# HIL-Friction Simulation for Experimental Testing of Position Control Schemes

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## 1 Introduction

The compensation of friction and active vibration damping have gained considerable importance over the past few years. One reason for this are the increasing demands for higher precision and speed in positioning mechanisms in the areas of machine tools and handling mechanisms (robotics). Furthermore, the requirements of modern actuator-systems in the automotive area, for example in electronic brakes and steering, have also intensified the difficulties in handling friction and compliance.

For this reason the Cologne Laboratory of Mechatronics (CLM) began to develop suitable compensation algorithms several years ago. These were easy to realize and robust when implemented [1]. In addition, a new approach for estimating and compensating friction was developed and implemented successfully for a typical electromechanical positioning system [2,3].

The comparison of simulation studies with experimental results led to the development of a model library for the realistic simulation of friction. These models are based on the latest trends from international papers on the subject of friction in controlled systems [4] and were extended to cover several effects – like load dependent friction – which are relevant in real-life applications [5]. The library can be used to conduct extensive parameter studies of the behavior of a friction compensator in combination with different friction characteristics in a drive system.

Despite the advances in friction modeling, experimental studies of the system behavior remain essential. Therefore, the positioning system mentioned above was extended to allow experimental studies of the robustness of friction compensators. The application of Hardware-in-the-Loop (HIL) Simulation makes it possible to reproduce almost any friction characteristic in an experiment.

## 2 Positioning system and extensions for the HIL-Simulation of friction

Figure 1 shows the electromechanical positioning system (EMPS) used at the Cologne Laboratory of Mechatronics for the experimental testing of controllers with active vibration damping and friction compensation. The elastic vibrations are caused by the compliant coupling of the motor and ball screw, and friction is generated mainly by the backlash free ball screw drive which converts the rotation of the screw to the translational movement of the carriage. The output voltage $u_{tacho}$ of a tachometer and the counter value of an incremental encoder can be used by the control to determine the angle and angular velocity of the ball screw. The motor is driven by the input voltage $u_{servo}$ of a servo amplifier with integrated current control. An extensive description of this test bed is available in [6].

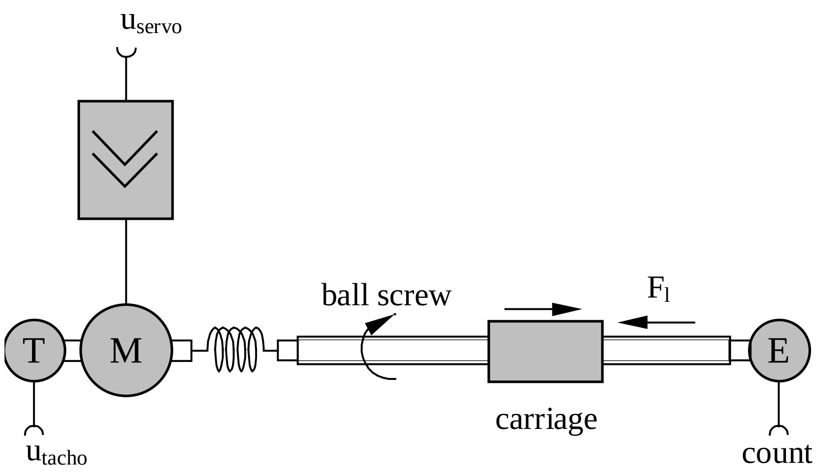


**Figure 1:** *EMPS for active vibration damping and friction compensation*

The extensions made to the EMPS in order to test a control with different friction characteristics are shown in figure 2.

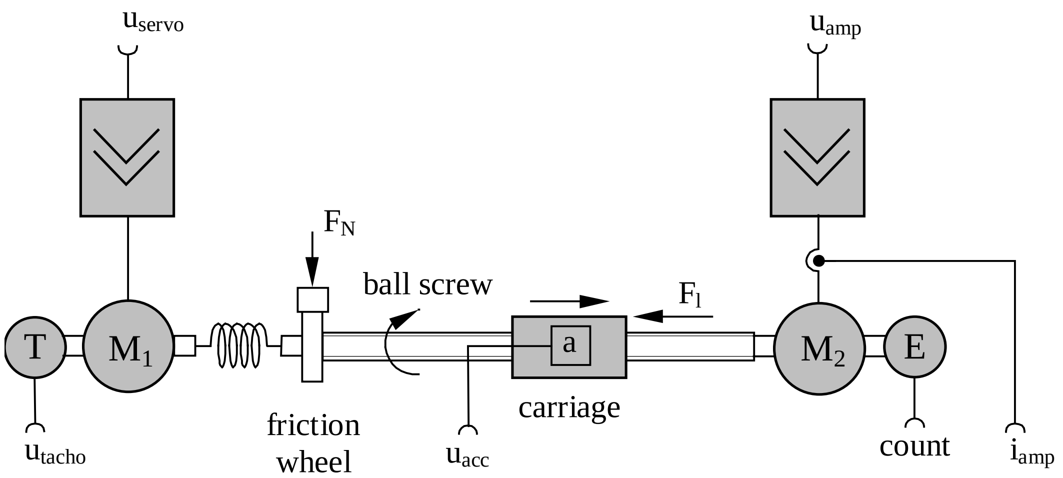


**Figure 2:** *EMPS with friction wheel, load motor and accelerometer for HIL-Simulation of additional friction*

The load side friction of the ball screw drive can be increased mechanically by the friction wheel. It is equipped with two diametrically opposed friction linings so that no radial forces act on the ball screw. These forces would produce an undesired deflection of the ball screw and would lead to a falsification of the angle measurement through the incremental encoder.

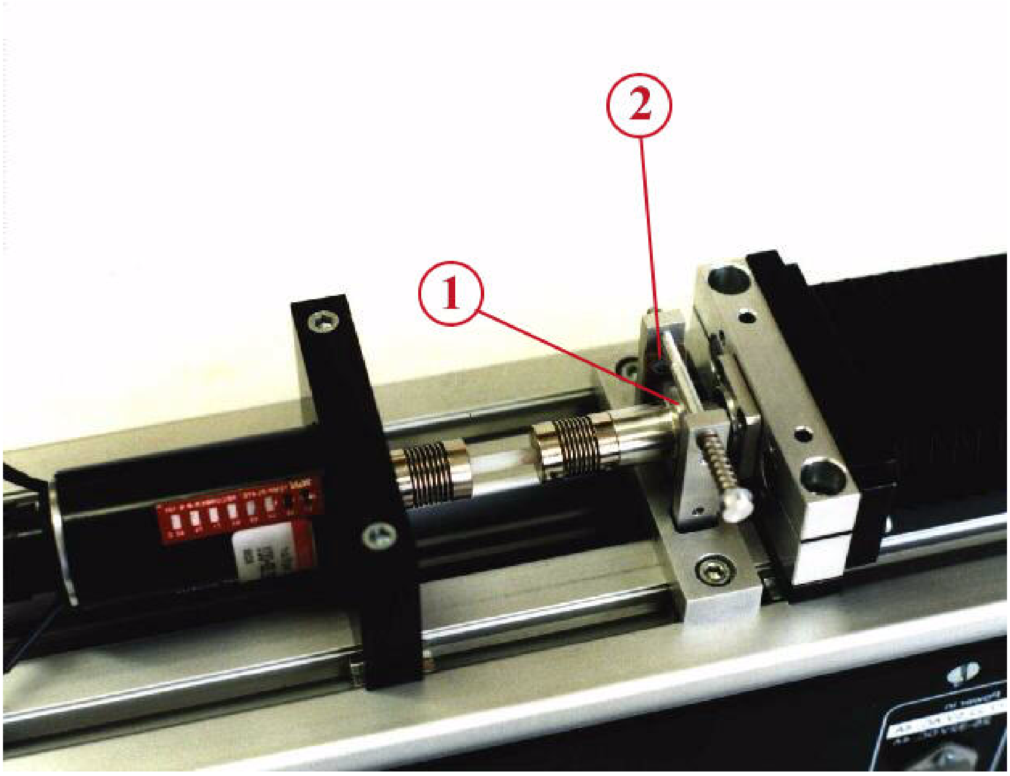


**Figure 3:** *Friction wheel (1) with friction linings (2)*

However, the use of the friction wheel allows only a limited influence on the friction characteristic. Although it is possible to apply a maximum static friction torque within the limit of the maximum torque of the drive motor, the shape of the velocity dependent friction characteristic is determined by the properties of the friction linings. In order to apply an arbitrary velocity dependent friction characteristic to the test bed, a load-side motor is used. The motor is controlled via the reference voltage $u_{amp}$, the measured current $i_{amp}$ and a digital current control to generate a specific load torque. The combination of this load torque with the friction torque of the ball screw drive and the friction wheel shall yield the desired velocity dependent friction characteristic. The static friction is generated exclusively by the mechanical components. The velocity dependent friction is generated by a signal processing setup, a simplified version of which is shown in figure 4. The difference between the actual and desired friction characteristic is applied to the reference input of the current control as the reference torque of the load motor. The load side angular velocity is computed by an observer which estimates the velocity via the measurement signals of the incremental encoder and the accelerometer.

In combination with the EMPS this configuration constitutes a HIL-Simulator for velocity dependent friction.

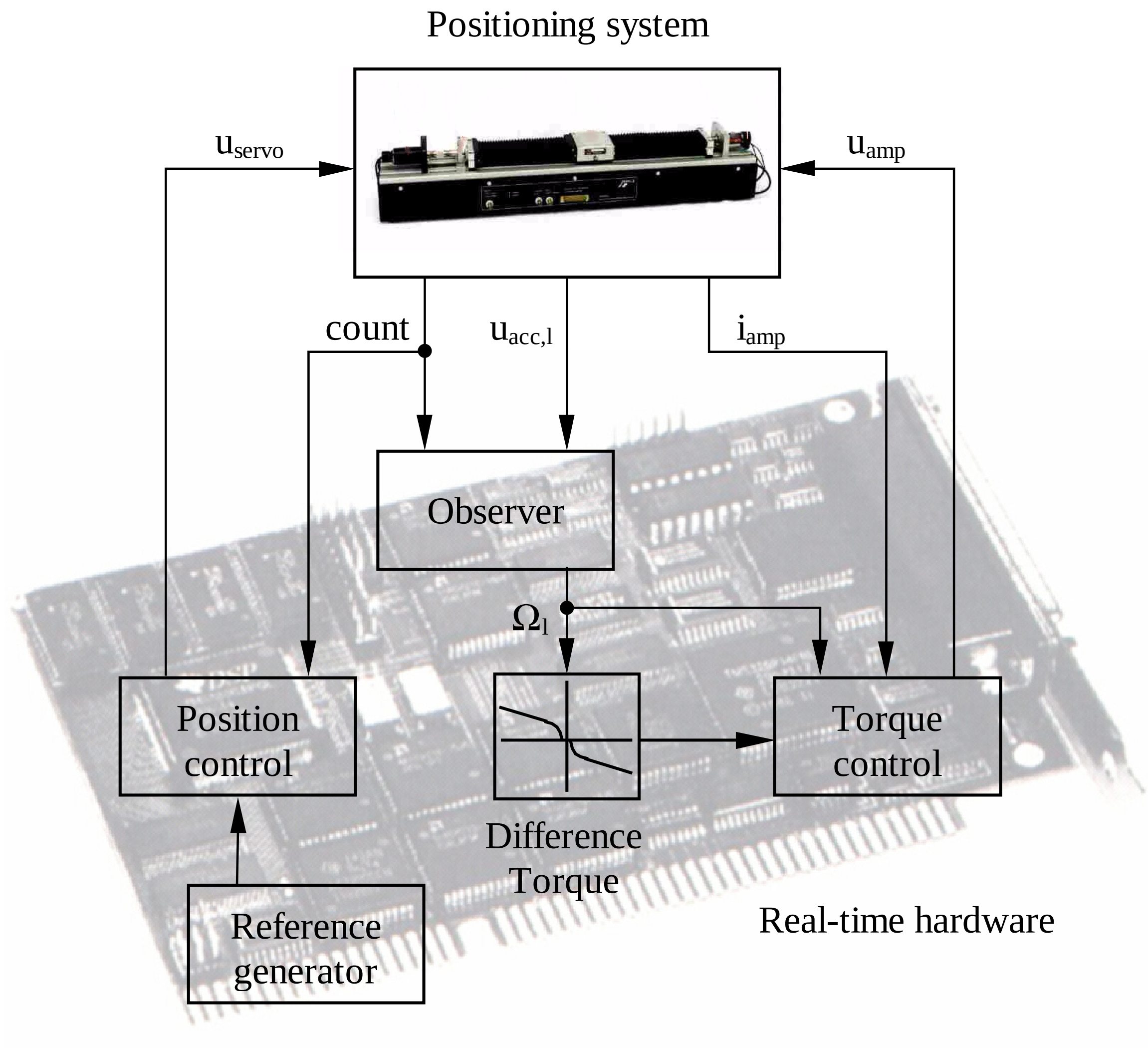


**Figure 4:** *HIL-Simulation of velocity dependent friction.*

## 3 HIL-Simulation of friction

Section 2 described the expansion of the EMPS to a HIL-Simulation test bed for the examination of position control schemes with vibration damping and friction compensation with different friction characteristics. Figure 5 will be used to explain how the interaction between the components can produce almost any desired friction characteristic.

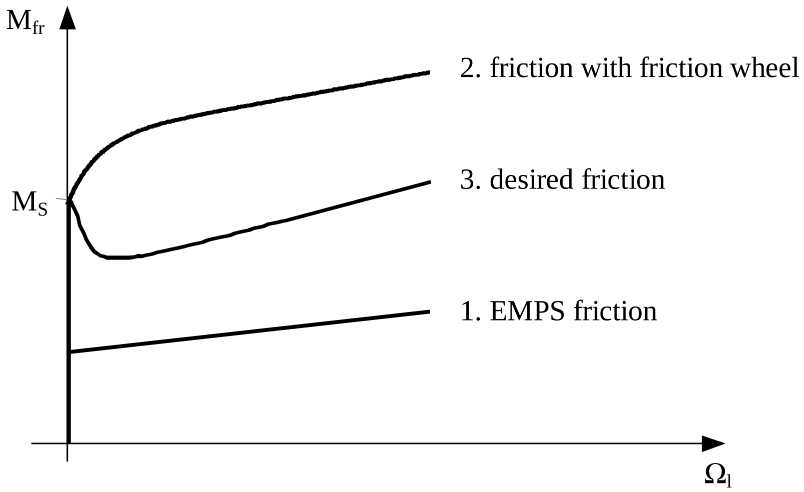


**Figure 5:** *Friction in the HIL test bed*

Curve 1 shows the friction characteristic of the EMPS without the friction wheel. To obtain the desired friction characteristic 3 the friction wheel is tightened so that the maximum static friction torque $M_s$ is equal to the desired value. This results in friction characteristic 2 where the velocity dependent friction characteristic does not necessarily correspond to the requirements of a given experiment. Then this characteristic can be adjusted through the application of the difference torque between characteristic 2 and 3 by the load motor. The difference torque is computed after the measurement of the actual characteristic with the friction wheel and the definition of the desired characteristic and is applied via the torque controller (figure 4).

This HIL-Simulation of the difference friction torque is realized with the development environment used at the Cologne Laboratory of Mechatronics, which is composed of MATLAB/Simulink and dSPACE TDE. Figure 6 shows the block diagram used for the rapid prototyping of a controller for the EMPS and the load motor control to perform the first experimental studies with automatic code generation.

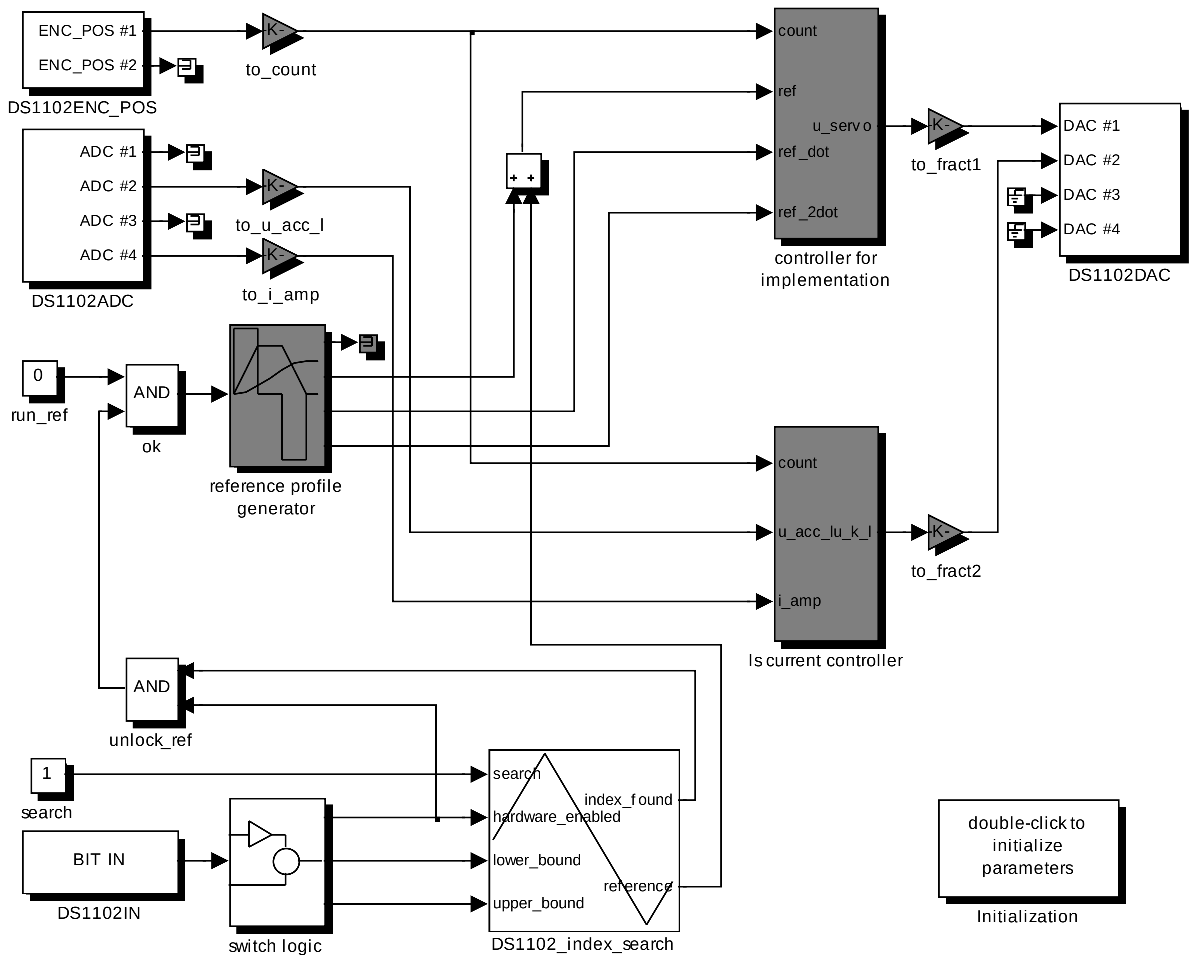


**Figure 6:** *Simulink block diagram for rapid control prototyping*

The load motor current control and the calculation of the difference torque which is necessary for the desired friction characteristic are grouped in the subsystem "ls current controller". The other blocks and subsystems are the controller to be tested with reference signal generator, the peripheral interfaces and the index search (homing) for the zero position of the carriage.

The inputs of the current control are the measured ball screw angle, carriage acceleration and motor current from the incremental encoder and A/D converter interfaces. The output is the motor voltage needed for the difference torque. This voltage is scaled to the input voltage $u_{amp}$ of the servo amplifier (see figure 2) before it is applied to the D/A interface. The contents of the subsystem "ls current controller" are shown in the following block diagram

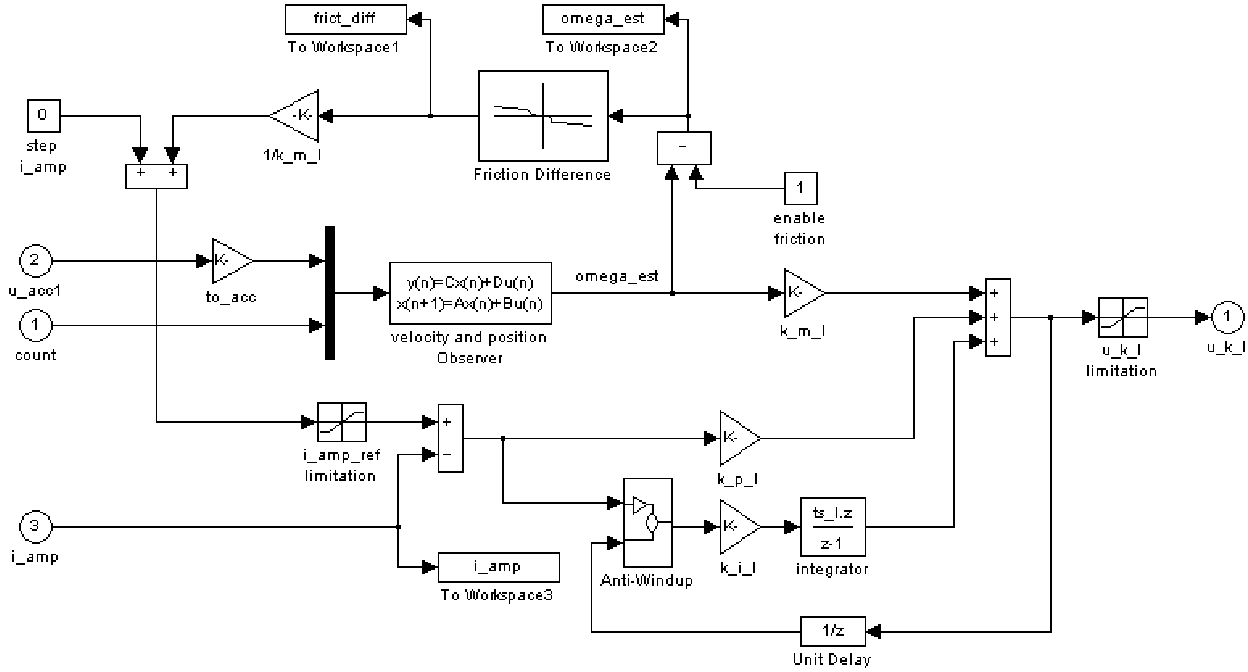


**Figure 7:** *Extended load motor current control*

The lower part of the block diagram contains a conventional PI current control. The screw's angular velocity $\Omega_{est}$ is determined from the incremental encoder signal count and the accelerometer voltage $u_{acc}$ by a subsystem observer (center part of the block diagram). The observer was designed for a simplified plant model using the pole placement method. By augmenting the plant model with a disturbance model for accelerometer offset voltages, errors in the estimated angular velocity due to possible offsets in $u_{acc}$ are compensated. The angular velocity $\Omega_{est}$ is used to counteract the back-EMF of the load motor and to generate the difference torque for the desired velocity dependent friction (upper part of the block diagram). The difference torque is converted to a current via the load motor torque constant, and the range of this current is limited before it is applied to the current control as a reference signal. Details on the design and implementation of the current control can be found in [7].

Finally we will take a closer look at the computation of the difference torque. This is performed in the subsystem "Friction Difference", which is shown in figure 8.

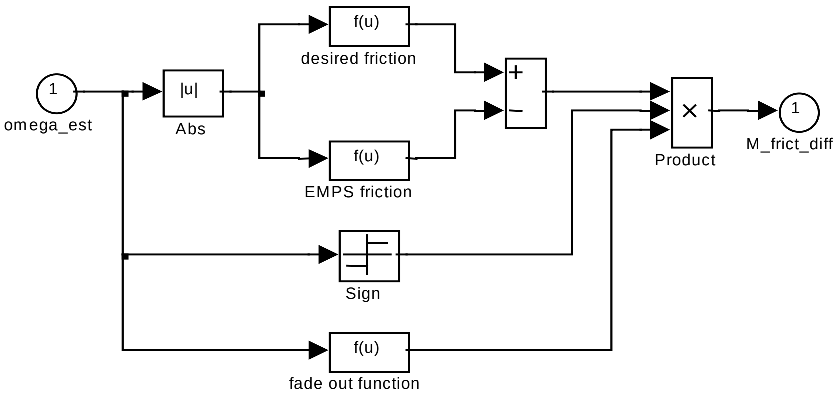


**Figure 8:** *Subsystem for the generation of the difference torque for velocity dependent friction.*

The magnitude of the angular velocity $\Omega=|\Omega_{est}|$ is fed to two nonlinear function blocks for the computation of the Stribeck characteristics for the desired and actual friction characteristics. The output signals of the function blocks are computed via the equation

$$M_R(\Omega) = M_K + (M_S - M_K) \cdot e^{-\frac{\Omega}{\Omega_{Strib}}} + b \cdot \Omega$$

with the given parameters for the static friction torque $M_S$, the kinetic friction torque $M_K$, the Stribeck velocity $\Omega_{Strib}$ and the velocity proportional damping constant b for each block. The definition of these parameters is covered in detail in the next chapter. After the calculation of the difference between the two characteristics the difference torque with the correct sign is the product of the difference with the sign of the angular velocity $\Omega_{est}$. In addition, a so-called fade-out function prevents the application of the torque in a defined interval around the angular velocity value $\Omega_{est}=0$. This is required because of the delays in the estimation of the angular velocity and the current control. In the event of a sign change of the angular velocity the fade-out prevents the application of a difference torque with an incorrect sign. In the area of the fade-out the friction torque is applied solely by the mechanical components (friction wheel and EMPS).

The code for the realization of the block diagram in figure 6 with the subsystems in the figures 7 and 8 is generated automatically for the real-time hardware shown in figure 4.

# 4 Graphical User Interface

The use of the hardware described in the previous chapters together with the realization of the simulation and the application of friction in software makes it possible to create various friction characteristics to test position controllers. However, a flexible and reproducible design of experiments requires an easy-to-use interface. The graphical user interface developed for this purpose will be demonstrated by using an example for the measurements of the actual friction characteristic (Friction Measurement) and the design of the desired characteristic (Friction Design). Figure 9 shows the program window after starting the user interface. The interface and all underlying functions were programmed using the development environment at the Cologne Laboratory of Mechatronics (MATLAB/Simulink and dSPACE TDE). The modular design makes it easy to extend the interface with new control and compensation algorithms.

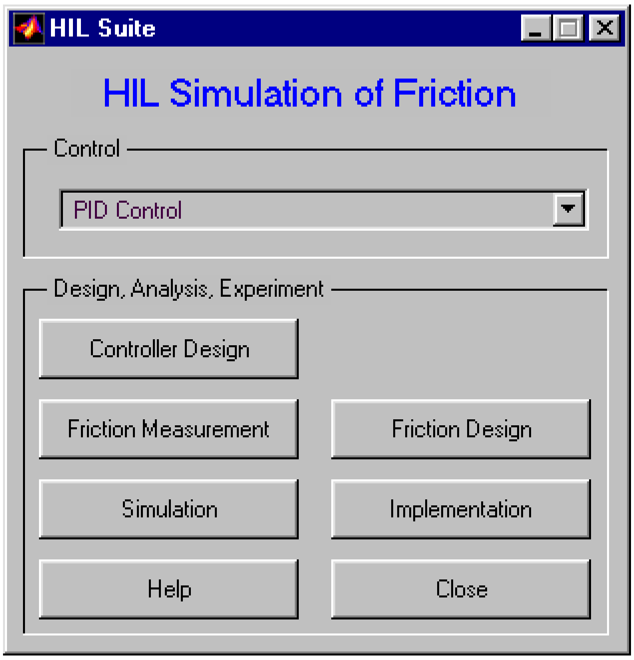


**Figure 9:** *User interface for compensator testing*

A click on the pushbutton labeled "Friction Measurement" opens the window shown in figure 10 for measuring the current friction characteristic, e.g. the characteristic adjusted with the friction wheel. Clicking another pushbutton in the toolbar starts an experiment for the automatic measurement of this friction characteristic. The experiment is composed of a suitable set of parameters for the reference signal generator from figures 4 and 6 and the capture of time histories from the real-time hardware necessary for the computation of the friction characteristic. The result of such an experiment is displayed in the plot window in figure 10 as the noisy course. Afterwards the Stribeck function from equation (1) can be fit to the measured course (smooth plot in figure 10). The parameters ($M_S$, $M_K$, $\Omega_{Strib}$, and b) are displayed and then downloaded to the corresponding function block (EMPS friction) in figure 8 of the real-time hardware.

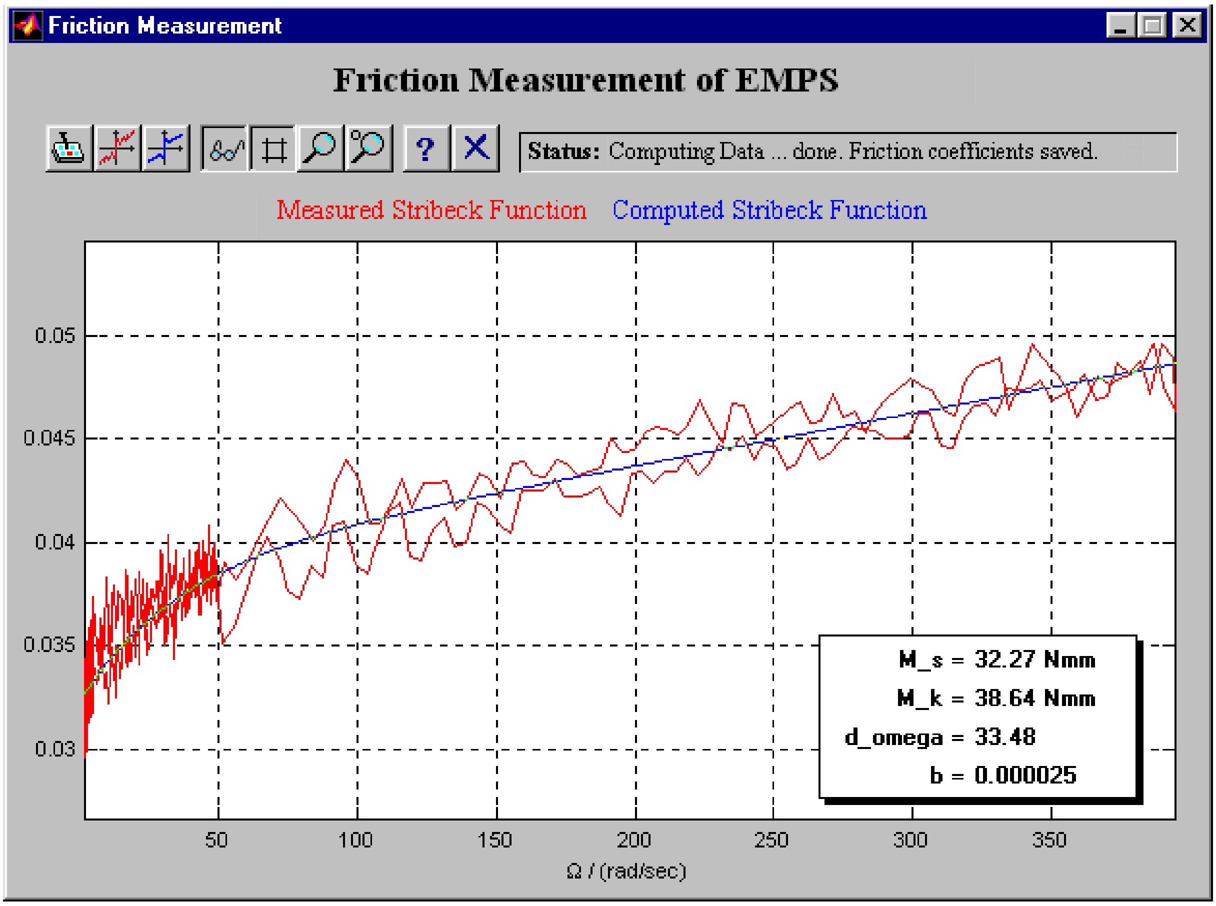


**Figure 10:** *User interface for the determination the actual friction characteristic*

To perform an experiment using the modified friction characteristic, the next step is the design of the desired characteristic. To do this the user clicks the button labeled "Friction Design" on the user interface in figure 9 in order to open the window shown in figure 11. The plot window displays the friction characteristic fitted to the measurement from earlier on. In the same window, after the activation of a matching function from the toolbar, reference points of a desired characteristic can be set with the mouse pointer. For these points the computation of the corresponding Stribeck parameters and the display of the desired characteristic is performed with a simple click on another button. The characteristic and the parameters can be further edited with the help of the mouse and dialog interfaces. Once the characteristic has its desired shape the computed parameters are also downloaded to the corresponding function block in the subsystem of figure 8 on the real-time hardware.

The result of the design of such a friction characteristic is described in the following chapter.

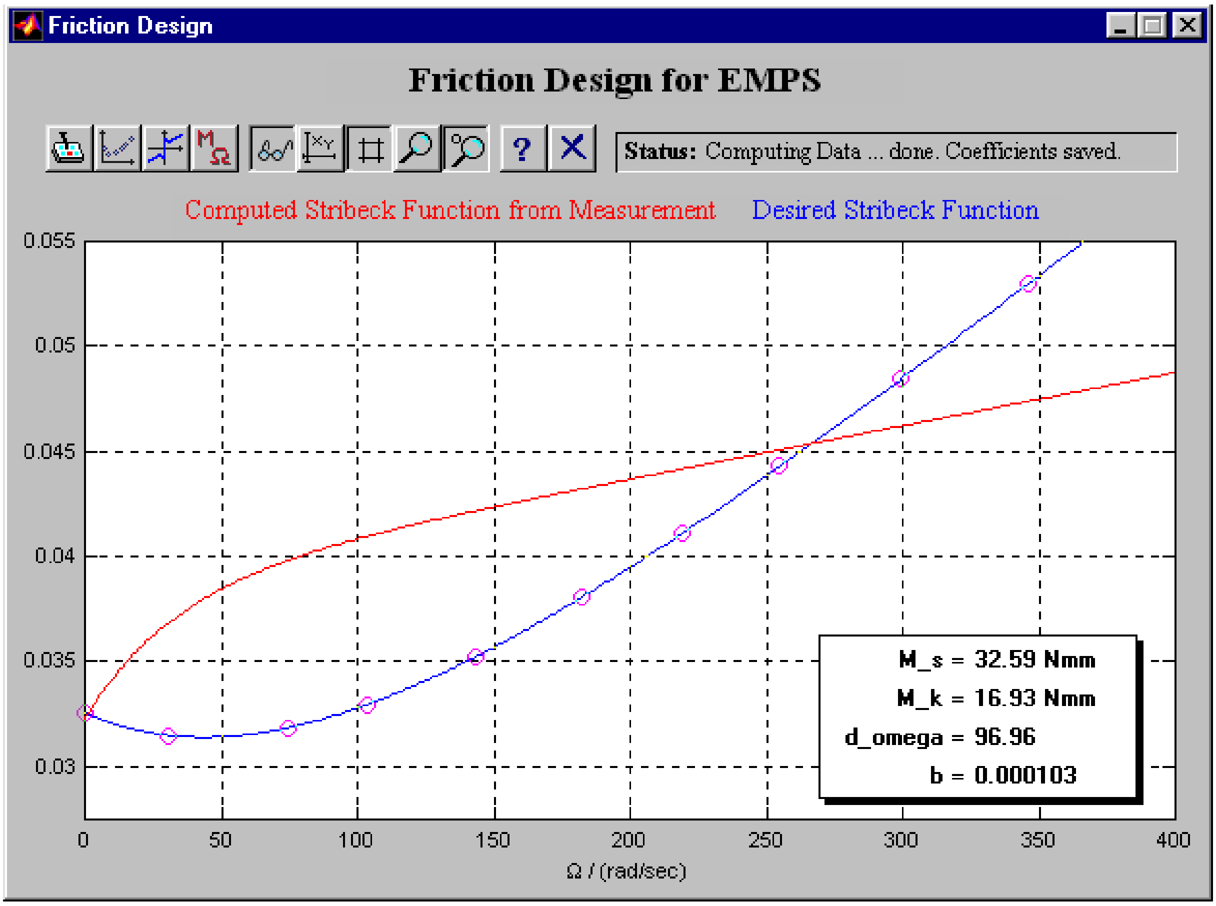


**Figure 11:** *User interface for the design of a desired friction characteristic*

## 5 Experimental Results

The first experiments show that the developed test bed works very satisfactory. The following example of a desired friction characteristic will demonstrate this. Figure 12 shows the actual characteristic (i.e. with friction wheel) and the desired characteristic. For small angular velocities, the load motor has to generate a rapidly changing load torque. In the velocity proportional area the slope of the desired characteristic has been increased so that a sign change of the difference torque occurs.

After downloading the parameters determined for both characteristics to the real-time hardware the friction measurement yields the characteristic shown in figure 13. Through the HIL-Simulation and the application of the difference torque to the load motor the resulting friction corresponds very well to the desired characteristic.

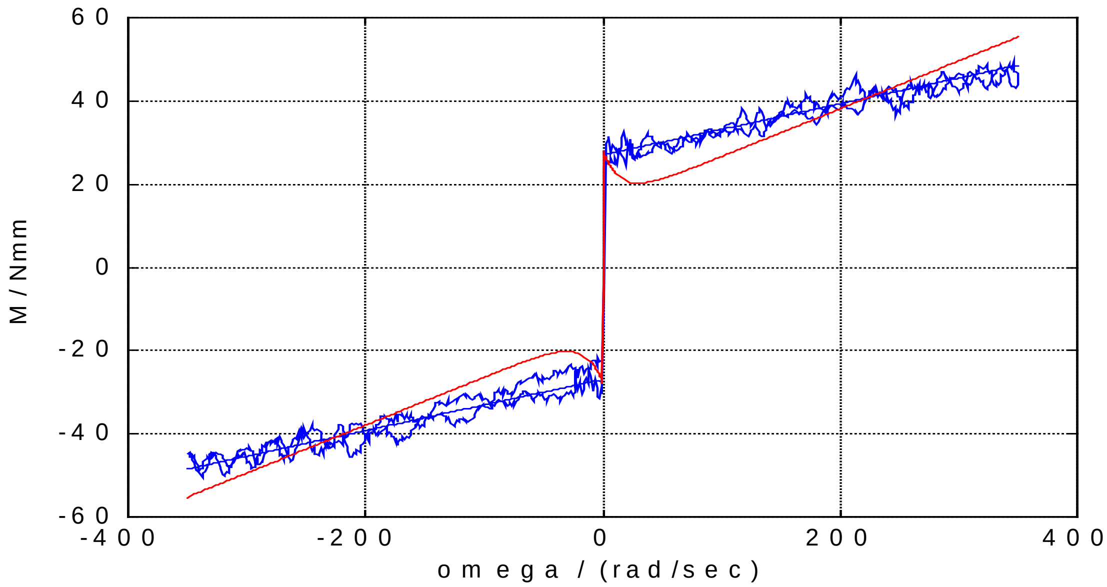


**Figure 12:** *Measured friction characteristic of the EMPS with friction wheel (noisy) and desired characteristic (smooth)*

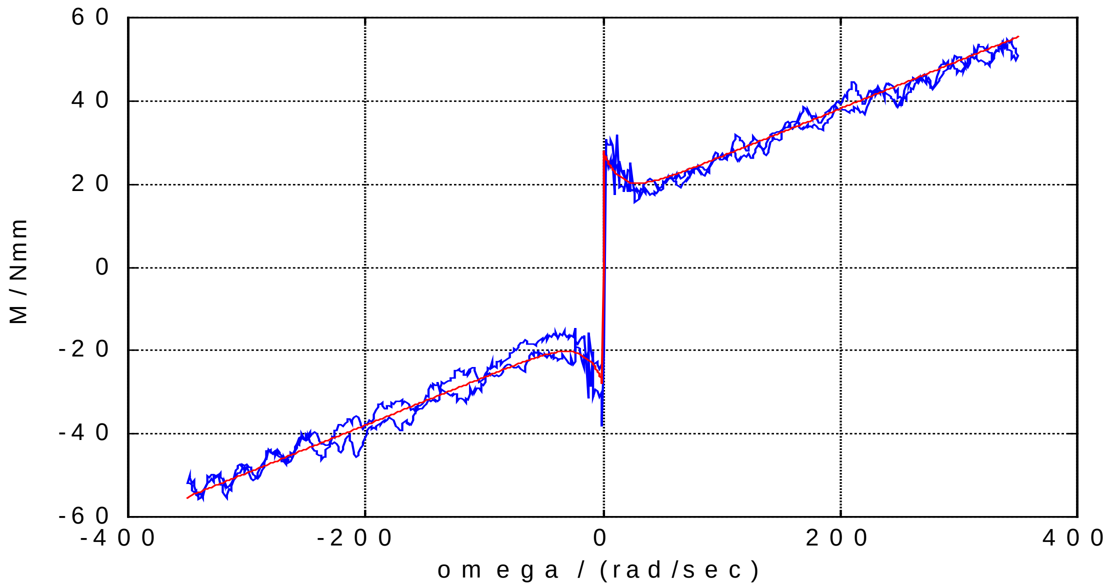


**Figure 13:** *Desired friction characteristic (smooth) and measured resulting characteristic after application of the simulated difference torque (noisy)*

The effect of the fade-out function in the block diagram from figure 8 becomes especially apparent in the load motor torque (figure 14), which is determined with the measurement of the current at the load motor. There is a step in the course of the difference torque (which is the reference signal of the current control) at the limits of the fade-out interval. For a large step size this can lead to undesired vibrations of the controlled system. Therefore, the fade-out interval should be kept as small as possible. The fade-out function had to be introduced to avoid wrong signs for the difference torque which might appear for zero crossings of the angular velocity due to noise in the estimated velocity signal.

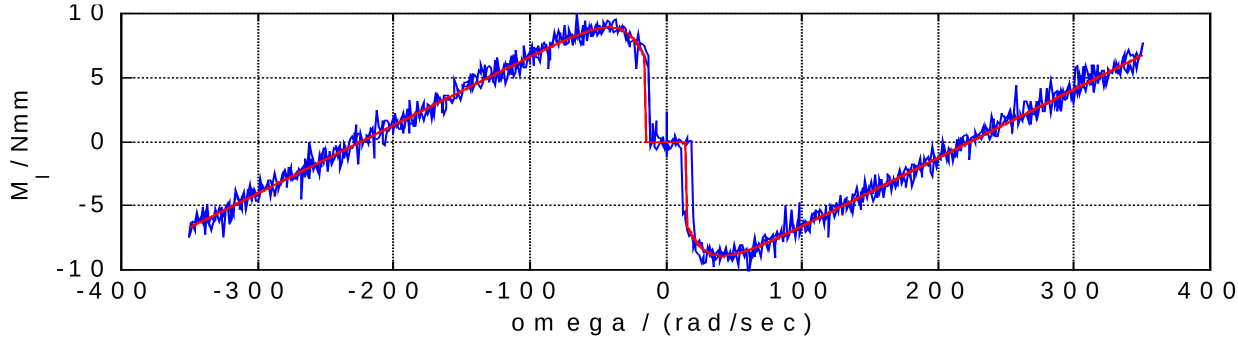


**Figure 14:** *Simulated difference torque (smooth) and measured load motor torque.*

## 6 Summary and Outlook

The first experiments with the test bed described above for the test of position controllers for drives with compliance and friction show very promising results. The use of the HIL-simulation method allows the realization of almost any desired friction characteristic in an experiment. The test bed forms the basis for subsequent experiments with different approaches for friction compensation. Future experiments will include a new approach developed at the Cologne Laboratory of Mechatronics as well as compensators based on predictive control and neural networks. The latter are provided by cooperating partner institutes at the University of Ottawa, Canada and at the Saga University, Japan. The goal of these experiments is to learn more about the implementation and robustness characteristics of the considered approaches for practical application. In addition, classical position control schemes will be examined for comparison.

At the same time as the experiments mentioned above the CLM examines further ways to improve and expand the test bed.

The measurement of the carriage acceleration forms the basis for an extension of the test bed with the HIL-simulation of further EMPS attachments. For example, with the acceleration measurement and the load motor the real-time simulation of an elastic positioning arm developed at the Laboratory of Mechatronics can interact with the real EMPS. This configuration is suited especially well for the testing of newly developed state space control schemes with decentralized observers [8]. The linear observer contained in the control structure is to be augmented and tested with the nonlinear compensation of the drive friction developed at the Laboratory of Mechatronics.

The use of the HIL-Simulation of friction has turned the test bed into a flexible environment for realistic experiments with compensators for drives with compliance and friction. Furthermore, the chosen methodology provides an example for the development and the test of modern drive controllers in the industry.

## Bibliography


[1] H. Henrichfreise und C. Witte, *Experimental Comparison of Observer-Based Friction Compensation Schemes for an Electromechanical Positioning System*. Internal report at the CLM, Faculty of Mechanical Engineering, University of Applied Sciences Cologne 1997

[2] H. Henrichfreise and C. Witte, *Observer-Based Nonlinear Compensation of dry Friction in a Positioning System*. IX. Geman-Polish Seminar, University of Applied Sciences Cologne, October 1997.

[3] H. Henrichfreise and C. Witte, *Beobachtergestützte nichtlineare Kompensation trockener Reibung in einem Positionierantrieb*. Automatisierungstechnik 46, Volume 3, pp. 128-135, R. Oldenbourg Verlag, Munich 1998.

[4] B. Armstrong-Hélouvry, P. Dupont und C. Canudas de Wit: *A Survey of Models, Analysis Tools and Compensation Methods for the Control of Machines with Friction*. Automatica, Vol. 30, No. 7, 1994, S. 1083-1138.

[5] A. Lüngen, *Erstellung einer Modellbibliothek für die Simulation von Reibung*. Diploma thesis at the CLM, Faculty of Mechanical Design Engineering, University of Applied Sciences Cologne 1998.

[6] H. Henrichfreise: *Prototyping of a LQG Compensator for a Compliant Positioning System with Friction*. 1. Workshop TRANSMECHATRONIK - Entwicklung und Transfer von Entwicklungssystemen der Mechatronik, HNI-Verlagsschriftenreihe, Band 23, Paderborn 1997. Herausgeber: Jürgen Gausemeier.

[7] J. Binder, R. Starbek, *Aufbau eines Versuches zur HIL-Simulation von Reibung in Positionierregelungen*. Project work at the CLM, Faculty of Mechanical Design Engineering, University of Applied Sciences Cologne 1998.

[8] R. Lummert, *Dezentrale Beobachter zur Realisierung robuster Regler für aktive Schwingungsdämpfung*. Diploma thesis at the CLM, Faculty of Mechanical Design Engineering, University of Applied Sciences Cologne 1998.