Journal of Natural Gas Science and Engineering 39 (2017) 28–43

Contents lists available at ScienceDirect

Journal of Natural Gas Science and Engineering

journal homepage: www.elsevier.com/locate/jngse

Modeling and analyzing the movement of drill string while being rocked on the ground

Xueying Wang [a], Hongjian Ni [a,*], Ruihe Wang [b], Tianke Liu [c], Jingguo Xu [d]

[a] Research Institute of Unconventional Oil & Gas and Renewable Energy, China University of Petroleum (East China), Qingdao 266580, China
[b] School of Petroleum Engineering, China University of Petroleum (East China), Qingdao 266580, China
[c] SINOPEC Oilfield Service Shengli Corporation, Dongying 257064, China
[d] No. 3 Drilling Company, BoHai Drilling and Exploration Company, CNPC, Tianjin 300280, China

ARTICLE INFO

Article history:
Received 26 August 2016
Received in revised form
8 November 2016
Accepted 6 January 2017
Available online 8 January 2017

Keywords:
Rocking drill string
Rocking depth
Rocking velocity
Weight transfer characteristic
Tool face stabilization

ABSTRACT

Automatic rocking drill string on the ground has proven to be capable of improving slide drilling performance. To make best use of the rocking technology and avoid potential risks, this paper proposes a model to calculate the response of drill string to the surface rocking motion. Friction between drill string and wellbore is finely incorporated in the model to simulate the redistribution of friction during rocking and friction-induced stick-slip motion. A finite difference method with second-order accuracy is used to solve the numerical model. The model was first used to demonstrate that commonly-used static torque and drag models are not applicable for directly determining the rocking depth or required surface torque. The influences on weight transfer and tool face stabilization of surface rocking parameters are investigated. The simulation results indicate that greater rocking velocity produces smoother weight transfer and rocking depth determines the range of weight on bit. Rocking velocity should match rocking depth to reduce the fluctuation of weight on bit. In the design of rocking depth, reactive torque and relationship between weight on bit and reactive torque should also be taken into consideration. The simulated results are in good agreement with field practice mentioned in relevant literature and field measurements. Therefore, the proposed model and solving method are useful for determining rocking parameters and further improving the efficiency of automatic rocking technology.

1. Introduction

Although the Rotary steering systems (RSS) have made considerable technology leaps in improving build rates and high temperature capabilities, the directional bent housing motor continues to dominate the land directional market (Gillan et al., 2011). With the motor/MWD system, longitudinal drag, mainly caused by non-rotating drill string lying on borehole wall, makes it difficult to smoothly transfer a proper amount of weight to the bit, or to achieve and maintain a desired tool face angle. The efficiency of slide drilling is usually unsatisfactory. In some cases, the rate of penetration (ROP) achieved with conventional sliding technology typically averaged 10%–25% of the average ROP in rotating mode. Among directional wells with horizontal departures over 3000 ft, this ratio could fall to 10% (Maidla and Haci, 2004).

Many techniques have been developed to reduce longitudinal drag, which include addition of mud lubricity agents, catenary trajectory, downhole vibrating tools, and the recently developed automatic surface rocking technology. The effectiveness of each technique varies with well conditions, and none of them is perfect. The automatic surface rocking technology, which turns the drill string by top drive to the right limit and then to the left limit by an amount that avoids interference with the tool face, has achieved success in improving the efficiency of slide drilling. The improvements include increasing slide drilling ROP by 20%–294%, reducing motor stalls to zero, and reducing time of orienting tool face by an order of magnitude (Maidla et al., 2009).

The rocking technique is a low-cost technology, and technically it can be adopted by any type of top drive. However, there are some problems preventing its expanding in use. One of the problems is that it is difficult to determine surface rocking parameters to maximize drag-reduction and keep tool face relatively stable. According to the features of the rocking technology (Maidla and Haci, 2004), it is quite necessary to know the response of drill string to

* Corresponding author.
E-mail address: hongjianni@qq.com (H. Ni).

http://dx.doi.org/10.1016/j.jngse.2017.01.011
1875-5100/

the surface torque rocking, like rocking depth and weight on bit (WOB). Commercial automatic rocking systems, like “Slider” (Maidla et al., 2009) and “DSCS” (Gillan et al., 2009), generally use differential pressure of standpipe with bit on and off bottom and MWD information to reflect reactive torque, WOB and tool face orientation. Differential pressure of standpipe provides an indication of reactive torque and WOB, but this indication can be misleading. Because there are many unrelated factors affecting standpipe pressure, such as cuttings build-up and partially plugged nozzles. Because of the limited data transmission rate of mud pulse telemetry, the tool face signal is produced at a rate of about once every twenty seconds. However, the high latency makes it difficult to determine the real-time tool face orientation with surface rocking drill string.

Many torque and drag models have been developed in the last decades, including “soft string” model (Aadnoy and Andersen, 2001; Johancsik et al., 1984; Maidla and Wojtanowicz, 1987; Paslay, 1994; Sheppard et al., 1987) and “stiff string” model (Adewuya and Pham, 1998; Ho, 1988; Rezmer-Cooper et al., 1999; Zifeng et al., 1993). These models can be used to calculate the drag while tripping in or out, and the torque while rotating drilling or reaming. However, they were not designed to capture the torque, drag and WOB during the processes of starting rotation and stop turning.

Drill string dynamic models, mainly used to analyse vibrations and predict wellbore trajectories, are able to calculate displacement and rotation along drill string, and then WOB and other parameters can be obtained. Some of them focus on the wave propagation problems for axial and torsional vibrations, and others are based on beam model together with the finite element method (Dykstra, 1996). However, these dynamic models mainly focus on the operating mode of rotating drilling, and longitudinal friction is usually neglected or simply assumed as linear viscous damping.

In this paper, friction is well incorporated based on the velocity of contact point between drill string and borehole wall to simulate the redistribution of friction during rocking and axial stick-slip motion. In the rocking technology, only upper drill string is slowly and periodically rotated, while high-stiffness bit and bottomhole assembly (the two main sources of vibration) are separated from upper drill string by a length of “actual sliding” drill string. Therefore, the basic model is derived from the “soft string” model, instead of the more powerful finite element method. Apart from this, the solving process of finite element method is complicated and time-consuming due to the iterative calculation of contact points in each time step.

Besides, the special surface boundary in torsional direction has not been studied before. The goal of this paper is by analysing the movement of drill string in axial and torsional directions during the processes of periodically starting rotating and stopping turning, to provide suggestions on the choose of optimal surface rocking parameters. Actually, to the best of the authors knowledge, there is no previous publicly-published work on such problems related to the rocking technology. Some conclusions about the rocking technology in this paper have never been mentioned by available literature.

2. Model details

2.1. Basic assumptions

(1) The deformation of drill string is within linear elastic range, and cross section stays annular all the time;
(2) Wellbore clearance is ignored, and drill string centreline is the same as that of wellbore;
(3) Drill string is regarded as special heavy cable with torque transporting ability, and the cross section of drill string is loaded by only axial force and torque, which means bending moment is ignored.

2.2. General derivations

To illustrate which effects related to the stiffness of drill string are ignored, the derivations were first obtained based on real drill string. Consider a small element cut out of the drill string in natural curvilinear coordinates $(\vec{e}_t, \vec{e}_n, \vec{e}_b)$ with density $\rho(s)$, linear buoyant weight $\rho_s(s)$, cross-sectional area $A(s)$, elastic modulus $E(s)$, shear modulus $G(s)$, and incremental length ds, as shown in Fig. 1. The parameter s is taken to denote the pipe distance from the bit, and t represents time. T, M, u and θ denote axial force, moment, axial displacement and rotation, respectively.

According to the equilibriums of forces and moments (moments for the centre of the upper cross section):

$$\begin{aligned}&\vec{T}(s+ds,t)-\vec{T}(s,t)+\vec{F}\left(s+\frac{ds}{2},t\right)ds-\\&\quad\left[f(s,t)+c_1\left(s+\frac{ds}{2},t\right)\frac{\partial u(s+ds/2,t)}{\partial t}\right]ds\,\vec{e}_t\\&\quad+\rho_s\left(s+\frac{ds}{2}\right)ds\,\vec{e}_g=\rho\left(s+\frac{ds}{2}\right)A\left(s+\frac{ds}{2}\right)\\&\quad ds\frac{\partial^2 u(s+ds/2,t)}{\partial t^2}\vec{e}_t.\end{aligned} \tag{1}$$

where $\vec{F}$ is the distributed normal contact force; c_1 is the coefficient of axial drag produced by drilling fluids; f denotes the axial friction.

$$\begin{aligned}&\vec{M}(s+ds,t)-\vec{M}(s,t)+\frac{ds}{2}\vec{e}_t\times\left\{2\vec{T}(s+ds,t)\right.\\&\quad-\left[f(s,t)+c_1\left(s+\frac{ds}{2},t\right)\frac{\partial u(s+ds/2,t)}{\partial t}\right]ds\,\vec{e}_t\\&\quad\left.+\rho_s\left(s+\frac{ds}{2}\right)ds\,\vec{e}_g+\vec{F}\left(s+\frac{ds}{2},t\right)ds\right\}\\&\quad-\left[m(s,t)+c_2\left(s+\frac{ds}{2},t\right)\frac{\partial\theta(s+ds/2,t)}{\partial t}\right]ds\,\vec{e}_t\\&\quad=\rho\left(s+\frac{ds}{2}\right)I_p\left(s+\frac{ds}{2}\right)ds\frac{\partial^2\theta(s+ds/2,t)}{\partial t^2}\vec{e}_t.\end{aligned} \tag{2}$$

where c_2 is the coefficient of torsional drag produced by drilling fluids; m denotes the torsional friction torque.

The first-order approximations of the Taylor series expansion in (s,t) of variables in above equations are taken, and high−order quantities are ignored, simplifying Equations (1) and (2):

$$\begin{aligned}&\frac{\partial\vec{T}(s,t)}{\partial s}+\vec{F}(s,t)-\left[f(s,t)+c_1(s,t)\frac{\partial u}{\partial t}\right]\vec{e}_t+\rho_s(s)\vec{e}_g\\&\quad=\rho(s)A(s)\frac{\partial^2 u(s,t)}{\partial t^2}\vec{e}_t.\end{aligned} \tag{3}$$

$$\begin{aligned}&\frac{\partial\vec{M}(s,t)}{\partial s}+\vec{e}_t\times\vec{T}(s,t)-\left[m(s,t)+c_2(s,t)\frac{\partial\theta(s,t)}{\partial t}\right]\vec{e}_t\\&\quad=\rho(s)I_p(s)\frac{\partial^2\theta(s,t)}{\partial t^2}\vec{e}_t.\end{aligned} \tag{4}$$

Decomposing forces and moments (omitting the parameters (s,