IMPROVING THE TRACKING PERFORMANCE OF MECHANICAL SYSTEMS BY ADAPTIVE EXTENDED FRICTION COMPENSATION

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Abstract. We discuss a tracking controller and show with simulation and experimental results that extended friction models can be successfully incorporated in computed-torque-like adaptive control schemes. The friction model used includes Coulomb, viscous and periodic friction with direction dependent parameters. To get small tracking errors, adaptation of the friction model parameters is necessary. The tracking performance is an order of magnitude better as with PD control. The robustness of the scheme for parameter inaccuracies is sufficient, but the adaptation gains are limited due to stability problems, caused by unmodeled dynamics.

Key Words. Nonlinear control systems; adaptive control; robots; tracking systems; robustness; friction compensation

1. INTRODUCTION

The presence of friction in mechanical systems where material parts move relative to each other and contact is necessary due to a guiding or bearing function of the parts, is unavoidable. It is not always possible to eliminate friction by using advanced tribological measures. When traditional techniques to eliminate backlash are used, the problem of friction becomes even more pronounced. In general, friction is a limiting factor for the tracking performance of mechanical control systems.

There are several ways to overcome the effects of friction

- the use of high gain feedback, but this has disadvantages, such as large input signals and no robust performance due to excitation of high frequency unmodeled dynamics,
- the use of additional dither signals, that prevent the system from stiction,
- compensation of friction by the controller; the accuracy of the compensation largely depends on the correctness of the structure of the friction model used for the compensation and on an accurate knowledge of the friction model parameters.

We focus on friction compensation to overcome the disadvantages of friction, but to use it effectively some problems have to be addressed.

The main problem is the formulation of accurate friction models. These models are difficult to obtain, due to the complexity of friction phenomena, and even the physical causes of friction are not well understood (Haessig and Friedland, 1991). One approach is to perform some measurements on the system in question and deduce an indication of the structure of the equations describing the effects of friction. Some experiments in this direction are performed (Armstrong, 1988), but the conclusions with respect to the structure of the friction model are closely related to the system investigated and can hardly be generalized.

Another approach, chosen in this work, is to use an elaborate friction model, and to adapt the parameters of the model.

When some terms in the model are not significant, the corresponding parameters will be small. After an initial period of use, the structure of the friction model can be simplified by deleting terms that are related with small parameters (i.e., insignificant terms) or have parameters of equal value, e.g., for direction dependent parameters. It is necessary to use a sufficiently rich model to encompass all effects that can appear and are related to friction. Yet, the number of parameters should not be too large, to avoid problems with the adaptation (overparametrization) and to avoid modeling of disturbances that are not related to friction.

Adaptive friction compensation has been used by (Canudas et al., 1987; Canudas de Wit, 1990; Canudas de Wit et al., 1991; Niemeyer and Slotine, 1988; Niemeyer and Slotine, 1991), but they use relatively simple friction models.

Our main contribution is the proof of concept for the viability of the use of a more elaborate friction model then generally used. A discussion of the robustness of the parameter estimates and of the obtainable tracking error, compared with a PD controller and with simple friction compensation, is also included.

We give this proof in the following order. First, we discuss the experimental system, the simulation model, the friction model and the adaptive control scheme used. Then, in Section 3, the setup for the numerical and real world experiments is given. Sections 4 and 5 present the simulation and experimental results. The discussion of the results follows in Section 6. Finally, Section 7 shows the conclusions and recommendations.

2. SYSTEM, MODELS AND CONTROLLER

We present the experimental system, a simple model of this system, and the friction model used, including some background why we choose this type of model. We close this section with the presentation of the control scheme used in the simulations and experiments to control the model and experimental system, respectively.
2.1 Experimental System

The system used for the experiments is a two degrees-of-freedom manipulator, moving in the horizontal plane, with two prismatic joints, a so called TT-robot or, emphasizing the Cartesian coordinates, an XY-table. For a schematic drawing of the XY-table, see Fig. 1.

![Fig. 1. Schematic drawing of XY-table](image)

The main characteristics of the system are:
- working area 1 x 1 [m],
- two permanent magnet DC motors,
- two current amplifiers,
- optical encoders for the motor positions,
- microcomputer based control.

2.2 Design Model

The equations for a simple model of the XY-table of Fig. 1 are

\[
\begin{align*}
\theta_1 x + g_x(x, x) &= f_x \\
\theta_2 y + g_y(y, y) &= f_y
\end{align*}
\]

where \( x \) and \( y \) are the two prismatic degrees-of-freedom, \( f_x \) and \( f_y \) the control forces in \( x \) and \( y \) direction, \( \theta_1 \) and \( \theta_2 \) are the inertia parameters in \( x \) and \( y \) direction, and \( g_x \) and \( g_y \) are disturbance forces due to Coulomb, viscous and other types of friction or due to other state dependent disturbances. Coriolis and centripetal forces are neglected, because there is almost no coupling between movements in \( x \) and \( y \) direction. Gravitational forces are absent because the manipulator moves in the horizontal plane. The absence of these forces makes the XY-table an ideal object for the study of the merits of friction compensation.

2.3 Friction Model

Earlier experiments performed to assess the robustness of adaptive control schemes, see (de Jager, 1992), are used to guide the selection of a favorable friction model. In these experiments it appeared that the tracking error in \( y \)-direction is larger than the error in \( x \)-direction. Also, it can be deduced from the characteristics of the tracking error in \( y \)-direction, that there is an harmonic disturbance force, which is the cause of the lower tracking accuracy for \( x \)-direction. This difference in accuracy is completely in contradiction with common expectations. The transmission of the torque from the motor to the end-effector in \( y \)-direction is much simpler than in \( x \)-direction and therefore the model in \( y \)-direction could be assumed to be much more accurate than in \( x \)-direction. However, this is not true.

The reason for this discrepancy between reality and expectation can be deduced from the harmonic nature of the disturbance force. The period of the force fluctuation is equal to the time needed for one complete revolution of the \( y \)-motor, and so of its shaft, bearings and belt wheel. Therefore, it seems logical to assume that the disturbance force stems from some imperfections and friction in the shaft and bearings. Another possible explanation could be the presence of imperfections in the magnetic and electrical fields in the motor due to, e.g., a lack of rotational uniformness or symmetry. In this case we are not modeling friction, but state dependent disturbances. The use of a reduction in the transmission for the \( x \)-motor alleviates these effects for the \( x \)-direction.

A solution for the periodic friction would be to eliminate it by replacing the shaft and bearings, but, incidentally, it provides a source of model error, which does not endanger the stability, but significantly reduces the performance. None of the control schemes used can cope directly with this type of disturbance, except by using larger gains in the PD part of the schemes, but those large gains do endanger the stability and can therefore not be applied in practice.

Another solution is canceling the disturbance force by compensation. This can be regarded as an extension of standard Coulomb friction compensation, it just requires an extended friction model.

The appearance of periodic or position dependent friction components has been observed previously and is reported by, e.g., (Armstrong, 1988), but for their system the harmonic friction component was small, in the order of 7% of the Coulomb friction. In our case this is not true, so we should explicitly consider periodic friction.

When the compensation is based on the angular position \( \alpha_0 \) of the shaft, only the amplitude \( b_0 \) and phase \( \phi_0 \) of the sinusoidal compensation force has to be determined. When adaptive controllers are used, one could try to use adaptive friction compensation (Canudas et al., 1987; Canudas de Wit et al., 1991; Niemeyer and Slotine, 1991) by estimating amplitude and phase. However, when the compensating force is of the form

\[
f_p = b_0 \sin(\alpha_0 q + \phi_0)
\]

the parameter \( \phi_0 \) does not appear linear in the control force, which is required for the adaptation part of the controller we want to use. The angular frequency \( \omega_0 \), is assumed to be known to avoid this problem. Fiddling with the phase to get a small error is possible, but tedious and should be repeated for each arrangement of belt wheels and belt, and must be repeated every time the connection between motor, belt wheel and end effector is changed, e.g., a belt change or even re-attachment of the belt. So, a much better solution is to incorporate the adaptation of the phase in the control scheme. For this purpose, write the previous expression for \( f_p \) as

\[
f_p = a_{p_1} \sin(\alpha_0 q + \phi_0) + a_{p_2} \cos(\alpha_0 q)
\]

and now the two amplitudes \( a_{p_1} = b_0 \cos(\phi_0) \) and \( a_{p_2} = b_0 \sin(\phi_0) \) and no phase has to be adapted. Both parameters appear linear in the control force. A disadvantage of this method is that both sine and cosine have to be computed, resulting in a slightly longer computation time.

So, including Coulomb, viscous and periodic friction in the model, we obtain for the friction force \( g^* \), for \( q \geq 0 \), and \( g^- \), for \( q < 0 \)

\[
\begin{align*}
g^*(q, \dot{q}) &= a_{\alpha_0}' \text{sgn}(q) + a_{\alpha_0} \dot{q} + a_{\alpha_0}'' \sin(\alpha_0 q) + a_{\alpha_0}''' \cos(\alpha_0 q) \\
g^-(q, \dot{q}) &= a_{\alpha_0}' \text{sgn}(q) + a_{\alpha_0} \dot{q} + a_{\alpha_0}'' \sin(\alpha_0 q) + a_{\alpha_0}''' \cos(\alpha_0 q)
\end{align*}
\]
where we assume that all parameters in the friction model are direction dependent.

### 2.4 Controller

Before giving a short description of the control scheme we introduce a general model of a mechanical system

$$M(q, \dot{q})\ddot{q} + C(q, \dot{q}, \theta)\dot{q} + g(q, \dot{q}, \theta) = f$$  \hspace{1cm} (3)

used in this scheme. Here, $M(q, \dot{q})$ is the $n \times n$ positive definite inertia matrix, with model parameters $\theta$, $C(q, \dot{q}, \theta)\dot{q}$ is the $n$ vector of Coriolis and centripetal forces, $g(q, \dot{q}, \theta)$ the $n$ vector of gravitational forces, Coulomb and viscous friction and other state dependent forces, $f$ the $n$ vector of generalized control forces (forces or torques). In this model each of the $n$ degrees-of-freedom has its own motor. Here, we neglect the dynamics of the motors and amplifiers, backlash, and flexibility of the joints and links.

The control scheme we will use is proposed by Slotine and Li (Slotine and Li, 1988). See also the comments in (Spong et al., 199). This scheme has an approximate feedforward component, based on an estimate of the manipulator dynamics and using a virtual reference trajectory, and a PD component. The generalized control force is just the sum of these components

$$f = \hat{M}(q)\ddot{q} + \hat{C}(q, \dot{q})\dot{q} + \hat{g}(q, \dot{q}) + K_s$$  \hspace{1cm} (4)

where $\hat{M} = M(q, \hat{\theta})$, $\hat{C} = C(q, \dot{q}, \hat{\theta})$ and $\hat{g} = g(q, \dot{q}, \hat{\theta})$ are the same as the corresponding terms in (3), with $\hat{\theta}$ an estimate of the model parameters $\theta$. $\hat{\theta}$, $\hat{q}_d = \hat{q}_d + A\hat{\theta}$ a virtual reference trajectory, $s = \hat{q} + A\hat{\theta}$ measure of tracking accuracy, $\dot{s} = \dot{q} - q$, $\dot{q}_d = q - \hat{q}_d$ the tracking error, and $\hat{q}_d(\hat{\theta}), \hat{q}_d(\hat{\theta}), \hat{q}_d(\hat{\theta})$ the desired trajectory. Adaptation of the model parameters used in $\hat{M}$, $\hat{C}$ and $\hat{g}$ is based on the fact that, with an appropriate choice of parameters, the generalized control force (4) is linear in the parameters $\hat{\theta}$ and can be expressed as

$$f = Y(q, \dot{q}, \dot{q}_d, \dot{q}_d)\hat{\theta} + K_s,$$  \hspace{1cm} (5)

Then the adaptation proceeds according to

$$\dot{\hat{\theta}} = \Gamma^{-1}Y(q, \dot{q}, \dot{q}_d, \dot{q}_d)s.$$  \hspace{1cm} (6)

When the initial estimates are chosen as $\hat{\theta}(0) = 0$ and the adaptation gain $\Gamma^{-1} = 0$, the controller of Slotine and Li becomes a PD controller acting on the tracking error $\dot{s}$. When the initial estimates are chosen as $\hat{\theta}(0) = \theta$ and the adaptation gain is 0, we obtain a computed-torque-like control scheme without adaptation.

We now apply this control scheme to the model of the XY-table (1). To rewrite (1) as (3), we define the following quantities

$$q = \begin{bmatrix} x \\ y \end{bmatrix},$$

$$M(q, \theta) = \begin{bmatrix} \theta_1 & 0 \\ 0 & \theta_2 \end{bmatrix},$$

$$C(q, \dot{q}, \theta) = \begin{bmatrix} 0 \\ 0 \end{bmatrix},$$

$$g(q, \dot{q}, \theta) = \begin{bmatrix} g_x(q, \dot{q}, \theta) \\ g_y(q, \dot{q}, \theta) \end{bmatrix},$$

$$f = \begin{bmatrix} f_x \\ f_y \end{bmatrix}.$$  \hspace{1cm} (7)

Here, the parameters $\theta_i$, $i > 2$, correspond in an obvious way to the parameters $d^c, d^c$ in (2).

This results in expressions for $Y$ in (5) as follows

$$Y = \begin{bmatrix} x, 0 \sin x \sin \omega_p x, \cos \omega_p x & 0 & 0 & 0 \\ y, 0 \sin y \sin \omega_p y, \cos \omega_p y & 0 & 0 & 0 \end{bmatrix}$$

for positive velocities, used for adaptation of the $\theta^c$ parameters and an equivalent expression for negative velocities to adapt the $\theta^c$ parameters. The parameters $\theta^c$ are ordered as $\theta^c = \theta_1, \theta_2, d^c, d^c$ and $\theta^c = \theta_1, \theta_2, d^c, d^c$ but, of course, the velocities for $x$ and $y$ direction change sign independent of each other.

### 3. SIMULATION AND EXPERIMENTAL SETUP

In this section specific information of the setup for the simulations and experiments is given, that enables the reader to verify the results given in the next sections.

The control task is to follow a periodic trajectory in the XY-plane, with position control in both coordinate directions. The desired trajectory in Cartesian end-effector space is

$$\begin{bmatrix} x_c(t) \\ y_c(t) \end{bmatrix} = \begin{bmatrix} a - R_y \cos \psi_o \\ b - R_x \cos(\psi_o + \psi_d) \end{bmatrix}$$

where $R_x = 0.2$ [m] is the "radius" of the trajectory, $\psi_d = a d^c$, with $a = \frac{\pi}{3}$ [rad/s], is the desired angular position, and $a = 0.8$ [m], $b = 0.8$ [m] specify the center of the working area of the manipulator, see Fig. 2. The constant angle $\psi_o$ is used to select the trajectory. We use $\psi_o = \frac{\pi}{2}$ and then the trajectory is a circle, if $\psi_o$ has another value the circle is deformed to an ellipse or even a straight (diagonal) line.

![Fig. 2. Desired trajectory](image-url)

The periodic nature of the task makes it easy to compute accurate and repeatable tracking error statistics, without influence of initial transients.

The continuous time adaptive controller is implemented in discrete time without a modification that compensates for the discrete implementation. The Euler method is used for the integration of the adaptation differential equation. Because only the position is measured (by code wheels) we estimate the velocity. We also estimate the position to diminish the effects of quantization. The position and velocity are estimated by a Kalman filter, to compensate for the time delay incurred by the controller computations.

The simulation model of the XY-table is almost implemented as a plug-in-replacement for the experimental system. Controllers developed for the simulation model can therefore directly be used in the control system of the XY-table, without the need for an additional translation step between different software implementations, e.g., scaling of measurements.
The design of the control parameters $K_c$ and $A$ is performed by choosing a favorable dynamics of the tracking error, characterized by the undamped characteristic frequency $f_\text{c}$ and damping coefficient $\beta_\text{c}$ of a second order system. The goal was to get a small tracking error without exciting high frequency dynamics that could endanger stability. The selection of $f_\text{c}$ was guided by the rule given in (Niemeyer and Slotine, 1991), but the gains had to be detuned to avoid stability problems.

For the nominal parameter values used in the model of the XY-table and used for the controller design, see Table 1. Parameter $a_\text{c}$ is the amplitude of a band limited pseudo white noise disturbance force used to model torque ripple and other random disturbances.

### Table 1 Nominal parameters of the simulation model and controller

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value $x$</th>
<th>Value $y$</th>
<th>unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\theta_1$, $\theta_2$</td>
<td>46.5</td>
<td>4.3</td>
<td>kg</td>
</tr>
<tr>
<td>$a_\text{c}^2$ = $a_\text{c}^2$</td>
<td>45.0</td>
<td>12.5</td>
<td>N</td>
</tr>
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<td>$a_\text{c}^2$ = $a_\text{c}^2$</td>
<td>6.0</td>
<td>10.0</td>
<td>N m$^{-1}$</td>
</tr>
<tr>
<td>$b_\text{c}^2$ = $b_\text{c}^2$</td>
<td>12.5</td>
<td>3.5</td>
<td>N</td>
</tr>
<tr>
<td>$\omega_\text{c}$</td>
<td>$1/9.7$</td>
<td>$1/10.5$</td>
<td>rad mm$^{-1}$</td>
</tr>
<tr>
<td>$\phi_\text{c}/\phi_\text{c}$</td>
<td>-815</td>
<td>-790</td>
<td>mm</td>
</tr>
<tr>
<td>$\phi_\text{c}/\phi_\text{c}$</td>
<td>-835</td>
<td>-820</td>
<td>mm</td>
</tr>
<tr>
<td>$a_\text{c}$</td>
<td>6.25</td>
<td>2.5</td>
<td>N</td>
</tr>
<tr>
<td>$f_\text{c}$</td>
<td>4.0</td>
<td>4.0</td>
<td>Hz</td>
</tr>
<tr>
<td>$\beta_\text{c}$</td>
<td>0.7</td>
<td>0.7</td>
<td>-</td>
</tr>
</tbody>
</table>

### 4. SIMULATION RESULTS

An overview of the simulation results for extended friction compensation is given. Five sets of results are presented, all for the second of two cycles of 3.5 [s] duration each. The results can be divided in two groups. Consideration of Figs. 3-5 indicates the effect of using more elaborate friction models. Figures 5-7 give an opportunity to assess the effect of using adaptation of parameters instead of fixed parameters in the scheme of Slotine and Li. We now present the five sets of results in more detail, starting with an assessment of the effects of extended friction compensation.

First, the results without extended friction compensation in $y$-direction are shown, where only the standard Coulomb friction is present in the computed torque part. See Fig. 3. Results without and with adaptation of the parameters are shown, both starting with the nominal parameters. The tracking error is mainly due to the lack of viscous friction compensation. The tracking error is reduced by the adaptation, i.e., the inertia and Coulomb friction parameters are given values, that change in time, to compensate somehow the effects of the viscous and the periodic friction.

Second, the results without periodic friction compensation in $y$-direction are shown, where only Coulomb and viscous friction are compensated. Both the results without and with adaptation of the parameters are shown, both starting with the nominal parameters. The tracking error is smaller by a factor of 2, due to the compensation of the viscous friction. Again, the use of adaptation can partly compensate for the unmodeled periodic friction.

Third, the results with extended friction compensation in $y$-direction, including Coulomb, viscous and periodic friction compensation. The almost ideal tracking error is given in Fig. 5. The results without and with adaptation of the parameters are presented, both starting with the nominal parameters. The remaining tracking error is almost completely caused by the torque ripple. When the torque ripple is absent the error is much smaller, but not equal to 0 due to:

- the quantization error in the position measurement,
- the prediction error in position and velocity of the one step ahead Kalman filter,
- inexact cancellation of the Coulomb friction, because the compensation can detect the instance of a change of sign of the velocity with an accuracy of 1 sample only, due to the discrete time implementation of the controller.

Comparison with the previous figure shows that the addition of periodic friction compensation results in a small, but noticeable, improvement in the performance. In relative terms, it is again a factor of 2. With adaptation the tracking error is only slightly smaller then without, which means that the parameter adaptation somehow cancels the effects of the 3 causes for the remaining tracking error mentioned above, although the first two causes are mainly of a random nature. Further improvement is hardly possible, due to the lack of structure in the pseudo white noise signal used to model the torque ripple.

To show the influence of the initial parameters estimates and the rate of convergence of the adapted parameters, or better: the rate of convergence of the tracking error due to the adaptation of the parameters, the results starting from an initial parameter estimate of 80% and 0% of all nominal parameters
are presented in Figs. 6 and 7. So, all parameters, including the inertia and Coulomb friction parameters, are assumed to be approximately known or even completely unknown.

Figure 6 shows the advantage of using adaptation. The tracking error is reduced by a factor of 2. The adaptation is fast, so an error comparable with the result given in Fig. 5 for exactly known parameters can be obtained after approximately 1 control cycle.

Last, the results starting from a zero initial estimate for all parameters show clearly the advantage of using a computed-torque-like control scheme. With a zero initial estimate for the parameters the control scheme of Slotine and Li degenerates to a pure PD feedback of the tracking error. The tracking error is an order of magnitude larger than the error obtainable with more advanced control schemes. Figure 7 also clearly shows that the parameters obtain values that reduce the tracking error significantly after 2 cycles when adaptation is used. In the long run, the error will be as small as in Fig. 5.

5. EXPERIMENTAL RESULTS

We present five experimental results, also for the second of two cycles of 3.5 [s] duration each, except for the last result. The results for PD feedback are obtained by using the controller with zero initial values of the parameters and no adaptation. From all other experiments only results with adaptation are shown and the initial values of the parameters are assumed to be completely unknown. Results with fixed parameters are not obtained, because, in contrast to the simulation, exact model parameters are not defined. The results are presented in order of increasing tracking performance.

A reference result for the tracking error in y-direction, presented in Fig. 8, is obtained with a PD controller. Compare this with the first plot in Fig. 7 to see the difference between simulation and experiment.

Figure 9 gives the result when only inertial forces and Coulomb friction, without direction dependent parameters, are compensated. The error is already small after two cycles.

In Fig. 10 the influence of the directional dependency of the Coulomb friction gives the largest improvement of the tracking error.

Good results are obtained with the full friction model, as shown in Fig. 11, although the improvement is not as large as suggested by the simulation results. See the second plot of Fig. 7 for comparison with the simulation.

To unfold the potential of extended friction compensation: the result of Fig. 12, where a longer period to obtain appropriate values for the parameters \( \theta \) was allowed (3 cycles), is the best that could be obtained experimentally. This result is comparable with the second plot of Fig. 5. A faster adaptation,
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This discrepancy means that evaluation of modifications of control schemes by simulations should always be checked by implementation of the modification in the controller software and validation of the simulation results with experiments. This indispensable step is, however, often omitted in the development and presentation of control schemes.

7. CONCLUSIONS AND RECOMMENDATIONS

From the simulation results and the experiments we conclude that the use of extended friction models can improve the tracking performance. Adaptation of the model parameters is necessary to get small tracking errors. The adaptation should be made fast to permit short adaptation times in case no previous knowledge of the parameters is available. In our setup this was not possible without influencing the stability. When previous knowledge of the parameters is available the allowable adaptation gains give a sufficiently fast parameter adaptation.

Further research in this area should focus on guidelines for the choice of the adaptation gain. The tuning rule proposed by (Niemeyer and Slotine, 1991) could not be used without additional adjustments of the gains. There is also a modest discrepancy between the simulation results and the experiments. To be able to evaluate modifications of control schemes with simulations only, a more accurate model must be made. To facilitate the interpretation of the results the authenticity of the model should be sufficient.

8. REFERENCES


