Method for Controlling Stick-Slip Vibrations in Slender Drilling Systems

Guilherme Sampaio and Hans I. Weber

Abstract Systems actuated trough a flexible shaft poses a big challenge to control strategies as the actuator is not connected directly to the end effector, causing propagation effects as well as an energy accumulation and dissipation in the shaft. This paper focuses on the top driven drilling system used in the oil and gas industry. In these systems, all kind of vibrations are found: longitudinal deformations (bit bouncing), flexional (rubbing), and torsional (stick-slip). This paper is about the torsional deformation of the highly flexible string modeled as a 20 DOF Lumped parameters system. A method for reducing stick-slip vibrations is presented and its results analyzed. The investigation includes the development of a reduced scale test rig adequate for torsional vibrations under damping. Results from the mathematical model and experimental tests are then compared.

Keywords Stick-slip · Torsional vibrations · Friction · Control · Drilling

1 Introduction

Top driven drilling used in the oil and gas industry is one of the most investigated application of systems driven by a highly flexible shaft. These systems pose a big challenge to control strategies, as the actuator is not linked directly to the end effector, causing propagation effects as well as an energy accumulation and dissipation in the shaft. These drilling systems are composed by a top drive linked to the drill bit through hundreds of meters of steel pipes. Drilling is one of the most expensive parts of oil prospecting and involves many risks of accidents, even though the methods in use are still very much based on trial and error experiences. Linear control theories

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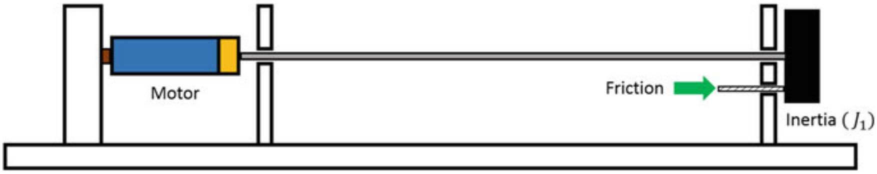

Fig. 1 Test setup

for example, PID have little success on real drilling applications because there are lots of uncertainties present, friction with the well, friction between rock and bit, etc. This paper will present a numerical and experimental study of a control technique that aims to reduce torsional vibrations maintaining a constant speed at the bit in a simple model of an oil drilling rig. A 2 m long test setup Fig. 1 composed by a DC motor, an elongated flexible shaft and a driven inertia that simulates the bit of a drilling structure, was constructed to test in the lab the controller in the presence of uncertainties and sensor noises.

The torsional system was modeled as a 20 degrees of freedom (DOF) flexible shaft. The contact of the pin that simulates the contact between bit and rock in the real system, is modeled as sum of a Coulomb static friction coefficient, a dynamic coefficient and a viscous friction, dependent of the angular speed.

The problem of modelling the torsional dynamics of a flexible shaft can be approached in different ways, the most common in literature are a simple torsional spring or spring-damper [1], lumped parameters or a finite element discretization of the shaft.

Khulief et al. [2] analyze self-excited stick-slip oscillations in drillstrings using a proposed dynamic model where the equation of motion of the rotating drillstring is derived using Lagrangian approach in conjunction with the finite element method and analyses torsional-bending and axial-bending nonlinear couplings.

Navarro-Lopez and Cortes [3] developed a lumped parameter model to investigate the influence of sliding motion on self-excited stick-slip oscillations and bit sticking phenomena. Hopf bifurcations were used to investigate the range of rotary speeds where the undesired torsional vibrations of the drillstring happen.

Rudat and Dashevskiy [4] present in the article, an innovative model based stick-slip control system using a lumped parameters model with parameters identified from real world applications trough Newton Gauss method and extended Kalman filter. The key idea of this paper is to run simulations on an embedded system down hole and transmit the updated model parameters in a lower bitrate trough mud pulses, an established technology. It shows that lumped masses model can reproduce nonlinear dynamics of drilling and shows, with experiments done in field tests, the effectiveness of the proposed approach.

Bayliss et al. [5] analyze a basic pole placement controller design for a Single Input Single Output (SISO) linear model of a drilling system, but recursively evaluated based on an online Recursive Least Squares (RLS) identification of the open-loop plant parameters. It presents a discussion on system architecture implications, and the simulation results with and without adaptive stick/slip mitigation method. The presented method in this paper relies on accurate measurements of the speed on Bit.

Ritto et al. [6] analyze the dynamics of a horizontal drill-string modeling uncertainty on the frictional force and how uncertainties on the frictional forces propagate through the system. A stochastic field with exponential correlation function is used to model the frictional coefficient.

Kreuzer and Steidl [7] present in the paper a method for controlling these vibrations by exactly decomposing the drill string dynamics into two traveling waves traveling in the direction of the top drive and in the direction of the drill bit. Authors state that by using two angular sensors placed 5 m away from each other it is possible to characterize traveling torsional waves in the drillstring and therefore use the top drive to eliminate them.

Kapitaniak et al. [8] experimentally investigates drillstring vibrations using a vertical reduced scale drilling rig. In the test setup presented by authors, one can investigate torsional, compression and helical bucking vibrations. The proposed experiment uses a 10 mm diameter steel drillstring and rotational speeds up to 54 RPM. The article describes methods used to obtain the mechanical properties of the setup as well as the use of finite element models to represent it.

2 Mathematical Model

A model with a 20 DOF Lumped parameters flexible shaft Fig. 2, was chosen to be used for the simulations in this paper. This model was chosen for its simplicity yet being capable of including the inertia of the shaft and dissipation from internal friction. This model also uses a complete DC motor model with electrical and mechanical parts.

The DC motor is modeled by the equations of the mechanical Eq. (1) and electrical parts Eq. (2) as well as the torque constant k_t that is the relation between T_m and i. All the parameters used for the motor were the ones obtained by [9].

$$J_m \frac{d^2\theta}{dt^2} = T_m - b_m \frac{d\theta}{dt} \tag{1}$$

$$L\frac{di}{dt} = -Ri + V - e \tag{2}$$

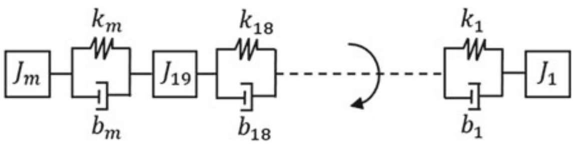

Fig. 2 Lumped parameters flexible shaft

In the lumped parameters model, each element or DOF is an elementary inertia-damper-spring system modeled as:

$$I\frac{d^2\theta}{dt^2} = -k\theta - b\frac{d\theta}{dt} \tag{3}$$

Writing the differential equations in Eq. (3) for each sub system and assembling them in a matrix form it becomes:

$$\bar{M}\ddot{\theta} + \bar{B}\dot{\theta} + \bar{K}\theta = \bar{\tau} \tag{4}$$

where θ is the state vector representing the angular displacements of the lumped masses, $\bar{\tau}$ is the external torques vector, the mass matrix ($\bar{M}$), the damping matrix ($\bar{B}$), and the spring matrix ($\bar{K}$) are:

$$\bar{M} = \begin{bmatrix} J_1 & & & & \\ & J_2 & & & \\ & & \ddots & & \\ & & & J_{19} & \\ & & & & J_m \end{bmatrix} \bar{B} = \begin{bmatrix} b_1 & -b_1 & & & \\ -b_1 & (b_1+b_2) & -b_2 & & \\ & -b_2 & \ddots & & \\ & & & (b_{18}-b_m) & -b_m \\ & & & -b_m & b_m \end{bmatrix} \tag{5}$$

$$\bar{K} = \begin{bmatrix} k_1 & -k_1 & & & \\ -k_1 & (k_1+k_2) & -k_2 & & \\ & -k_2 & \ddots & & \\ & & & (k_{18}-k_{19}) & -k_{19} \\ & & & -k_{19} & k_{19} \end{bmatrix} \tag{6}$$

This way, the mechanical lumped parameters of the drill string can be written substituting Eqs. (5) and (6) in Eq. (4). The model parameters were obtained experimentally, Table 1 shows the numerical values.

Table 1 Model parameters used on simulations

Properties	Value	Unit
String length (L)	1.7	m
String diameter (mm)	3	mm
Total inertia of J_1	0.01555819	kg m^2
String stiffness (K)	0.2548	N m/rad
Moment of inertia of motor (J_m)	0.37×10^{-3}	kg m^2
Armature inductance (L_{DC})	1.10×10^{-3}	H
Armature resistance (R_{DC})	0.33	Ω
Torque constant (K_t)	0.12	N m/A
Speed constant (K_e)	6.02×10^{-2}	V/(rad/s)

2.1 Torque on Bit Formulation

The contact between bit and rock is modeled, according to Armstrong-Hlouvry et al. [10] by the sum of a Coulomb static friction coefficient, a dynamic coefficient and a viscous friction, dependent of the angular speed. This contact appears in the model on the inertia J_1.

$$T_r = (T_C + (T_{brk} - T_C) \cdot exp(-c_v|\omega|))sign(\omega) + f\omega \qquad if|\omega| \geq \omega_{th} \tag{7}$$

and:

$$T_r = \omega \frac{(f\omega_{th} + (T_C + (T_{brk} - T_C) \cdot exp(-c_v\omega_{th})))}{\omega_{th}} \qquad if|\omega| < \omega_{th} \tag{8}$$

where: T_r is the friction torque, T_C is the Coulomb friction torque, T_{brk} is the static friction torque, c_v is the dynamic friction coefficients, ω is the angular speed, f is the viscous friction coefficient and ω_{th} is the velocity of threshold that is in the order of 10^{-4} included to avoid numerical problems (Fig. 3).

Table 2 shows the numerical values of friction used in this paper.

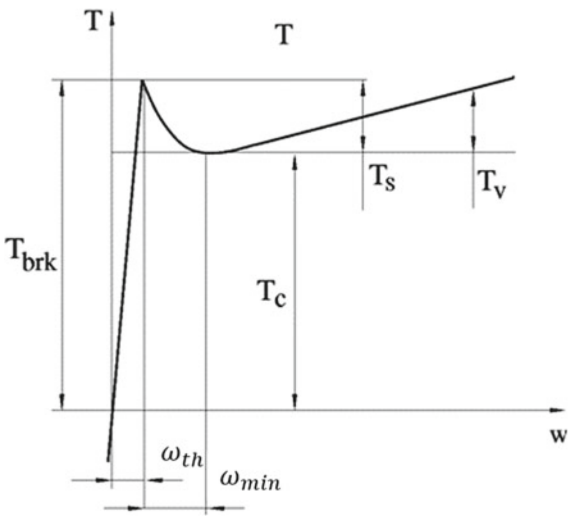

Fig. 3 Friction torque versus rotational speed

Table 2 Friction parameters

Parameter	Value	Unit
Breakaway friction coefficient (μ_{brk})	0.5	–
Coulomb friction coefficient (μ_c)	0.47	–
Normal force on bit (N_f)	26	N
Viscous friction coefficient (b)	0.001	$\frac{\text{N m}}{\text{rad/s}}$

3 Controlling the System with a Torque Source

By analyzing the structure of the friction law Eq. (8) and the results of the simulations, an investigation was started to analyze if adding a torque source to J_1 (a kind of downhole motor), it could be possible to modify the stick slip phenomenon. Using this supposition an analysis was made to verify if it is possible to mitigate the stick slip by controlling the torque on J_1 with the use of a DC motor.

A recent study from Shor et al. [11], showed that the propagation effects on torsional vibrations are important for the implementation of torsional vibrations mitigation techniques, which led to suppose that the phase of the proposed control for an imposed torque on J_1 should be important for the results. Therefore, the lumped masses system described in Sect. 2 was simulated with two torque sources, both DC motors referred as "Motor" and "J_1 Motor" in Fig. 4. The simulations started at $t = 0$ s with angular displacement and speed of the drill string being zero. At $t = 0$ the top drive motor is started at 2 rad/s, and around $t = 9$ s the energy accumulated in the drill string is enough to overcome the static friction force and the stick slip phenomenon begins. At $t = 15$ s a second DC motor attached to J_1 is energized applying a torque of aprox. -0.29 N m to J_1. This method only observes the output (angular speed at J_1) to start the motor in J_1, then this control is an open loop scheme, only applying a constant torque on J_1. The torque applied on J_1 therefore is: 0 N m for $t < 15$ s and $t > 30$ s, and -0.29 N m between $t = 15$ s and $t = 30$ s. Results are shown in Figs. 5 and 6.

As described by Shor et al. [11], the delay effects from the propagation of torsional vibrations along the drill string must be considered for the control structure of the

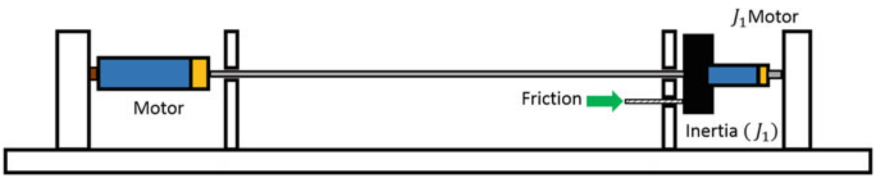

Fig. 4 Mechanical model with motor on J_1

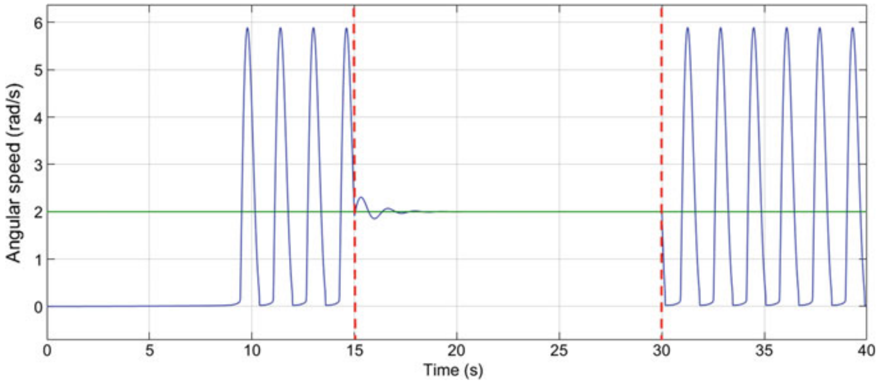

Fig. 5 Angular speed on top drive (green) and J_1 (blue)

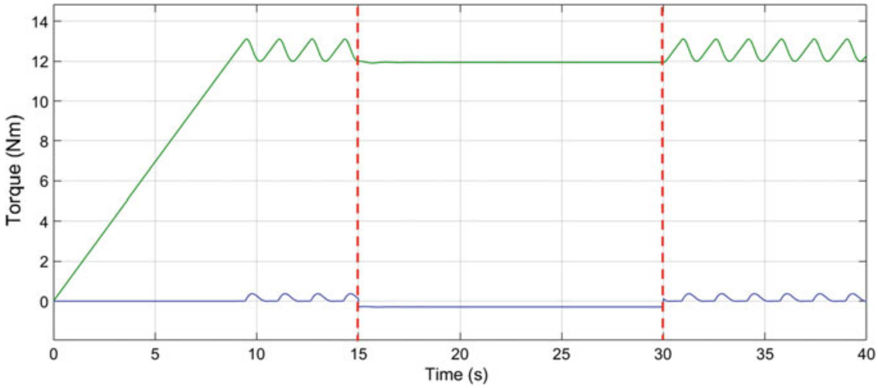

Fig. 6 Torque on top drive (green) and J_1 (blue)

problem. To prove that the developed mathematical model is capable of representing these effects, the system described in Figs. 5 and 6, was also simulated for a torque on J_1 being applied from $t = 14$ s to $t = 30$ s.

Results in Figs. 7 and 8 show that if the torque on J_1 is applied at a wrong moment it will have no effect on the stick-slip, only adding a small disturbance on the angular speed when it is applied. Simulations show that this approach to mitigate the stick slip only works if the torque on J_1 is applied when the angular speed of J_1 is decreasing, i.e. $\ddot{\theta}_1 < 0$.

In order to test this supposition, the same system was simulated again with the beginning of application of torque in J_1 at $T = 13.25$ s.

Simulation results in Figs. 9 and 10 show that the closer to the top drive speed is to the speed on J_1 when the torque on J_1 is applied, the better results are obtained. This is valid considering that the torque on J_1 is applied when the angular speed of J_1 is decreasing, i.e. $\ddot{\theta}_1 < 0$.

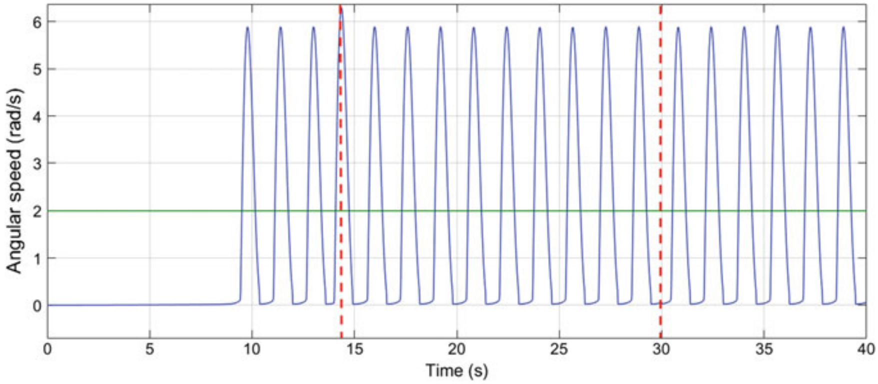

Fig. 7 Angular speed on top drive (green) and J_1 (blue)

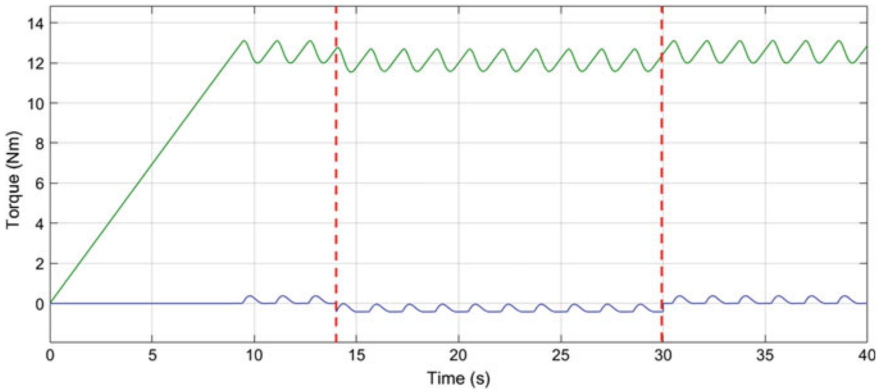

Fig. 8 Torque on top drive (green) and J_1 (blue)

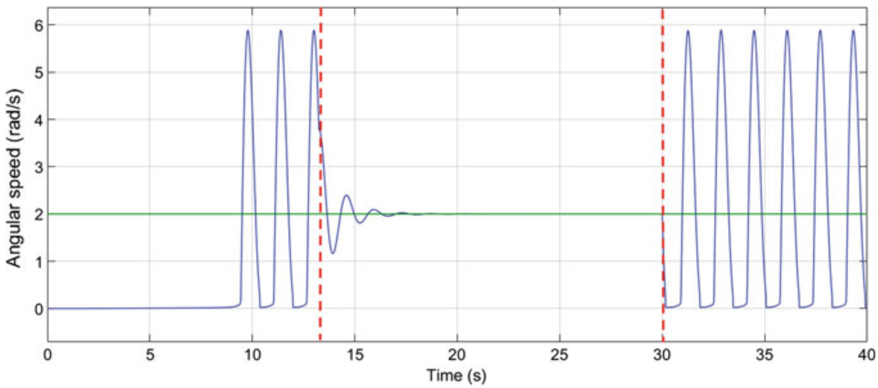

Fig. 9 Angular speed on top drive (green) and J_1 (blue)

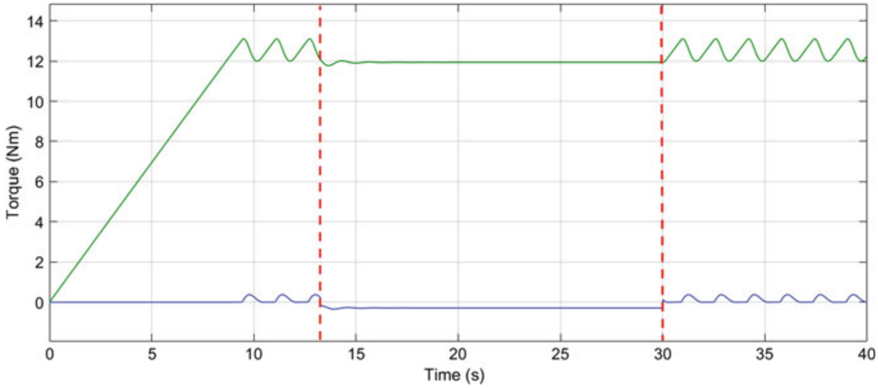

Fig. 10 Torque on top drive (green) and J_1(blue)

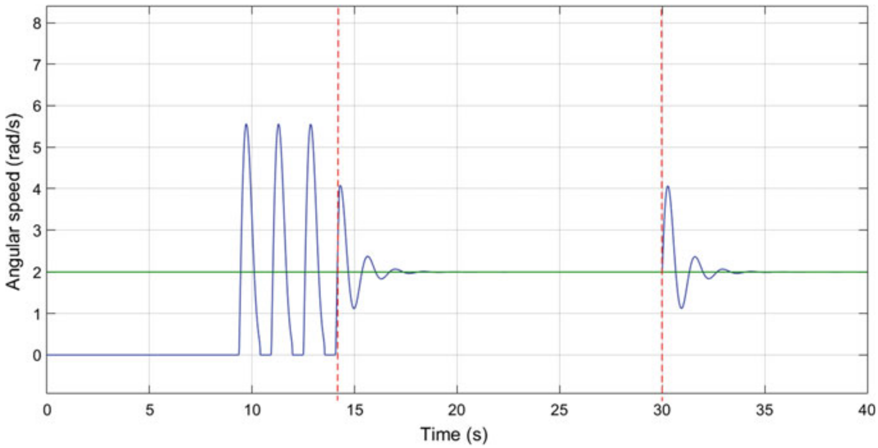

Fig. 11 Angular speed on top drive (green) and J_1 (blue)

One should note that despite this strategy is effective and shows good results, it depends on a precise measure of the speed on the bit (J_1) which limits its application for real life oil drilling problems with the existing technologies of bottom hole measurement available.

If, on the other hand, a positive torque is applied when the system is accelerating i.e. when the angular speed of J_1 is increasing, i.e. $\ddot{\theta}_1 > 0$, the results stay the same, the stick-slip is eliminated, but in this case another phenomenon is observed. When the motor in J_1 is turned off, the system has a torsional perturbation, but it does not come back to a stick-slip behavior. Figure 11 shows the angular speed of the top drive and J_1, and Fig. 12 shows the torque of both motors. As in the previous case, it is worth to note that the torque applied by the motor in J_1 is much smaller than the one from the top drive.

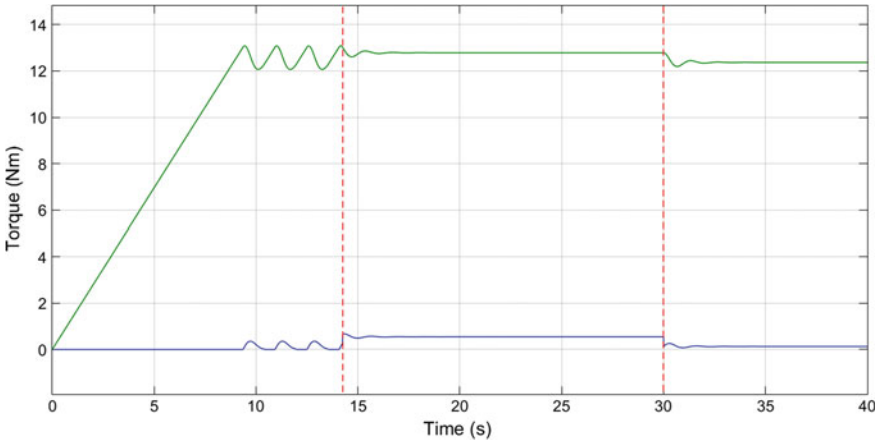

Fig. 12 Torque on top drive (green) and J_1 (blue)

4 Experimental Tests

In order to perform the experimental tests, a test bench was made Fig. 13. This apparatus is proposed to study only the rotational and torsional dynamics of the system, for that reason the motor and the inertia are mounted on bearings so that the setup can be used in an horizontal position, making it easy to operate. The drill string is 1.7 m long and is made of a 3 mm diameter steel rod. The DC motor is mounted on two ball bearings, so the torque applied by the motor is obtained through a force measured by a load cell positioned at a known distance from the motor. This force, the normal force of the brake on the inertia, and the rotational speed of the motor and of the inertia, are measured and these data are acquired by a National Instruments cDAQ system. A complete description of this experimental setup is shown in [9].

Table 3 shows the numerical values measured on the experimental setup used in this work.

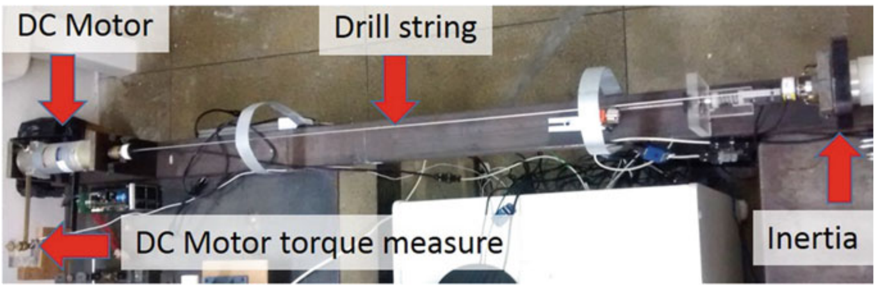

Fig. 13 Experimental setup

Table 3 Mechanical parameters of drillstring

Properties	Value	Unit
String length (L)	1.7	m
String density (ρ)	7850	kg/m^3
String diameter (mm)	3	mm
Young modulus (E)	210	GPa
Poisson ratio (v)	0.3	–
Inertia of J_1	0.01555	kg m^2

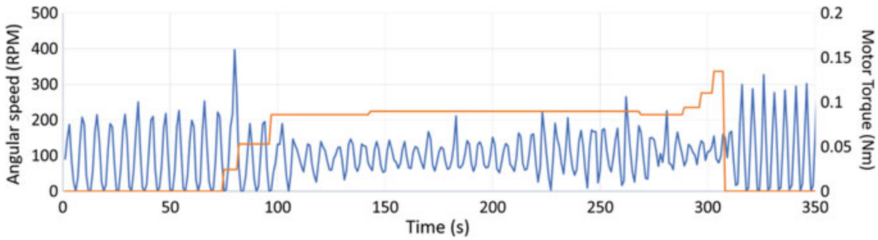

Fig. 14 Experimental results for torque applied on J_1 in green and speed at J_1 in orange

Figure 14 shows the results obtained with the apparatus shown in Fig. 13. In blue, on the left axis is the angular speed in RPM of J_1, in orange on the right axis is the amplitude of the torque in N m applied by the DC motor attached to J_1. Results shown are very similar to the ones obtained with the numerical model Fig. 5. The torsional vibration on the experimental test rig is not completely eliminated due to noise of the sensors and to non-modeled imperfections of the apparatus. But this approach shows that it can eliminate the stick-slip during the period the DC motor is being used.

5 Conclusions

It was shown that it is possible to mitigate the stick-slip phenomenon in the presented test setup and in the model, by applying a small torque on the bit, orders of magnitude lower than the one applied by the main motor, i.e. top drive.

Despite the construction challenges, the results presented in this paper show that the presence of a torque source on the bit of the drilling column (a kind of bottom hole motor, or mud motor) has a very important role on the torsional vibrations of the column. The presence of the bottom motor is almost not studied by the dynamics and vibrations community.

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