m$. Note that we define $\theta_t = x_t/R$.

From (1)-(5) is obtained the block diagram of the MSS that is represented in Fig.2. The total moment of inertia obtained by summing the motor and screw moments of inertia was noted by J_t .

2.1. FRICTION MODELING

The friction models considered in the MSS represented in figure 2 are as follow: one friction model on the motor side expressed by (6), where the Coulomb friction is represented by F_m and the viscous friction by B_m , and a second friction model on the table side given by (7) which is a more complex expression function of position, velocity and acceleration.

$$C_m = F_m \text{sign}(\dot{\theta}_m) + B_m \dot{\theta}_m \quad (6)$$

In order to identify the friction behavior and compensate its effects at the pre-sliding motion, it was designed a velocity hysteresis model. This model is composed by the typical Coulomb and viscous friction models that is a very commonly used combination of frictions [12], the hysteresis effect, the Stribeck speed and an angular depending friction force that occurs during the table displacement. The hysteresis effect is a pre-sliding characteristic and unlike other models where it is introduced through the position term [2], in this case the hysteresis effect depends of the velocities. The Stribeck speed is considered in this friction model in order to translate the starting phase of displacement. The Coulomb friction is represented by F_t and the viscous friction by B_t , the Stribeck effect by the terms C_1 and C_2 with V_s the Stribeck velocity. The acceleration $\ddot{\theta}_t$ from the terms C_1 and C_2 permits to introduce an hysteresis effect while the Stribeck effect represented by the exponential term with V_s does not occur during the decreasing phase of the speed, as shown in Fig.3. The angular depending friction force is represented by a magnitude term C_p and an initial phase ψ . The equation (8) defines a fictive rotation angle of the linear displacement of the table.

The friction force for the table side is given as follows:

$$F_f = F_t \text{sign}(\dot{\theta}_t) + B_t \dot{\theta}_t + C_1 + C_2 + C_p \cos(\theta_t + \psi) \quad (7)$$

$$\theta_t = \frac{x_t}{R} \quad (8)$$

$$C_1 = \begin{cases} C_{s1} \frac{1+\text{sign}(\dot{\theta}_t)}{2} e^{-\left(\frac{\dot{\theta}_t}{V_s}\right)^2}, & \text{if } \ddot{\theta}_t > 0 \text{ and } \dot{\theta}_t > 0 \\ 0, & \text{if } \ddot{\theta}_t \leq 0 \text{ or } \dot{\theta}_t \leq 0 \end{cases} \quad (9)$$

$$C_2 = \begin{cases} 0, & \text{if } \ddot{\theta}_t \geq 0 \text{ or } \dot{\theta}_t \geq 0 \\ C_{s2} \frac{1-\text{sign}(\dot{\theta}_t)}{2} e^{-\left(\frac{\dot{\theta}_t}{V_s}\right)^2}, & \text{if } \ddot{\theta}_t < 0 \text{ and } \dot{\theta}_t < 0 \end{cases} \quad (10)$$

The advantage of the velocity hysteresis model in comparison to other dynamic friction models is that the major effects of the friction are considered by a reduced number of parameters, six parameters for a symmetrical friction model and seven parameters for an asymmetrical friction model.

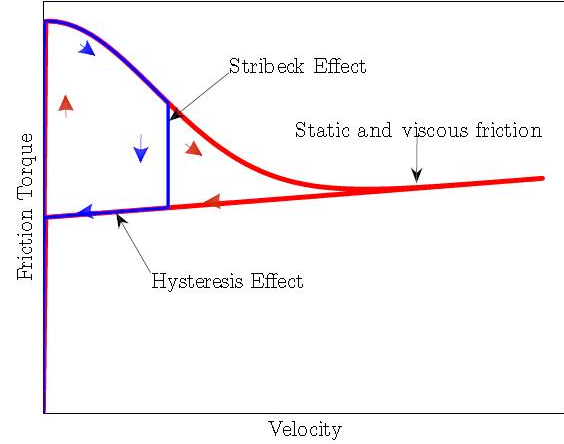


Figure 3. Friction model with hysteresis

3. PARAMETER IDENTIFICATION

Classical methods such as least square or recursive identification [6] were tested for the experimental plant. The least square method allows to find an approached solution for the vector of parameters, but the results are not accurate because of the strongly perturbed signals. The recursive identification method or the open loop identification method based on the output error, where the output is considered the motor position, provide good results only for the friction parameters from the motor side. However, using the principle of the recursive identification method and the vector of parameters obtained from the least square method can be applied a closed loop identification method based on the output error used in [6] for mechanical systems with charge.

The recursive method applied for the present servomechanism is described in [3] and [4]. The identification was available for the reference position value of the system, the current, the velocity and the acceleration of the motor known. The reference position and the current are signals measured using an interface provided by the micro-controller manufacturer. The data acquisition are obtained with a sampling period of 5ms and a trapezoidal profile velocity. The current signal is treated by a classic fifth order median filter in order to reduce the measurement noise, and the velocity and the acceleration of the motor are computed using a non causal second order derivative filter. The identification is obtained for a measured vector with reduced dimension (only 100 measured samples) because of the controller memory size.

3.1. CLOSED LOOP IDENTIFICATION

The closed loop identification method is schematized in Fig.4. The model of the mechanical sub-system is integrated into a closed loop control. The simulation is performed with the same PID numerical control law implemented on the positioning controller. The assembly between the output of the numerical PID controller and the torque provided by the motor is modeled by a first order function, while the motor torque is given by $\tau_m = \frac{ke}{T_{ff}s+1}\hat{I}$, where $k_e = 45.5 [Nm/A]$ is the torque constant, $T_{ff} = 40 \mu s$ the time constant and $\hat{I}$ a reference current with an imposed saturation of $[-5A, 5A]$. The mechanical sub-system was described in section 2. The current of each phases of the brushless motor is controlled by internal loops, and a quantization process is applied in order to describe the motor position encoder which is acquired with a quantification error of $\Delta\theta$ and memorized under a θ_{mz} value. θ_{dv} is the reference motor position which was not measured because of the controller memory size.

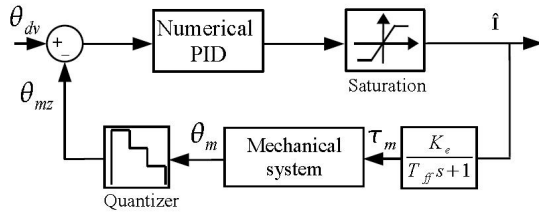


Figure 4. Closed loop simulation model

The simulation allows to recognize for a set of parameters of the mechanical sub-system model the evolution of all sizes and in particular the simulated current of the motor $\hat{I}$. The current estimation $\hat{I}$ is very sensitive at parameters variation than the simulated position θ_m , thus $\hat{I}$ is considered as output of the simulated model and according to its values the optimization criterion MSE is calculated. This optimization criterion was used by [18], representing the normalized mean square error and computed as follows:

$$MSE = 100 \frac{\|I - \hat{I}\|^2}{\|I - \bar{I}\|^2} \quad (11)$$

with I the measured current, $\bar{I}$ the mean value of the measured current, $\hat{I}$ the simulated current considered as output of the model.

4. PRECISION OF THE IDENTIFIED PARAMETERS

The accuracy of identified parameters is an important key in improving the mathematical model designed

for a mechanical system. However not all parameters can be accurately estimated [16], and in this section is proposed a method of study the parameters obtained by closed loop identification method.

The precision of the identified parameters is calculated using statistical properties applied to the measured signal [5]. It is supposed that the system has the same structure as the model. The measured output signal $y(i)$ is defined by (12), where $\hat{y}(i, \hat{\Theta})$ represents the estimated signal and $\varepsilon(i)$ the noise that affects the output.

$$y(i) = \hat{y}(i, \hat{\Theta}) + \varepsilon(i) \quad (12)$$

The error between the vector of the estimated parameters $\hat{\Theta}$ and the optimal value Θ^* is calculated by (13) and the convergence criterion by (14).

$$e = \hat{\Theta} - \Theta^* \quad (13)$$

$$C = \sum_{i=1}^n (y(i) - \hat{y}(i, \Theta^*))^2 = \sum_{i=1}^n [\hat{y}(i, \hat{\Theta}) + \varepsilon(i) - \hat{y}(i, \hat{\Theta}) - \frac{\partial \hat{y}^T}{\partial \hat{\Theta}}(i, \hat{\Theta})e]^2 \quad (14)$$

The term $\frac{\partial \hat{y}^T}{\partial \hat{\Theta}}$ indicates the sensitivity vector and is noted by $s(i)$.

$$s(i) = \frac{\partial \hat{y}}{\partial \hat{\Theta}}(i, \hat{\Theta}) \quad (15)$$

For an optimal value of $\hat{\Theta}$, the criterion error is:

$$C = \sum_{i=1}^n [\varepsilon(i) - s^T(i)e]^2 = \sum_{i=1}^n [\varepsilon(i)^2 - 2\varepsilon(i)s(i)^T e + e^T s(i)s(i)^T e] \quad (16)$$

$$B = \sum_{i=1}^n s(i)s(i)^T \quad (17)$$

$$\beta^T = \sum_{i=1}^n \varepsilon(i)s(i) \quad (18)$$

Replacing $s(i)$ in (14) and using the relations (17) and (18), the criterion is reduced to the following expression:

$$C = \sum_{i=1}^n \varepsilon^2(i) - 2\beta^T e + e^T B e \quad (19)$$

The optimal value of Θ^* is obtained for $\frac{\partial C}{\partial \hat{\Theta}} = 0$ and using the relation (20) was shown that the square between the real and estimated parameters depends sometimes by noise, another time by the sensitivity functions of the model.

$$e = B^{-1}\beta \quad (20)$$

The mean square error e is calculated by (20).

$$E[e] = E[B^{-1}\beta] = B^{-1}E[\beta] \quad (21)$$

For a centered noise $\varepsilon(i)$, the estimator is not biased (22), and the covariance matrix $\hat{\Theta}$ can be calculated by (23).

$$E[e] = 0_d \quad (22)$$

$$\begin{aligned} E[(\hat{\Theta} - \Theta^*)(\hat{\Theta} - \Theta^*)^T] &= E[B^{-1}\beta\beta^T B^{-1}] = \\ &= B^{-1}E[\beta\beta^T]B^{-1} = \\ &= B^{-1}E[(\sum_{i=1}^N \varepsilon(i)s(i))(\sum_{i=1}^N \varepsilon(i)s(i))^T]B^{-1} \end{aligned} \quad (23)$$

Introducing the expression of β into $E[\beta\beta^T]$ and considering this time that the noise $\varepsilon(i)$ is white with a variance σ^2 , the covariance matrix of $\hat{\Theta}$ is presented under the expression (24).

$$\begin{aligned} E[(\hat{\Theta} - \Theta^*)(\hat{\Theta} - \Theta^*)^T] &= B^{-1}(\sum_{i=1}^n \sigma^2 s(i)s(i)^T) \\ B^{-1} &= B^{-1}\sigma^2 B B^{-1} = \sigma^2 B^{-1} \end{aligned} \quad (24)$$

The variance of the estimator and its precision depend of the noise power and its sensitivity functions of the model. The criterion is applied for the precision calculation of the identified parameters using the relation (25).

$$CRT = \sum_{i=1}^n [\varepsilon^2(i) - \beta e] \quad (25)$$

The noise variance σ^2 which intervenes in the precision calculation of the parameters vector $\hat{\Theta}$ is given by:

$$\sigma^2 = \frac{CRT}{n-d} \quad (26)$$

where n is the number of measured samples and d the number of identified parameters. Finally, the precision of the identified parameters is calculated according to P_{Θ} [1].

$$P_{\Theta} = 100 \frac{(\text{diag } \sigma^2)^{0.5}}{\hat{\Theta}(i)} \quad (27)$$

5. RESULTS AND COMMENTS

Following the closed loop identification method presented in section 3, the numerical values of the optimization are shown in table 1 for two displacements performed with different velocities but same sampling period of 5ms and trapezoidal velocity profile. The first optimization was realized for a displacement of 0.3mm operated with a velocity of 0.4r/s (data 1), and the second optimization for a displacement of 0.015mm with a velocity of 0.2r/s (data 2). The precision of the identified parameters is given in column 3 and 5 respectively. The precision percentages shows good results, even if it can be observed a difference between the precision for the same parameter. The velocity sliding and the vibrations could affect easily the friction parameters so that small differences appear in the analysis of data.

In Fig.5 are indicated the comparison between the measured and estimated current for the two different cases of displacement. By the blue line was represented the square of error. There is a relative good similarity between the curves. The MSE criterion for the data 1 (0.3mm displacement) gives 8.70% and for the data 2 (0.015mm) 5.35%.

	data 1	P_{Θ} [%]	data 2	P_{Θ} [%]
J_t	$1.44 \cdot 10^{-5}$	2.02	$1.49 \cdot 10^{-5}$	2.00
B_m	$0.65 \cdot 10^{-4}$	2.37	$0.70 \cdot 10^{-4}$	6.43
F_m	$5.31 \cdot 10^{-3}$	1.27	$6.80 \cdot 10^{-3}$	1.08
M_t	417	1.64	422	2.63
B_t	$0.85 \cdot 10^{-5}$	2.01	$0.77 \cdot 10^{-5}$	2.13
F_t	7.79	1.90	9.77	1.77
C_{s1}	2.23	3.51	2.88	7.29
C_{s2}	1.12	1.75	1.08	3.14
V_s	0.5	3.12	0.48	4.05
K_l	$1.60 \cdot 10^{-2}$	2.22	$1.36 \cdot 10^{-2}$	7.26
C_p	$0.11 \cdot 10^{-3}$	1.58	$0.12 \cdot 10^{-3}$	3.51
ψ	$-5.83 \cdot 10^{-2}$	2.17	$-6.73 \cdot 10^{-2}$	2.98

Table 1. Results of the identified parameters (J_t [kgm²], B_m [Nm s], F_m [Nm], M_t [g], B_t [Ns/m], F_t [N], C_{s1} [N], C_{s2} [N], V_s [rad/s], K_l [N/rad], C_p [N], ψ [rad])

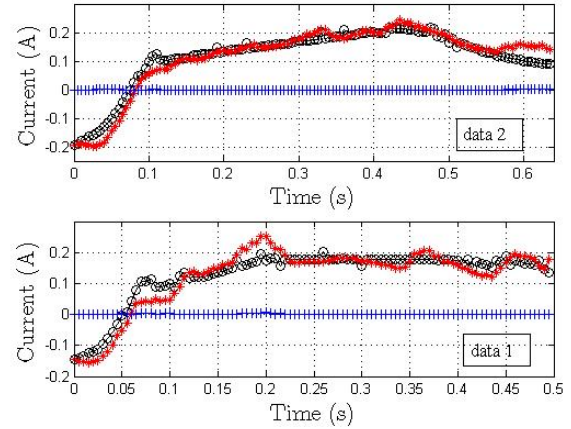


Figure 5. Comparison between measurement (red) and simulation (black)

In Fig. 6 is represented the comparison between the friction torque estimations from data 1 (red) and data 2 (blue) respectively. It can be observed that for a smaller displacement the friction torque obtained by data 2 is higher than the friction torque resulted from data 1. At low velocity, the observed switching on the two friction torque estimations is due to the combination of the static friction and Stribeck friction terms.

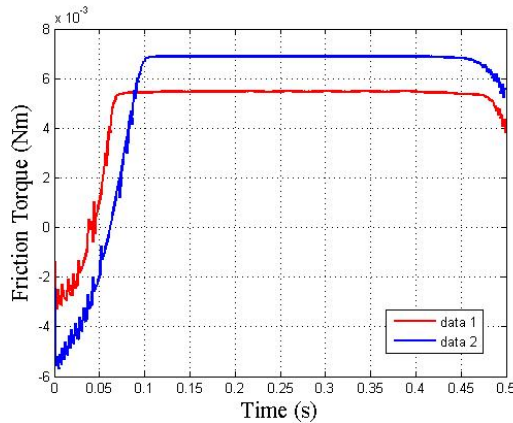


Figure 6. Comparison between friction torques

6. CONCLUSION

In this paper was proposed an identification method for a modeled micropositioning system. The nonlinear parts of the system were modeled by friction, a simple friction model for the motor side, and a velocity hysteresis friction model dependent on the position, velocity and acceleration for the table side. The parameters were identified by applying an optimization method that minimize the criterion error of the model of physical behavior, and the optimization was realized for a convergence of about 200-450 iteration steps. The accuracy of each identified parameters was determined by a computation method based on the sensitivity analysis. The velocity hysteresis friction model by introducing the dependence on the input angular position vector allowed to ameliorate the identification results for signals measured at low velocity, knowing that in past works [4] the friction model was tested only for measured signal at high velocities. The results obtained with the closed loop identification method based on the control input provided by the mechanical system, encourage future works instead of the measured signals characterized by reduced dimensions and noise.

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