AN OBSERVER BASED VELOCITY CONTROL APPROACH FOR A DC SERVO-DRIVE SUBJECT TO FRICTION

Anthony Chidolue Nnaji
Department of Electrical and Electronics Engineering,
Enugu State University of Science and Technology, Enugu Nigeria
All correspondence to; chidoluen@yahoo.com

Abstract
PID controllers for position and velocity control have become very popular in systems automation and control of industrial processes. This is easily attributable to their robustness, easy of design and implementation and simplicity. In this paper, a PID-based linear controller is designed for the position control of a DC motor driven system. The mathematical model structure of the DC motor with gear and load discs was obtained and used for position control simulations in Matlab/Simulink environment.Parameters of the PID controller were son chosen to meet performance criteria. The simulation results of the position control showed great correspondence to the expected reference position when compared to the case where there was no form of control to the output position.
Index Terms: Linear controller, Position control, Simulation, DC motor.

Introduction

DCservo actuating systems have been widely used in the process and automation industry. This wide applicability is due to their low cost, design ease and their being simple to model and implement position or velocity control [1], [2], [3], [4]. Contacting mechanical systems experiencing relative motion such as the DC servos are usually influenced by friction nonlinearity. The friction phenomena has been defined as the tangential force between two contacting surfaces in contact experiencing a relative motion [5], [6], [7]. The first recorded attempt in the study of friction was by Da Vinci, though the first formulation of the mathematical relationship between friction and load was by Coulomb. This later became the popular Coulomb friction

A. C. Nnaji, is with Electrical and Electronic Engineering Department, Enugu State University of Science and technology,

Enugu, Nigeria

law. Friction is a nonlinear phenomenon with multi-faceted effect on systems. Some of these effects are limit cycle oscillation, wear, tracking errors in position and velocity control systems [8], [9], [10]. Friction features manifest mainly in two distinct regimes namely the pre-slide and gross-slide. In the pre-slide regime, it is seen as the force experienced by a system under the influence of an external force until the contacting surfaces are on the verge of relative motion to each other. On the other hand, in the gross-slide regime, there is relative motion between the two surfaces in contact. In the gross-slide the friction force is a function of the velocity while in the pre-slide it is a function of the displacement. Some of the main features of friction are; Stribeck effect, varying breakaway force, pre-slide hysteresis with non local memory, frictional lag and non drift property [11], [12], [13]. The DC servo systems are nonlinear due to system friction, however a linear version can often be obtained and used for control purposes especially for the region of operation within the linear friction model could be applied and high precision is not paramount. Thus the effectiveness of the linear controller are often more pronounced in the pre-slide regime and velocity reversals. As a result, the design and implementation of a model based friction compensation scheme adequate for simulation analysis and control of the servo drive is desired for accurate control of the system in high precision control systems. The influence of the non-linear friction on the DC servo driven system results in tracking errors in the position and velocity, limit cycle oscillations and wear.

Linear controllers, usually of the PID type and their combinations are effective for the control of linear systems as they help achieve system stability, error reduction and reduction in transient response time. This is the main reason they are widely used for such control actions in system automation and industrial processes. However, in the presence of system non-linearity such as friction, such linear controller are ineffective for precision control of such systems. this ineffectiveness is more pronounced in the pre=slide friction regime and the reversal or zero velocity region.

In this paper a model based friction observer in combination with a linear controller for velocity control of a DC servo driven system is implemented.

The layout of the paper is thus: In section II, a model for the Dc servo drive was obtained.

Friction models and their observers used for control implementations are described in section III. In section IV, velocity control simulations are performed, while the results of the various control simulations are discussed in section V. Finally, section VI contains the conclusion of the paper.

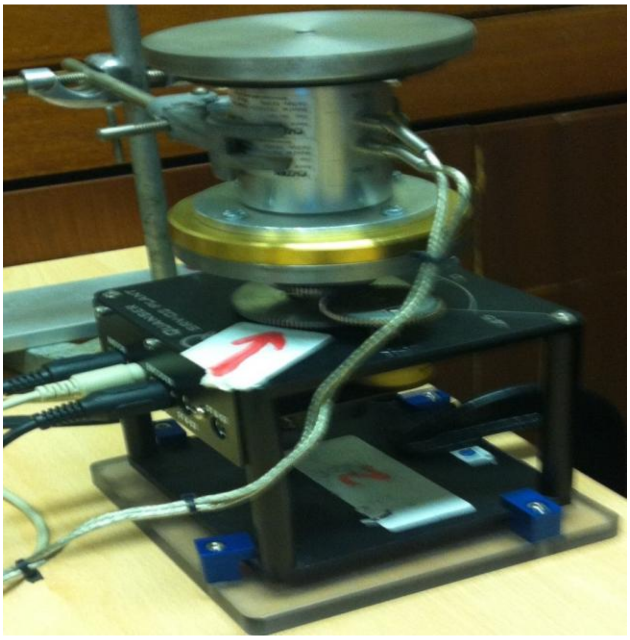

Fig. 1. The DC-servo motor friction test-bed

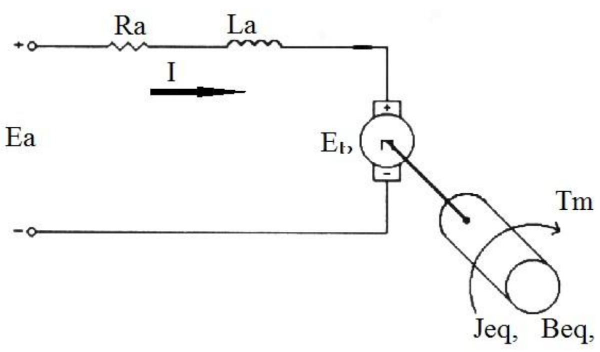

Fig. 2. The schematic representation of the DC-servo motor

Model of the DC Servo System

The DC motor driven system for the velocity control implementation is shown in figure 1.

The general model of the electric motor (Permanent Magnet DC type) driven system can be obtained from the schematic shown in figure 2. The model of the servo driven system [20] is given as

$$\frac{\omega_l}{E_a(s)} = \frac{AK_m}{(J_{eq}s + B_{eq})R_a + (AK_m)^2} \tag{1}$$

TABLE I

PARAMETERS OF THE FRICTION TEST-BED

Parameter	Description	Value	Unit
K_m	Motor torque constant	0.00767	N-m/A
A	Gear box transmission ratio	70:1	-
J_l	Moment of inertia of load	$2.6583x10^{-5}$	kgm^2
Jm	Moment of inertia of the motor armature	$3.87x10^{-7}$	kgm^2
K_b	Back emf constant	0.00767	N-m/A
R_a	Armature resistance	2.6	ω

which is a first order system.

where E_a is the motor armature voltage of the motor, J_{eq} is the equivalent inertia and B_{eq} the equivalent frictional damping, K_m is Motor torque constant, ω_l the angular velocity, A is the gear transmission ratio and R_a the armature resistance.

Equations 1 is the transfer functions for velocity output. The frictional damping term *Beq*comprising the frictions in the motor, gear and load is here modeled as a linear function of the output velocity. This linear friction function will be replaced by the relevant models for control simulations. The parameters of the DC motor system are contained in the specification sheet as
supplied by the manufacturer, however relevant values are as presented in table I.

Friction Models and their Observers

Many models of friction exist ranging from the simple static models to the complex dynamic ones [14]. Static models do not capture friction dynamics even though they are simple to implement. The dynamic model of friction capture friction dynamics to varying degrees depending on the complexity of the model. The LuGre model [15] has gained much popularity in control field due to its model structure being simple and the ease with which the parameters can be
identified. As a result, it is often used for position or velocity control simulation and experiments. The LuGre model is given as

$$F_f = \sigma z + \alpha \dot{z} + f_v v \tag{2}$$

$$\dot{z} = v - \sigma \frac{|v|}{g(v)} z \tag{3}$$

$$g(v) = F_c + (F_s - F_c)e^{-(\frac{v}{v_s})^2} \tag{4}$$

where $F_f$is the friction force, $f_v$the viscous damping coefficient, z the micro displacement between the surfaces (or bristle deflection), σ is the stiffness of the material, α is the material damping coefficient, $g(v)$ is the non-linear term modeling the Stribeck effect at low velocities, F_c is the coulomb (kinetic) friction force, F_s is the static friction force (the minimum amount of force required to initiate relative motion between the two bodies in contact), v is the relative velocity between the surfaces in contact and v_s the Stribeck velocity.
Despite its wide acceptance the model does not capture adequately the non-local memory

hysteresis friction in the pre-slide regime [16]. It also does exhibit position drift when subjected to an external vibration force of less value than the stiction force [17].

The Generalised Maxwell Slip (GMS) friction model [18] is a multi-element, multi state model unlike the LuGre which is single state. Two different sets of state equations describe the pre-slide
and gross-slide regimes of friction. In the pre-slide regime

$$\dot{z}_i = v \tag{5}$$

and for slipping

$$\dot{z}_i = \mathrm{sgn}(v) C_i \left(1 - \sigma_{0i} \frac{z_i}{g_i(v)}\right) \tag{6}$$

i= 1,2,...,N

During stick, the elements remain stuck till the deflection $z_i = \frac{1}{\sigma_{0i}} g_i(v)$, and also during slip, the element continues to move until the velocity v goes through zero (reversal). The friction force
is thus given as;

$$F_t = \sum_{i=1}^{N} (\sigma_{0i} z_i + \sigma_{1i} \dot{z}_i) + f_v v \qquad (7)$$

where z_i is the deflection of the ith element, $g_i(v)$ is the velocity weakening Stribeck function, C_i is an attraction parameter showing how rapidly the z_i tracks changes in $g_i(v)$, σ_{0i} is the stiffness of the ith element and N is the number of massless bristle elements.

The GMS model is capable of modelling most friction features including the non-local hysteresis friction and it does not exhibit position drift as does the LuGre model. The applicability of this model is however limited due largely to difficulties in implementation and parameter estimation. This is because of the large number of parameters needed to model efficiently friction features. The friction observers used in this paper are based on the LuGre and the GMS models are shown to be dissipative [19]. In practice, it is usually the norm to combine a linear controller with the friction model observer for velocity control implementations. Stand alone linear control of DC servo driven system is usually not effective for high precision control systems due to the friction non-linearity. This is because linear controllers as the name suggests are best suited for the control of linear systems. The simulation block diagram for the velocity control implementationof the system under investigation is shown in figure 4.

Implementation Example of Velocity Control

To illustrate the ability of model-based friction compensation, consider the servo system givenas

$$Js\omega_o = u - F_f$$

Where J is the inertia of the system, ω_o the output velocity, u the control command and F_f system friction modelled by the LuGre friction model. Given the absence of an input velocity (i.e $\omega_r = 0$), though the system is subjected to a disturbance velocity signal as shown in the figure 3. Thus the system output velocity is $\omega_o \neq 0$. Since the reference velocity is zero, the objective is to eliminate the error between input reference ω_r and the response ω_o so as to make $\omega_o \longrightarrow 0$. The choice for the nature of the disturbance signal was informed by the fact thatfriction is most disturbing near-zero velocities and velocity reversals. As such the disturbance signal is chosen to be low enough with many reversal velocities. To capture the capability of the LuGre and GMS friction model structures to compensate friction effects in this region, a disturbance signal of the form shown in figure 3 was used as suggested in [19]. This disturbance signal was obtained by passing a white noise signal with zero mean and deviation of unitythrough a filter $H(s)$ of the form

$$H(s) = \frac{1}{40} \frac{s^2}{(s + 0.2)^4} \qquad (8)$$

For the control implementation of the observer based feedback control system shown as figure 4, the values of the linear controller are $P = 4$ and $I = 16$. First a PI-control implementation in the absence of friction was performed and the results illustrated in figure 5, while figure 6 captured the linear controller implementation in the presence of friction. From these two figures

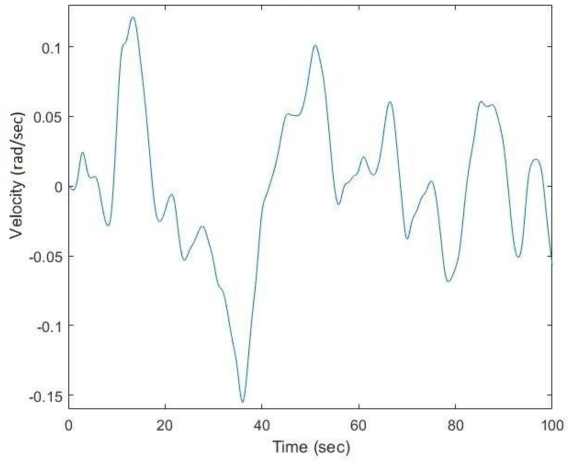

Fig. 3. A disturbance signal used for the velocity control simulations

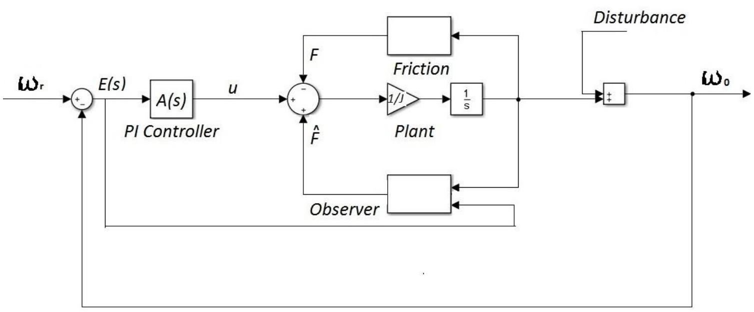

Fig. 4. Block diagram of the observer-based friction compensator for the velocity control simulation example

it is clear that friction effects on the system is more pronounced in the near zero and velocity reversals and this rendered the linear PI controller inadequate for high precision velocity systems control. The error measure was more pronounced around zero velocities as compared with the case of no friction. As a measure of the effectiveness of the control scheme, mean square error (mse) and the maximum error values were used. For all the velocity control schemes studied these two measures were obtained and presented in table II.

Subsequently, the LuGre model observer was introduced as shown in the figure 4 to provide added control for the non-linear friction and the results shown in figure 7, while the mse and maximum error values are in table II. In the same token GMS model-based observer figure 8, the mse and maximum error were also presented in the table II. The parameters for the

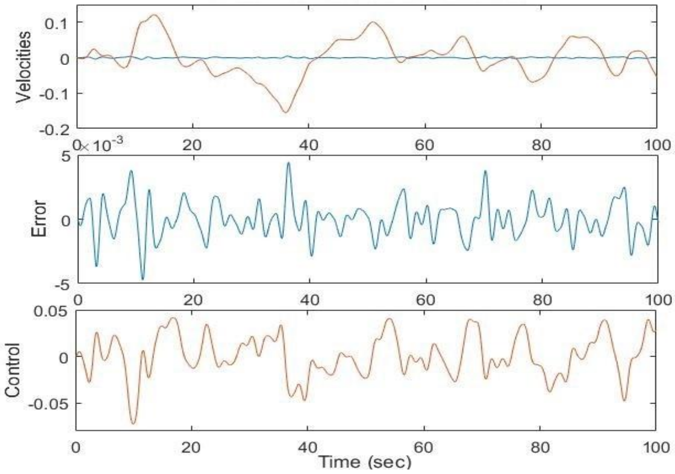

Fig. 5. Linear controller performance in the absence of friction: top- Disturbance (red), and output (blue) velocities, middle-

The control error signal, and bottom- The PI control signal

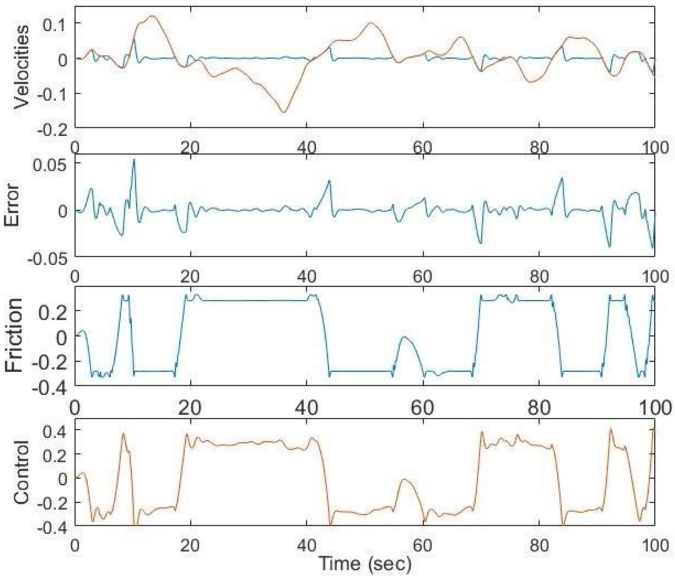

Fig. 6. Linear controller performance in the presence of friction: top- Disturbance (red), and output (blue) velocities, mid-upperThe error signal showing increased error due to friction, mid-lower- The system friction, and bottom- The PI control signal

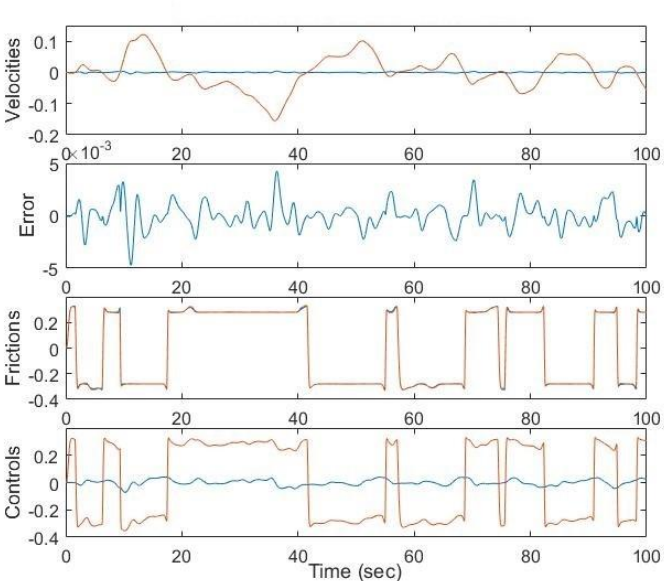

Fig. 7. Observer based velocity control of a system subject to friction using the LuGre model: top- Disturbance 'red', and output 'blue' velocities, mid-upper- The error signal showing increased error due to friction, mid-lower- The system friction 'red' and estimate friction 'blue', and bottom- The linear control signal'blue' and the modified control law 'red'

TABLE II

PERFORMANCE INDICES FOR THE VARIOUS CONTROL SCHEMES STUDIED UNDER VELOCITY CONTROL

Controller type	mse $(x10^{-6})$	Maximum error $(x10^{-4})$
PI no Friction	1.7187	47
PI with Friction	101.42	543
LuGre model observer	1.3738	47
GMS model observer	1.6553	50

LuGre model-based observer used for the simulation are: $F_s = 0.33$, $F_c = 0.28$, $v_s = 0.01$, $\sigma_1 = 20$, $\sigma_0 = 1000$, and f_v= 0.0176. For the GMS model-based observer using 4 slip elements: $F_s = 0.33$, $F_c = 0.28$, $v_s = 0.01$, $\sigma_{0i} = 25$, $\sigma_{1i} = 0.0015$, $C_i = 0.025$, f_v= 0.0176, and α_i= 0.25.

From the table II it is shown that with the LuGre and GMS models, the compensation scheme was able to reduce the error originating from frictional effects in the system as indicated in the table.

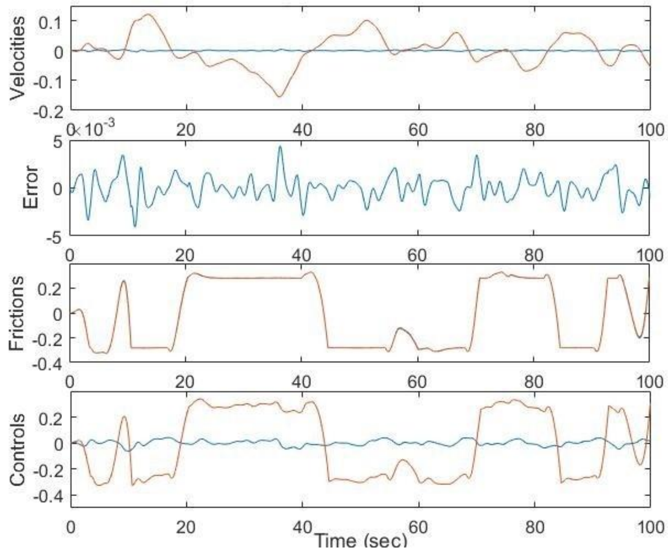

Fig. 8. Observer based velocity control of a system subject to friction using the GMS model: top- Disturbance 'red', and output 'blue' velocities, mid-upper- The error signal showing increased error due to friction, mid-lower- The system friction 'red' and estimate friction 'blue', and bottom- The linear control signal'blue' and the modified control law 'red'

Analysis of Simulation Results

The low velocity signal used for the simulation was necessary so as to capture the effects of friction non-linearity especially at low velocities and reversals. From the simulation results of figure 5, the linear controller implementation was effective for the system velocity control as the controlled error result showed. This is typical of linear control in the absence of friction. The control of systems with friction is most problematic in the low and reversal velocity regions and as such linear controllers are not effective when used as stand-alone controllers for nonlinear systems involving these velocity regions. This scenario was shown in figure 6 where the introduction of friction non-linearity to the system rendered the otherwise appropriate linear controller inefficient for velocity control. This deterioration in the control as measured by the mse and maximum error is of a unit order (10 times worse) than the case for no friction. The control effort also was shown to increase by a factor of about 10. With a model-based friction observer to compensate friction non-linearities incorporated in the control loop of figure 4, improvements are easily observed. From the results of figures 7 and 8, the controlled error was seen to be well below the error obtained for the cases of no friction of figure 5 and the PI control in the presence of friction, figure 6. The control signal needed to achieve this improved control is quite small also. The error reduction of the model-based observer compensation were also better as seen in the table II and figures 7. It should be noted that the improvement of the modelbased compensation strategy over the simple control even in the case of no friction suggests the capability of model-based compensators to improve linear system performance also. Low velocities and reversal point velocities control measures of the compensators from the various performance results of figures 5 to 8 indicate the ability of the observer based compensation approach for superior velocity control result than the PI linear controller schemes. A critical look at the error results in the relevant figures and tables also indicate that the performance improvement of the observer based compensation to be mostly in the low and reversal velocitieswhere friction is mostly non-linear.

Conclusion

Velocity control of a DC servo actuating system under friction influence using a combination of friction observer and linear PI controller was presented. The implementation results in this paper showed the remarkable improvement in system performance when an observer is integrated in the control loop rather than using only the PI controller. This thus buttresses the fact that linear control approach to systems with non-linearity such as friction are not very effective for highprecision control implementations.

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