Lateral Vibration of Oilwell Drillstring During Backreaming Operation

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ABSTRACT: Oilwell drillstring vibrations are renowned by their high damaging effects in the drillstring elements. In the case of backreaming operations, i.e. when the drillstring is pulled out of the wellbore, abnormal lateral vibrations have been reported by operators in the field. Such lateral vibrations of the drillstring can cause several problems during wellbore construction, e.g. BHA electronic equipment failure, falling rocks into the well, drillstring blockage, which result in high financial losses when they occur. This work focuses on the study of the dynamics of oilwell drillstrings during the withdrawing operation (backreaming). For that, a non-linear mathematical model is used for representing the drillstring vibrations considering fluid pumping and string rotation simultaneously. The proposed model focuses on the effects of lateral vibrations on the lower portion of the drill string, commonly known as Bottom Hole Assembly (BHA). The modelling approach is based on analytical, nonlinear and lumped parameters, which considers the effects of drilling fluid damping, stabilizer and drill collar contact with the borehole wall. The numerical results are compared to experimental results obtained in a scaled test rig.

KEY WORDS: Rotor dynamics; Drillstring dynamics; Backreaming operation; Mechanical vibration.

1 INTRODUCTION

During the borehole drilling of oilwells, when the final depth is reached, there comes the task of pulling out the drillstring. There are two possible ways of pulling out the drillstring: dry operation (no rotation and no pumping of fluid) and backreaming operation (rotation and pumping of fluid). The dry operation represents a faster operation, but the friction between the drillstring and the borehole becomes so high that it usually jeopardizes the pulling-out operation. The backreaming operation is more commonly adopted, but their optimum conditions are still controversial in literature. In some applications, the backreaming operation represents a much better friction condition for pulling out the drillstring but, in other applications, the severe lateral vibration of the drillstring can cause serious damage to the Bottom Hole Assembly (BHA), and even borehole collapse with drillstring imprisonment [1].

The challenge of modelling the drillstring during backreaming operation is the correct representation of the coupling between the torsional and lateral movements of the drillstring and its longitudinal movement in contact with the borehole. The friction between the drillstring and the borehole during longitudinal movement, together with occasional impacts due to lateral vibration, lead the system to a strong nonlinear behaviour.

In literature, one can find few studies on the subject, mainly focused on axial vibration. In [2,3], axial vibration of the drillstring is studied to improve the dynamic behaviour of the system during stick-slip phenomenon and to optimize the design of the stabilizers. Friction between the drillstring and the borehole is studied in [4], based on a low order mathematical model and experimental correlation to results shown in [5]. The results show that friction plays an important role in the dynamics of the system and in the impact level of the drillstring against the borehole walls. The control of the rotating speed of the drillstring has also been investigated to minimize lateral vibrations. In [6], the adopted mathematical model took into account the longitudinal movement of the drillstring and results showed that vibration is self-excited. Hence, in some cases, vibration was reduced by increasing the rotating speed of the drillstring. However, in other cases, the lateral vibration observed was severe.

Despite the advances in the study of lateral vibrations of drillstrings, there are no related studies on the backreaming operation, considering drillstring rotation and fluid pumping. In this work, one develops a mathematical model for the drillstring during backreaming. The modelling approach is based on analytical, nonlinear and lumped parameters, which considers the effects of drilling fluid damping, stabilizer and drill collar contact with the borehole wall. The results of numerical simulations show the occurrence of abnormal lateral vibrations during the drillstring withdrawal, but this effect decreases with higher friction coefficients, and for rotating speeds near the natural frequency of the system. A comparison between numerical and experimental results obtained in scaled test rig showed good agreement for conditions of low friction coefficients.

2 MATHEMATICAL MODEL

The following mathematical model was based on the two degree-of-freedom model presented by Jansen [7,8], which takes into account the pumping fluid, contact of the stabilizers with the borehole walls, contact of the drillstring with the borehole walls, and excitation due to unbalance. The model is then complemented with additional degrees-of-freedom of torsion and longitudinal movement. The basic hypotheses of the
model are:

- **Pumping Fluid**: drag in the annular gap between the drillstring and the borehole is proportional to the square of the rotating speed. There fluid also adds inertia to the system, as showed in [9];
- **Stabilizers**: the hydrodynamic effects in the gap between the stabilizers and the borehole walls are neglected;
- **Borehole**: the cross section of the borehole is circular, and contact between the drillstring and the borehole obey the Coulomb law;
- **Drillstring Vibration**: the adopted rotating speeds are close to the first natural frequency of the drillstring associated to the first bending mode. Hence, lateral vibration of the drillstring will be limited to that of the bending mode of a simply supported beam, which means that both stabilizers will be in contact at the same time;
- **Longitudinal Movement**: Coulomb friction is considered in the longitudinal direction of movement. The weight of the drillstring, and its effects, will not be considered in the model.

The mathematical model is composed of a cylindrical beam simply supported, represented by lumped parameters. The beam represents the BHA and the supports are the stabilizers (Fig. 1). Vibration is analyzed in the mid section of the beam (section A-A of Figs. 1 and 2). The model has two degrees-of-freedom to represent the torsion of the drillstring, and one degree-of-freedom to represent the lateral movements of the BHA in two directions \( x_1 \) and \( x_2 \), \( \phi \) is the eccentricity of the center of mass, and \( \Omega \) is the rotating speed of the drillstring. \( t \) is time, \( q \) is the radial deflection of the BHA geometric centre, \( S_0 = \frac{1}{2}(D_b - D_S) \) is the gap between the stabilizer and the borehole \( (D_b \) is the borehole diameter, \( D_S \) is the stabilizer diameter), and \( p = q \cos \beta + S_0 \cos \gamma = q \cos \beta - S_0 \cos \varphi \) is the distance between the geometric centre of the BHA and the geometric centre of the stabilizer. It is important to note that, if \( q < S_0 \), there will be no restoring force in the system (no contact between the stabilizer and the borehole wall – Fig. 3).

The equations of the restoring force during contact can be simplified by a first order Taylor series of the angular terms of the equations, as follows:

\[
F_{k,rad} = -k (q - S_0) \\
F_{k,tan} = -k \varphi \left( S_0 - \frac{S_0^2}{q} \right)
\]

where \( q \) is the radial deflection of the BHA geometric centre, \( S_0 = \frac{1}{2}(D_b - D_S) \) is the gap between the stabilizer and the borehole \( (D_b \) is the borehole diameter, \( D_S \) is the stabilizer diameter), and \( p = q \cos \beta + S_0 \cos \gamma = q \cos \beta - S_0 \cos \varphi \) is the distance between the geometric centre of the BHA and the geometric centre of the stabilizer. It is important to note that, if \( q < S_0 \), there will be no restoring force in the system (no contact between the stabilizer and the borehole wall – Fig. 3).
This is justified by the low friction values observed in practice during operation. Transforming to the Cartesian system of coordinates:

\[
F_{k,1} = -F_{k,ad} \frac{x_1}{q} + F_{k,tan} \frac{x_2}{q} \\
F_{k,2} = -F_{k,rad} \frac{x_2}{q} - F_{k,tan} \frac{x_1}{q}
\]

where \( q = \sqrt{x_1^2 + x_2^2} \).

The restoring force due to the contact of the BHA with the borehole wall is given by spring-damper model:

\[
F_{w,rad} = -k_w(q-c_o) - c_w \dot{q}
\]

\[
F_{w,tan} = -S \mu_c F_{w,rad}
\]

where \( \mu_c \) is the friction coefficient between the BHA and the borehole wall, \( k_w \) and \( c_w \) are the equivalent stiffness and damping coefficient of the borehole wall, respectively, and \( S = \text{sign}(v_y - v_{tan}) \) (Fig. 4). The BHA velocity and the tangential velocity in the contact point are given by:

\[
v_y = \sqrt{x_1^2 + x_2^2}
\]

\[
v_{tan} = \frac{D_c}{2} \Omega
\]

During the longitudinal movement of the drillstring, the contact force against the borehole wall is given by [10]:

\[
F_{c,axial} = \text{sign}(\dot{x}_3)(F_{k,rad} \mu + F_{w,rad} \mu_c) \tan \psi
\]

where \( \tan \psi = \dot{x}_3/\Omega R_c \). Such contact force is only considered in the model when there is contact between the BHA and the borehole wall, or between the stabilizer and the borehole wall.

By adopting the following adimensional parameters:

\[
\beta = \frac{m + m_f}{m} \quad \xi = \frac{c_f c_o}{m} \quad \zeta = \frac{S_0}{c_o}
\]

\[
\eta = \frac{\Omega}{\omega} \quad \rho = \frac{k_w}{k_T} \quad \tau = \omega t \quad \nu = \frac{c_w}{m \omega}
\]

\[
\omega = \sqrt{\frac{k_T}{m}} \quad c_o = \frac{1}{2}(D_h - D_c) \quad y_i = \frac{x_i}{c_o}
\]

one can write the equations of motion of the drillstring for the backreaming operation, as follows:

\[
\begin{align*}
\beta \dot{y}_1 + \xi \dot{y}_1 + \alpha F_{k,rad} y_1 - \alpha F_{k,tan} y_2 + \gamma F_{w,rad} y_1 \\
- \gamma F_{w,tan} y_2 = c \cos(\phi - \eta\tau) \left( \dot{\phi} - \Omega \right)^2 - \frac{c}{\omega^2} \sin(\phi - \eta\tau) \dot{\phi} = 0
\end{align*}
\]

\[
\begin{align*}
\beta \dot{y}_2 + \xi \dot{y}_2 + \alpha F_{k,rad} y_2 - \alpha F_{k,tan} y_1 + \gamma F_{w,rad} y_2 \\
+ \gamma F_{w,tan} y_1 = -c \sin(\phi - \eta\tau) \left( \dot{\phi} - \Omega \right)^2 - \frac{c}{\omega^2} \cos(\phi - \eta\tau) \dot{\phi} = 0
\end{align*}
\]

\[
\begin{align*}
\ddot{\phi} + 2\xi \dot{\phi} \dot{\phi} + \ddot{\phi} \left( \phi - \eta\tau + \frac{T_0}{K_T} \right) + \frac{c}{K_T} \dot{y}_1 \sin(\phi - \eta\tau) \\
- \dot{c}_h \dot{y}_2 \cos(\phi - \eta\tau) = F(\alpha F_{k,tan} + \gamma F_{w,tan})[\ddot{R} - c \cos(\phi - \eta\tau)] \\
- \dot{F}(\alpha F_{k,rad} + \gamma F_{w,rad}) \sin(\phi - \eta\tau)
\end{align*}
\]

where:

\[
F_{k,rad} = \left( 1 - \frac{\zeta}{a} \right)
\]

\[
F_{k,tan} = \phi \left( \frac{\zeta}{a} - \frac{c^2}{c} \right)
\]

Figure 3: Contact force between the stabilizer and the borehole wall, and restoring force on the BHA [8].

Figure 4: Representation of the relative velocities in the BHA and in the borehole wall.
\[ F_{w,\text{rad}} = \rho (a - 1) + \nu b \tag{17} \]
\[ F_{w,\text{tan}} = -S \mu_c F_{w,\text{rad}} \tag{18} \]

and:

\[
\begin{cases} 
\alpha = 1 \text{ and } \gamma = 0 , \text{ if } a > \zeta \text{ and } a \leq 1 \\
\text{(impact of stabilizer)}
\end{cases}
\]

\[
\begin{cases} 
\alpha = 1 \text{ and } \gamma = 1 , \text{ if } a > 1 \\
\text{(impact of stabilizer and BHA)}
\end{cases}
\]

\[
\begin{cases} 
\alpha = 0 \text{ and } \gamma = 0 , \text{ if } a \leq \zeta \\
\text{(no impact)}
\end{cases}
\tag{19}
\]

The equations of motion are integrated in time for different operating conditions of rotating speed of the drillstring, longitudinal velocity of the drillstring, and friction coefficient of the borehole wall. The adopted parameter values are those of the scaled test rig. Some of the parameters are uncertain (e.g. friction coefficients and damping coefficients) and they must be adjusted accordingly to the experimental results.

3 TEST RIG

The test rig was built in scale according to the characteristics of a real 16” oilwell. The drillstring is represented by a steel shaft with diameter of 12 mm, mounted in the vertical direction, simply supported by a self aligning ball bearing (Fig. 5). In the extremity of the shaft above the bearing, it is mounted a inertia cylinder to represent the inertia of the drillpipe and BHA. The cylinder+shaft is driven by an electric motor through a flexible coupling, which represents the torsional stiffness of the drillpipe. The rotating speed of the shaft is measured by an encoder mounted in between the ball bearing and the inertia cylinder.

The electric motor and the ball bearing are rigidly mounted on the structure of the test rig. Hence, the relative longitudinal motion between the BHA (shaft) and the oilwell is reproduced by the movement of an acrylic tube in which the shaft is mounted and it is free to rotate and translate (Fig. 5). This tube is mounted on a linear guide driven by an electric servo motor. The rotating speed of the electric motor of the shaft is controlled by a frequency inverter, whereas the velocity of the servo motor of the linear guide is controlled by a servo drive. Analogue inductive proximity sensors, mounted in a position equidistant to the stabilizers mounted in the shaft, are responsible for measuring the behaviour of the shaft inside the tube.

The dimensions of the shaft, the inertia cylinder, and the flexibility of the coupling were determined according to parameters observed in the oil field (offshore platforms). For example, the ratios \( \omega_1/\Omega \) (first bending natural frequency of the BHA/rotating speed) and \( \omega_2/\Omega \) (first torsion natural frequency of the drillpipe/rotating speed) were the same as those observed in practice. The dimensions of the stabilizers mounted in the shaft also obey the proportions observed in practice. The velocity of the longitudinal movement of the tube is chosen according to the ratio \( L_{BHA}/t \) (BHA length/time) observed in practice. Table 1 presents the geometric and operational characteristics of the test rig.

4 RESULTS

Experimental tests were performed for different rotating speeds of the shaft and different longitudinal velocities of the tube. The tube was filled with drilling fluid with density of 1070 kg/m\(^3\) and dynamic viscosity of 0.01 Pa.s. The lateral vibration of the shaft was measured during the tests and the resultant RMS acceleration is presented in Fig. 6 in adimensional form, i.e. the adimensional acceleration is given by \( \text{RMS}(\ddot{e}_1)/\omega_2^2c_0 \).

As one can see in Fig. 6, good agreement is observed between numerical and experimental results, especially in the case when the tube has vertical movement, thus showing that the developed model for the system is suitable for describing the system dynamics. It is also observed that above 2400 rpm there is a sudden reduction in the lateral acceleration of the shaft, irrespective of the vertical velocity of the tube. Considering
that the natural frequency of the shaft (associated to the first bending mode) is 39.9 Hz (equivalent to 2396 rpm), this reduction in lateral acceleration of the shaft occurs after crossing the critical velocity of the system. A possible explanation for this phenomenon is the inversion of phase of the unbalance (self-centring effect).

The vertical velocity of the tube does not affect significantly the results. One can observe that there is a slight increase of the lateral acceleration as one increases the vertical velocity of the tube. Such low effect of the vertical motion on system dynamics can be explained by the low friction coefficient between the stabilizers and the tube (estimated during model correlation as $\mu = 0.013$). Considering the model correlated to the experimental results, one simulated numerically the system for the same operating conditions but with a higher friction coefficient between the stabilizers and the tube: $\mu = 0.2$. The results are presented in Fig. 7.

As one can see in Fig. 7, the lateral acceleration of the shaft increased in all tested conditions with higher friction coefficient between the stabilizers and the tube (around 30% higher). In addition, the effect of the vertical velocity of the tube in the system dynamics (increase of lateral acceleration) is more evident, and the decrease of lateral acceleration above the critical speed is still noticeable.

In summary, the results show that supercritical operation condition is interesting (irrespective of the longitudinal velocity) because this condition presents lower lateral accelerations of the shaft, and the longitudinal velocity tends to increase the lateral vibration of the shaft.

4.1 Simulations with Impacts of the Shaft against the Tube

In all the tested operation conditions, one observed that the shaft did not shock against the tube (a common occurrence during practical backreaming operation). This no-impact operation was probably caused by the small unbalance of the shaft in the test rig (basically the residual unbalance of the cylinder shaft). In practice, the unbalance is higher due to the irregular mass distribution in the BHA caused by instrumentation. Hence, the model adopted previously was numerically solved with higher unbalance (100 times higher), and one observed that the shaft (BHA) began to touch the inner side of the tube (borehole) – Fig. 8.

In Fig. 8, one presents the deflection of the shaft during time in adimensional form (ratio between the displacement $q$ and distance $c_0$). That means, if $q/c_0$ equals 1, then the shaft touches the inner surface of the tube. One can clearly see that, in the conditions tested in the experiments (Fig. 8(a)), the shaft never touches the tube, whereas adopting a higher unbalance value (Fig. 8(b)), the shaft continuously touches the tube.

In order to evaluate the level of impact of the shaft against the tube in each operation condition, one adopted the following definition of total impulse:

$$ I = \int_0^T (F_{k,\text{rad}} + F_{w,\text{rad}}) \, dt $$

(20)
By calculating the total impulse for different operation conditions under the same unbalance, one obtained the results presented in Fig. 9. As one can see, as the rotating speed increases, the total impulse of the shaft against the tube increases significantly. After crossing the critical speed of 2396 rpm, there is a sudden reduction of the total impulse (similarly to what happened to the lateral acceleration of the shaft in the experiments). However, for high rotating speeds (much above the critical speed), the total impulse further increased.

Such results show that the supercritical operation is still interesting when shocks occur between the BHA and the borehole. However, it is clear that one cannot increase the rotating speed indefinitely.

Considering the vertical velocity of the tube, again no significant effect on the results is observed, except in the case of 3940 rpm (rotating speed above the critical speed). In this case, a low longitudinal velocity of the tube results in a total impulse 50% lower than that observed in higher longitudinal velocities. This is an indication that, in case of an operation with impacts, the longitudinal velocity of the BHA must be as small as possible to reduce the total impulse against the borehole walls.

5 CONCLUSION

This work presents a simplified mathematical model for the oilwell drillstring during backreaming operation. A comparison between numerical and experimental results showed that the developed model is suitable to describe the dynamics of the system under such operating conditions (backreaming). The obtained results also showed that:

- a supercritical operation is an interesting operating condition due to the lower lateral accelerations (and total impulse) presented by the shaft, irrespective of the longitudinal velocity adopted in the backreaming operation. However, the rotating speed cannot increase indefinitely, otherwise the total impulse, and eventually the lateral acceleration of the shaft, will further increase;

- the longitudinal velocity of the shaft during the adopted backreaming conditions does not affect significantly the results. When there is no impact against the borehole walls, an increase of the longitudinal velocity slightly increases the lateral acceleration presented by the shaft. When there is impact against the borehole walls, a significant reduction of total impulse is only observed when the longitudinal velocity is small and the rotating speed is above the critical speed.

Such are promising results in terms of optimum backreaming operating conditions. However, due to the highly nonlinear characteristics of the system, other metrics shall be adopted to evaluate system response and better understand the system dynamics.

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