Practical Identification Method for Striebeck Friction

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Abstract: In order to achieve high position tracking performances in positioning and robotic systems, the friction phenomena must be taken into consideration during the development of control algorithm. For effective compensation, the value of the friction force must be known in every domain of operation of the machine. This study presents a method for low velocity friction measurement and parameter estimation for the well-known Striebeck friction. The developed measurement procedure can be implemented in simple industrial controller architectures, such as microcontrollers. Experimental results are also presented in the paper to show the applicability and performances of the proposed method.

Keywords: Friction measurement, parameter identification, industrial controller

1 Introduction

Nowadays many types of high resolution position sensors and precise, fast actuators are available at relatively low cost. With these devices very high precision position control tasks became achievable for industrial applications. However in many practical applications it was observed that the high precision position tracking control performances are severely influenced by friction in a negative sense. This is why the search for new friction modeling and identification techniques became a popular research trend.

Many models were developed to explain the friction phenomenon. These models are based on experimental results rather than analytical deductions and generally describe the friction force (F_f) in function of velocity (v).

The classical *static + kinetic + viscous friction model* is the most commonly used in engineering. This model has three components: the constant Coulomb friction model which depends only on the sign of velocity, the viscous component, which is proportional with the velocity and the static term, which represents the force necessary to initiate motion from rest and in most of the cases its value is grater than the Coulomb friction:

$$F_f = F_S \eta(v) + F_C sign(v) + F_V v, \quad \eta(v) = \begin{cases} 1, v = 0 \\ 0, v \neq 0 \end{cases} \tag{1}$$

The servo-controlled machines are generally lubricated with oil or grace (hydrodynamic lubrication). Tribological experiments showed that in the case of lubricated contacts the simple static +kinetic + viscous model cannot explain some phenomena in low velocity regime, such as the *Striebeck effect*. This friction phenomenon arises from the use of fluid lubrication and gives rise to decreasing friction with increasing velocities. This phenomenon can be modelled only by introducing nonlinear terms in the friction model.

In order to compensate the influence friction force, friction models are needed that describes well the frictional behavior and can be incorporated in control algorithms. To describe the Striebeck phenomena, in [1] nonlinear models were proposed that can explain the nonlinear behavior of friction at low velocities. Form theoretical point of view an important result was formulated in [2], in which it was shown that the dynamics of a single input mechanical system with Coulomb friction has a well defined, absolutely continuous solution (Carathedory solution). The influence of static friction force on control was also treated by many authors. The most important papers are connected to the friction generated limit cycles. In [3] the limit cycles were studied in flexible servo systems influenced by Coulomb+static friction. In [4] the PID type controllers' behaviors were studied in the presence of static friction.

To measure the frictional parameters, friction identification and measurement methods were also discussed in many studies. In [5] a time domain identification method is proposed for static friction models which are not necessarily linear in parameters. The method needs no information of acceleration and mass, the only assumption is that the initial and final velocity during the identification must be identical. Genetic algorithm based identification for the Striebeck friction parameters was described in [6]. Frequency domain identification methods were also proposed to identify friction. The study [7] presents a frequency domain identification method for simultaneous identification of velocity and position dependent friction. Neural network based identification methods are also popular to capture the frictional behavior. In [8, 9] Support Vector Machines were proposed for friction modeling and identification. The advantage of this type of model identification method is that it can identify nonlinearities from sparse training data.

The rest of the paper is organized as follows: Section 2 presents the Striebeck friction phenomena in detail and the proposed friction model, which describes this frictional behavior and will be used for the identification. Section 3 presents the proposed measurement and identification method. In Section 4 experimental measurements are given.

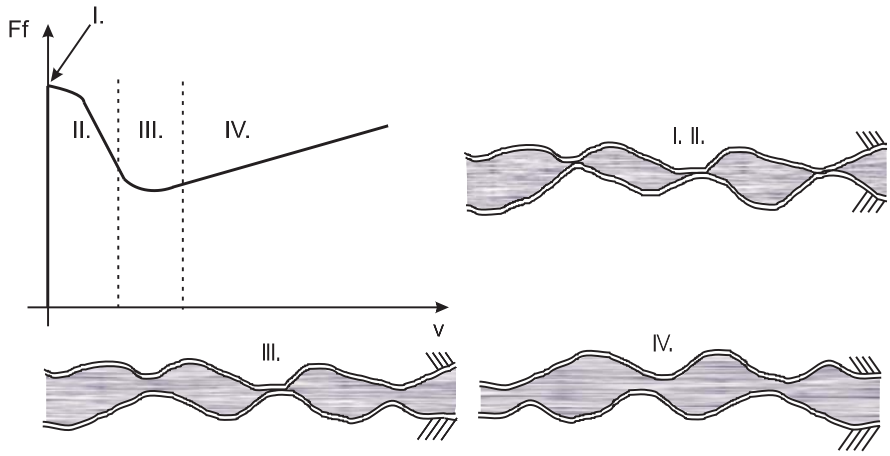

Figure 1
Striebeck Friction Regimes

2 Low Velocity Friction Modeling

Friction is a result of complex interactions between the surface and the near-surface regions of two moving bodies in contact. In many industrial mechatronic systems the surfaces in contact are lubricated with oil or grace. Due to the interaction of moving surfaces rises the highly nonlinear and discontinuous friction force in the low velocity regime which also contributes to nonlinear behavior of friction.

2.1 Striebeck Friction

To describe this low velocity friction phenomenon, four regimes of lubrications can be distinguished (see Figure 1). *Static Friction: (I)* the junctions deform elastically and there is no excursion until the control force does not reach the level of static friction force. *Boundary Lubrication: (II)* this is also solid to solid contact, the lubrication film is not yet built. The velocity is not adequate to build a solid film between the surfaces. A sliding of friction force occurs in this domain of low velocities. The friction force decreases with increasing velocity but generally is assumed that friction in boundary lubrication is higher than for fluid lubrication (regimes three and four). *Partial Fluid Lubrication: (III)* the lubricant is drawn into the contact area through motion, either by sliding or rolling. The greater the viscosity or motion velocity, the thicker the fluid film will be. Until the fluid film is not thicker than the height of aspirates in the contact regime, some solid-to-solid contacts will also influence the motion. *Full Fluid Lubrication: (IV)* When the lubricant film is sufficiently thick, separation is complete and the load is fully supported by fluids. The viscous term dominates the friction phenomenon, the solid-to-solid contact is eliminated and the friction is 'well behaved'. The value of the friction force can be considered as proportional with the velocity.

From these domains results a highly nonlinear behavior of the friction force. Near zero velocities the friction force decreases in function of velocity and at higher velocities the viscous term will be dominant and the friction force increases with velocity. Moreover it also depends on the sign of velocity with an abrupt change when the velocity pass through zero.

For the moment no predictive model of the Striebeck effect is available. Several empirical models were introduced to explain the Striebeck phenomena: Tustin model, exponential in velocity $e^{-|v|/v_S}$, Gaussian model $e^{-(v/v_S)^2}$, Lorentzian model $1/(1+(v/v_{sw})^2)$. The constant value v_S is the Striebeck velocity which describes the shape of the Striebeck curve. To obtain a more precise friction model. These terms can be introduced in the velocity dependent friction models as follows:

Tustin model: $$F_f = (F_C + (F_S - F_C)e^{-|v|/v_S})sign(v) + F_V v \qquad (2)$$

Gauss model: $$F_f = (F_C + (F_S - F_C)e^{-(v/v_S)^2})sign(v) + F_V v \qquad (3)$$

Lorentz model: $$F_f = (F_C + (F_S - F_C)1/(1+(v/v_{sw})^2))sign(v) + F_V v \qquad (4)$$

2.2 Friction Model for Identification

The model for adaptive compensation, introduced in this study, was developed based on the Tustin model.

For the simplicity, only the positive velocity domain is considered, but same study can be made for the negative velocities. Assume that the mechanical system moves in $0 \ldots v_{max}$ velocity domain.

Consider a linear approximation for the exponential curve represented by two lines: d_{1+} which cross through the $(0,F_f(0))$ point and it is tangent to curve and d_{2+} which passes through the $(v_{max}, F_f(v\ v_{max})$ point and tangential to curve (see Figure 2). These two lines meet each other at the v_{sw} velocity. In the domain $0 \ldots v_{sw}$ the d_{1+} can be used for the linearization of the curve and d_{2+} is used in the domain v_{sw}

... v_{max}. The maximum approximation error occurs at the velocity v_{sw} for both linearizations.

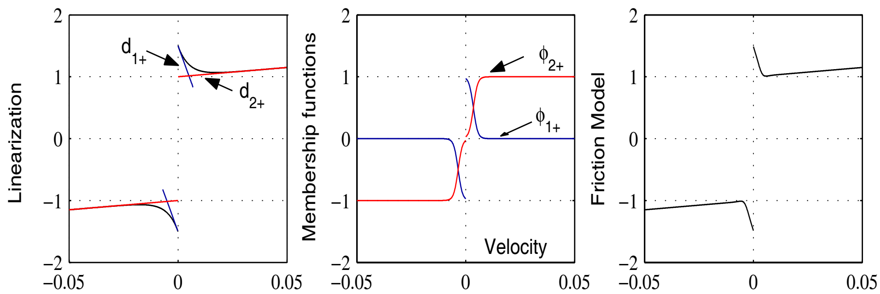

Figure 2
Linearization of the static friction model

If the positive part of the friction model (2) is considered ($v>0$), the obtained equations for the d_{1+} and d_{2+}, using Taylor expansion, are:

$$d_{1_+}: F_{L1f_+}(v) = F_S + \left.\frac{\partial F_f(v)}{\partial v}\right|_{v=0} v = F_S + (F_V - (F_S - F_C)/v_s)v \tag{5}$$

$$d_{2_+}: F_{L2f_+}(v) = F_f(v_{\max}) + \left.\frac{\partial F_f(v)}{\partial v}\right|_{v=v_{\max}} (v - v_{\max})$$
$$= F_f(v_{\max}) + (F_V - (F_S - F_C)/v_s)e^{-v_{\max}/v_S}(v - v_{\max}) \tag{6}$$

Thus the linearization of the exponential friction model with bounded error can be described by two lines in the 0 ... v_{max} velocity domain:

$$d_{1_+}: F_{L1f_+}(v) = a_{1+} + b_{1+}v,\ if\ 0 \le v \le v_{sw} \tag{7}$$

$$d_{2_+}: F_{L2f_+}(v) = a_{2+} + b_{2+}v,\ if\ v_{sw} \le v \le v_{\max} \tag{8}$$

with:

$$v_{sw} = \frac{a_1 - a_2}{b_2 - b_1} = \frac{-F_S + F_f(v_{\max}) + (F_V - (F_S - F_C)/v_s)e^{-v_{\max}/v_S}}{(F_V - (F_S - F_C)/v_s)(-1 + e^{-v_{\max}/v_S})} \tag{9}$$

Now consider two exponential membership functions parameterized in the following way:

$$\phi_{1_+}(v) = \frac{e^{-\beta(v - v_{sw+})}}{1 + e^{-\beta(v - v_{sw+})}} \quad \phi_{2_+}(v) = \frac{1}{1 + e^{-\beta(v - v_{sw+})}} \tag{10}$$

where β_+ is a large positive constant.

These types of membership functions are well known from the theory of neural and neuro-fuzzy systems. They were adapted for the linearized friction model to assure a smooth transition in the linearized model from on line (d_{1+}) to other (d_{2+}).

By applying the F_{L1f_+} from (7) on the membership function ϕ_{1+} from (10) and F_{L2f_+} on ϕ_{2+} a new model can be obtained that has the same behavior as the Tustin friction model. Moreover it is linearly parameterized if the parameters of the lines are considered. Thus for the positive velocity domain the friction model reads as:

$$F_{f+}(v) = a_{1+}\phi_{1_+}(v) + b_{1+}v\phi_{1_+}(v) + a_{2+}\phi_{2_+}(v) + b_{2+}v\phi_{2_+}(v) \tag{11}$$

With same train of thoughts a similar model can be determined for negative velocity domain. By combining the negative and positive velocity domains the obtained friction model reads as:

$$F_f = \underline{\theta}_F^T \underline{\xi}_F(v) \ldots where: \underline{\theta}_F = (a_{1+}\ b_{1+}\ a_{2+}\ b_{2+}\ a_{1-}\ b_{1-}\ a_{2-}\ b_{2-})^T \tag{12}$$

$$\underline{\xi}_F(v) = (\phi_{1_+}\mu(v)\quad v\phi_{1_+}\mu(v)\quad \phi_{2_+}\mu(v)\quad v\phi_{2_+}\mu(v)\quad \phi_{1\_}\mu(-v)\quad v\phi_{1\_}\mu(-v)\quad \phi_{2\_}\mu(-v)\quad v\phi_{2\_}\mu(-v))^T$$

3 Friction Measurement and Identification

3.1 Friction Measurement

For the friction force measurements it is assumed that the load is driven by a DC servo motor and the position and velocity is measured using an incremental encoder. The friction force can appear inside the motor, in the gearbox between the load and the motor and at the load side. The friction to identify is the sum of all these friction forces. As it was presented in the previous sections, the relationship between the friction behavior F_f and the velocity v is a mapping $F_f = F_f$ (v). The identification task in this is to obtain the parameters of the model (12) from a finite number (N) available measurements $(v_i, F_{fmi}),\ i=1..N$, where $F_{fmi} = F_{fi} + d_i$. The term d_i is the measurement error of the i'th measurement data.

The method is presented for the positive velocity regime. In order to solve the problem, a three step identification method was developed:

1 Measure the friction force using only velocity sensors.

2 Determine the parameters of the lines d_{1+} and d_{2+} given by (7) and (8).

3 Determine the parameters β_+ and v_{sw+} of the membership functions (10).

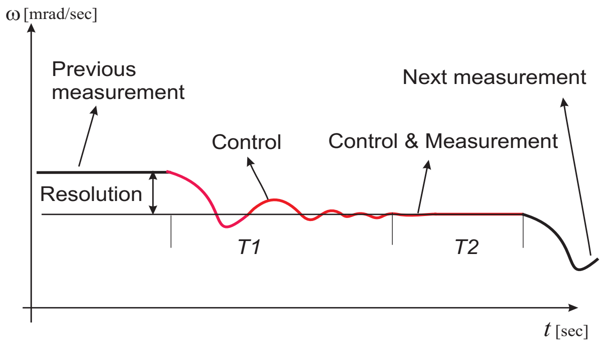

Figure 3
Velocity control for friction measurement

The dynamics of the positioning system reads as:

$$m\dot{v} = u - F_f(v) \tag{13}$$

with m mass of the load and u is the control input force.

It can be seen that if the velocity is kept constant, the friction force is proportional with control signal u, $F_f(v) = u$. Hence if the positioning system is stabilized to different angular velocities v_i, the value of the friction force will be proportional with the velocity.

The method needs high precision velocity control. It is known that the linear PI control algorithm assures only poor transients performances for velocity tracking but guarantees precise final tracking accuracy, if the reference velocity is kept constant. It suggests that for parameter identification it is enough to use standard PI algorithm for velocity control: $u = K_P((v_{ref} - v) + 1/T_i \int (v_{ref} - v)dt)$. The well tuned PI controller guarantees precise velocity stabilization.

The measurement algorithm can be summarized as follows (see Figure 3):

- *Stabilize the velocity to* v_{refi}

- *Wait a time period T1 to get rid of transients.*

- *After the transients, calculate the average of the speed (v) and the control signal (u) over a time period T2 to get rid of measurement noise.*

- *Save the measurement data (*v_i, u_i*).*

- *Repeat the sequence for the next velocity* v_{refi+1}

Note that during the data collection the closed loop control algorithm remains active.

For precise velocity stabilization, precise velocity measurements are needed. Denote the encoder resolution with N_E. This value is generally given in Pulses Per Turn (*PPT*). The velocity can be measured with the encoder by counting the encoder pulse over a period of time T (*pulse counting method*). The velocity measurement accuracy is: $\Delta v_C = 2\pi / N_E T [rad/\sec]$. However at low velocities higher accuracy can be obtained using the *pulse timing method:* count the number of pulses of a high frequency external timer over a single encoder pulse. In this case the accuracy of this measurement method is $\Delta v_T = Nv^2 / 2\pi f [rad/\sec]$ where f is the frequency of the timer. For the better accuracy at low speed regime the pulse timing method and at the high speed regime the pulse counting method should be used. The optimal switching between the methods is at the speed $v_{TC} = 2\pi / N_E \sqrt{f/T}\, [rad/\sec]$, where $\Delta v_T = \Delta v_C$. By combining these two methods the velocity can be measured using standard industrial encoders ($N_E \leq 5000\ PPT$) with high accuracy even at low speed regime.

3.2 Friction Identification

3.2.1 The Parameters of the Lines

The first line (d_{1+}), given by (7), characterizes the friction phenomena at low velocities, where the friction force has a downward behavior in function of velocity. At high speeds the friction increases almost linearly with the velocity, the second line (d_{2+}) should be fitted on this part of the Striebeck curve. Hence, let us consider two subgroup of measurement data: the first N_1 measurements at the decreasing part of the curve, and the final N_2 measurements where the friction force increases with velocity.

The parameters of d_{1+} and d_{2+} can be determined as a solution of the following optimization problems:

$$\min_{a_{1+}, b_{1+}} \sum_{i=1}^{N_1} (F_{fi}(v_i) - (a_{1+} v_i + b_{1+}))^2 \tag{14}$$

$$\min_{a_{2+}, b_{2+}} \sum_{i=N_2}^{N} (F_{fi}(v_i) - (a_{2+} v_i + b_{2+}))^2 \tag{15}$$

Applying standard optimization techniques such as the the *Least Squares (LS)* method, the friction parameters can easily be calculated.

3.2.2 The Parameters of the Membership Functions

The membership functions (10) depend exponentially on β_+ and v_{sw}. If the parameters of the lines are known, the value of v_{sw} can be calculated from the relation (9), hence only the parameter β_+ should be identified to obtain the precise expression of the membership functions. In the case of noisy measurements, it is desirable to obtain the model by minimizing the sum of quadratic deviations between the measurement points and the model to be fitted. To define the cost function, all the N measurements were used. The optimization problem is formulated as follows:

$$\min_{\beta_+,\omega_{sw+}} \sum_{i=0}^{N} (F_{fi}(v_i) - (a_{1+}\phi_{1_+}(v) + b_{1+}v\phi_{1_+}(v) + a_{2+}\phi_{2_+}(v) + b_{2+}v\phi_{2_+}(v))^2 \tag{16}$$

The parameters of the lines are known from the previous step of the identification. The membership functions cannot be written in a linearly parameterized form, thus the *LS* method is not applicable to determine its parameters. To find the parameters of the membership functions as a result of the optimization problem (16), Genetic Algorithm can be applied. Each chromosome represents the β_+ parameter and the algorithm minimizes cost function (16).

4 Experimental Results

4.1 Experimental Setup

The experimental setup consists of a permanent magnet *24V* DC servo motor with 38.2 [mNm/A] torque constant. The motor drives a metal disc with known inertia (J = 0.015 kgm^2) through a *1:66* gear reduction (*N=66*). Friction is introduced via a metal surface, which is held against the disc (see Figure 4.). The contact between the disc and the metal surface is lubricated with grease. The reaction torque generated by the friction component related to the motor side also can be written as a sum of three terms $F_f = F_{fR} + F_{fG} + F_{fL}/N$. F_{fR} denotes the friction component inside the motor, F_{fG} denotes the friction component inside the gearhead, F_{fL} is the friction component at the load side.

The friction measurement and control algorithm are implemented on a PIC18 type microcontroller with *40 MHz* clock frequency. The used C compiler for the implementation of the control algorithms allows floating point representation. The microcontroller is connected to an IBM-PC computer through RS232 serial port. The PC is used only for data monitoring and off-line data processing.

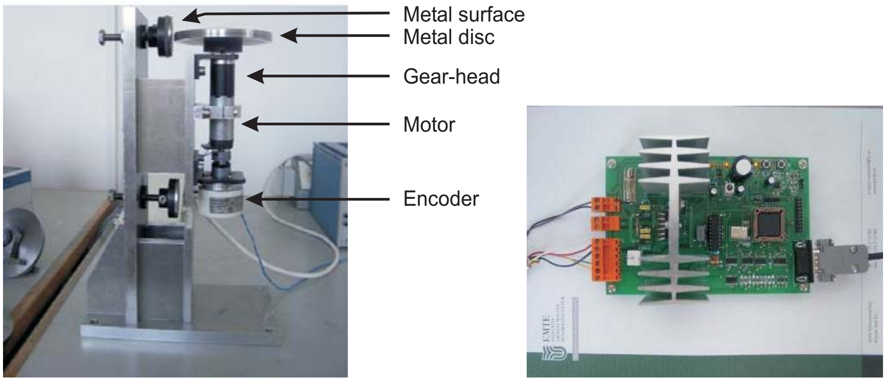

Figure 4
The experimental setup and the control circuit

The DC servo motor is driven by a H-bridge amplifier. The armature current is controlled by a high speed, analog current controller. The microcontroller is interfaced to the current servo amplifier through a 11 bit DAC. The command signal calculated by the control algorithm running on the controller represents the reference for the current controller. Hence the positioning system is controlled by a cascade control architecture.

The angular position and velocity of the mechanical system are measured using a *5000 PPT* two channel rotational encoder. The encoder is interfaced through a signal conditioner circuit to microcontroller which also determines the direction of rotation. The impulses of the encoder are counted using the embedded 16 bit timers of the controller. The pulse counting method uses the *Timer 0* block of the controller which has external clock input. The counting period is set to *5 msec*. The pulse timing method is implemented using the *Capture* block of the controller, which generates an interrupt when positive signal edge appears on its external input. The high frequency timer, necessary for the measurement in pulse timing mode is derived from the microcontroller clock frequency. The switching between the two methods are implemented in the velocity and position measurement software module.

4.2 Measurement Results

To obtain the low velocity friction characteristics, the friction force was measured in *0 ... 0.5 [rad/sec]* velocity domain (at the load side). The speed resolution was chosen 5 [mrad/sec]. Accordingly, totally $N=100$ measurements data were collected. A *PI* type control algorithm stabilizes the motor speed for each reference speed with $K_P=15$ proportional gain and $T_i=0.24$ *[sec]* integral time constant. The algorithm was implemented with *5 [msec]* sampling period. When the reference speed value is changed, for $T1=50$ sampling period no data was collected in order to get rid of transients, and after that for *T2=16* sampling period the average of the velocity values and control signal values were calculated to obtain one measurement point.

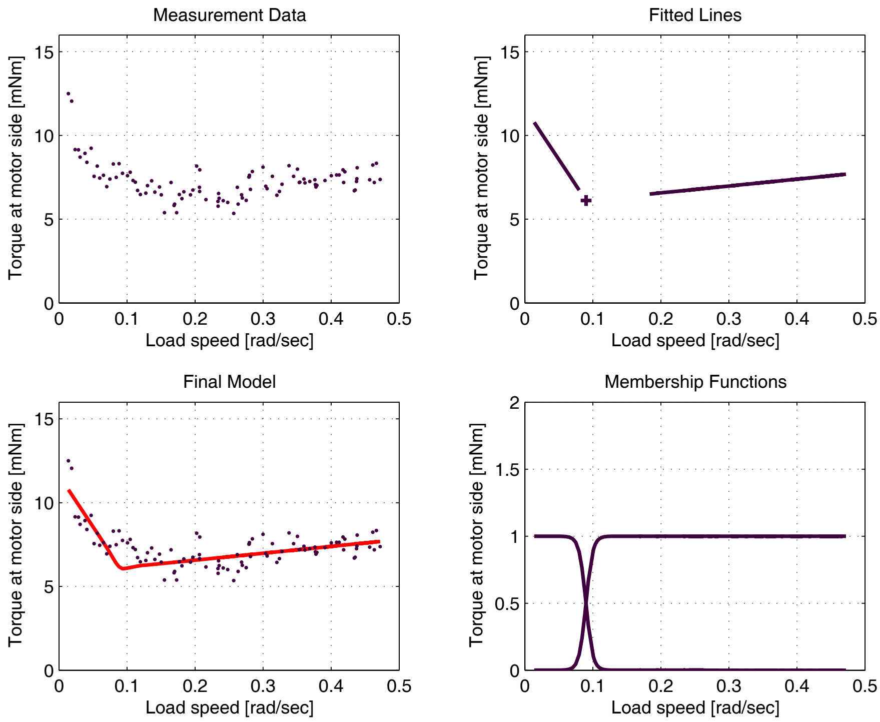

Figure 5

Identification results

The measurements clearly capture the increasing and decreasing part of the Striebeck curve. On the first *15* measurements at low velocities a line was fitted using *LS* method to obtain the parameters a_{1+} and b_{1+}. On the last *50* measurements another line was fitted to obtain the a_{2+} and b_{2+} parameters, which characterize the high velocity regime. The v_{sw+} parameter of the membership function was determined from the relation (9). The β_+ parameter was determined using Genetic Algorithm which minimized the cost function (16). For the identification all the *100* measurements data were used. For the Genetic Algorithm the following parameters were chosen: *80* number of individuals, *0.5* insertion rate and *0.01* mutation probability. With these parameters Genetic Algorithm showed good convergence during the identification and the value of the parameter β_+ was obtained at the *30*'th generation.

Note that the friction was determined at the load side and the velocity is the velocity of the load. It can be seen (Figure 5) that the obtained model fits well the measurement data.

The following parameter values obtained during the identification are presented below:

$a_{1+} = 11.6$ [mNm] $\quad b_{1+} = -61.2$ [mNmsec/rad]

$a_{2+} = 5.7$ [mNm] $\quad b_{2+} = 4$ [mNmsec/rad]

$v_{sw+} = 0.085$ [rad/sec] $\quad \beta_{+} = 167$

The obtained friction model can easily be introduced in position tracking control algorithms for efficient compensation of friction force.

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