NON-CONVENTIONAL METHOD FOR FRICTION COMPENSATION IN DC DRIVE POSITION TRACKING

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Abstract: Friction, especially its nonlinear components, may degrade the tracking performance of any servo drive systems. Focused on high precision positioning of DC electrical drive the paper proposes an easy method to compensate for the friction by using a non-model based compensator developed around a simple control structure. The approach to inhibit the friction influence is based on a compensation signal provided by the mathematical model of the servo positioning DC drive in which the load torque (including friction component) is considered zero. Build on the difference between the command signals of real servo tracking drive and, respectively, of the same servo drive, but no load status, the compensation variable depends only on the usual and measurable states of an electrical drive (current and velocity)   The control problem of the servo drive with friction is solved via the gain scheduling of a P controller parameter as function of position reference scale. Good results concerning the tracking errors are achieved for large range of reference test signals, including the micrometer scale.

Keywords: friction, non-model-based friction compensation, stiction, Stribeck effect, tracking;

1. INTRODUCTION

Being the major cause of positioning error, friction should be well modeled and compensated to achieve precise control performances for servo systems. In (Armstrong-Helouvry, *et al.*, 1994), a detailed discussion on friction phenomena and its compensation methods in different systems is presented and many references are cited. As a thumb rule, control strategies that attempt to compensate for the effects of friction are classified in model-based and non-model-based techniques.

Model-based approach. Its implementation requires choosing the appropriate friction model, identification of its parameters, and finally friction compensation using the identified model. This also assumes that the force or torque actuation of adequate bandwidth is available and stiffly coupled to the friction element. Then, friction compensation can be obtained by adding the opposite of the predicted friction to the control signal.

Non-model-based approach. In this case no friction model is used and the compensation is achieved by:
1) Changing the position controller parameters;
2) Applying dither;
3) Using a non-model-based observer for friction.

This paper deals with a non-conventional friction compensation method by adding a compensation component to the control signal of a tracking DC servo drive. Processed as difference between control signals of real electrical drive with friction and, respectively, of same drive but without any disturbance torque, the compensation signal doesn't need any information about friction. Thus, any kind of friction observer is obsolete. The control law synthesis is accomplished by a suitable gain scheduling for a P position controller placed before a filter. The filter design was made considering the mathematical model of the DC tracking servo, under

the assumption of no load torque.

The paper is organized as follows. Section 2 presents the friction model chosen to emulate the friction behavior. In section 3 the control structure for friction compensation is presented. Some simulation results are illustrated in section 4 along with several conclusions concerning the compensation method approached, all presented in last section of the paper.

2. FRICTION MODELING

The mathematical model chosen for describing the friction nonlinear behavior was the LuGre model which best captures friction phenomena experimentally observed (Canudas de Wit, *et al.*, 1995). Hereafter, the equations depicting the model and the meaning of state variables and parameters will be presented. Thus, the following equations hold for friction model.

$$(1)\ \frac{dz}{dt} = v - \frac{\sigma_0 |v|}{g(v)} z$$

$$(2)\ F = \sigma_0 z + \sigma_1 \dot{z} + \alpha_2 v$$

where z is the pre-sliding displacement or, more accurately, the average deflection of the bristles, v is the velocity, σ_0 and σ_1 are the stiffness of bristle and, respectively, the damping, α_2 is viscous friction and F denotes the friction . The function $g(v)$ is the function describing Stribeck's effect. A parameterization proposed for $g(v)$ is given as follows:

$$(3)\ g(v) = F_C + (F_S - F_C)e^{-(v/v_S)^2}$$

where F_C is Coulomb force, F_S static friction and v_S is Stribeck's velocity. Whereas the LuGre model includes the Stribeck effect and also offers a smooth transition at velocity reversal, the reason of its choice for next investigations in control of systems with friction is fully justified.

3. COMPENSATION STRATEGY

Generally, the load torque T_{LOAD} of an electrical drive (DC or AC) is the sum of inertial torque $T_{inertia}$, external torque T_{ext}, and friction torque $T_{friction}$. Considering a classical DC electrical servo drive for a numerical positioning system it can be assumed that the friction torque prevails and the statement $T_{LOAD} \simeq T_{friction}$ works out especially for low velocities and crossing zero velocity regimes. Starting from the model equations of DC motor (eq.4) and supposing

there is no load torque, $T_{LOAD} = 0$, the transfer function of the system can be written as relation of eq.5.

$$(4)\ \begin{aligned} u_A &= R_A i_A + L_A \frac{di_A}{dt} + K_\phi \omega \\ J\frac{d\omega}{dt} &= K_\phi i_A - T_{LOAD} \end{aligned}$$

$$(5)\ H_{Mcc}(s) = \frac{1}{K_\phi (T_m T_A s^2 + T_m s + 1)}$$

In eq.5, the parameters involved are: $T_m = \frac{JR_A}{K_\phi^2}$, the electromechanical time constant and $T_A = \frac{L_A}{R_A}$, electrical time constant. For system positioning, the control law adopted consists in a P controller and a filter $H_R(s)$ along with a causal imposed system around the inverse DC motor model, $H_{Mcc}^{-1}(s)$. The P controller and filter parameters are adjusted according to achieve a stabilized and accurate positioning for tracking servo. In upper side of fig. 1 is presented the structure of the position controlled DC servo drive with no load torque with all blocks mentioned before. Lower side of fig.1 illustrates the real simulated process of DC servo tracking in which the friction is emulated through LuGre model and represents the load torque.

The transfer function of the filter is a first order system:

$$(6)\ H_R(s) = \frac{1}{1 + sT_F}$$

where T_F is the time constant of the filter being the same for both controlled structures (with friction torque and with no load torque).

The causal system $H_{Mcc}^{-1}(s)$ is conceived like a resonant compensator around natural frequency of DC motor system ω_N.

$$(7)\ H_{Mcc}^{-1}(s) = \frac{s^2 + 2\xi\omega_N s + \omega_N^2}{s^2 + 2\xi\omega_C s + \omega_C^2}$$

where ω_N is obtained from eq.5 as:

$$(8)\ \omega_N = \frac{1}{\sqrt{T_m T_A}}$$

and the ratio ω_C / ω_N is chosen in such way to ensure a good stability and high positioning performance (fast response and minimized tracking error).
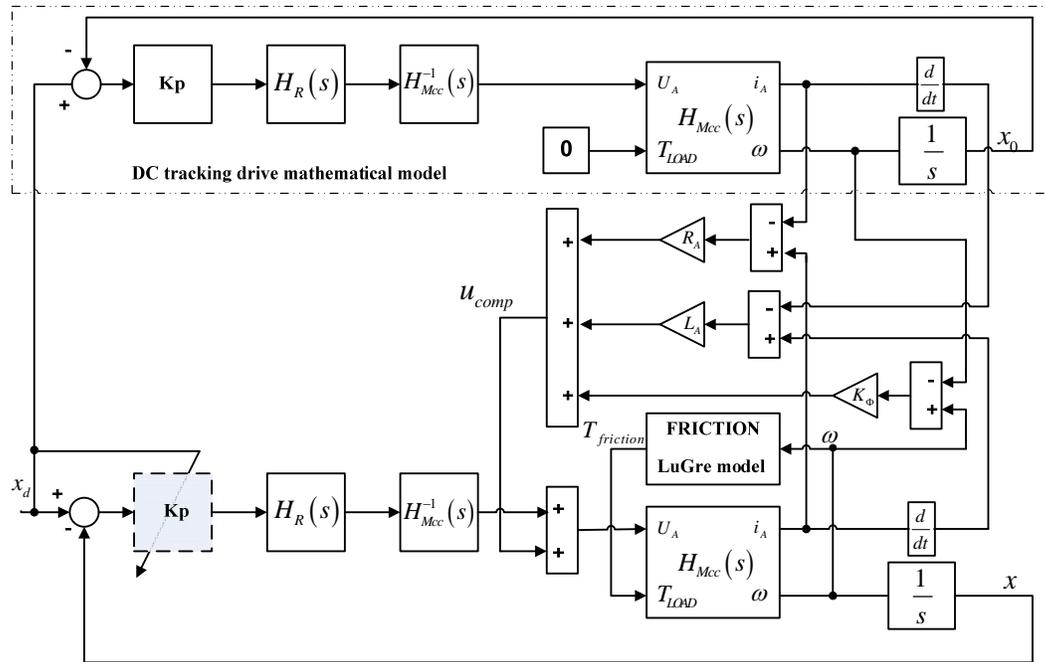
Fig.1. Block diagram of friction compensation scheme

Herein, like in case of filter synthesis $H_R(s)$, all the calculus is made for the mathematical model of no load DC servo tracking and it is also available for real servosystem with friction.

In both cases concerning the filter and the compensator synthesis, their design must be a compromise between the degree of noise rejection required in high precision positioning and the detrimental effects of the filter and the resonant compensator on the speed of the position feedback loop.

The friction compensation signal u_{comp} is processed like an electrical quantity resulting from the difference between the command signals U_A of each structure considered (mathematical no load DC tracking servo and real positioning servo system). Both signals are deduced using the measurable state variables of the system (the current and the velocity). The following equations describe the methodology to get the compensation signal:

$$u_{A0} = R_A i_{A0} + L_A \frac{di_{A0}}{dt} + K_\Phi \omega_0$$
$$(9)$$
$$\hat{u}_A = R_A \hat{i}_A + L_A \frac{d\hat{i}_A}{dt} + K_\Phi \hat{\omega}$$

$$(10) \quad u_{comp} = \hat{u}_A - u_{A0}$$

where 0 index refers the state variables of the no load DC drive model. The variables involved in 2nd relation of eq.9 are measured via transducers or estimated by an observer.

Applying this friction compensation signal, the real servo drive acts like the mathematical model within the load torque is eliminated. The only issue to take into account is the control methodology for different position reference scales. The way to accomplish this goal is using a gain scheduling P controller. Initially, the parameter K_P of the controller is determined according to the position closed loop stability and good performance imposed for basic structure of servo drive (no load mathematical model). By varying parameter K_P according to the operating point of real system a flexible and easy way of controlled positioning is achieved. It has been noticed that good results are obtained for a linear gain scheduling related to the magnitude order of position reference. This fact eases much the control matter and leads to high precision of positioning for large scale of position trajectories.

4. SIMULATION RESULTS

In fig.2 are presented the position responses for a reference trajectory chosen in micrometer scale and having such profile to force the velocity within worst friction regimes (small velocities combined with velocity reversal – see fig.3). A zoom capture is presented in the right of fig.2 to underline the tracking errors for the 2 controlled positioning: with and without compensation.
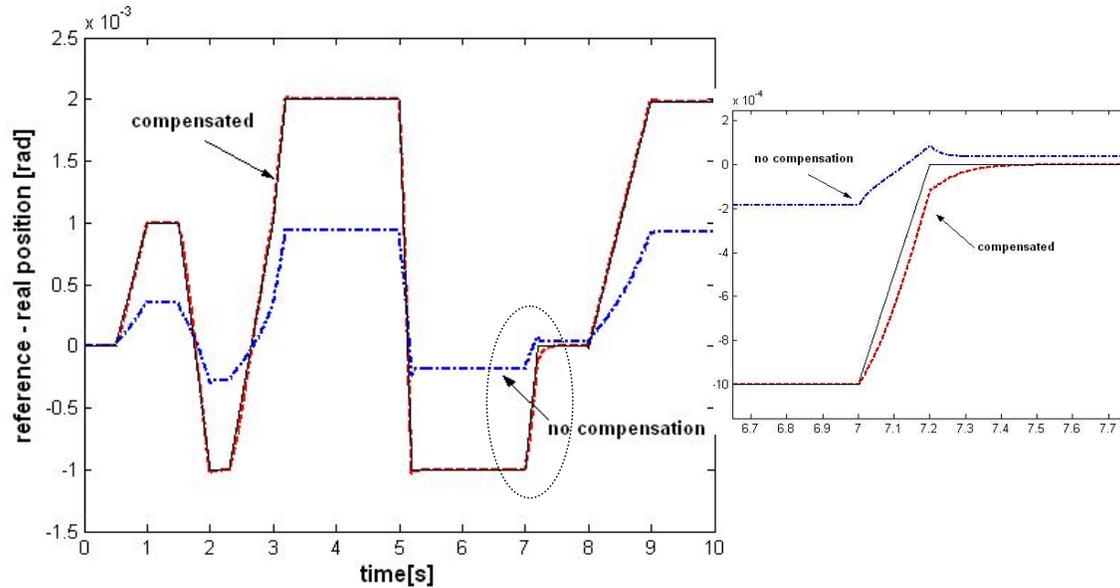
Fig.2. Simulation results for trajectory reference (position response with and without compensation)
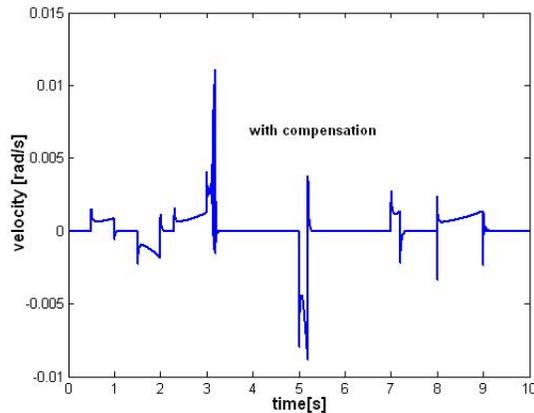
Fig.3. Velocity for controlled positioning with compensation signal applied

In fig.3 it must be noticed that the velocity range is 10 times smaller against the value of Stribeck velocity (chosen by LuGre model at 0.1rad/s). It implies that the set point of this test is placed in the vicinity of zero cross velocity which represents the most unpredictable regime for friction behavior. Nevertheless, all the other tests performed to verify the validity of compensation strategy proposed have passed regardless the magnitude order of position references. The choice of P controller gain was made according to relation:

$$(11) \quad K_p x_d = \left(10^{-1} \div 1\right)$$

where x_d is the position reference. A fuzzy inference system or a look-up table can be used to provide a smoothly varying function that maps the value of position reference onto appropriate controller gain values with respect of eq.11.

5. CONCLUSIONS

A non-conventional compensation strategy to alleviate the friction impact in a classical DC positioning drive has been discussed in this paper. The method approached is easy to implement, doesn't need a model-based observer for friction and also, doesn't need the classical inner loops for current and velocity (nevertheless, for some circumstances of working regime current limitation block is required to alleviate great gradient of velocity – see fig.3). The positioning control requires just a P controller which adjustment is easy to accomplish in order to provide a flexible and robust control structure.

6. REFERENCES

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