Tracking Control of Mechatronic Systems based on Precise Friction Compensation

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Abstract: The paper presents a control algorithm for high precision position tracking of the DC motor driven mechatronic systems near Striebeck velocities. The algorithm is based on a linearized friction model, which is introduced in the control algorithm and its aim is to compensate the effect of nonlinear friction force. An identification method, to obtain the parameters of the linearized friction model is also presented. Experimental measurements are provided to show the performances of the proposed control algorithm.

Keywords: tracking control, friction modelling, friction compensation

1 Introduction

Many industrial applications require precise positioning of a mechanical system, namely moving an object in a given position in space with a given orientation. Some common applications are material manipulation with industrial robots or cranes, positioning in Hard Disk Drives or optical drives. Other applications require a controlled motion in space along a predefined path, for example welding, spray painting with robots, path tracking of unmanned vehicles or guided missiles (trajectory tracking).

These tasks are commonly solved with feedback control. The aim of the control algorithm is to calculate the command signal for electric motors or other type of actuators which drive the mechanical system in order to obtain zero or acceptably small difference between the real and desired position.

Mechatronics deals with the integrated design of a mechanical system and its embedded electronic control system. In the perspective of the control algorithm it means that in order to obtain precise positioning, the model of the controlled mechanical system should also be taken into account in the control algorithm.

Friction is a nonlinear phenomenon which is universally present in the motion of bodies in contact. In servo controlled machines friction has an impact in all regimes of operation. In high precision positioning systems it is inevitable to know the value of the friction force to assure good control characteristics and to avoid some undesired effects such as limit cycle and steady state error, tracking lags.

The nonlinear and dynamic behavior of friction is accentuated near zero velocities. Many practical applications require precise motion control in low velocity regime. As examples can be mentioned the space telescopes that should track the motion of a star [1] and positioning applications, when the start point and the end point are near to each other.

Accordingly, in high precision position control systems the friction force should be taken into consideration at the formulation of the control law. To describe the friction phenomena, in [2] 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 [3], 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). To measure the frictional parameters, friction identification and measurement methods were also discussed in many studies. In [4] 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. Neural network based identification methods are also popular to capture the frictional behavior. In [5] Support Vector Machines were proposed for friction modelling and identification. The advantage of this approach is that it can identify nonlinearities from sparse training data. The compensation of the frictional effects from positioning systems is also discussed in many papers. In the paper [6] a nonlinear observer is developed to compensate the Coulomb friction. In the works of Laura et al. the extended Kalman filter technique is proposed for friction estimation [7].

In this paper a model based compensation method is presented. The rest of the paper is organised as follows: Section 2 is divided in three subsections. The first subsection presents the friction model, which is used in the control algorithm. Afterward, the second subsection presents a parameter identification for the introduced model. The third subsection in Section 2 presents the proposed tracking control algorithm. In Section 3 experimental measurements are given. The final Section sums up the conclusions of this study.

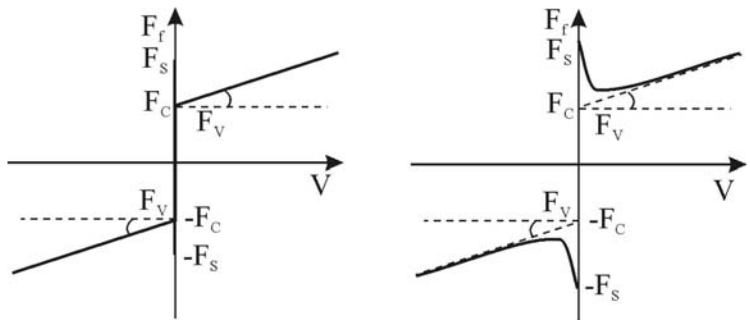

Figure 1
Static+kinetic+ viscous and Striebeck friction

2 Friction Modelling, Identification and Compensation

2.1 Friction Modelling near Striebeck Velocities

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 term ($F_C sign(v)$), which depends only on the sign of velocity, the viscous component ($F_V v$), which is proportional with the velocity and the static term (F_S), which represents the force necessary to initiate motion from rest and in most of the cases its value is grater than the Coulomb friction: (see Figure 1).

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

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.

For the moment no predictive model of the Striebeck effect is available. Several empirical models were introduced to explain the Striebeck phenomena, such as the Tustin model: (see Figure 1).

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

The model introduced in this paper is based on Tutin friction model and on its development, the following aspects were taken into consideration:

- allows different parameter sets for positive and negative velocity regime
- easily identifiable parameters
- the model clearly separates the high and low velocity regimes
- can easily be implemented and introduced in real time control algorithms

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 $(v_{max}, F_f(v_{max}))$ 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 $v_{sw} \ldots v_{max}$. The maximum approximation error occurs at the velocity v_{sw} for both linearizations.

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:

$$d_{1_+} : F_{L1f_+}(v) = F_S + \left.\frac{\partial F_f(v)}{\partial v}\right|_{v=0} v = F_S + (F_V - (F_S - F_C)/v_s)v \tag{3}$$

$$\begin{aligned} d_{2_+} : F_{L2f_+}(v) &= F_f(v_{\max}) + \left.\frac{\partial F_f(v)}{\partial v}\right|_{v=v_{\max}} (v - v_{\max}) \\ &= F_f(v_{\max}) + (F_V - (F_S - F_C)/v_s)e^{-v_{\max}/v_S}(v - v_{\max}) \end{aligned} \tag{4}$$

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

$$d_{1_+} : F_{L1f_+}(v) = a_{1+} + b_{1+}v, \text{ } if \text{ } 0 \le v \le v_{sw} \tag{5}$$

$$d_{2_+} : F_{L2f_+}(v) = a_{2+} + b_{2+}v, \text{ } if \text{ } v_{sw} \le v \le v_{\max} \tag{6}$$

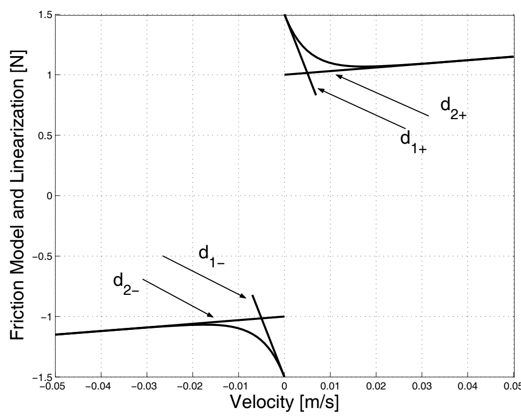

Figure 2
Linearization of Striebeck friction

with:

$$v_{sw} = \frac{a_1 - a_2}{b_2 - b_1} = \frac{-F_S + F_f(v_{\max}) + (F_V - (F_S - F_C)/v_s)e^{-v_{\max}/v_S}}{(F_V - (F_S - F_C)/v_s)(-1 + e^{-v_{\max}/v_S})} \tag{7}$$

Same study can be made for negative velocities. Based on linearization, the friction can be modelled as follows:

$$F_f(v) = \begin{cases} a_{1+} + b_{1+}v, \; if \;\; 0 \le v \le v_{sw+} \\ a_{2+} + b_{2+}v, \; if \;\; v_{sw+} \le v \le v_{v\max} \\ a_{1-} + b_{1-}v, \; if \;\; v_{sw-} \le v \le 0 \\ a_{2-} + b_{2-}v, \; if \;\; v_{v\max} \le v \le v_{sw-} \end{cases} \tag{8}$$

It can be seen that the model is linearly parameterized and it can be implemented with low computational cost.

2.2 Friction Measurement and Parameter Identification

2.2.1 Friction Measurement

For the friction force measurements it is assumed that the load is driven by a servo motor and the torque developed by the motor is proportional with the command signal. 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 subsection, 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 (8) 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 on the i'th measurement data.

The method is presented for positive velocity regime.

The dynamics of the positioning system reads as:

$$\begin{aligned} \dot{x} &= v \\ m\dot{v} &= u - F_f(v) \end{aligned} \tag{9}$$

with m mass of the load and u is the control input force, x denotes the position.

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 command signal.

The method needs high precision velocity control. It is known that the linear PI control algorithm assures only poor transient 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 control.

The measurement algorithm can be summarized as follows:

- 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 velocity control algorithm remains active.

2.2.2 The Parameters of the Lines

The first line (d_{1+}), given by (5), 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+}), given by (6), 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:

$$\min_{a_{1+},b_{1+}} \sum_{i=1}^{N_1} (F_{fi}(v_i) - (a_{1+}v_i + b_{1+}))^2 \qquad \min_{a_{2+},b_{2+}} \sum_{i=N_2}^{N} (F_{fi}(v_i) - (a_{2+}v_i + b_{2+}))^2 \tag{10}$$

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

2.3 Position Tracking Combined with Friction Compensation

The tracking control problem for the system (9) can be formulated as follows: given a reference position trajectory $x_d=x_d(t)$, a twice differentiable function of time, determine the command signal u which guarantees, that $x(t)-x_d(t) \rightarrow 0$, as $t \rightarrow \infty$.

In order to solve the tracking problem, define the following tracking error metric, which combines the time dependent position and velocity errors: ($e_x = x - x_d, e_v = v - v_d$):

$$S = e_v + \lambda e_x \tag{11}$$

where $\lambda > 1$ is a constant parameter.

Based on the plant model (9), the tracking error dynamics reads as:

$$\dot{S} = (u - F_f(v))/m - \ddot{x}_d + \lambda e_v \tag{12}$$

Consider the control law as:

$$u = -m(-\ddot{x}_d + \lambda e_v + K_S S) + \hat{F}_f(v), \quad K_S > 0 \tag{13}$$

Note that if we have $\hat{F}_f(v) - F_f(v) \approx 0$ yields $\dot{S} + K_S S = 0$, the combined position and velocity error converges to zero.

Hence the tracking problem can be solved, if the control algorithm contains a precise friction compensator term, which can 'cancel' the effect of frictional force on the system dynamics. $\hat{F}_f(v)$ can be modelled according to the relation (8) where the model parameters can be obtained as it was described in subsection 2.2.

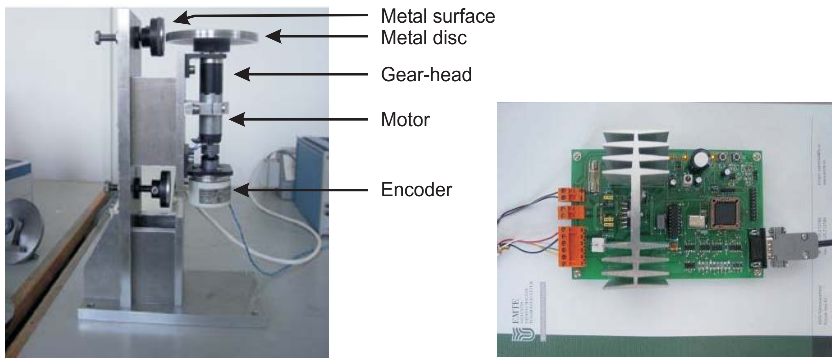

Figure 3
The experimental setup and the control circuit

3 Experiemental Results

3.1 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 3). 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 $F_f = F_{fR} + F_{fG} + F_{fL}/N$, where 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.

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.

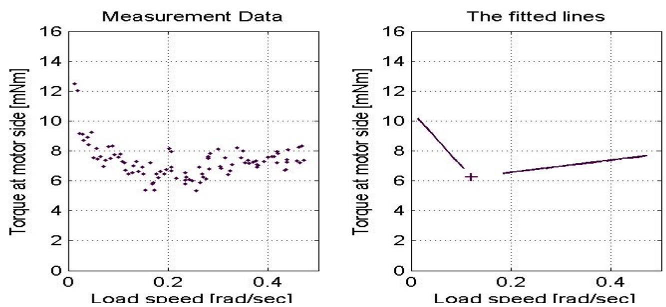

Figure 4

Friction measurement results and the fitted lines

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.

3.2 Measurement and Identification 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.

The measurements clearly capture the increasing and decreasing part of the Striebeck curve (see Figure 4). On the first *15* measurements at low velocities a line was fitted using *LS* method, which optimized the cost function (10), 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 was determined from the relation (7).

Note that the friction was determined at the load side and the velocity is the velocity of the load. It can be seen (Figure 4) that the obtained model fits well the measurement data.

For the positive velocity regime, the following parameter values obtained during the identification are presented below:

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

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

$v_{sw+} = 0.085$ *[rad/sec]*

3.3 Compensation Results

In order to test the proposed friction compensation method, the control law (13) was implemented on the microcontroller, with the following parameters: $K_S=1$, $\lambda=10$. Because the motion is rotational, the mass m is replaced by the inertia of system (J). The desired track contains has acceleration, constant speed, and deceleration regimes in both positive and negative velocity regimes.

Two experiments were carried out. In the first experiment, the friction compensator term ($\hat{F}_f(v)$) was neglected from the control law. In the second experiment, the friction compensator term was introduced in the control law according to the equation (8). The experimental results are presented in Figures 5 and 6. For the numerical evaluation, the following error sum is considered. $E_S = \frac{1}{N}\sum_{i=1}^{N}|S_i|$, where N represents the number of measurements ($N=1000$). Without friction compensation, we obtained $E_S=0.3558$. With friction compensation we obtained E_S=0.0656. Accordingly, the proposed friction compensation method outperforms the classical linear control algorithms, in which the friction is not taken into consideration.

Conclusions

A friction identification and method was proposed for mechatronic systems, which operates at low velocity regimes near Striebeck velocities. The introduced control algorithm is developed for position tracking tasks and it is based on a linearized friction model, which can easily be implemented on simple industrial controller and its parameters can be identified with standard LS techniques. Experimental results clearly shows the advantages of the proposed tracking control algorithm.

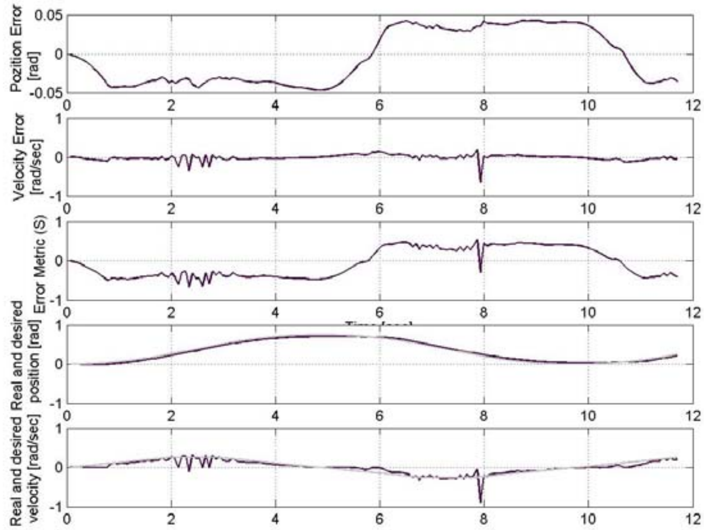

Figure 5
Tracking without friction compensation

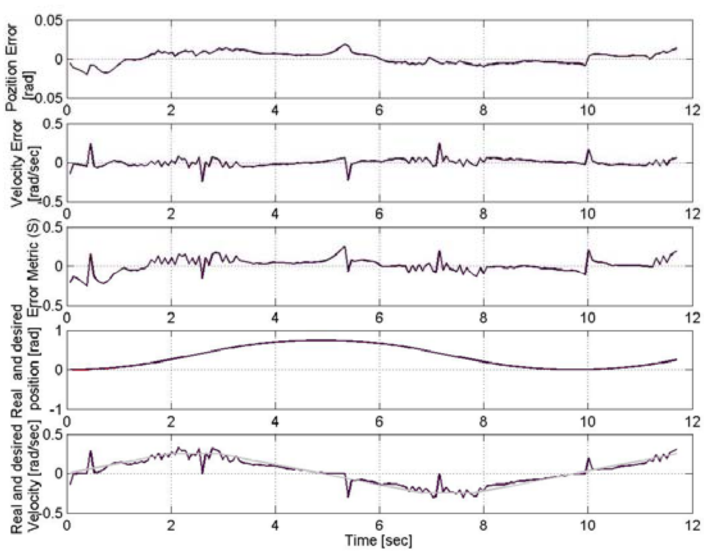

Figure 6
Tracking with friction compensation

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