

Article

Design of an Intelligent Cascade Control Scheme Using a Hybrid Adaptive Neuro-Fuzzy PID Controller for the Suppression of Drill String Torsional Vibration

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Abstract: Eliminating the excessive stick-slip torsional vibrations of drill strings by utilizing an effective controller can significantly increase penetration rates and reduce drilling operation costs. The present study aims to develop a fast and intelligent cascade control structure for a multi-degree-of-freedom drill string system under torsional vibrations. The proposed control configuration consists of two control loops in a cascade arrangement. The first controller estimates the top drive velocity reference from the actual drill bit velocity and its reference. The role of the second controller is to regulate the top drive velocity to its reference by generating the necessary torque for the drilling operation process to progress. Each control loop is designed based on a hybrid adaptive neuro-fuzzy PID and feedforward term. The latter assures fast regulation when there are sudden changes in operation. To evaluate the performance of the suggested cascade feedforward neuro-fuzzy PID (CFF-NFPID) control structure, extensive simulations were conducted using Matlab/Simulink. The simulation results clearly showed that the proposed CFF-NFPID controller provided high-performance control with variations in the weight on the bit and the desired drill bit rotary speed in comparison with that of cascade feedforward fuzzy PID, sliding-mode, cascade feedforward PID, cascade PID, and conventional PID controllers. Furthermore, the control robustness is very suitable despite the change in the system parameters.



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Keywords: drill strings; stick-slip; cascade control scheme; adaptive neuro-fuzzy; feedforward term

1. Introduction

The control of drill string systems in oil wells to improve extraction has received significant attention in the gas and petroleum industry. A drill string is composed of a rotary table, a series of hollow drill pipes, and a bottom hole assembly (BHA). Numerous drill collars and intermediate stabilizers connecting to a drill bit are the main components of a BHA [1,2]. The rotary table provides the required torque on the bit (TOB) to the bottom of the borehole through the components of the drill string system [3,4]. A drill string system is regarded as a complex dynamic system with many indefinite and unstable parameters due to the changes in the characteristics of a drill string that occur during drilling operations. Furthermore, different challenges within the perforation process can lead to failure or an elevation of its overall cost. The interaction between the drill string and the borehole causes many undesired vibrations. Therefore, these vibrations are the most common cause of perforation process failure [4,5]. A permanently stuck bit, which occurs in cases where the bit is unable to rotate, is the most detrimental phenomenon aside from stick-slip at the bit. The stick-slip phenomenon occurs when the top rotary table system maintains a constant rotary speed while the bit's rotary speed fluctuates between zero and

several times the surface rotary speed [1–5]. Many researchers have suggested various passive control methods for mitigating these undesirable phenomena. The most common passive control approaches discussed in the literature include drill string reconfiguration, bit selection with redesign, optimization of the BHA configuration, and the use of anti-vibration down hole tools [3]. These approaches prevent stick–slip vibrations by changing the weight on the bit (WOB) and/or the bit velocity. However, the effectiveness of these approaches is not perfect under all conditions [3–5]. As an example of passive approaches, the authors of [6] introduced a zone with optimal operating conditions that ensured the stability of a drill string system during its operation. The zones bordering the optimal zone underwent stick–slip phenomena, backward whirls, forward whirls, and a low rate of penetration (ROP). Nevertheless, these zones are likely to disappear entirely when drilling into hard formations. Therefore, the control of vibration in drill string systems based on the adjustment of the drilling parameters typically fails, which leads to the degradation of the effectiveness of passive control approaches [3–5].

Because of the improvements in real-time measurement, different closed-loop control strategies have been designed and applied. Initially, a control scheme based on a proportional–integral (PI) regulator was created and implemented in [7,8]. A cascade control scheme was designed in [9] to improve the performance provided by conventional control schemes. In [10], Christoforou and Yigit proposed an optimum state feedback control mechanism to regulate a drill string’s rotational motion. To limit stick–slip vibrations, Serrarens et al. developed a linear H^∞ control strategy [11]. Moreover, in [12], a linear quadratic controller was designed through a linearized model of a drill string. The previous active controllers were considered linear control methods. The performance of this type of controller is limited due to the high nonlinearity of drill string systems. By taking the modeling errors in the control design process in terms of uncertain WOB into consideration, Vernon et al. developed synthesis control strategy for drill string systems [13]. The design of this control strategy requires the linearization of the dynamic model of the drill string system around an operating point. Therefore, the functionality of the controller cannot be confirmed, especially in the case of more degrees of freedom (DOFs).

To avoid the limitations of linear control schemes, in [14], Al-Hiddabi et al. investigated a nonlinear dynamic inversion control method. The main objective of this method was to suppress lateral and torsional vibrations. In [15], Navarro-Lopez and Liceaga-Castro introduced the dynamic sliding-mode theory to design an effective controller (SMC) that avoided different bit-sticking problems. Yang Liu modified the SMC to achieve a tolerance for parameter uncertainties and robustness to variations in the WOB [16]. Nevertheless, the design of nonlinear control schemes is based on a system model in addition to much information about the system, which degrades the control performance.

Recently, artificial intelligence techniques have been increasingly utilized in industrial control applications due to the rapid advancement of signal processing capabilities and the progression of industrial informatics [17–26]. The ability to develop effective control schemes without depending on a mathematical model of the considered system is a key benefit of artificial intelligence. Fuzzy logic control (FLC) [17–21], artificial neural networks (ANNs) [22,23], and adaptive neuro-fuzzy inference systems (ANFISs) [24–26] have been widely employed in the literature to develop intelligent controllers. These techniques are appropriate for developing robust regulators of unknown or uncertain systems. Laib et al. designed an optimal hybrid interval Type-2 fuzzy PID+I logic controller (OH-IT2FPID+I) using a practical swarm optimization algorithm for a multi-degree-of-freedom oil well drill string system [27]. The proposed OH-IT2FPID+I provided high control performance under several critical conditions. However, the FLC’s efficacy was restricted, since it depended on expert knowledge to determine the controller’s optimal gains. For these reasons, ANFISs have recently been employed instead of FLC to improve control performance because they provide more DOFs to deal with nonlinearities [24–26]. ANFIS is a type of ANN that exploits the learning ability of neural networks with the performance of fuzzy linguis-

tics [26]. On the other hand, numerous researchers have attempted to combine FLC with conventional PID control to enhance the control performance [28–30].

To improve the control performance for multi-DOF oil well drill string systems and eliminate stick–slip phenomena, a cascade control scheme for drill string systems was investigated and modified using hybrid neuro-fuzzy PID and feedforward terms (CFF-NFPID). The suggested control configuration involved two control loops. The first controller estimated the top drive velocity reference from the drill bit velocity and its reference. The second controller aimed to regulate the top drive velocity to its reference by generating the required torque. Each control loop was designed based on hybrid adaptive neuro-fuzzy PID and feedforward terms. Adaptive neuro-fuzzy control was combined with conventional PID control to develop a hybrid neuro-fuzzy PID control scheme (H-NFPID). Furthermore, a feedforward term for each control loop was designed based on a mathematical model of a multi-DOF oil well drill string system. The main objective of the proposed CFF-NFPID was to regulate the rotary velocities of the drill string components—especially the rotary table velocity and drill bit velocity—to a desired value in order to overcome several bit-sticking problems and stick–slip vibration. The proposed control structure was evaluated and tested through numerical simulations in Matlab/Simulink with variations in the WOB, the desired drill bit rotary speed, and parametric uncertainties. Furthermore, the performance of the CFF-NFPID controller was quantitatively compared with that of a cascade feedforward fuzzy PID (CFF-FPID) controller, a cascade feedforward PID (CFF-PID) controller, two SMC algorithms, a cascade PID (C-PID) controller, and a conventional PID controller.

The remainder of this article is structured as follows: In Section 2, a discontinuous torsional model for a drill string, which includes bit–rock interactions, is discussed. The modified cascade control structure based on a hybrid neuro-fuzzy PID control structure and feedforward terms is presented in Section 3. This section consists of two parts. The first part discusses the hybrid neuro-fuzzy PID for each control loop, while the second part presents a mathematical demonstration of the suggested feedforward terms. The simulation results are illustrated and analyzed in Section 4. Finally, the main conclusions of the work are drawn and are followed by references.

2. The Mathematical Model of a Drill String System

In the last few years, numerous studies have focused on describing physical phenomena in actual boreholes in order to develop an accurate dynamic mathematical model for drill string systems [31]. The preferred dynamic model for designing active control strategies is called the lumped parameter model. In this mathematical model, a drill string system is conceptualized as multiple rotational mass–spring–damper systems. These systems can be accurately represented using a set of ordinary differential equations. This finite-dimensional structural model represents a comprehensive illustration of drill string dynamics from single to multiple degrees of freedom (DOFs) [31,32].

Figure 1 depicts the mathematical model of a drill string system with four DOFs. This representation consists of four damped inertias, which are depicted as four discs. These inertias are interconnected through shafts using three torsional dampers with damping coefficients d_i and three torsional springs with stiffness coefficients k_i (where $i = t, tl, tb$) [31]. The lumped parameter model consists of a sequence of inertias that represent the drill string elements. Here, the subscripts b , l , p , and r represent the drill bit, drill collars, drill pipe, and rotary table, respectively. An electric motor with torque T_m drives the rotary table via a gearbox. The system is characterized by large damping inertias to avoid rapid variations in angular velocity. The drill string may reach lengths of several kilometers by linking together a sequence of connected drill pipes [32]. The bottom part (BHA) comprises a heavier pipe, which is called a drill collar.

The drilling mud surrounding the drill bit serves to dissipate the drill bit's energy and the substantial torque applied to it [33]. Moreover, friction torque T_{fb} , which reflects rock–bit interactions, is also imposed on the drill bit. Figure 2 depicts the geometric and

material characteristics of the drilling system. In addition, the bit is considered free, while the rotary table remains fixed.

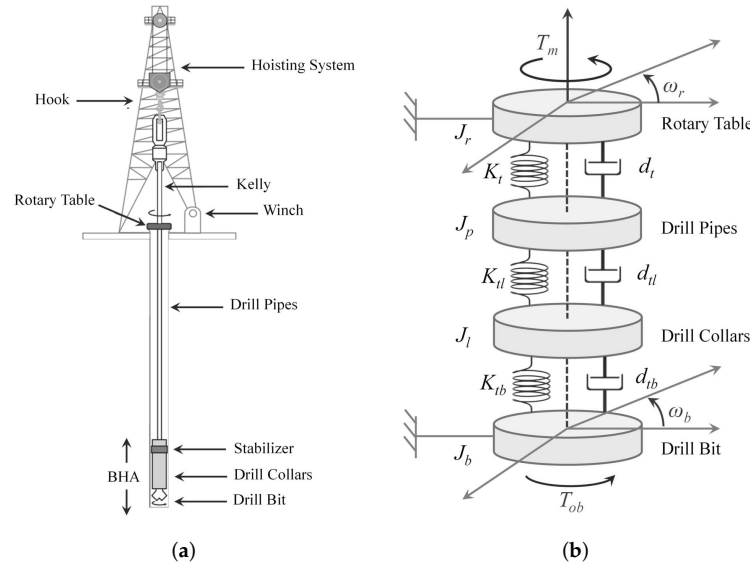


Figure 1. The studied system: (a) a drilling rig with a drill string system; (b) mathematical model of the drill string system [15].

The drill string system's dynamics are represented by Equation (1) with the aforementioned characteristics of a drill string. The state vector of the drill string system is defined in Equation (2) [34].

$$\begin{cases} \ddot{\Omega}_r &= -\frac{d_t}{j_r}(\dot{\Omega}_r - \dot{\Omega}_p) - \frac{k_t}{j_r}(\Omega_r - \Omega_p) - \frac{d_r}{j_r}(\dot{\Omega}_r) + \frac{T_m}{j_r} \\ \ddot{\Omega}_p &= \frac{d_t}{j_p}(\dot{\Omega}_r - \dot{\Omega}_p) + \frac{k_t}{j_p}(\Omega_r - \Omega_p) - \frac{d_{tl}}{j_p}(\dot{\Omega}_p - \dot{\Omega}_l) - \frac{k_{tl}}{j_p}(\Omega_p - \Omega_l) \\ \ddot{\Omega}_l &= \frac{d_{tl}}{j_l}(\dot{\Omega}_p - \dot{\Omega}_l) + \frac{k_{tl}}{j_l}(\Omega_p - \Omega_l) - \frac{d_{lb}}{j_l}(\dot{\Omega}_l - \dot{\Omega}_b) - \frac{k_{lb}}{j_l}(\Omega_l - \Omega_b) \\ \ddot{\Omega}_b &= \frac{d_{lb}}{j_b}(\dot{\Omega}_l - \dot{\Omega}_b) + \frac{k_{lb}}{j_b}(\Omega_l - \Omega_b) - \frac{T_{ob}(x)}{j_b} \end{cases} \quad (1)$$

$$\mathbf{X} = [\dot{\Omega}_r, (\Omega_r - \Omega_p), \dot{\Omega}_p, (\Omega_p - \Omega_l), \dot{\Omega}_l, (\Omega_l - \Omega_b), \dot{\Omega}_b]^T \quad (2)$$

where j is the inertia, Ω is the angular position, $\dot{\Omega}$ is the angular velocity, k is the stiffness, and d is the damping. The torques on the system are the drive torque T_m , the torque on the bit (TOB) T_{ob} , and the rock-crushing torque T_{fb} . Equation (2) is simplified as follows:

$$\mathbf{X} = [x_1, x_2, x_3, x_4, x_5, x_6, x_7]^T \quad (3)$$

By combining Equations (1) and (3), the mathematical model based on the state equations for the system model can be rewritten as follows:

$$\begin{cases} \dot{x}_1 &= \frac{1}{j_r}[-(d_r + d_t)x_1 - k_t x_2 + d_t x_3 + T_m] \\ \dot{x}_2 &= x_5 - x_7 \\ \dot{x}_3 &= \frac{1}{j_p}[d_t x_1 + k_t x_2 - (d_{tl} + d_t)x_3 - k_{tl} x_4 + d_{tl} x_5] \\ \dot{x}_4 &= x_3 - x_5 \\ \dot{x}_5 &= \frac{1}{j_l}[d_{tl} x_3 + k_{tl} x_4 - (d_{lb} + d_{tl})x_5 - k_{lb} x_6 + d_{lb} x_7] \\ \dot{x}_6 &= x_5 - x_7 \\ \dot{x}_7 &= \frac{1}{j_b}[d_{lb} x_5 + k_{lb} x_6 - (d_{tb} + d_b)x_7 - T_{ob}(x)] \end{cases} \quad (4)$$

The torque on the bit $T_{ob}(x)$ is defined as follows:

$$T_{ob} = T_v(x_7) + T_{fb}(X) \quad (5)$$

where $T_v(x_7)$ is the reaction torque of the drilling mud on the drill bit and is defined as

$$T_v(x_7) = d_b x_7 \quad (6)$$

The interaction between the drill bit and the rock dissipates most of the energy generated by the drive motor [5]. The main cause of vibrations in a drill string is rock–bit interaction. Consequently, the creation of an interaction model requires a detailed analysis of the induced vibration [35].

Friction models have been proposed in the literature [16,36–38]. The Stribeck model in [16,36,39] and Karnopp’s model in [36] have been the most frequently employed in recent years. However, the Stribeck model suffers from instability and zero-speed discontinuity. To overcome these weaknesses, Karnopp’s model is exploited to establish a rock–bit interaction model in which cutting and frictional contact are taken into account [36] (see Figure 2).

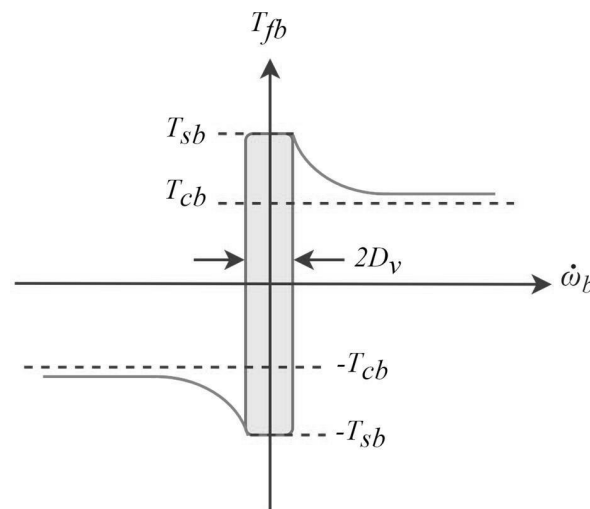


Figure 2. Karnopp’s rock–bit interaction model [36].

The reaction torque is estimated using the following mathematical model:

$$T_{fb}(x) = \begin{cases} T_{eb}(x) & \text{if } |x_7| < D_v, \quad |T_{eb}| \leq T_{sb} \\ T_{sb} \operatorname{sign}(T_{eb}(x)) & \text{if } |x_7| < D_v, \quad |T_{eb}| > T_{sb} \\ T_{cb} \operatorname{sign}(x_7) & \text{if } |x_7| \geq D_v \end{cases} \quad (7)$$

where

$$T_{eb} = d_{tb}(x_5 - x_7) + k_{tb}x_6 - d_b x_7 \quad (8)$$

$$T_{sb} = w_{ob} R_b u_{sb} \quad (9)$$

$$u_b(x_7) = u_{cb} + (u_{sb} - u_{cb}) e^{\frac{\gamma_b}{V_f} |x_7|} \quad (10)$$

$$T_{cb} = w_{ob} R_b u_b(x_7) \quad (11)$$

where $T_{eb}(X)$, T_{cb} , and T_{sb} are the breakaway torque, the torque due to Coulomb friction, and the torque due to static friction, respectively. w_{ob} is the weight on the bit (WOB), and R_b is the radius of the drill bit. $u_b(x_7)$, u_{sb} , and u_{cb} are the coefficients for the dry friction, static friction, and Coulomb friction of the drill bit, respectively.

The multi-variable form of the lumped parameter model of the drill string is as follows [38]:

$$\dot{x}(t) = M_1 X(t) + M_2 T_m + M_3 T_{fb}(X) \quad (12)$$

where the matrices M_1 , M_2 , and M_3 can be easily deduced by solving Equations (3), (4) and (12).

3. The Proposed Cascade Control Scheme

In this section, a recursive cascade control scheme is designed to mitigate stick–slip vibrations by controlling the rotary velocities of drill string elements. The controller regulates the drill bit velocity and top drive velocity to predefined values. The proposed control scheme has two feedback controllers, as illustrated in Figure 3. Each control loop is designed based on a hybrid neuro-fuzzy PID controller (H-NFPID) and feedforward term. From the error between the velocity of the drill bit and the reference velocity, the first controller estimates the variation in the top drive velocity reference (Δx_1^*). Then, Δx_1^* is added to the feedforward term of the first control loop x_F to generate the top drive velocity reference x_1^* as follows:

$$x_1^* = x_F + \Delta x_1^* \quad (13)$$

The role of the second control loop is to regulate the top drive velocity to its reference and generate the torque (u) required to suppress torsional vibrations. The required torque is calculated by adding the feedforward term u_F of the torque to the variation in the torque Δu generated by the second controller.

$$u = u_F + \Delta u \quad (14)$$

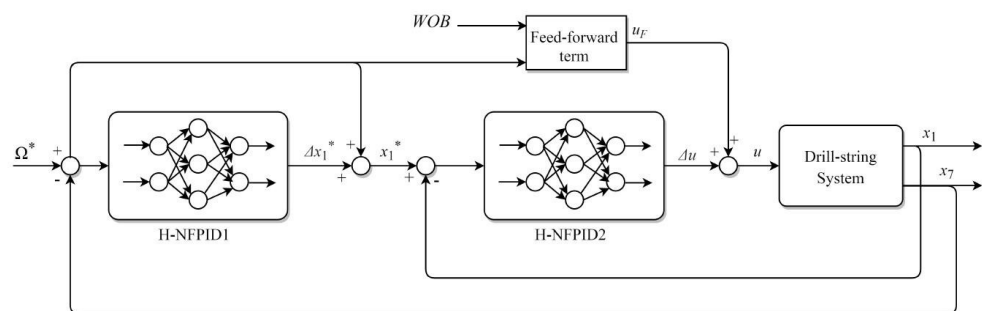


Figure 3. The proposed cascade control scheme.

3.1. Hybrid Neuro-Fuzzy PID Controller

The adaptive neuro-fuzzy inference system (ANFIS) combines the benefits of neural networks and fuzzy logic systems. The ANFIS was first designed by Mr. Jang in 1995 [24–26]. A fuzzy logic system offers a structural framework for neural networks based on membership functions and fuzzy rules (if–then). A connection structure and learning capabilities are provided by neural networks to the fuzzy logic system [26]. To improve the system efficiency, the role of artificial neural networks is to train and optimize the fuzzy parameters in membership functions [26]. The structure of an ANFIS with two inputs and one output is depicted in Figure 4. The ANFIS consists of five layers, and each layer is composed of several nodes. To fuzzify the inputs, several computing nodes with activation functions based on fuzzy logic membership functions are used to make up Layer 1. The role of Layer 2 is to select the minimum value of the inputs. In Layer 3, the normalization of each input is carried out. Then, the outputs of Layer 3 are applied to the defuzzifiers in Layer 4. In the last layer, the output signals are generated by accumulating all of the fuzzifiers’ outputs. In this study, the ANFIS was integrated with a conventional PID controller to design two hybrid neuro-fuzzy PID controllers (H-NFPID) for the first and the second control loops. The structures of H-NFPID1 and H-NFPID2 are shown in Figure 5. In Figure 5a, the error tracking and its derivative obtained from the values of the drill bit and reference velocities are considered as the inputs of the H-NFPID1 controller:

$$e_1 = k_{e1}(\Omega^* - x_7) \quad (15)$$

$$C_{e1} = k_{ce1} \frac{d(\Omega^* - x_7)}{dt} \quad (16)$$

where k_{e1} and k_{ce1} represent the normalizing coefficients for the inputs of H-NFPID1, while Ω^* is the desired velocity.

The output of H-NFPID1 can be written as follows:

$$\Delta x_1^* = k_{p1}X + k_{i1} \int X dt + k_{d1} \frac{de_1}{dt} \quad (17)$$

where k_{p1} and k_{i1} are the denormalizing proportional coefficient and integral coefficient, respectively, k_{d1} is the derivative coefficient, and X is the output of ANFIS1.

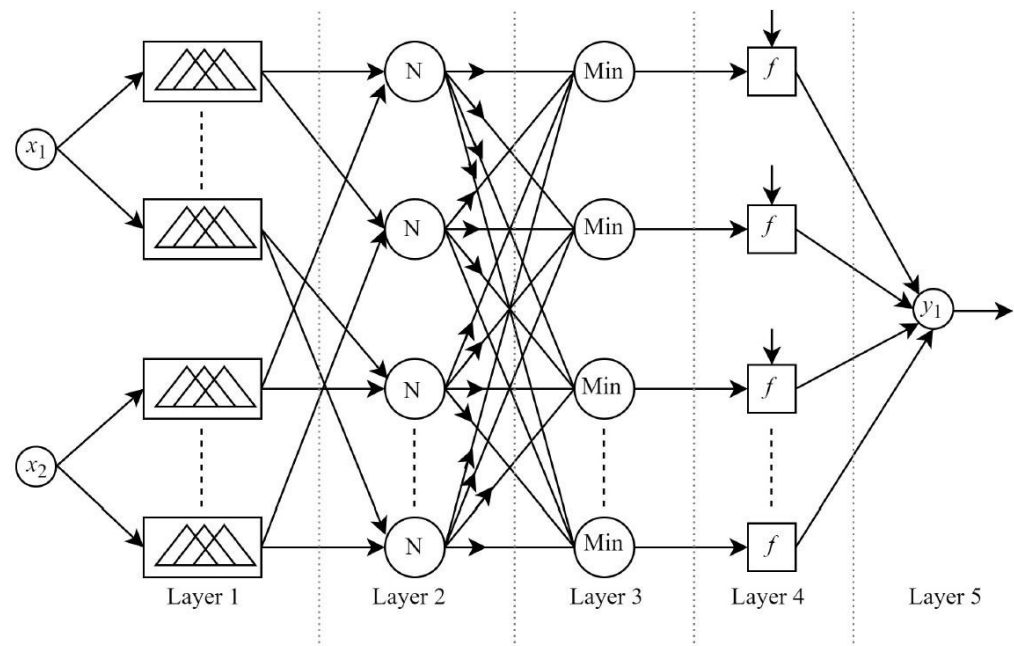


Figure 4. The structure of the ANFIS.

As shown in Figure 5b, the inputs of the H-NFPID2 controller are the error tracking and its derivative for the angular velocities of the rotary table velocity:

$$e_2 = k_{e2}(x_1^* - x_1) \quad (18)$$

$$C_{e2} = k_{ce2} \frac{d(x_1^* - x_1)}{dt} \quad (19)$$

where k_{e2} and k_{ce2} represent the normalizing coefficients for the inputs of H-NFPID2, while x_1^* is the desired angular velocity for the rotary table. The output of H-NFPID1 can be written as follows:

$$\Delta u = k_{p2}U + k_{i2} \int U dt + k_{d2} \frac{de_2}{dt} \quad (20)$$

where k_{p1} , k_{i1} , k_{d1} , and U are the denormalizing proportional coefficient, the denormalizing integral coefficient, the derivative coefficient, and the output of ANFIS2, respectively. The objective of the derivative term used in the proposed H-NFPID1 and H-NFPID2 controllers is to decrease the settling time and enhance the stability of the drill string. On the other hand, ANFIS1 and ANFIS2 were designed with minimum fuzzy rules for the first and second control loops, respectively, in order to decrease the computational burden. Here, three Gaussian functions were employed as input membership functions for the two ANFIS controllers, in addition to only nine fuzzy rules.

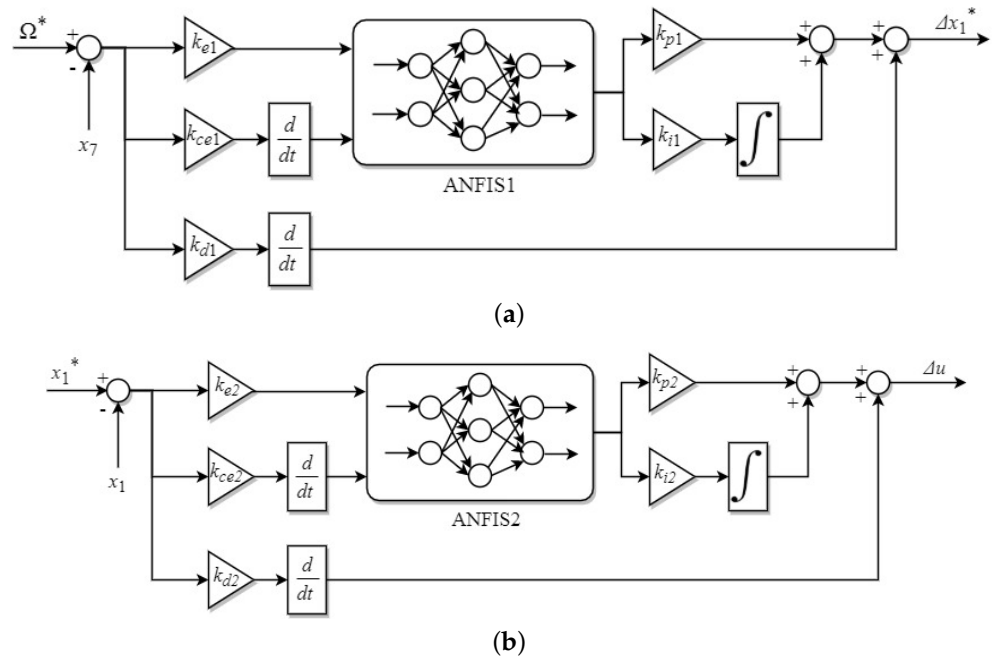


Figure 5. The proposed hybrid neuro-fuzzy PID controller: (a) ANFIS1; (b) ANFIS2.

3.2. Design of the Feedforward Term

One of the main contributions of this study is the use of a feedforward term in each control loop in the case of a cascade control scheme for a drill string system. The major benefit of the feedforward term is the provision of a fast dynamic response when there are changes in the angular velocity reference and the WOB. To estimate the feedforward term, the system model should be studied in the steady state without stick-slip vibrations as follows [40]:

$$\dot{x}_1 = \dot{x}_2 = \dot{x}_3 = \dot{x}_4 = \dot{x}_5 = \dot{x}_6 = \dot{x}_7 = 0 \quad (21)$$

$$x_1 = x_3 = x_5 = x_7 = \Omega^* \quad (22)$$

In Equation (22), the feedforward term used in the first control loop is the angular velocity reference.

$$x_F = \Omega^* \quad (23)$$

Then, by substituting Equations (21) and (22) into (4), the feedforward term used in the second control loop is defined as follows:

$$u_{FF} = (c_t + c_r)\Omega^* + k_t x_2 - c_t \Omega^* \quad (24)$$

where x_2 can be written according to Equations (21), (22), and (4) as follows:

$$x_2 = \frac{1}{k_t} [-c_t \Omega^* + (c_t + c_{tl})\Omega^* + k_{tl} x_4 - c_{tl} \Omega^*] \quad (25)$$

Equation (16) can be simplified as follows:

$$x_2 = \frac{k_{tl} x_4}{k_t} \quad (26)$$

According to Equation (4), x_4 in the steady state can be defined as

$$x_4 = \frac{1}{k_{tl}} [-c_{tl} \Omega^* + (c_{tl} + c_{tb})\Omega^* + k_{tb} x_6 - c_{tb} \Omega^*] \quad (27)$$

The simplified version of Equation (27) is described by Equation (28):

$$x_4 = \frac{k_{tb}x_4}{k_{tl}} \quad (28)$$

In addition, x_6 should be defined in the steady state as follows:

$$x_6 = \frac{1}{k_{tb}} [-c_{tb}\Omega^* + (c_{tb} + c_b)\Omega^* + T_{fb}(X)] \quad (29)$$

where the nonlinear torque on the bit T_{fb} can be estimated in the case without stick–slip vibrations and in the steady state as follows:

$$T_{fb}(X) = W_{ob}R_bu_b(\Omega^*) \quad (30)$$

Finally, the torque feedforward term for the second control loop is given using the following simplified equation:

$$u_F = c_r\Omega^* + c_b\Omega^* + W_{ob}R_bu_b(\Omega^*) \quad (31)$$

Regarding Equation (31), the feedforward term for the second control loop is estimated according to some system parameters, the angular velocity reference, and the WOB. The WOB can be adjusted by the operator.

4. Simulation Results

To confirm that the proposed CFF-NFPID control approach can be practically implemented in drill string systems, extensive simulations were carried out using Matlab/Simulink. The four-DOF drill string mathematical model presented in Equation (4) was implemented in an embedded Matlab function as program lines to simulate the behavior of a drill string system. This mathematical model considered nonlinear rock–bit interactions, including the mud-drilling effect and friction torque. Furthermore, the studied controllers were implemented using components of Matlab/Simulink toolboxes. The global system parameters and the coefficients of the controllers are illustrated in Table 1. The performance of the proposed CFF-NFPID approach was tested under three conditions: (i) a stepwise and slow change in the reference velocity; (ii) variation in the WOB at a constant reference velocity; (iii) parametric uncertainties at a constant reference velocity. To evaluate the efficacy of the CFF-NFPID strategy, a quantitative comparison with several controllers (CFF-FPID, CFF-PID, SMC [15], SMC [16], cascade PID, and conventional PID) was conducted.

Table 1. Numerical values of the drill string system parameters [15].

Parameter	Element(s)	Symbol	Value	Units
Inertia	rotary table	J_r	930	kg·m ²
Inertia	drill collar	J_l	750	kg·m ²
Inertia	drill pipes	J_p	2782.25	kg·m ²
Inertia	drill bit	J_b	471.97	kg·m ²
Stiffness	drill pipes and rotary table	k_t	698.06	N·m/rad
Stiffness	drill pipes and drill collar	k_{tl}	1080	N·m/rad
Stiffness	drill bit and drill collar	k_{tb}	907.48	N·m/rad
Damping	drill pipe and rotary table	d_t	139.61	N·m·s/rad
Damping	drill collar and drill pipe	d_{tl}	190	N·m·s/rad
Damping	drill bit and drill collar	d_{tb}	181.49	N·m·s/rad
Damping	drill bit and mud drilling	d_b	50	N·m·s/rad
Weight on bit	drill bit	WOB	100	kN
Radius	drill bit	R_b	0.156	m
Factor	drill bit	γ_b	0.9	—
Limit velocity		D_v	10 ^{−6}	rad/s
Drive torque		T_m	10	kN·m

Table 1. Cont.

Parameter	Element(s)	Symbol	Value	Units
Coefficient of static friction		μ_{sb}	0.8	—
Coefficient of Coulomb friction		μ_{cb}	0.5	—
Coefficient of viscous damping		d_r	425	N·m·s/rad

4.1. Effect of Applying the Proposed CFF-NFPID Approach to a Drill String System

Figure 6 shows the behavior of a drill string system with a WOB of 97 kN and without a closed control loop. It can be observed that the drill string system suffered from stick–slip phenomena in this situation. The velocities of the rotary table and drill bit periodically oscillated from zero to a value that surpassed the rotary table’s velocity in addition to the variation in the drill bit velocity. These oscillations negatively affected the drill string components and could cause failure or complete damage. For this reason, the utilization of closed-loop control in drill string systems is imposed as a compulsory solution for dampening these oscillations. As described in this section, the proposed CFF-NFPID control scheme was implemented to eliminate stick–slip vibrations. Furthermore, the performance of the proposed CFF-NFPID strategy was tested when undergoing sudden and slow changes in the drill bit reference when the WOB was set to 100 kN, as shown in Figures 7 and 8.

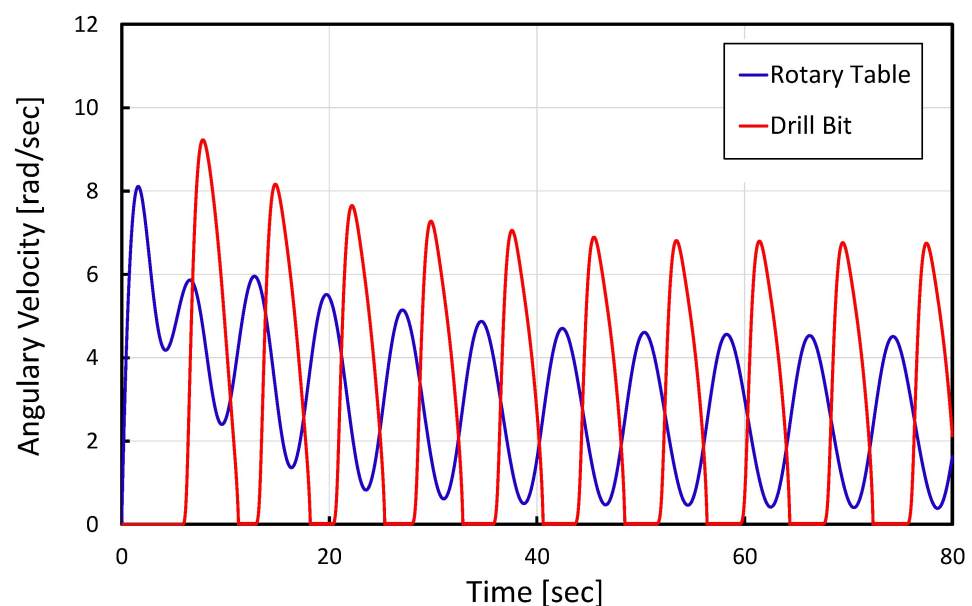


Figure 6. Simulation results with a WOB of 97 kN and without a closed control loop.

As illustrated in Figure 7a, for the time range from 0 s to 150 s, the reference angular velocity was set to 12 rad/s. The proposed controller dampened the stick–slip vibration and guaranteed a quick stabilization of both the rotary table and the drill bit to the reference angular velocity at $t = 25.33$ s and $t = 24.73$ s, respectively, with a very small peak overshoot. Then, a stepwise decrease in the change in the drill bit reference from 12 rad/s to 8 rad/s was applied at 150 s. The angular velocities of both the rotary table and the drill bit were quickly stabilized to the desired drill bit reference without undershoots at $t = 25.33$ s and $t = 24.73$ s, respectively. Furthermore, a large and sudden increase in the change in the drill bit reference from 8 rad/s to 14 rad/s occurred at 300 s. In addition, the CFF-NFPID strategy provided a fast stilling time for the angular velocities of both the rotary table and the drill bit at $t = 15.40$ s and $t = 12.50$ s, respectively, without overshoots. The simulation response of the drive torque with sudden changes in the drill bit reference is presented in Figure 7b. It can be clearly observed that the drive torque increased or decreased depending on the change in the angular velocity reference with a small overshoot/undershoot (less

than 18%). As presented in Figure 8a, a slow decrease in the variation in the reference angular velocity from 12 rad/s to 8 rad/s occurred from 75 s to 175 s. Moreover, a slow increase in the variation in the reference angular velocity from 8 rad/s to 12 rad/s occurred from 250 s to 350 s. Despite these changes, the proposed CFF-NFPID controller guaranteed accurate angular velocity tracking. On the other hand, the drive torque increased or decreased depending on the slow variation in the angular velocity reference with very small deviations, as presented in Figure 8b.

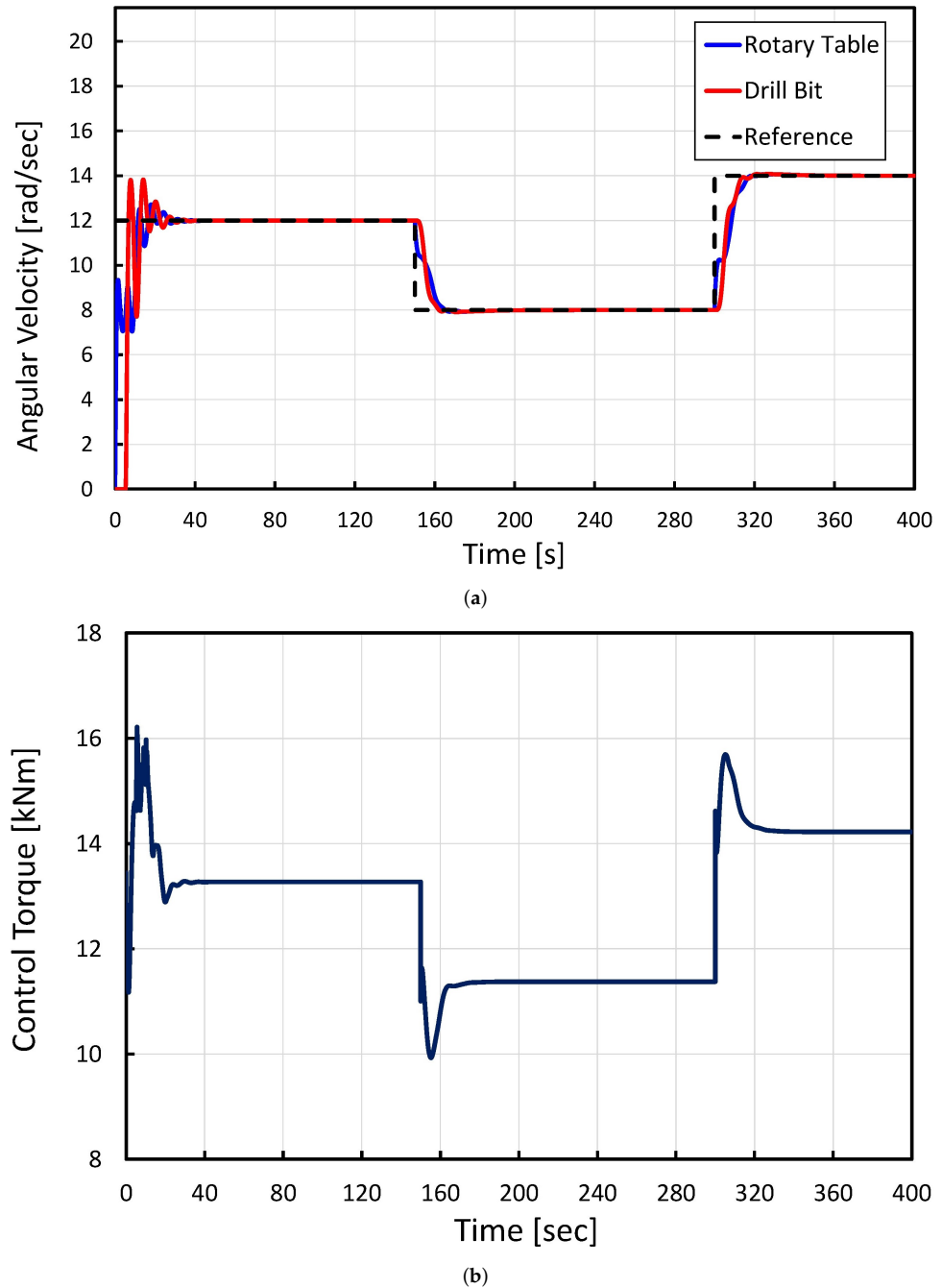
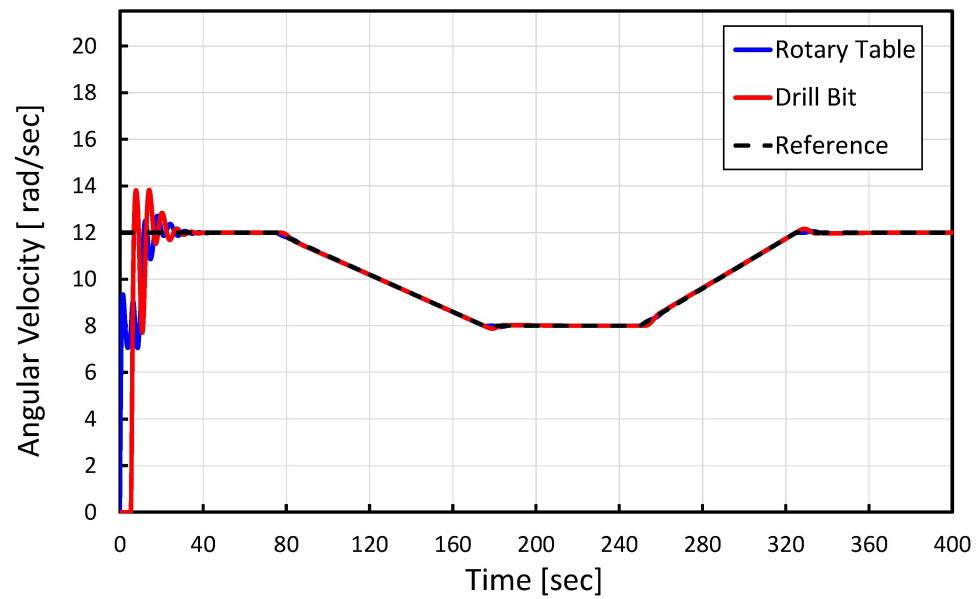
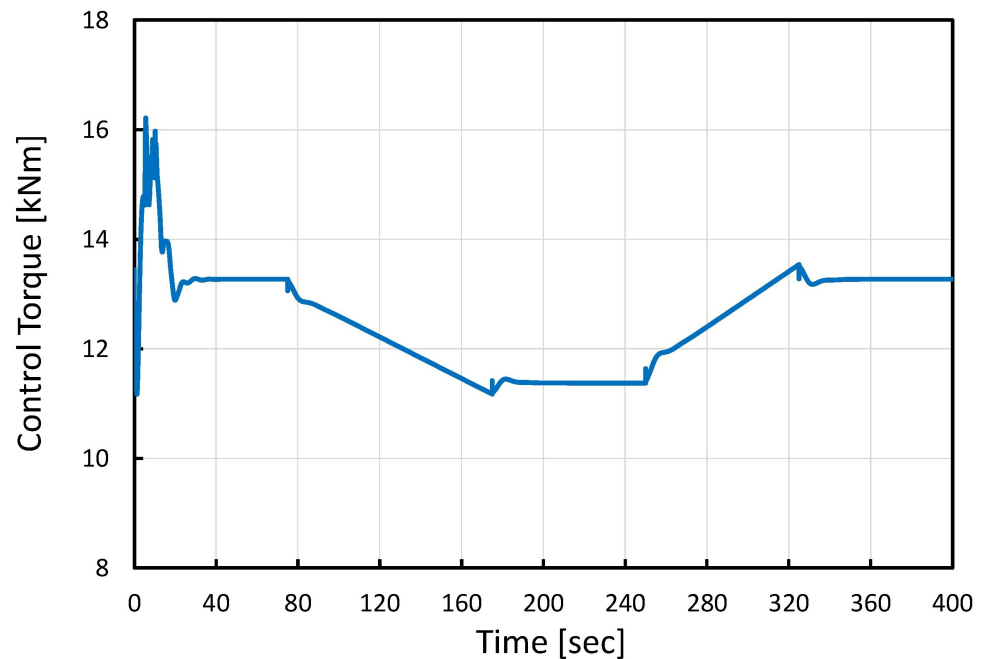


Figure 7. Drill string stabilization when using the CFF-NFPID controller with sudden change in the drill bit reference and WOB = 100 kN: (a) angular velocities; (b) control torque.



(a)



(b)

Figure 8. Drill string stabilization when using the CFF-NFPID controller with slow changes in the drill bit reference and WOB = 100 kN: (a) angular velocities; (b) control torque.

4.2. Quantitative Comparison under Critical Conditions

The performance of the CFF-NFPID controller was compared with that of the CFF-FPID, CFF-PID, and SMC controllers proposed in [15], the SMC proposed in [16], and the cascade PID and conventional PID controllers when there were sudden changes in the desired drill bit velocity to evaluate the benefits of using the suggested control scheme. All of the controllers were simulated on the drill string system with the parameters listed in Table 1. Figure 9 shows the simulated responses of the different controllers when there were sudden variations in the drill bit's angular velocity reference. Furthermore, a quantitative performance analysis in terms of the peak overshoot/undershoot and the settling time of the different controllers is summarized in Table 2. It can be noted that the proposed CFF-NFPID approach provided high-performance control in the dynamic state (fast settling

time and very small overshoot/undershoot) in comparison with the other controllers. In the steady state, all of the controllers succeeded in regulating the velocities of both the drill bit and rotary table to follow the desired velocity. It can also be noted in Table 2 that employing the cascade PID controller improved the control performance in comparison with that of the conventional PID controller. On the other hand, the CFF-PID controller provided enhanced control performance in comparison with that of the SMC algorithms proposed in [15,16]. In addition, we noticed that, with the exception of the proposed CFF-NFPID controller, the CFF-FPID controller provided better performance than that of the other controllers in all cases of changes in the reference.

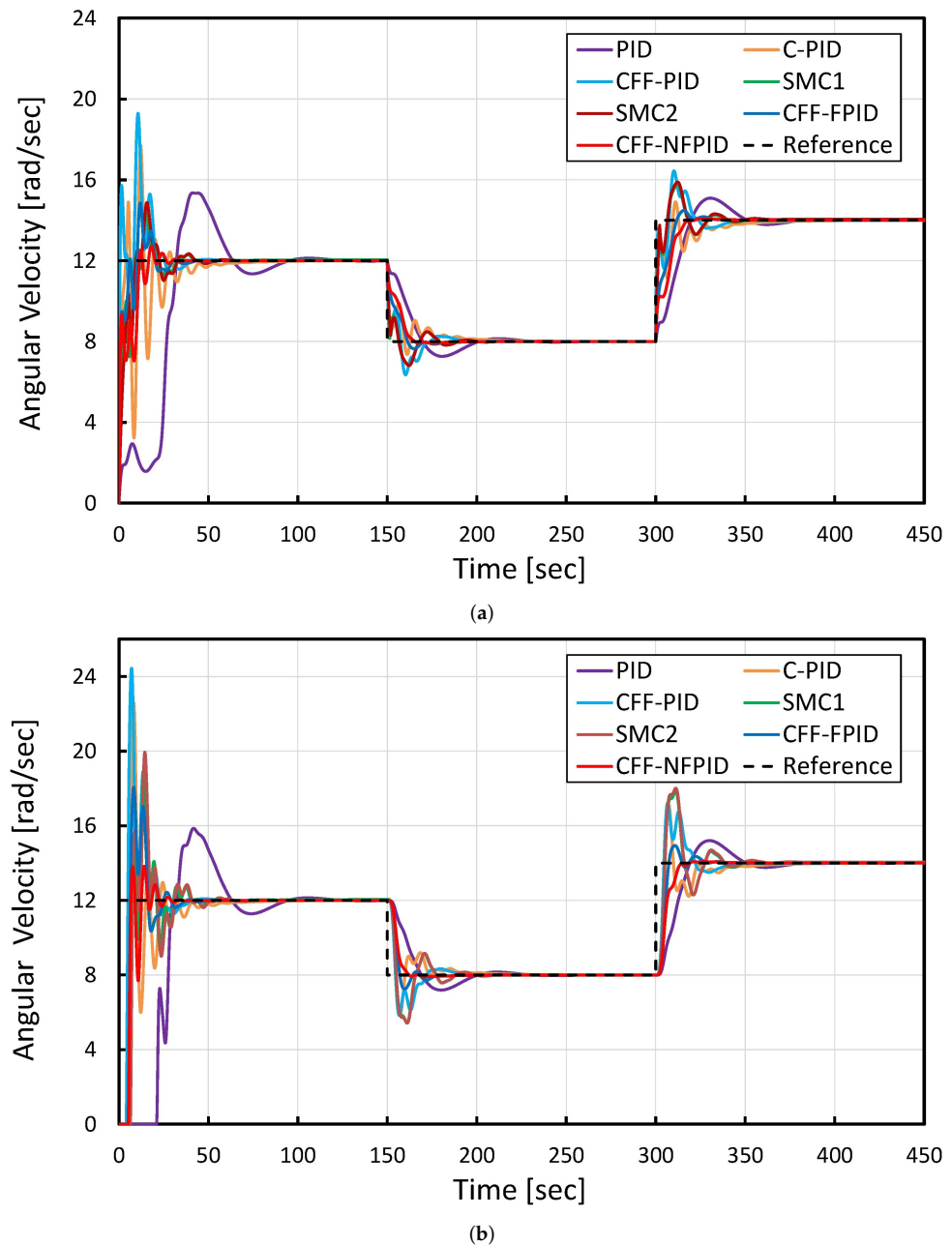


Figure 9. Simulated waveforms of the angular velocity during variations in the desired angular velocity with different controllers: (a) rotary table; (b) drill bit.

Table 2. Quantitative comparison with a fixed WOB and variations in the desired angular velocity.

	Stepwise Increase in the Angular Velocity Reference 0→12 rad/s				Stepwise Decrease in the Angular Velocity Reference 12→8 rad/s				Stepwise Increase in the Angular Velocity Reference 8→14 rad/s			
	Angular Velocity of the Rotary Table		Angular Velocity of the Drill Bit		Angular Velocity of the Rotary Table		Angular Velocity of the Drill Bit		Angular Velocity of the Rotary Table		Angular Velocity of the Drill Bit	
	T_r [s]	U_p [rad/s]	T_r [s]	U_p [rad/s]	T_r [s]	U_p [rad/s]	T_r [s]	U_p [rad/s]	T_r [s]	U_p [rad/s]	T_r [s]	U_p [rad/s]
PID	87.87	3.86	87.94	3.35	50.05	0.82	51.02	0.74	45.65	1.21	45.89	1.09
C-PID	50.63	5.71	54.12	10.60	42.15	0.65	39.35	2.16	41.43	0.92	38.77	3.16
CFF-PID	35.4	7.51	34.1	12.08	35.12	1.72	34.51	2.28	34.43	2.57	33.89	3.38
SMC [15]	39.45	2.25	47.56	6.73	36.02	1.51	33.92	1.15	36.03	3.87	26.63	1.79
SMC [16]	40.56	3.83	48.77	7.96	36.29	2.57	34.65	1.19	36.62	4.02	35.21	1.89
CFF-FPID	28.68	2.85	32.61	6.07	18.11	0.36	24.61	0.76	18.21	0.48	24.22	0.94
CFF-NFPID	25.33	0.72	24.73	1.82	13.13	0.00	10.78	0.00	15.40	0.00	12.50	0.00

Secondly, another quantitative comparison was conducted when considering a sudden change in the WOB from 10 kN to 8 kN at 150 s and from 8 kN to 12 kN at 300 s. The reference angular velocity was set to 12 rad/s. The simulation waveforms obtained with different controllers are shown in Figure 10. The detailed control performance of all controllers is presented in Table 3. High-performance control in terms of the settling time and peak overshoot was guaranteed by the proposed controller in comparison with the other controllers. On the other hand, the C-PID controller improved the performance obtained with the conventional PID controller. The CFF-PID controller provided a fast settling time and low peak overshoot/undershoot in comparison with the SMC algorithms, as we noted in the previous test. Due to the fact that the design of the feedforward term requires some system parameters to be set, a quantitative comparison in the case of parametric uncertainty was conducted. The purpose of this critical test was to show if the proposed control scheme was able to keep the superior performance that it had in the previous tests. The obtained waveforms are presented in Figure 11, and the detailed control performance is summarized in Table 4. It can be clearly observed that the control performance of all controllers was degraded in this critical scenario. However, the proposed CFF-NFPID controller still provided better performance than that of the other controllers. Generally, the simulation results that were obtained proved the greater ability of the proposed control scheme to suppress stick-slip vibrations. In addition, the CFF-NFPID controller achieved high control performance in terms of the response time overshoot/undershoot in different critical scenarios in comparison with the other controllers.

Table 3. Quantitative comparison using a constant desired angular velocity and variations in the WOB.

	Stepwise Decrease in the WOB 100→120 kN				Stepwise Increase in the WOB 120→80 kN			
	Angular Velocity of the Rotary Table		Angular Velocity of the Drill Bit		Angular Velocity of the Rotary Table		Angular Velocity of the Drill Bit	
	Settling Time [s]	Overshoot [rad/s]	Settling Time [s]	Overshoot [rad/s]	Settling Time [s]	Overshoot [rad/s]	Settling Time [s]	Overshoot [rad/s]
PID	51.02	2.29	50.56	1.84	56.27	4.59	56.37	3.67
C-PID	19.02	1.17	15.56	2.08	27.01	2.31	30.83	4.13
CFF-PID	16.89	0.98	19.48	2.06	23.36	1.95	20.59	4.11
SMC [15]	19.32	0.91	26.87	2.19	22.66	1.78	31.86	4.36
SMC [16]	19.31	0.89	26.86	2.19	22.65	1.78	31.85	2.37
CFF-FPID	14.13	0.62	17.86	2.07	20.09	1.09	20.92	4.13
CFF-NFPID	11.82	0.79	12.45	2.06	15.75	1.09	16.23	4.12

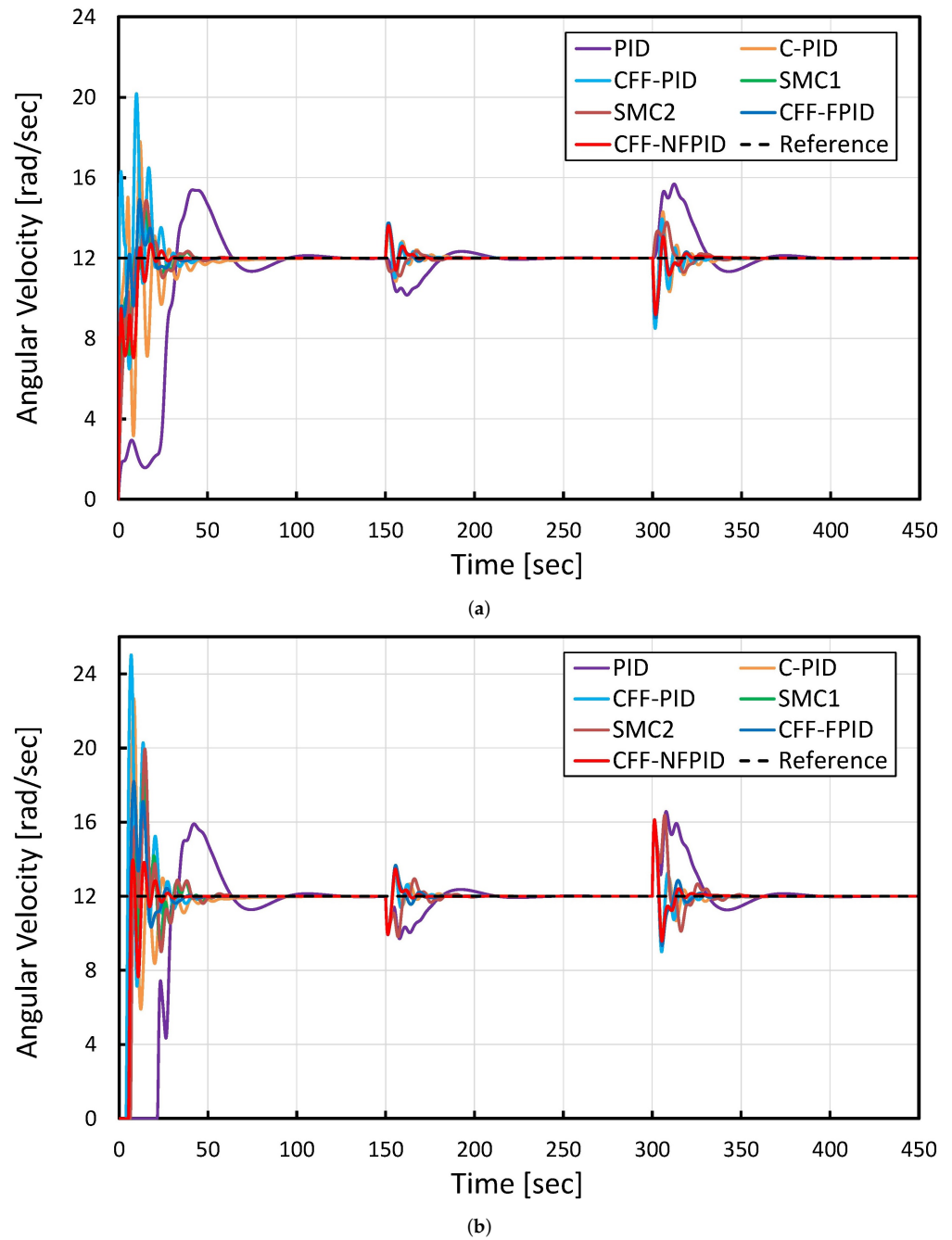


Figure 10. Simulated waveforms of the angular velocities during variations in the WOB and constant desired velocity with different controllers: (a) rotary table; (b) drill bit.

Figure 11 shows the simulated responses of the OH-IT2FPID+I controller with parametric uncertainties. In this case, the parameters J_r , k_t , d_t , and d_r were changed, and the reference velocity was set to 12 rad/s. It is clear that the parametric uncertainties did not degrade the performance of the proposed OH-IT2FPID+I controller. This led to the achievement of a small overshoot and fast response time.

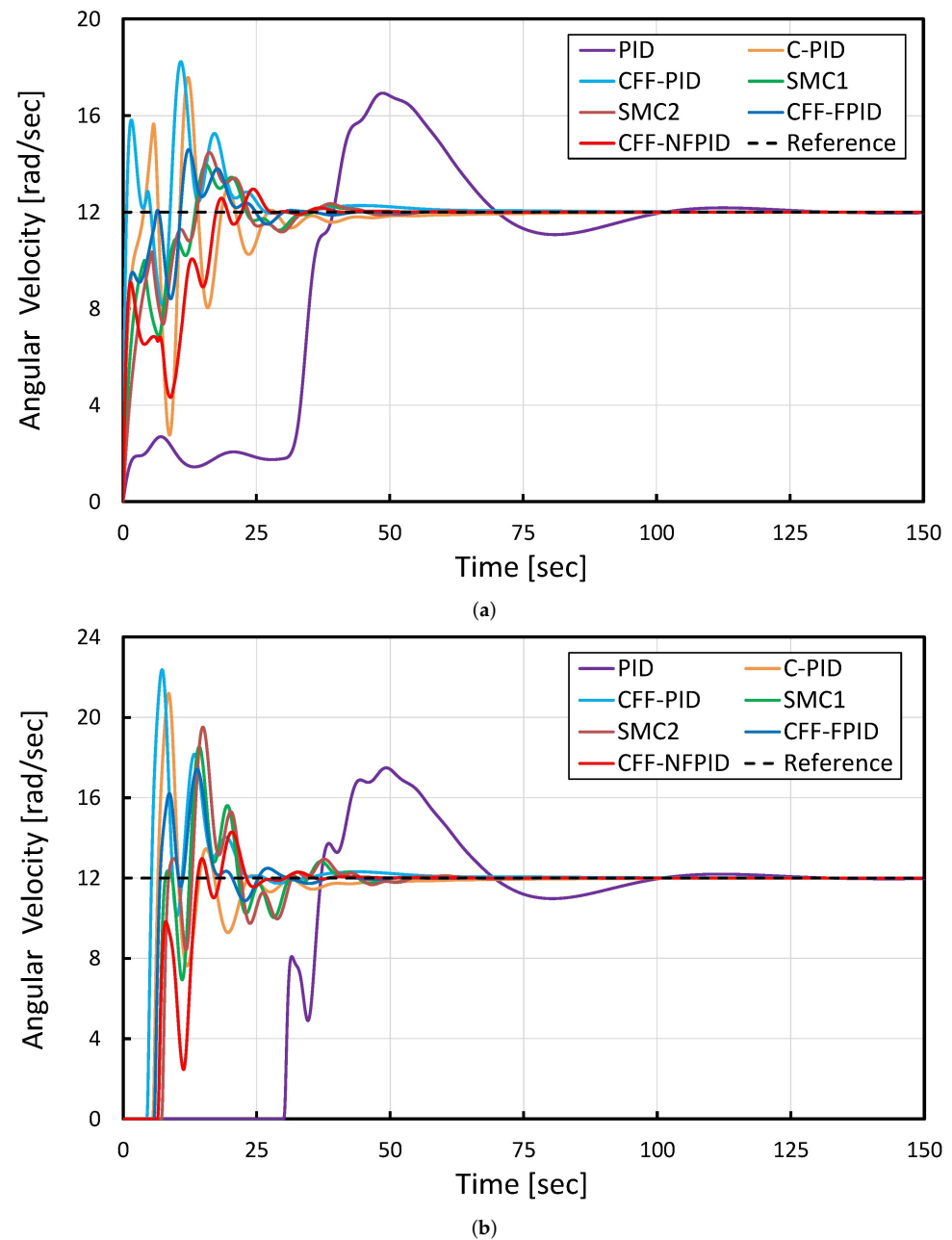


Figure 11. Simulated waveforms of the angular velocities in the case of parametric uncertainty with different controllers: (a) rotary table; (b) drill bit.

To demonstrate the effectiveness of using the proposed control scheme, the OH-IT2FPID+I controller was compared with a conventional PID controller, the SMC from [16], the SMC from [15], and the OH-IT1FPID+I controller. Figure 9 shows the simulation time response of the velocities with variations in the reference angular velocity when using different controllers. The results showed a large undershoot with large decreases in the reference velocity in the SMC algorithms proposed in [15,16], which could cause instability in a drill string system. Figure 10 shows the simulation time response of the velocities with variations in the WOB when using different controllers. In Figure 11, it is clear that the proposed OH-IT2FPID+I controller produced high-performance control with parametric uncertainty in terms of the peak overshoot and settling time. Table 4 depicts a quantitative analysis of the performance in terms of the peak overshoot and settling time for all controllers in the different situations. It is clear that the proposed OH-IT2FPID+I controller was more effective than the other controllers in all of the tested situations.

Table 4. Quantitative comparison using a constant desired angular velocity and parametric uncertainties.

	Stepwise Increase in the Angular Velocity Reference 0→12 rad/s in the Case of Parametric Uncertainty			
	Angular Velocity of the Rotary Table		Angular Velocity of the Drill Bit	
	Settling Time [s]	Undershoot [rad/s]	Settling Time [s]	Undershoot [rad/s]
PID	95.95	5.49	96.19	4.93
C-PID	42.23	5.58	44.31	9.19
CFF-PID	48.97	6.23	49.61	10.37
SMC [15]	39.62	1.95	46.59	6.56
SMC [16]	40.76	2.47	47.98	7.51
CFF-FPID	28.76	2.59	35.75	5.41
CFF-NFPID	26.63	0.95	33.84	2.29

5. Conclusions

In this study, a fast and intelligent cascade control scheme based on hybrid adaptive neuro-fuzzy PID and feedforward terms (CFF-NFPID) was presented for an oil well drill string system. The controller's main objective was to mitigate the torsional stick-slip vibration phenomena in oil well drill strings by controlling the velocities of the drill string components to remain at a desired value. The key feature of the CFF-NFPID controller is the estimation of the top drive velocity reference from the drill bit velocity and its reference via the first control loop. Then, a second control loop was employed to regulate the top drive velocity to its reference. Each control loop was designed based on a hybrid adaptive neuro-fuzzy PID controller and feedforward term. The feedforward (FF) term provides an ideal control action that helps to achieve a fast settling time of the measured signal to the reference signal. The effectiveness of the proposed control scheme was evaluated via simulations in Matlab/Simulink. The obtained results showed the outstanding performance of the proposed CFF-NFPID controller in comparison with that of the fuzzy feedforward (CFF-FPID) controller, sliding-mode controller, cascade feedforward PID controller, cascade PID controller, and conventional PID controller when there were variations in the weight on the bit and the desired drill bit rotary speed. Although the design of feedforward terms requires exact system parameters, the proposed CFF-NFPID controller maintained its superiority over other controllers under parametric uncertainties. Future work will involve combining the predictive capabilities of model predictive control with the proposed cascade control scheme to significantly improve the response time, stability, adaptability, operational efficiency, safety, and cost savings.

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Abbreviations

The following abbreviations are used in this manuscript:

ANN	artificial neural network
ANFIS	adaptive neuro-fuzzy inference system
BHA	bottom hole assembly
C-PID	cascade PID
CFF-PID	cascade feedforward PID
CFF-FPID	cascade feedforward fuzzy PID
CFF-NFPID	cascade feedforward neuro-fuzzy PID
DOF	degrees of freedom
FF	feedforward
FLC	fuzzy logic control
H-NFPID	hybrid neuro-fuzzy PID
NFPID	neuro-fuzzy PID
PI	proportional–integral
PID	proportional–integral–derivative
ROP	rate of penetration
RPM	revolutions per minute
SMC	sliding-mode control
WOB	weight on bit

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