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Decentralized Overlapping Control Design with Application to Rotary Drilling System

M. Z. Doghmane[1,2], M. Kidouche[3] and A. Ahriche[4]

[1]The Laboratory of Applied Automatic, University M'hamed Bougara, Boumerdes 35000, Algeria; [2]Exploration Division, SONATRACH, Hassi Messaoud 30500, Algeria; [3]Department of automation, University M'hamed Bougara, Boumerdes 35000, Algeria; [4]Electrical engineering department, University M'hamed Bougara, Boumerdes 35000, Algeria

ABSTRACT

The main objective of this paper is the development of overlapping decomposition strategy for controller design of rotary drilling systems based on the state feedback technique. The lamped model of the system has been considered, Graph-based representation has been used to rearrange the states of the model, and then a new model has been decomposed into subsystems after analyzing the interconnection terms. The expansion–contraction principle, combined with Lyapunov theory, is used to investigate the global stability of the decomposed system so that controllability of the designed controller is guaranteed and its robustness is improved. Moreover, the designed strategy has been validated with high-frequency mode stick-slip vibrations data measured in an operational rotary drilling system of an exploration well drilled in an Algerian hydrocarbon field.

KEYWORDS

Overlapping decomposition; Rotary drilling systems; Graph-based representation; Expansion–contraction; Lyapunov theory; High-frequency mode stick-slip vibration

1. INTRODUCTION

The appearance of vibrations, along the rotary drilling systems' string, is considered to be one of the major causes of drilling cost augmentation and performance limitation. Indeed, these types of vibrations affect proportionally the non-productive time by creating fatigue to the drill pipes and premature failure to the bit [1,2]. For the sake of cost reduction and failure prevention, and to avoid such dynamics, considerable research studies have been steered in the last decades to diminish the drill string vibration severity [3]. Several methodologies have been proposed, from industrial and academic point of views [4].

From practical side, the drilling performance is monitored through system's parameters such as rate of penetration (ROP), rotary speed, weight, and torque on bit (WOB and TOB) [5]. The ROP optimization means that the rest of drilling parameters are adjusted to drill the geological formation most efficiently. However, equipment safety should be respected during vibrations' mitigation, in other words, achieving improved performance (*i.e.* by reducing vibrations) with reduced costs, may not be practically realizable with respect to the desired safety level [6]. From theoretical side, vibrations are excited by the regenerative effect of the rock-bit interaction, hydrodynamics of the mud, eccentricity of the BHA, and frictions. Thus, they can be classified into three types: Torsional, Axial, and lateral vibrations. The torsional vibration is induced by the nonlinear relationship between the torque and the angular velocity at the drill bit. Whereas the torsional flexibility of the drilling assembly exacerbates nonuniform oscillatory behavior, which causes the rotational speeds to be much higher than the nominal velocity of the rotary table [7]. The appearance of these drill string vibration types can be synchronous; it is possible to observe all the three vibrations at the same time. Stick-slip, whirling, and bit bounce are the severe situations of the preceding vibration status, it has been shown that slip section of the high-frequency torsional vibration mode is the stimulator of lateral and axial vibrations, and it is potentially harmful if not controlled [8]. Despite all the analyses provided in drill string dynamics and rock-bit interaction term, there is a lack of comprehensive modeling of the entire drill string and borehole assembly (BHA) for real-time vibration control design [3,9]. Consequently, vibration mechanisms are subject to intensive research by academics and the industrials [10], where an accurate modeling of these vibrations is crucial for reliable control and monitoring of drilling performance [11]. Down hole measurements made while drilling reveal severe vibrations, which often lead to rotational stick-slip and cause additional wear on the drill string and the bit, a penetration rate decrease and drill bits' precocious failure [2].

Drilling systems' models are highly nonlinear, which makes the analytical expression of its dynamic behavior more complex. Decomposition of complex systems is an efficient technique to deal with problems in diverse domains, *i.e.* automated high way systems [12–14], electric power systems [15,16], mechanical designs [17,18], applied mathematics [19,20], large space structures [21], and aeronautical system [22]. Supported by mathematical background of the inclusion theories, this methodology is essentially based on expanding the input and output spaces to larger spaces wherein the overlapped subsystems become disjoint. Then, design decentralizes controller and contracts it to the original system for implementation [12,22]. For the case of large flexible system such as drill string, overlapping decentralized control can be one of the most effective control robustness improvement methods since it can be applied for strongly connected systems [17,23]. Besides, the short-time response of this controller can improve rotary system performance with respect to a wide variety of structured and unstructured perturbations exposed to the bit and the BHA.

This study is focused on stick-slip mitigation since it is the stimulator of other types of vibrations [24]. The emergence of vibrations in drilling systems has seen a significant increase in the last few years, as an example, in the Algerian explorations industry approximately one out of every three rigs has experienced drill string vibration failure [25]. For this reason, we propose in this study the application of new control strategy based on overlapping decomposition to mitigate torsional vibrations in rotary drilling systems; this technique has been proved to be efficient in increasing controller robustness and decreasing response time [23,26]. The dynamic behavior of the system has been described by a high-fidelity model, where new rock-bit interaction term has been considered.

2. TWO-ELEMENT MODEL

In this study, the parameters of the model have been taken from operational rotary drilling system of oil exploration field [5]. The rotating part of drilling system is composed essentially of driving motor (top drive), drill string, including thin-walled pipe section, and BHA that contains heavy thick-walled collars and bit attached at the bottom end of the drill string (Figure 1(a)). In order to centralize the bit, many stabilizers are placed at the collar section; they increase the buckling load-carrying capacity of the collar section, and they are used to control well trajectories for directional drilling [27]. The drilling mud circulates in the annulus between the BHA and borehole; it raises cuttings to the surface, keeps pore fluid in their places, in addition to maintaining wall stability and cooling the bit.

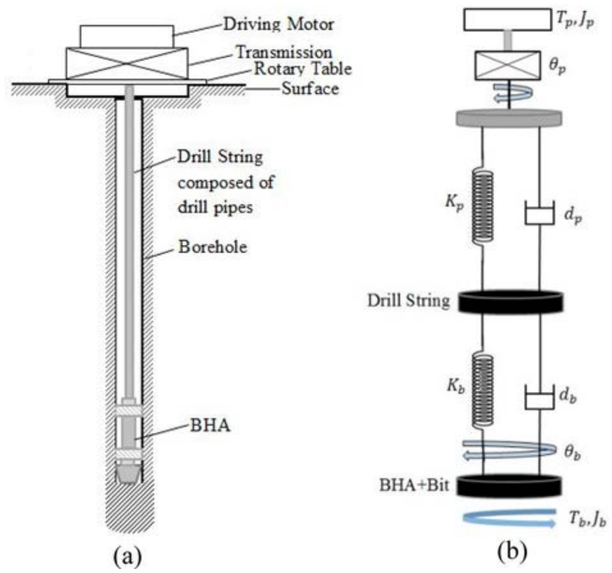

Figure 1: Rotary drilling system: (a) Schematic diagram, (b) equivalent mass-spring-damper two-element model

One of the most used models for rotary drilling system is mass-spring-damper model, wherein the behavior of drill pipes is reproduced by a torsional spring, and the drill collar is represented by a rigid body (Figure 1(b)). The inertial masses, locally damped by d_p and d_b, are connected to each other by a linear spring with torsional stiffness k and torsional damping μ [28]. The equation of motion is described by an ordinary differential Equation (1).

$$\begin{cases} J_p\ddot{\theta}_p + d_p(\dot{\theta}_p - \dot{\theta}_b) + k(\theta_p - \theta_b) + \mu_p\dot{\theta}_p = u_T \\ J_b\ddot{\theta}_b - d_b(\dot{\theta}_p - \dot{\theta}_b) - k(\theta_p - \theta_b) + \mu_b\dot{\theta}_b = Tob(\dot{\theta}_b). \end{cases} \tag{1}$$

θ_p and θ_b are the angular displacements of the top drive and the BHA, respectively. The angular velocity of the top drive is considered as the input signal u_T; it is proportional to top drive torque. The transmission box usage is to adjust the angular velocity $\dot{\theta}_b$; the frictional torque $Tob(\dot{\theta}_b)$ represents, in this study, the torque on bit combined with the nonlinear frictional forces along the drill collars [29]. By setting $x_1 = \dot{\theta}_p$, $x_2 = \dot{\theta}_b$ and $x_3 =$

$(\theta_p - \theta_b)$, Equation (1) is written as

$$\begin{cases} \dot{x}_1 = -\dfrac{\mu_p + d_p}{J_p}x_1 + \dfrac{d_p}{J_p}x_2 - \dfrac{K}{J_p}x_3 + \dfrac{1}{J_p}u_T \\ \dot{x}_2 = \dfrac{d_p}{J_b}x_1 - \dfrac{\mu + d_b}{J_b}x_2 + \dfrac{K}{J_b}x_3 + \dfrac{1}{J_b}Tob \\ \dot{x}_3 = x_1 - x_2 \end{cases} . \quad (2)$$

For the system composed of three elements (two elements for drill string and one element for drill bit) [3], Equation (2) is

$$\begin{cases} \ddot{\theta}_1 = \dfrac{d_1}{J_1}(\dot{\theta}_{td} - \dot{\theta}_1) + \dfrac{K_1}{J_1}(\theta_{td} - \theta_1) - \dfrac{d_2}{J_1}(\dot{\theta}_1 - \dot{\theta}_2) \\ \qquad -\dfrac{K_2}{J_1}(\theta_1 - \theta_2) - \dfrac{\mu}{J_1}\dot{\theta}_1 \\ \ddot{\theta}_2 = \dfrac{d_2}{J_2}(\dot{\theta}_1 - \dot{\theta}_2) + \dfrac{K_2}{J_2}(\theta_1 - \theta_2) - \dfrac{d_3}{J_2}(\dot{\theta}_2 - \dot{\theta}_3) \\ \qquad -\dfrac{K_3}{J_2}(\theta_2 - \theta_3) - \dfrac{\mu}{J_2}\dot{\theta}_2 \\ \ddot{\theta}_3 = \dfrac{d_3}{J_3}(\dot{\theta}_2 - \dot{\theta}_3) + \dfrac{K_3}{J_3}(\theta_2 - \theta_3) + \dfrac{1}{J_3}(\mu - Tob)\dot{\theta}_3 \end{cases} \quad (3)$$

$Tob(\dot{\theta}_3)$ is the torque on bit; it represents the nonlinearity of the system. $\theta_i, \dot{\theta}_i, J_i, d_i$ are the angular displacement, angular velocity, equivalent mass moment of inertia, equivalent viscous damping coefficient at the bottom of the element i of the drill string, respectively. $\theta_{td}, \dot{\theta}_{td}$ are the angular displacement, the angular velocity of the top drive, respectively, K_i is the torsional stiffness coefficient [30]. If we set $\dot{x}_1 = \dot{\theta}_1$, $\dot{x}_2 = \dot{\theta}_2$, $\dot{x}_3 = \dot{\theta}_3$, $u_T = \dot{\theta}_{td}$ and $x_4 = \theta_{td} - \theta_1$, $x_5 = \theta_1 - \theta_2$, and $x_4 = \theta_2 - \theta_3$, Equation (3) becomes

$$\begin{cases} \dot{x}_1 = -\dfrac{d_1 + d_2 + \mu}{J_1}x_1 + \dfrac{d_2}{J_1}x_2 + \dfrac{K_1}{J_1}x_4 \\ \qquad -\dfrac{K_2}{J_1}x_5 + \dfrac{d_1}{J_1}u \\ \dot{x}_2 = \dfrac{d_2}{J_2}x_1 - \dfrac{d_2 + d_3 + \mu}{J_2}x_2 + \dfrac{d_3}{J_2}x_3 \\ \qquad +\dfrac{K_2}{J_2}x_5 - \dfrac{K_3}{J_2}x_6 \\ \dot{x}_3 = \dfrac{d_3}{J_3}x_2 - \dfrac{d_3 + \mu + Tob}{J_3}x_3 + \dfrac{K_3}{J_2}x_6 \\ \qquad \dot{x}_4 = u - x_1 \\ \qquad \dot{x}_5 = x_1 - x_2 \\ \qquad \dot{x}_6 = x_2 - x_3 \end{cases} \quad (4)$$

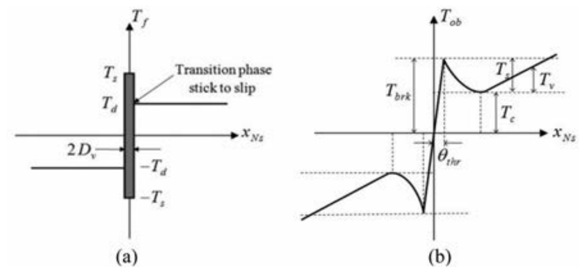

Figure 2: Bit-rock interaction models: (a) Karnopp discontinuous model, (b) Continuous model with Karnopp term

Table 1: Parameters of the TOB obtained from well-1

Parameter_Definition	Value	
K	Positive parameters	0.5
μ	The bit torque friction coefficient	0.25
N	The force vector	98077 g
r	The contact radius vector	0.0825 m
Ω_0	The transition speed for the string	0.2 rad/s
Ω_1	The transition speed of top drive	31.4159 rad/s
D	The linear damping vector	0.28
p	The start friction parameter	1.5

The general mathematical mass-spring-damper model of rotary drilling system composed of N_s elements is derived by the same methodology. The nonlinear function Tob is given by Equation (5); it is clear that it is continuous around $x_{Ns} \approx 0$ and has higher fidelity to Stick-Slip term (Figure 2(b))

$$Tob = \mu N_r \left(\frac{x_{Ns}}{\sqrt{x_{Ns}^2 + \Omega_0^2}} + \frac{p\Omega_0 x_{Ns}}{x_{Ns}^2 + \Omega_0^2} \right) - Dx_{Ns}\left(\frac{x_{Ns}}{\Omega_1} - 1 \right), \quad (5)$$

The most useful simplified nonlinear form of Equation (5) is given by Equation (6) [31].

$$Tob = \frac{2\bar{K}x_{Ns}}{x_{Ns}^2 + \bar{K}^2}. \quad (6)$$

The numerical values used in Equation (5) are taken from real rotary drilling system used to drill well-1, and they are summarized in Table 1, where $N_s = 3$ in this study.

Figure 2 shows a comparison between nonlinear torque on bit given by Equation (5) and the Karnopp model; we notice that Karnopp function can simulate torsional vibration term near zero angular velocity without discontinuity problem. However, the proposed Tob is more representative for stick-slip phenomenon case, thus a more appropriate model that gathers these two forms and overcome discontinuity is used [32].

3. LYAPUNOV STABILITY ANALYSIS

Consider the nonlinear model of rotary drilling system described by Equation (4), the nonlinearity term is given by Equation (5). This system is rewritten as

$$\dot{X} = f(X) = [\boldsymbol{A}_0 + \boldsymbol{G}(t)]X. \tag{7}$$

For the system in Equation (7), the following theorem has been given.

THEOREM *The matrix $\tilde{\mathbf{A}} = \mathbf{A}_0 + \mathbf{G}$ is Hurwitz if and only if, for any $\mathbf{Q} = \mathbf{Q}^{\mathrm{T}}$ there is $\mathbf{W} = \mathbf{W}^{\mathrm{T}}$ that satisfies the Lyapunov equation $\mathbf{W}\tilde{A} + \tilde{\mathbf{A}}^{\mathrm{T}}\mathbf{W} = -\mathbf{Q}$. Moreover, if $\tilde{\mathbf{A}}$ is Hurwitz, then W is the unique solution* [33].

Proof From Lyapunov' theorem, the solution can be seen as verifying that

$$\boldsymbol{W} = \int_0^{\infty} e^{(\tilde{A}^T t)} \boldsymbol{Q} e^{(\tilde{A}t)}\, dt, \tag{8}$$

is positive definite so that it guarantees the Lyapunov sufficient condition of stability.

First, we suppose $\tilde{\boldsymbol{A}}$ is Hurwitz, then we choose $\boldsymbol{Q} = \boldsymbol{Q}^T > 0$, and solve the Lyapunov equation $\boldsymbol{W}\tilde{\boldsymbol{A}} + \tilde{\boldsymbol{A}}^T\boldsymbol{W} = -\boldsymbol{Q}$ for $\boldsymbol{W}$.

We propose the following Lyapunov function candidate $L(X) = X^T\boldsymbol{W}X$ for the system $\dot{X} = f(X)$ so that

$$\begin{aligned}\dot{L}(X) &= X^T\boldsymbol{W}f(X) + f^T(X)\boldsymbol{W}X \\ &= -X^T\boldsymbol{Q}X + 2X^T\boldsymbol{WG}X,\end{aligned} \tag{9}$$

where $X^T\boldsymbol{WG}X + X^T\boldsymbol{G}^T\boldsymbol{W}X = 2X^T\boldsymbol{WG}X$, the stability condition is verified if

$$\dot{L}(X) \le -X^T\boldsymbol{Q}X + 2||\boldsymbol{W}|| \cdot ||\boldsymbol{G}|| \cdot ||X||^2, \tag{10}$$

and for any $\gamma > 0$, there exists $r > 0$ such that $||\boldsymbol{G}(X)|| < \gamma, \forall ||X|| < r$. We have

$$\begin{aligned}X^T\boldsymbol{Q}X \ge \lambda_{\min}(\boldsymbol{Q})||X||^2 \Leftrightarrow -X^T\boldsymbol{Q}X \\ \le -\lambda_{\min}(\boldsymbol{Q})||X||^2,\end{aligned} \tag{11}$$

$$\dot{L}(X) \le -[\lambda_{\min}(Q) - 2\gamma||\boldsymbol{W}||]||X||^2, \quad \forall X < r, \tag{12}$$

It is enough to choose

$$\gamma < \frac{\lambda_{\min}(Q)}{2\boldsymbol{W}}. \tag{13}$$

So $\dot{L}(X) < 0, \forall 0 < ||X|| < r$, condition in Equation (13) is always satisfied since the torque on bit is bounded by the top drive maximum torque, it is enough to choose any value for γ less than the maximum torque on bit. If we take $c = \min(X^T\boldsymbol{W}X) = \lambda_{\min}(\boldsymbol{W})r^2, ||X|| = r$, then

$$\{X^T\boldsymbol{W}X < c\} \subset \{||X|| \prec r\}. \tag{14}$$

From Equation (14), we conclude that all trajectories with initial condition included in the set $\{X^T\boldsymbol{W}X < c\}$ converge to the origin as t tends to ∞ [34]. Thence, the system is proved to be globally stable and the set $\{X^T\boldsymbol{W}X < c\}$ is an estimation of the attraction region. Furthermore, dynamic of the system in Equation (4) can be decomposed into overlapped subsystems with common states of rearranged model obtained from the graph theory, and then an expanded system will be established by disassociating these subsystems with mutual state variables. Lyapunov theory can be applied to the expanded system to achieve stability of the original system [35]. The discussed overlapping decomposition technique is particularly attractive in decentralized stabilization of the dynamic system [36,37].

4. GRAPH-BASED REPRESENTATION

The nonlinear mathematical model of rotary drilling system can be represented by using the graph theory. From the mathematical model of the system, the corresponding graph is constructed where the nodes represent the state variables and the edges represent the interconnection terms. Thus, the state variables and interconnection terms are represented by vertices and dependency relation by arcs (Figure 3(a)). The mathematical model is rearranged according to the graph-based representation shown in Figure 3.

5. EXPANSION–CONTRACTION PRINCIPLE

Expansion–contraction theory is based on finding a transformation matrix system for which the expanded system can be decoupled without having a big influence on the system dynamics. For each decoupled subsystem, a decentralized controller is designed unconnectedly, and then the resultant controllers can be implemented to the original system [18]. The inclusion principle was introduced in the 1980s based on the mathematical background of complex system analysis. Originally, it provides mathematical theories in which two dynamic systems with different dimensions may have an equivalent behavior. The objective is to find system of matrix $\boldsymbol{V}, \boldsymbol{U}$ and $\boldsymbol{M}$ such that $\tilde{X} = \boldsymbol{V}X$, where the original and expanded states of two-element model of rotary drilling system are, respectively $X = \begin{bmatrix} x_4 & x_1 & x_5 & x_2 & x_6 & x_3 \end{bmatrix}^T$ and $\tilde{X} = \begin{bmatrix} x_4 & x_1 & x_1 & x_5 & x_2 & x_2 & x_6 & x_3 \end{bmatrix}^T$. The

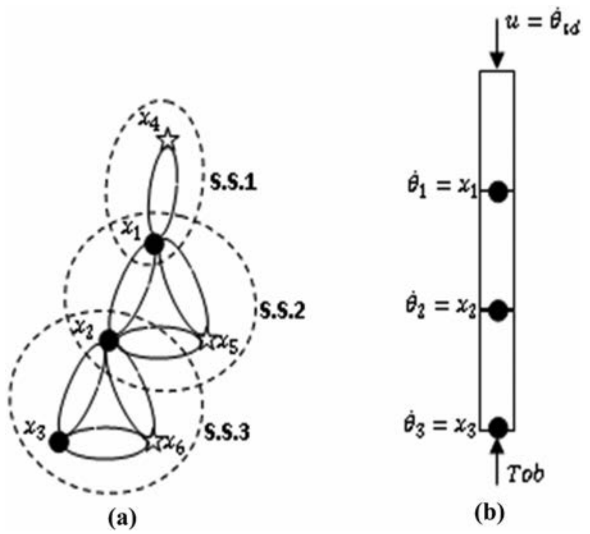

Figure 3: Two-element model of rotary drilling system: (a) Graph-based representation, (b) schema of the model

matrix $\boldsymbol{V}, \boldsymbol{U}$ and $\boldsymbol{M}$ are of appropriate dimensions and they are found by using expansion–contraction theorem [27], where $\tilde{\boldsymbol{A}}$ is

$$\tilde{\boldsymbol{A}} = \boldsymbol{VAU} + \boldsymbol{M}, \tag{15}$$

The simplified system can be seen as combination of three interconnected subsystems where the dynamic matrices for each subsystem are S.S.1: $\boldsymbol{A}_{11}$, S.S.2: $\boldsymbol{A}_{22}$, and S.S.3: $\boldsymbol{A}_{33}$. Thus, the interaction terms are

$$\boldsymbol{A}_{12}, \boldsymbol{A}_{13} = \boldsymbol{A}_{31} = 0, \boldsymbol{A}_{21}, \boldsymbol{A}_{23}, \boldsymbol{A}_{32}. \tag{16}$$

Figure 4 shows the effect of eliminating interconnection terms $\boldsymbol{A}_{12}$, $\boldsymbol{A}_{21}$, $\boldsymbol{A}_{23}$, and $\boldsymbol{A}_{32}$ on the dynamic of the system. We notice that eliminating the interconnection terms $\boldsymbol{A}_{21}$ and $\boldsymbol{A}_{32}$ did not influence the dynamic of the system; however, $\boldsymbol{A}_{12}$ and $\boldsymbol{A}_{23}$ cannot be neglected. Thus, decentralized controller can be designed to the expanded system by neglecting the weak terms, and then it can be contracted for implementation to the original system; this can improve the robustness of the designed controller [23]. The subsystems can now be written as given by Equation (17)

$$\begin{cases} S.S.1 : \dot{\tilde{X}}_1 = \boldsymbol{A}_{11}\tilde{X}_1 + \boldsymbol{A}_{12}\tilde{X}_2 + \tilde{B}_1 u_1 \\ S.S.2 : \dot{\tilde{X}}_2 = \boldsymbol{A}_{21}\tilde{X}_1 + \boldsymbol{A}_{22}\tilde{X}_2 + \boldsymbol{A}_{23}\tilde{X}_3 + \tilde{B}_2 u_2 \\ S.S.3 : \dot{\tilde{X}}_3 = \boldsymbol{A}_{32}\tilde{X}_2 + \boldsymbol{A}_{33}\tilde{X}_3 \end{cases} . \tag{17}$$

The state feedback controller for subsystem 1 is designed after supposing that the interaction term of subsystem 2 on system 1 is negligible in comparison to the dynamic of the whole system. Therefore, the state feedback controller of subsystem 1 is $u_1(t) = -\tilde{\boldsymbol{K}}_1(t)\tilde{X}_1(t)$, and for

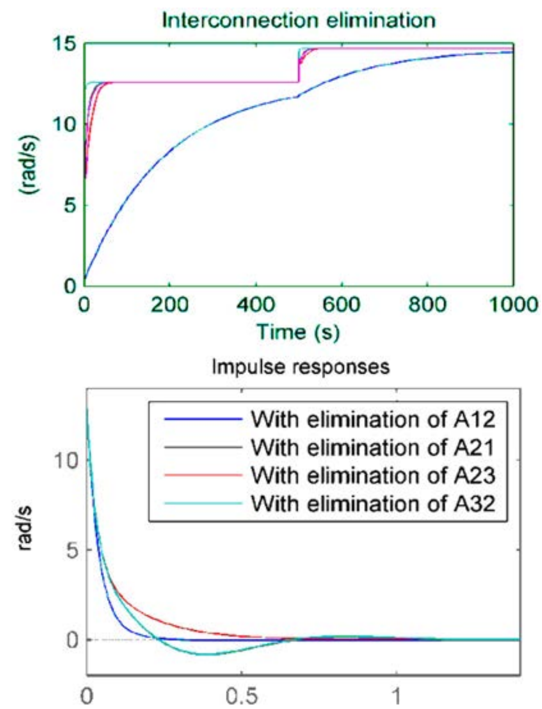

Figure 4: Comparison between system responses with and without interconnection terms

subsystem 2 is $u_2(t) = -\tilde{\boldsymbol{K}}_2(t)\tilde{X}_2(t)$, and the resulting gain matrix for this expanded system is then

$$\tilde{K}(t) = \left[\begin{array}{c:c} \tilde{K}_1(t) & 0 \\ \hdashline 0 & \tilde{K}_2(t) \end{array}\right], \tag{18}$$

The contracted controller based on Expansion– Contraction theory can be found as

$$K(t) = \left[\begin{array}{cc:c} \tilde{K}_{11}(t) & \tilde{K}_{12}(t) & 0 \\ \hdashline 0 & \tilde{K}_{23}(t) & \tilde{K}_{33}(t) \end{array}\right]. \tag{19}$$

Figure 5 shows the repartition of total time required to finish drilling well-1, as it is shown only 28% of time has been spent for drilling; however, 6.33% of the total time has been spent for fishing logging tools that have been stuck in the borehole. The main cause of the stuck is the bad borehole quality, which can be avoided by mitigating the torsional vibrations by applying the designed controller.

6. RESULTS AND DISCUSSION

Torsional vibration, up to full Stick-Slip, is a significant risk to drilling systems. It can be classified into two forms, low and high frequency. Low frequency is known as the conventional torsional vibration and described as the first harmonic mode of the drill string, thus, it is a low risk type.

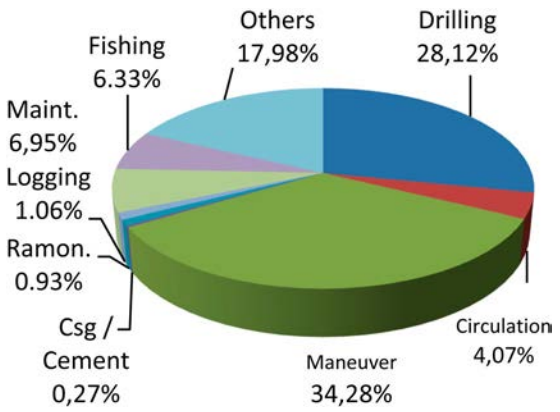

Figure 5: Time analysis of operation progress of 16-inch section of well-1

The high-frequency form is created when torque requirement at the bit exceeds what the BHA can supply. The highest frequency torsional vibrations are correlated with the 4th to 5th harmonic modes. Whereas, they depend on mass, stiffness, and string length. The recorded data demonstrate an important boost in vibration energy at the 5th mode harmonics; it is an abnormal vibration mechanism and requires a robust controller to be mitigated.

Analysis of the downhole data confirms the appearance of high-frequency torsional vibrations preceding axial vibrations from bit chatter. Therefore, it can be said that the high-frequency oscillation in torque is the primary excitation factor and its reduction will decrease severity of all vibrations types. Figure 6 demonstrates that the rate of penetration has decreased with depth, and this is even with increasing top drive velocity and decreasing weight on bit. Figure 7 shows the occurrence of 5th mode harmonic torsional vibrations; wherein, the spectrogram plots represent its fast Fourier transform, which highlights how the frequency of vibration changes over the course of the drilled section.

The results of designing decentralized controller on the rotary drilling system are shown in Figure 8, as in Figure 8 (top left) the overlapping decomposition has improved the time response of the system ($t = 4$ s) in comparison to original system ($t = 180$ s), moreover, the oscillations at the bit have been reduced in reasonable time. Thus, the robustness of state output feedback controller has ameliorated.

In Figure 8 (top right): the system response did not track the reference input for the original system (without decomposition strategy); however; for the designed controller the system responds quickly ($t = 3.4$ s) to the top drive angular velocity variations.

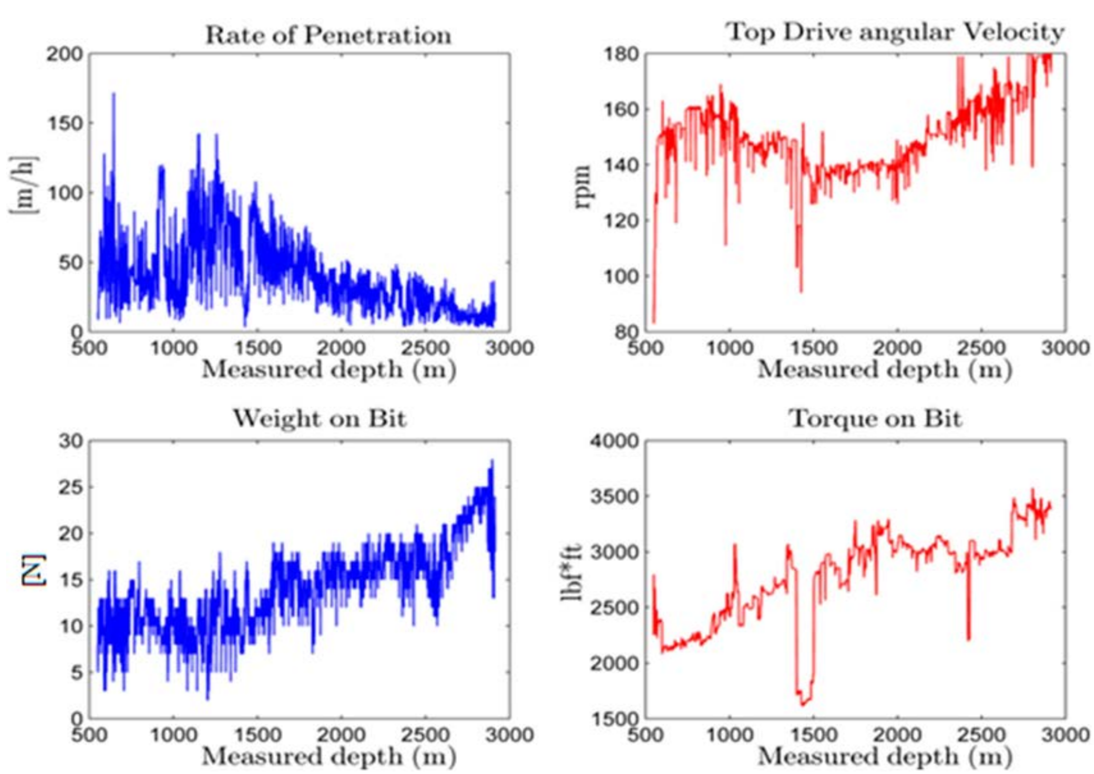

Figure 6: Measured Parameters of Top drive and estimated curves of bit for 16-inch section

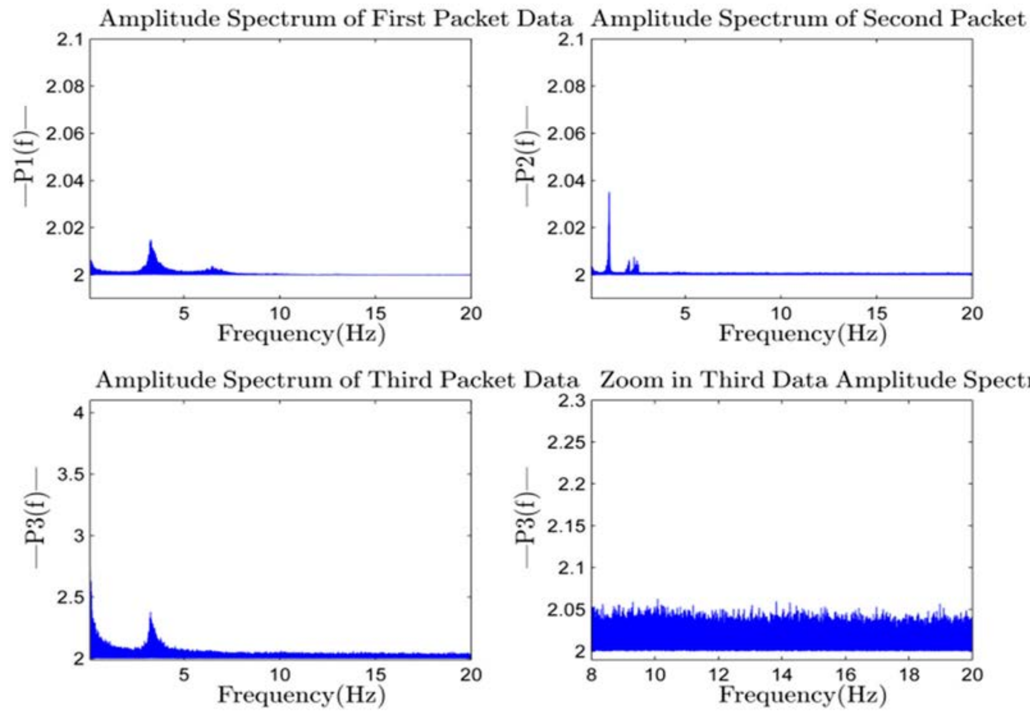

Figure 7: Fast Fourier transform of measured data in the 16-inch section

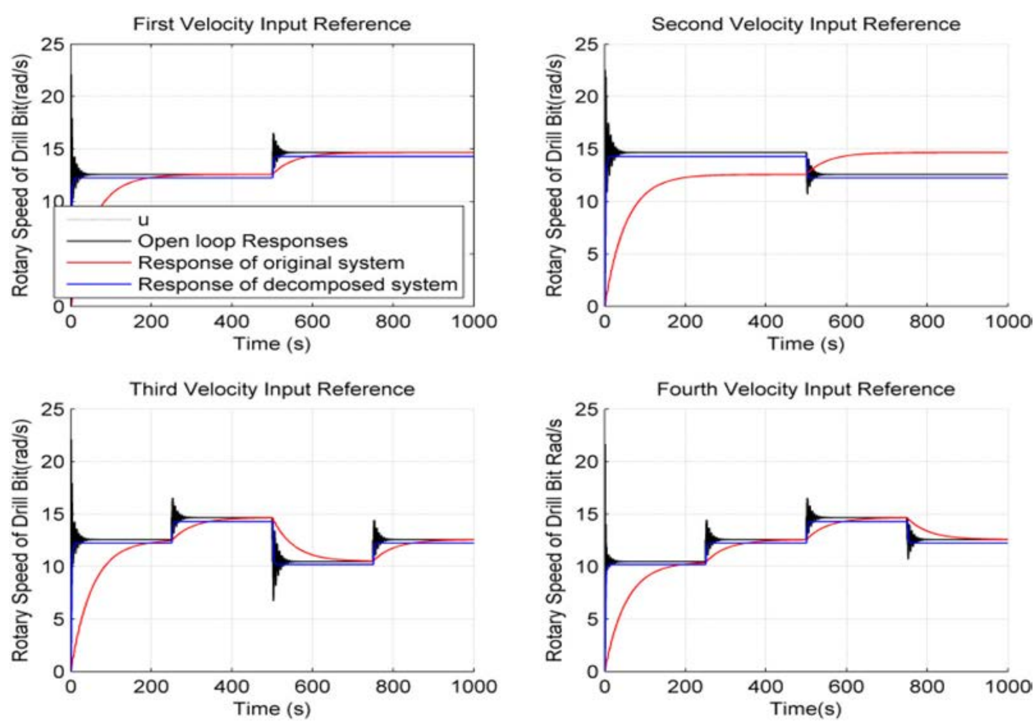

Figure 8: Results of decentralized controller responses to for different velocity input scenarios

In Figure 8 (bottom left): the designed controller follows a more varying input reference, wherein the time response is very short in comparison to the original system; this will prevent the drill bit from damage and will reduce the appearance of torsional vibration, consequently, it will improve borehole quality and reduce drilling cost.

In Figure 8 (bottom right): the controlled system reacted to an increasing velocity from the top drive quicker than the original system, which means that this fast response has mitigated the fluctuation in bit velocity. This means that it reduced the torsional vibrations, and tracked the desired velocity quickly, thus, it prevented the system from the regenerative effects of slip phase.

7. CONCLUSIONS

In this paper, an overlapping decentralized controller has been proposed to mitigate the high-frequency torsional vibrations, and consequently reduce stick-slip phenomenon for a rotary drilling system. The controller design was based on applying expansion–contraction theory to the overlapped structure of the model. The controller has been designed on the expanded system, then it has been contracted for implementation to the original one. Thus, the robustness of control system has been improved and time response was minimized. The overlapped controller has been designed based on the graph theory and expansion–contraction technique, the optimality of the controller has been checked by analyzing the results with/without decomposition strategy and using field data measurements of well-1. Moreover, practical comparison has been done by using data of vibration tool logged recently in the Algerian Sahara. These data have confirmed the presence of harmful high-frequency torsional harmonic modes (4th and 5th modes). Since the designed controller' robustness was improved and time response was reduced, this controller can be implemented in real-time control system to reduce these types of torsional vibrations in smart rotary drilling systems.

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AUTHORS

Mohamed Zinelabidine Doghmane was born in Bousàada, Algeria. He holds a Ph.D degree in Electrical engineering From Faculty of Hydrocarbons and Chemistry, university M'hamed Bougara of Boumerdes, Master's degree in Petroleum Geophysics from the National Algerian Petroleum Institute, master's degree in Applied Automatic from the department of Automation, University of Boumerdes, an engineering degree from the National institute of electrical and electronics engineering (Ex-INELEC) Boumerdes, Algeria. Now, he is working with SONATRACH (Algérie), and he is member in research laboratory of Applied Automatic, University of Boumerdes. His current research interest is optimization of control design for complex systems in Petroleum. Engineering.

Corresponding author. Email: mohamedzinelabidine.doghmane@sonatrach.dz

Professor Madjid Kidouche was born in Bordj-Menaiel, Algeria. He received his Electrical Engineering, Master of Sciences, and Ph.D degrees all in control theory. He joined M'hamed Bougara University of Boumerdes, Algeria in 1990 where he is Professor in the automation department and electrification of industrial process. He is a research group head on "Control of complex dynamical systems" at Applied Automatic Control Laboratory. He has been actively involved in several research projects in the fields of control and power system analysis. He is the author and co-author of numerous research publications in international conferences and journals. His research interests include control of dynamic nonlinear systems, stability of large scale systems, and sliding mode control.

Email: mkidocuche@univ-boumerdes.dz

Aimad Ahriche was born in Algiers, Algeria. He received his B.S. degree in electrical engineering from the University of Jijel, Jijel, Algeria, in 2002 and his Master's degree in electrical engineering from the University of Boumerdes, Boumerdes, Algeria, in 2008 and his Ph.D degree in electrical engineering from the same university in 2015.He has been as Associate Professor with the Department of Maintenance, University of Boumerdes, Algeria, since 2008. His current research interests include development of renewable energy systems, power converters, and AC drives. He is expected to attend full professor degree in the next few years.

Email: A.Ahriche@univ-boumerdes.dz