HIL SIMULATION TECHNIQUE FOR NON-MODEL-BASED CONTROL OF DC SERVO-DRIVE WITH FRICTION

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Abstract: This paper is a contribution to the development of a real time friction simulator (RTFS) using a Hardware-In-the-Loop (HIL) structure built around the DS1103 PPC Controlled Board (dSPACE). The simulator is designed to generate friction torque on its mechanical shaft and to provide the characteristics of a given friction model in order to implement and test different compensation strategies for a DC position tracking servo drive. Considering the LuGre model of friction, a non-model-based friction compensator is proposed by using three different structures of a disturbance observer which estimates friction for a position controlled DC electrical drive. Copyright © CONTROL 2006

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1. INTRODUCTION

Friction is to be considered an essential process for micrometer scale tracking servo systems. Tracking errors, limit cycles and undesired stick-slip motion, static friction (stiction) effect in standstill, Coulomb friction, nonlinearity and negative derivation of static characteristic of friction during transition from stiction to Coulomb friction – called Stribeck effect – are features that made friction to be responsible for great obstacles in high precision positioning systems such as a DC servo drive. Control strategies that attempt to compensate for the effects of friction inherently require a suitable friction model to predict and compensate friction. An approach for studying friction behavior in order to validate a model or a compensation scheme as part of a high precision servo system control is the real time friction simulator (RTFS) built around the hardware-In-the-Loop (HIL) structure presented hereafter.

The modern controlled systems' development is based on the use of HIL simulation techniques, for physical emulation of the systems, with which the investigated equipment interacts (Hanselman, 1993). HIL systems contain a physical sub-system, which is controlled in closed loop by a real time software simulator, built on the basis of the emulated system's mathematical model. This physical sub-system, connected to the investigated equipment, offers static and dynamic characteristics, practically identical with the ones of the emulated system.

The HIL system this paper dealing with is the friction torque simulator, connected to a DC servomotor. The simulator "offers" a mechanical shaft, where the static and dynamic behavior of the friction is obtained, according to the conditions yielded by the model took into account. In this case, the investigated equipment is the following assembly: electrical actuator (DC servo-drive) + power electronic circuit + electrical network + associated control systems. However, HIL systems allow experimental investigation of high precision control algorithms required in positioning servo systems such those used in robotics and emulate friction behavior for motion control kind of applications.

This paper aims to present the HIL technique (conceiving a methodology for HIL configuration) in connection with the performance evaluation of a position tracking servo-system with friction effects.
implemented using an experimental setup based on dSPACE 1103 board. Moreover, a non-model-based friction compensator is proposed considering that the friction torque could be assimilated to a disturbance load torque. According to the compensation scheme adopted three different observer structures for friction estimation were developed by authors.

II. HIL STRUCTURE FOR RTFS

Let us consider a tracking servo-system with friction, in which the DC servo-motor, the power electronic circuit, the electrical network and the control system form an investigated physical system (IPS). In this case, the HIL system contains the real time physical system (RTPS) as a friction torque simulator, which emulates the real friction, offering a mechanical shaft, to which IPS is coupled. The interaction variables between IPS and RTPS are the shaft speed \( \Omega \) and the shaft torque, \( T_{friction} \), denoted hereafter as \( z_1 \) and \( z_2 \). Thus, the HIL system is represented by the interconnected assembly RTPS+IPS which allows the experimental study of IPS in the emulated environment created by RTPS (figure 1). Subsequently, RTPS has two main components: RTSS (Real Time Software Simulator) and the physical part (PP) of RTPS which offers the “natural” environment for IPS. Concerning the real time soft simulator (RTSS) as part of RTPS it must be noticed that \( z_1 \) becomes \( T_{ref} \) as piloting variable processed via the chosen mathematical model for friction. The electromechanical tracking system (ETS) is the physical part (PP) of RTPS which receives the reference signal \( T_{ref} \) from RTSS and provides a shaft which has the static and dynamic characteristic of the typical friction torque load (\( T_{friction} \) \( \Omega \)) where \( \Omega = z_2 = z_2 \) is the response variable from IPS; the ETS controller imposes that the feedback \( T_{friction} = z_1 \) to be equal with \( T_{ref} \).

The IPS part of HIL structure (shading filled part in figure 1) includes beside the speed strategy controlled rectifier, a block featuring the software implemented strategy for friction control. The others HIL architecture blocks are: LOAD – torque controlled AC machine emulating the friction torque (\( T_{friction} \)), DC servo – servo-tracking electric drive tested with RTPS (the actuator mechanical coupled with induction machine LOAD emulating the friction torque), \( T_r \) – transducer for mechanical variables measured on the shaft (position \( x \), angular velocity \( \Omega \)), CIT – torque strategy controlled inverter and \( R\Omega - \) speed strategy controlled rectifier.

1I.1 RTSS configuration

The RTSS is based on mathematical models of the friction. The choice made by authors for friction modeling was the LuGre model. The model is related to the bristle interpretation of friction and has succeeded to incorporate the link between tribology and modeling for control features (Canudas de Wit, C. et al., 1995). It includes Strubeck effect, rate dependent friction phenomena such as varying break-away force and frictional lag. Friction is modeled as the average deflection force of elastic springs associated to the contact. When a tangential force is applied, the bristles will deflect like springs. If the deflection is largely enough the bristles start to slip. The average bristle deflection (the new state of friction process - \( z_1 \)) for a steady state motion is determined by velocity. It is smaller at low velocities, which implies that the steady state deflection decreases with increasing velocity. The model has the form:

\[
\frac{dz}{dt} = v - \frac{\sigma_0 |v|}{g(v)} z \\
F = \sigma_0 z + \sigma_1 z + \alpha v
\]

where \( z \) is the pre-sliding displacement or, more accurately, the average deflection of the bristles, \( \sigma_0 \) and \( \sigma_1 \) are the stiffness of bristle and, respectively, the damping, \( \alpha \) is viscous friction. The function \( g(v) \) is the function describing Strubeck’s effect. A parameterization proposed for \( g(v) \) is given hereafter:

\[
g(v) = F_c + (F_s - F_c)e^{-(v/v_s)^2}
\]

where \( F_c \) is Coulomb force, \( F_s \) static friction and \( v_s \) is Strubeck’s velocity.

![Fig.1. HIL system architecture for friction simulator](image-url)
Whereas the LuGre model includes the Stribeck effect and also offers a smooth transition at velocity reversal, the reason of its choice as a tool for further analysis of friction compensation and control for high precision positioning applications developed on HIL system is fully justified.

II.2. ETS (PP) configuration and control

The Physical Part (ETS in HIL architecture) of friction simulator is based on the current control of an AC motor (LOAD in figure 1). This current is considered as the electrical image of the torque developed on the shaft. The set point of the loop is obtained according to a mathematical model that includes different friction characteristics (depending on LuGre model). The state description of the friction model is:

\[ T_{\text{ref}} = T_{\text{friction}} = F(\Omega) \]  

(4)

The output variable \( T_{\text{ref}} \) is a mechanical variable, which characterizes the friction modeled behavior on its shaft. Depending on this reference variable, the simulator has been used with a torque control strategy, when \( T_{\text{ref}} \) is the shaft torque reference and \( Tr \) is a speed transducer (or a speed estimator). The typical electric drive dynamic model that governs the HIL investigated physical system is given by its motion equation:

\[ J \frac{d\Omega}{dt} = T(\Omega) - T_{\text{LOAD}}(\Omega) \]  

(5)

where: \( \Omega \) - the shaft rotation speed, \( J \) - the total moment inertia, \( T(\Omega) \) - the active torque characteristic, \( T_{\text{LOAD}}(\Omega) \) - the static characteristic of the load. Considering an electromechanical high performance servo tracking system (IPS in figure 1) working at low velocity and demanding an accurate positioning, the major influence among all components of load torque belongs to friction torque. The quantitative substitute in (5), \( T_{\text{LOAD}}(\Omega) = T_{\text{friction}} \) is fairly well understood. In our experimental studies, the torque control strategy for the LOAD AC machine has been adapted by imposing \( T_{\text{ref}} \) in terms of a friction model. The simplified block diagram of the implemented RTFS is given in figure 2.

Fig.2. Simplified block diagram of the implemented RTFS

The LOAD AC machine control is made in closed loop structure, according to the torque imposed to the VLT 5005 FLUX converter (IT). A second VLT 5005 FLUX converter (R \( \Omega \)) is used to regulate the position of the mechanical shaft by supplying power to DC servo machine controlled also in closed loop structure via a control law imposed by a friction regulation or a non-model-based scheme of compensation.

II.3. RTFS development

The hardware block diagram of the RTFS is presented in figure 3. The DS1103 PPC controlled board (dSPACE) is equipped with a Power PC processor for fast floating-point calculation at 400 MHz. Programming has been done via Matlab/SIMULINK® in Real-Time-Interface (RTI). The control has been entirely developed in SIMULINK® which is a user friendly, block-oriented, dynamic simulation environment. At first, an off-line model was developed, implemented and tested and then with only a few changes the model has been implemented into the dSPACE controller.

Fig.3. Hardware structure of the implemented RTFS

III. FRICTION COMPENSATION AND CONTROL

In figure 4 a classical feedback compensation block diagram is presented (\( H(s) \) is a linear PID controller).

Fig.4. Feedback compensation scheme

There are many ways to compensate for friction (Armstrong-Helouvry et al., 1994, Canudas de Wit et al., 1999). Focused on systems for motion control, especially for DC servo position controlled system mentioned above, a feedback non-model-based observer compensation scheme has been taken into account for the application proposed in this paper. Hereby, the friction observer is replaced by a disturbance load torque estimator expressed either in a stochastic manner as a filter or in a determinist conception. Generally, the load torque \( T_{\text{LOAD}} \) of an electrical drive (DC or AC) is the sum of inertial
torque $T_{\text{motor}}$, external torque $T_{\text{ext}}$ and friction torque $T_{\text{friction}}$. Considering a classical DC electrical servo drive for a numerical positioning system it could be supposed that the friction torque prevails and the assumption $T_{\text{load}} = T_{\text{friction}}$ works out especially for low velocities and crossing zero velocity regimes. Around this hypothesis the following friction estimator structures are proposed in order to develop the feedback compensation scheme to diminish the negative effects of friction.

### III.1. Filter estimator

The analysis of block diagram in figure 5 which represents mathematical model of a DC motor shows that the load torque can be estimated as follows:

$$
\hat{T}_{\text{load}}(s) = \frac{k}{J_s + k_s \left[ K_e I(s) - J_s \Omega(s) \right]} \quad (6)
$$

$$
\hat{T}_{\text{load}} = H_0(s)T_{\text{load}}; \quad H_0(s) = \frac{1}{T_s s + 1};
$$

$$
T_{\text{load}} = \frac{J}{k_s}; \quad \omega_d = \frac{1}{T_0} \quad (7)
$$

![Fig.5. Block diagram of DC motor with load torque estimator](image)

Beside the transfer function $H_0(s)$ defined as strictly causal subsystem, the dynamic performances of the estimator must be imposed. Subsequently, according to the $H_0(s)$ order the estimators could be denoted as 1st, 2nd, ..., n-th order. The estimator synthesis considers that along the achievement of dynamic performances a good rejection of measurement noise due to current and speed acquisition must be accomplished. Therefore, a low-pass filter (LPF) has been associated to $H_0(s)$ in order to realize a much better approximation of an ideal Butterworth filter response. As the filter order increases the approximation gets better. In figure 6 is presented a first order estimator. The estimator pass band is imposed by the cutoff frequency $\omega_d$ (in eq.7) chosen so that a compromise between the acquisition noise level and dynamic performances of the estimator is reached.

![Fig.6. First order load torque estimator](image)

### III.2. Load torque observer theory applied to DC servo motor

Starting from the model equations of DC motor:

$$
U_t = R_i i_t + L_s \frac{di_t}{dt} + K_e \Omega
$$

$$
J \frac{d\Omega}{dt} = K_i i_t - T_{\text{load}} \quad (8)
$$

the state-space representation is:

$$
\dot{x} = Ax + Bu \quad y = Cx \quad (9)
$$

where:

$$
x = \begin{bmatrix} i_t \\ \Omega \end{bmatrix}; \quad u = U_t;
$$

$$
A = \begin{bmatrix} -\frac{R_i}{L_s} & -\frac{K_e}{L_s} \\ \frac{K_e}{J} & 0 \end{bmatrix}; \quad B = \begin{bmatrix} 1 \\ 0 \end{bmatrix}
$$

As $(A,C)$ is observable (the current is measurable), the motor states could be estimated by a state observer governed by:

$$
\dot{\hat{x}} = A \hat{x} + Bu + GC (x - \hat{x}) \quad (11)
$$

where $G = \begin{bmatrix} g_1 \\ g_2 \end{bmatrix}$ is the observer gain determined so that the estimation error $e = x - \hat{x}$ tends to 0 faster than the motor’s dynamics. From eq.11 the estimation error response using the state equation can be expressed as:

$$
\dot{e} = (A - GC)e \quad (12)
$$

For load torque estimation purposes the previous theoretical issues must be extended to a system with disturbances. There are considered two extensions according to how the disturbance is assimilated: as an unknown input or as a state variable.

**Load torque as unknown input.** In this case eq. 9 and 10 are rewritten as follows:

$$
\dot{x} = Ax + Bu + D T_{\text{load}} \quad (13)
$$

where $x, A, B, u$ are given by eq. 10 with:

$$
D = \begin{bmatrix} 0 \\ 1 \\ J \end{bmatrix} \quad (14)
$$

Using the observer described by eq. 11, the estimation errors for current and speed ($\hat{e}_i, \hat{e}_\Omega$) become:

$$
\dot{\hat{e}}_i = -\frac{R_i + g_1 L_s}{L_s} \hat{e}_i - \frac{K_e}{L_s} \hat{e}_\Omega
$$

$$
\dot{\hat{e}}_\Omega = \frac{K_e + g_2 J}{J} \hat{e}_i - \frac{T_{\text{load}}}{J} \quad (15)
$$

Adopting a fast observer dynamics $\dot{\hat{x}} \geq 0$ and resorting to a variable substitution in previous equations system the expression of estimated load
torque will be:

\[
\hat{T}_{LOAD} = \frac{K_p \left( K_s - g_2 J \right) + F_c \left( R_s + g_3 L_s \right)}{K_p} e_u \quad (16)
\]

*Load torque as state variable.* Considering the load torque as a disturbance variable state the system order of state-space representation increase and eq.9 and 10 are written as follows:

\[
\begin{align*}
\dot{x}_a &= A_a x_a + B_a u \\
y_a &= C_a x_a
\end{align*}
\]

(17)

where:

\[
x_a = \begin{bmatrix} i_s \\ \Omega \\ T_{LOAD} \end{bmatrix}; \\
A_a = \begin{bmatrix} -\frac{R_s}{L_s} & -\frac{K_s}{L_s} & 0 \\ 0 & 0 & -\frac{1}{J} \end{bmatrix}; \\
B_a = \begin{bmatrix} 1 \\ 0 \\ 0 \end{bmatrix}
\]

(18)

\[
y_a = i_s; \quad C_a = [1 \quad 0 \quad 0]
\]

The extended observer for this case is depicted now by the modified relation contained in eq.11:

\[
\hat{x}_a = A_a \hat{x}_a + B_a u + G_a C_a (x_a - \hat{x}_a)
\]

(19)

where \( G_a = [g_1, g_2, g_3]^T \) and \( \hat{e}_a = (A_a - G_a C_a)e_a \) is the estimation error state-space equation. The mathematical model of the load torque estimator in the extended version is:

\[
\begin{align*}
\dot{\hat{x}}_1 &= -\frac{R_s + g_3 L_s}{L_s} \hat{x}_1 - \frac{K_s}{L_s} \hat{x}_2 + U_d + g_2 I_d \\
\dot{\hat{x}}_2 &= \frac{K_s}{J} \hat{x}_1 - \frac{1}{J} \hat{x}_3 + g_3 I_d \\
\dot{\hat{x}}_3 &= -g_3 \hat{x}_1 + g_3 I_d
\end{align*}
\]

(20)

Proceeding in same manner described before and knowing that \( \hat{x}_1 = \hat{i}_s; \quad \hat{x}_2 = \hat{\Omega}; \quad \hat{x}_3 = \hat{T}_{LOAD} \) the estimated load torque will be:

\[
\hat{T}_{LOAD} = g_3 \int e_u dt
\]

(21)

**III.3. Friction identification**

The identification of LuGre model parameters has been performed using a step-by-step procedure. Considering figure 7 which represents a velocity-friction static map the experimental identification trials have been done as follows.

- **step1** – Coulomb and viscous friction, \( F_c, \alpha_s \); gross motion \( v \gg v_s \) assumed (neglected Striebeck effect) and friction steady-state behavior i.e.

\[
F \approx F_c \quad \text{sgn} \quad v + \alpha_s v
\]

(22)

- **step2** – stiction and Striebeck velocity, \( F_s, v_s \); motions under controlled oscillations, small velocities \( v \ll v_s \) assumed, meaning:

\[
F \approx F_c + (F_s - F_c) e^{-v/v_s}
\]

(23)

- **step3** – stiffness and damping, \( \sigma_1, \sigma_2 \); motions in presliding regime, very low velocities \( v \ll v_s \).

All experimental tests for data selection have been made for DC system for motion control application in a classical structure with two inner loops (closed loop identification) – see figure 8.

![Fig.8. Simulink® diagram for LuGre parameters identification](image)

**IV. EXPERIMENTAL RESULTS**

First of all, a simulation scheme (classical DC motor driven by a sinusoidal reference in the presence of a disturbance torque imposed like a friction torque via LuGre model) is used to validate the estimation accuracy for the three types of observer developed and configured in previous section of the paper (filter estimator type and the 2 determinist approaches). The authors have emulated the worst working regime for the systems with friction – velocity reversal – to emphasis the observers’ capacity to be tuned properly in order to obtain a further accurate compensation for friction.

In figure 9 are illustrated the simulation results of the system for friction torque, velocity, position and estimation error of the filter estimator. Moreover, the stick-slip motion and stiction has been pointed out by velocity characteristic.

![Fig.9. Friction torque and estimation error for a first order torque estimator (up) and reference signal vs. velocity and position (down)](image)
The next simulations are focused on friction compensation strategy using non-model-based observers. Before using friction compensation, one must analyze its effects in a closed-loop system and avoid situations in which this compensation may result in adverse response characteristics, such as limit cycles. Choosing the reference position as $x_r = 0.05$ - small tracking scale – the LuGre friction model clearly predict limit cycles as have been experimentally observed in literature (Canudas de Wit et al., 1995). Fig.10 shows the main simulation results for the DC servo drive with a PID controller inducing limit cycles.

![Fig.10. Simulation of PID position control limit cycles: position and velocity (left) and friction-position hysteresis for limit cycles (right)](image)

The proposed scheme for friction compensation has been already reviewed in section 3 – fig.4. The block diagram presented herewith depicts a typical closed-loop servomechanism system in which friction compensation has been added. In this system, feedback consists of a state-feedback part ($H(s)$ is a PID controller) and of an off-line non-model-based friction compensation term, $\hat{F}$ (Js$^2$ as feedforward part is also added to complete a classical feedback compensation strategy).

It must be noticed that the PID controller has a filtered integral action for pole placement and stability reasons. The PID controller tuning rules and observer gains choice has been made according to dynamic constrains imposed for observer response performance versus the transient behavior of DC motor.

Figure 11 shows the position response after compensation applying and also the friction rejection with a detailed zoom-in capture. The PID parameters and observer gain are also included within figures (tau refer the I filter constant time).

![Fig.11. Simulation results (position and friction torque) for a PID compensation strategy using a first order disturbance torque estimator](image)

V. CONCLUSIONS

This paper illustrates the development of a RTFS based on HIL structure to test friction behavior and a compensation strategy proposed by authors for a DC servo tracking drive. Using the DS1103 PPC controller board (dSPACE) it was possible to realize a friction simulator with good performances concerning the real-time controlling and monitoring facilities.

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