NONLINEAR FRICTION COMPENSATION DESIGN FOR SUPPRESSING STICK SLIP OSCILLATIONS IN OIL WELL DRILLSTRINGS

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Abstract: Friction appears in all mechanical systems and has a significant impact on control. This paper presents simple but effective nonlinear friction compensation method for a drillstring system, by considering the stick- slip motion of the bit on the bottom hole assembly (BHA) in oil wells. Due to nonlinear problem at near- zero velocity where the torque exerted on bit is a function of the bit speed, the drillstring controlling is not an easy task. The control strategy proposed in this work involves a combination of a feedback model based compensation of friction and linear controller which enables to reduce the stick slip oscillations. The effectiveness of this approach is demonstrated through the resultants simulations.

Keywords: Control systems, nonlinear systems, stick-slip oscillations, drilling system, friction compensation, feedback.

1. INTRODUCTION

In the drilling processes there are different types of vibrations, one of them and for which we are interested in this work is torsional oscillations (Stick-Slip). These oscillations are induced by nonlinear frictional torques between the drill bit at the rock surface. The amplitude of this torsional excitation can be two to four times the target or average angular speed (typically between 3.14 and 15.7rad/sec) set by the top-drive. The torsional excitation can give rise to enormously destructive fluctuating torques in the drill-string that, once out of control; invariably cause damage to the bit or drill-string. Even small amplitude slip-stick vibrations are thought to be a major cause of bit wear. Different control solutions have been presented in the literature to combat this instability as by using the H_∞ technique on the local linearized model about suppressing stick-slip in oil well drillstrings (Serrarens *et al.*, 1998), by applying optimal control by (Smit, 1995) and by using classical controller as PI (Tucker and Wang, 1999).
A wide variety of studies have been reported, which formulate nonlinear friction models, identifying it's parameters, and suggest compensation techniques for the friction. The friction model has been widely studied by numerous researchers (Candus *et al.* 1993).

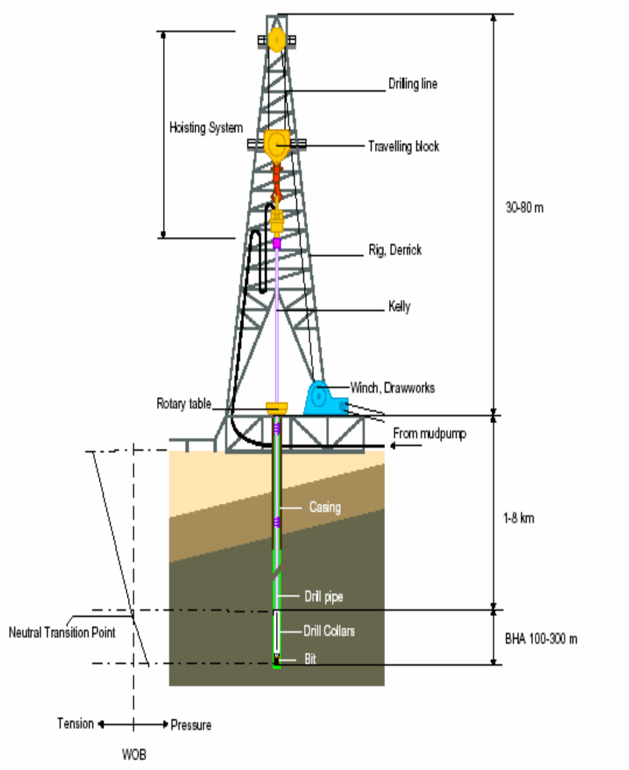

Fig. 1. Drilling equipment

The control strategy proposed in this paper consists of using compensation for the friction in drillstring system and together with a nonlinear controller.
The approach is based on the principle that it only feds back the friction in the control loop. The PI controller is used for reducing the steady state and the tracking error in the system.
Nevertheless, it has never been applied to drillstring system, at least to our knowledge. The effectiveness of this proposed approach is demonstrated by simulations on a drilling system.
An idealized system of a rotary drilling installation is shown in Figure 1, which one may be find in (Serrarens, 2002). Starting at the top end of the complete system, a heavy weight disc-shaped mass called the *rotary table* is driven by an electric motor. The rotary table acts as flywheel to approximately maintaining a constant speed of the subsequent system elements. These elements can be identified as the drillstring, the BHA and finally the very bottom end of the structure, a cutting tool called a *bit*. The medium to transport the energy from the motor to the bit is formed by a drillstring, mainly consisting of a number of relatively thin walled pipes.
The remainder of this paper is organized as follows. Section 2 presents a brief description of the elements of the rotary drilling model. In Section 3, nonlinear friction technique with PI controller is proposed. Friction method is applied on rotary drilling system for designing a nonlinear controller in Section 3. Simulation results are shown in Section 4 for the case of step input. Finally, Section 5 ends with some conclusions.

2. ROTARY DRILLING MODELLING

Figure 2 shows the main components of the model which contains two damped inertias mechanically coupled by an elastic intertialess (drillstring). The drillstring is modelled as a continuous mechanical system with infinity natural resonance frequencies. It is assumed that the drillstring is homogenous along its entire length and simply modelled as a single linear torsional spring with stiffness k.
The lowest part of the drillstring, BHA is composed of thick-walled tubulars loaded in compression, and, without buckling; it provides weight on the bit (WOB) required to generate sufficient cutting force. The BHA can be several hundreds of meters in length, and often contains specialized downhole tools. The connection of the drillstring to the rotary table is made by an interface element called the *Kelly*. This square or hexagonally shaped rod connected to the top of the drillstring, which fits into a square or hexagonal hole in the rotary table, allowing vertical motion of the drillstring.
The rotary drilling system consists of two main components:

- The drillstring and BHA
- The driven system

2.1 The drillstring and BHA

A simple model of torsional drillstring vibrations is obtained by assuming that the drillstring behaves as a torsional pendulum, i.e. the drill pipes are represented as a torsional spring, the drillcollars behave as a rigid body and the rotary table rotates at a constant speed. The corresponding equation of motion may be found in (Serrarens *et al*., 1998) and is

$$J_1\dot{\Omega}_1 + C_1\Omega_1 - k\phi = -T_{tob}\left(\Omega_1\right) \tag{1}$$

Furthermore, the quantities $\phi = \varphi_2 - \varphi_1$, $\Omega_1 = \dot{\varphi}_1$ and $\Omega_2 = \dot{\varphi}_2$ are defined, where φ_1 is the angular displacement of the bit and φ_2 is the angular displacement of the rotary table, J_1 is the equivalent of mass moment of inertia of the collars and the drillpipes, C_1 is the equivalent viscous damping coefficient of BHA, and T_{tob} is a nonlinear function which will be referred to be the torque on-bit.

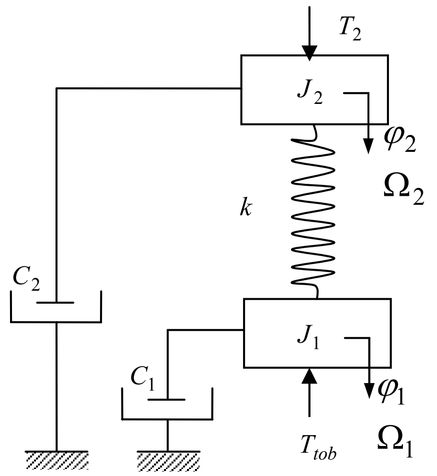

Fig. 2. Drilling rotary model

2.2 The driven system

The mechanical behaviour of the drive system is dominated by three components: the rotary table, a bevel gearbox with combined gear ratio of $1:n$, and an electric motor. The equations of this system are given in (Serrarens *et al*., 1998) as

$$J_2 = J_{rot} + n^2 J_m$$

$$J_2\dot{\Omega}_2 + C_2\Omega_2 + k\phi = T_2 \tag{2}$$

$$C_2\Omega_{ref} + u = T_2 \tag{3}$$

where J_2 represents the inertia of the rotary table (J_{rot}) augmented with inertias of the electric motor (J_m) and the transmission gear box ratio of the real system, C_2 is the viscous damping of the rotary table, Ω_{ref} is the desired velocity of the bit and finally T_2 is the torque delivered by the motor to the system.

In general, the electrical behaviour of the motor has to be described in terms of nonlinear differential equations that depend on the type of the motor. Many drilling rigs are equipped with a separately excited DC motor, and in that case the equations reduce to the linear relationships (Serrarens *et al.*, 1998):

$$L\dot{I} + RI + V_{emf} = V_m$$
$$V_{emf} = K\Omega_2$$
$$T_2 = KI$$

where $I, R, L,$ and V_m are respectively defined as motor current, motor resistance, motor induction and motor input voltage. The back-electromotive force (back-emf) V_{emf} is linearly related to the rotary table speed, K is the motor constant multiplied by the transmission ratio such as $K = nK_m$.

2.3 The stick-slip oscillations

The modelling of exact friction characteristic is a quite difficult problem, because the friction characteristic can be changed easily due to the environment's changes, for example, the variation of the load, lubrication, and the surface roughness (nature of the rocks).
In the oil drillstring, these oscillations are driven by nonlinear friction (T_{tob}) at near-zero bit velocities. T_{tob} represents the combined effects of reactive torque on the bit and nonlinear frictional forces along the BHA. Little is known about these effects, and several functions have been proposed for analyzing stick-slip vibrations (Serrarens, 2002; Jansen, 1993; Cull, and Tucker 1999). The connection between the friction torque (T_{tob}) as a function of the bit speed is given in (Serrarens, 2002) by the following nonlinear function:

$$T_{tob}(\Omega_1) = T_{tobdyn}\,\frac{2}{\pi}\left(\alpha_1\Omega_1 e^{-\alpha_2|\Omega_1|} + a\tan(\alpha_3\Omega_1)\right) \quad (4)$$

where $T_{tobyn} = 0.5kNm, \quad \alpha_1 = 9.5,\ \alpha_2 = 2.2$ and $\alpha_3 = 35.0$.

3. NONLINEAR FRICTION COMPENSATION METHOD

Many schemes form model based friction compensation have been proposed. If a good friction model is available it is possible to use a model based compensation scheme as shown in Figure 3. The system is provided with friction compensation and friction model. The linear controller is presented by PI (Proportional Integral) control, which is used to achieve the following performances: Increased damping, reduced rise time for step or rapid inputs, friction compensation (without much stick slip oscillation). PI may be any control structure of the form:

$$u(t) = k_p e(t) + k_i \int e(\tau)d\tau$$

where k_p , k_i are control parameters, $u(t)$ and $e(t)$ are the system input and error, respectively.

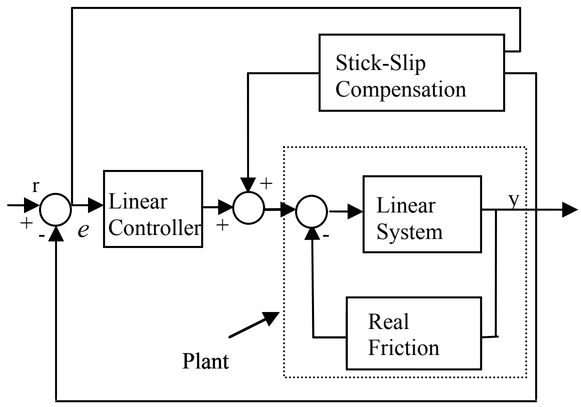

Fig.3. Block diagram of the model-based friction compensation scheme

4. CONTROLLER DESIGN FOR THE ROTARY DRILLING SYSTEM BASED ON FRICTION COMPENSATION

Figure 4 shows the block diagram of the system to be controlled. This system involves a plant which is driven by a PI controller. The plant has inherent driven system, drillstring, BHA and finally sticks slip friction.

Consider the nonlinear equation of the rotary drilling system given in (1)-(4), which may be written in the form:

$$\dot{x}(t) = f(x(t)) + g(x)u(t) \quad (8)$$
$$y(t) = h(x(t))$$

where $x(t) = [\Omega_1 \quad \phi \quad \Omega_2]^T \in R^3$ is the state vector, $y(t) \in R$ is the measured output variable,

$$f(x) = \begin{bmatrix} -\frac{C_1}{J_1}\Omega_1 + \frac{k}{J_1}\phi - \frac{1}{J_1}T_{tob}(\Omega\ ) \\ \Omega_2 - \Omega_1 \\ -\frac{C_2}{J_2}\Omega_2 - \frac{k}{J_2}\phi + \frac{C_2}{J_2}\Omega_{ref} \end{bmatrix}$$

$$g(x) = \begin{bmatrix} 0 & 0 & \frac{1}{J_2} \end{bmatrix}^T$$

and $h(x) = \Omega_1$

The control signal u is composed of a linear and a nonlinear part:

$$u = u_{lin} + \hat{T}_{tob}$$

where u_{lin} is the linear part and $T\hat{o}b$ is the estimated torque force.

$$u_{lin} = k_p e + k_i \int e dt$$

where $e = \Omega_1 - \Omega_{ref}$

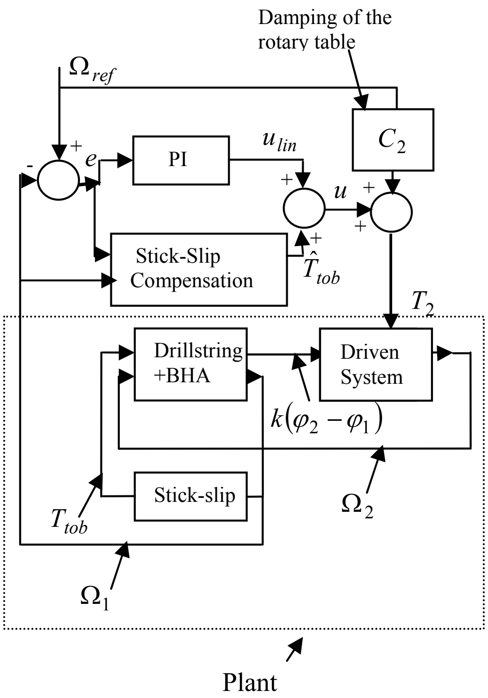

Fig.4. The system to be controlled

5. SIMULATION RESULTS

Simulations have been performed to investigate the efficiency level of the proposed nonlinear friction compensation method. The numerical values of the drilling system are listed in Table 1 and are taken from (Serrarens *et al*., 1998).

Table 1: Numerical values of drilling system

Parameter	Description	Value	Unit
J_1	BHA + 1/3 drillstring inertia	374	$[kgm^2]$
J_2	Rotary table + drive interia	2122	$[kgm^2]$
C_1	BHA damping	0-50	$[Nms/rad]$
C_2	Rotary table damping	425	$[Nms/rad]$
k	Drillstring stiffness	473	$[Nm/rad]$
w_r	Eigen frequency	1.125	$[rad/\sec]$

Figure 5 shows the velocity of bit without the controller, one may observe the oscillations that are undesirable because they are often regarded as one of the most damaging modes of vibration when drilling with low speed.

Furthermore, one may notice that the reference velocity is involved in the system through equation (3).

The drilling system, whose schematic in figure 1, is commanded by a step set point with final value $10 rad/\sec$. This step signal presents the desired velocity of the bit (Ω_{ref}).The PI parameters have been found based on the local linearized system as $k_p = 2$, $k_i = 9$.

The angular velocity of the rotary table, shown in Figure 6, reaches the desired velocity with time response of about 25s. Figure 7 shows that the angular velocity of the bit reaches the desired rotary drilling velocity (10 rad/sec) without much oscillation and with the same settling time like the angular velocity of the rotary table.

However, at the stick slip vibrations, as the control system attempts to track a desired behavior, without any more oscillations. Thus, the importance of nonlinear compensation is demonstrated.

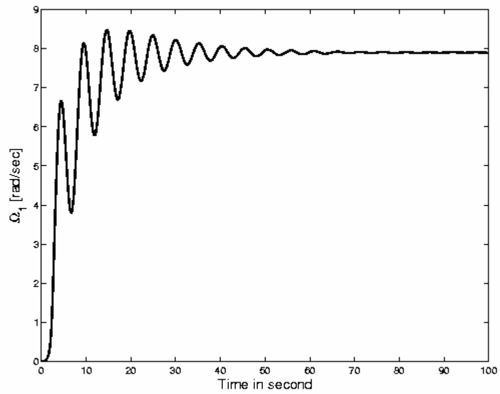

Fig.5. Angular velocity of the bit without the controller

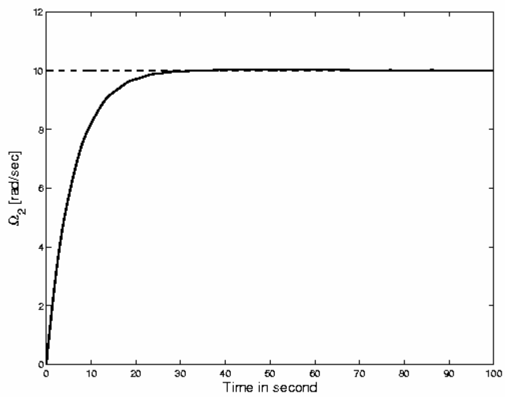

Fig.6. step response of the angular velocity of the rotary table (rad/sec)

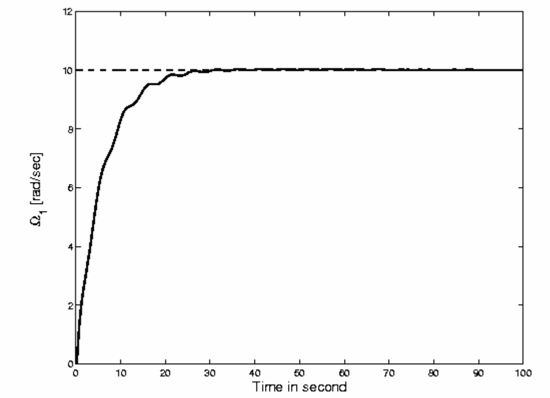

Fig. 7. Step response of the angular velocity of the bit (rad/sec) vs time (in sec) in closed loop

6. CONCLUISIONS

In this work, an effective method has been proposed to compensate the vibrations in oil well drillstrings, which has not seen applied up to now, at least to our knowledge. The nonlinear controller combines stick-slip compensation and a proportional integral controller.

In the controller development in this paper, it was assumed that all of the system parameters are known.

By the way, the modelling of exact friction characteristic is not an easy problem, because the friction characteristic can be changed easily due to the environment's changes, for example, the variation of the load, lubrication, and the surface roughness (nature of the rocks). However, this assumption is not always true. As for example the perturbations occur in the drillstring length during the process of drilling. The method has the potential to be extended by using an adaptive nonlinear controller. This will be reported in some near future work.

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