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Study of Lateral Vibration of Drilling String with Mass Imbalance Using Finite Element

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***Abstract*—The objectives of this study were to analyze the first three critical frequency, critical speeds and mode shapes of the drilling string due to lateral vibration. Drilling string vibrations pose a major challenge to the operational efficiency of the drilling process in the oil and gas industry. The mass imbalance in the drilling string caused by Measured While Drilling (MWD) tools and drill collar sag disrupts the accuracy of the drilling operation and can give rise to huge vibration thereby damaging the down hole components. Since the surface readings are not reliable measured of drilling string lateral vibration, it becomes necessary to study drilling string vibration under Finite Element method. In this project, ANSYS software was used to build the geometry of a drilling string model and appropriate boundary conditions were applied to the model such that the drilling string experiences lateral vibrations. The component of mass imbalance was introduced to the drilling string by adding an equivalent weight eccentrically on one side of the drilling string. Modal analysis was performed to determine the mode shapes and the first three critical frequencies of the drilling string. The effect of mass imbalance was studied by comparing the first three critical frequencies before and after the addition of the imbalance. The results showed an increase in natural frequency to 2.03Hz, 2.23Hz, and 4.63Hz were imposed on the deformation of drilling string. Since the normal operating speeds of a drilling string rotary table is around 100-200 RPM, it was clear that the drilling string without mass imbalance, the resonance of the drilling string occurs at 85 and 262 RPM which is clearly out of the operating speeds. But when a mass imbalance is introduced to the drilling string, the resonance occurs at 121, 133 and 277 RPM. The two operating speeds cause resonance which can critically damage the drilling string. The drilling string which consists of the stabilizers and bit are subject to almost no displacements at all. The best practices of this study are very useful for identify the safe operating ranges of rotary speed for the drilling string and identify the lateral displacement of a critical component for a range of frequencies, thereby avoiding damage to the drilling string.**

***Keywords*—Drilling string; lateral vibration; mass imbalance; frequency; modal analysis.**

I. Introduction

In the field of oil and gas industry, wells are drilled through the reservoir to produce the recoverable hydrocarbons to the surface. The drill bits are the cutting tools which are used to drill these oil wells. These drill bits are rotated around using a series of tubular pipes, collectively known as the drill string, which can be a thousand meters long. To turn the bit around, the entire drill string is rotated from the surface using the rotary table. As the drill string moves down the hole, it is subjected to a variety of stresses, including tension, compression, vibration, torsion, friction, formation pressure and circulating fluid pressure, all of which will lead to the failure of the drill string if not appropriately monitored.

Besides stresses and pressure, mass imbalance in the drilling operation is also a major concern. A rotating body is said to be unbalanced when its center of mass doesn't coincide with the axis of rotation. When rotating a shaft with unevenly distributed mass, its center of mass does not correspond with the axis of rotation. Thus it causes vibrations. There is a wide range of potential excitation sources such as mass imbalance, misalignment, and kinks or bends, the cutting action of the drill bit, stabilizer blades, mud motors and the friction factor between borehole and drill string. The effect of mud motor imbalance as a source of drill string vibrations is such that high torque low-speed motors have rotor orbiting eccentrically about the center of the motor. This imparts large dynamic loads to the drilling components, such as bottom hole assembly (BHA).

The vibrations, in particular, may disrupt the accuracy of the drilling operations and can cause damage due to fatigue. Drill string vibrations can be divided into three types- axial, torsional, and lateral. Axial vibrations can cause bit bounce, which can damage bit cutters and bearings. Torsional vibrations can cause irregular downhole rotation. Lateral vibrations are the most destructive type of vibration and can create large shocks as the BHA impacts the wellbore wall. The interaction between BHA and drill string can cause backward whirl sometimes, which is a severe form of vibration, creating a high-frequency large magnitude bending moment fluctuations that result in high rates of component fatigue. Therefore, it is vital to study and carry out vibration analysis, to understand its effects and develop ways of preventing problems it may cause.

In previous studies, finite element analysis of the proposed BHAs consisting of two motor configurations was used to determine the critical speeds, as well as the speeds that cause the stabilizers to bounce from side to side [1]. Further, it is studied that mass imbalance is a major source of downhole lateral vibrations [2]. Factors that contribute imbalance include bore misalignment, initial curvature, and gradual wear during service. Field experiments conducted by the author focused on drill collars, as they are a common source of imbalance. The author had neglected the effect of drilling bit by replacing it with a bullnose. Lateral vibrations caused by the collar/ wellbore collision were measured at various locations on the drill string using sensors. Costa and Ribeiro in [3] performed finite element modeling of the mechanical behavior of unbalanced drill collars, while Jamal and Seyed in [4] conducted finite element analysis of a drill string for lateral vibrations in deviated wells, which took account of the contact points between drill string and borehole as well as the effect of mud fluid.

Recent studies [5]–[9] have also suggested that drill string failures due to vibration have been encountered more commonly in vertical wells than in deviated wells. The reason for this is that the use of stabilizers in deviated wells which have reduced the torsional and axial vibration significantly by dumping it. The bit bouncing is an essential source of axial and torsional vibrations. It is found that the drill collar weight normal to the drill string axis, and the wellbore wall effect is the stabilizing force in the lateral modes of vibrations. Lateral vibrations depend mostly on drill collar sizes and position of stabilizers.

This paper investigates the lateral vibration of a drilling string with mass imbalance using the finite element method. The objectives of this study were to examine and analyze the first three critical frequencies, critical speeds, and mode shapes of the drilling string due to lateral vibration.

II. Material and Method

Lateral vibrations of drill string can cause fatigue failures, excessive wear, and Measurement While Drilling (MWD) tool failure. A rotating body is said to be unbalanced when its center of mass doesn't coincide with the axis of rotation. When rotating a shaft with unevenly distributed mass, its center of mass does not coincide with the axis of rotation. Thus it causes vibrations as shown by Fig. 1.

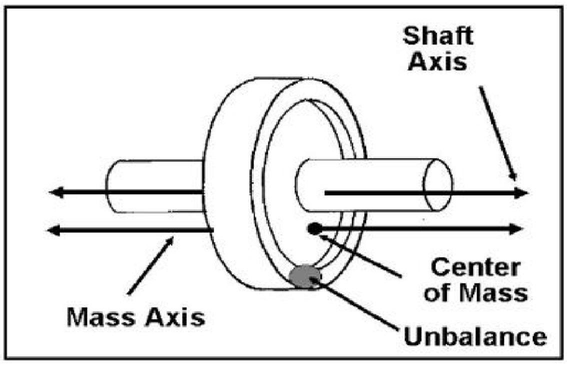

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Fig. 1 Schematic of mass imbalance [2].

As previously mentioned, there are several sources of mass imbalance on the drill string such as bore misalignment, initial curvature of the drill string, or gradual wear during service. A common source of imbalance occurs at the drill collars due to MWD tools.

A. *Case Configurations and Data*

Fig. 2 shows the geometry of the drill string design (not subject to scale) used for simulations.

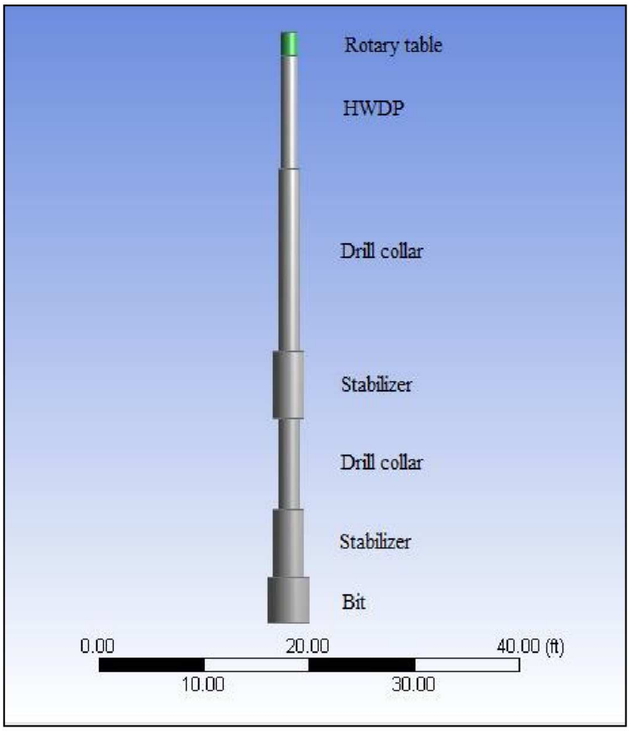

Fig. 2 Drill string design.

A drill string of certain specification [10] is used as a reference for the present vibration analysis. The data for this drill string was obtained from a field experiment which was subject to failure due to resonance. The field experiment was such that the wellbore was inclined at 5 degrees and the effective length of the drill string was around 700 feet. The first critical frequency was encountered around 3-3.4 Hz. Burgess conducted a finite element analysis for the drill string configuration and found through static analysis that the mode shape curves tend to zero at 113 feet. This implies that the drill string subject to lateral vibration was only 113 feet, which are taken as the cutoff point. Therefore, in this project, the drill string is modeled up to 114 feet, thereby eliminating the effect of well deviation on the model. However, the wellbore effect is taken into account through the boundary conditions, which will be specified subsequently.

To analyze the effect of different length and weight on bit and drill string vibration, three configurations of a drill string have been put forward as given in Tables 1-3, in which the data refers to that in [10] as experimental natural frequencies are available for comparison. Properties of the drill string and drilling fluid are respectively shown in Table 4.

TABLE I
BASE CASE DRILL STRING CONFIGURATION-1

Component	Length (ft)	O.D (in)	I.D (in)	Aggregate length (ft)
Bit	3.25	6.25		3.3
Stabilizer	6.40	4.75	2.25	9.7
Drill collar	30.8	4.75	2.25	40.5
Stabilizer	6.60	4.75	2.25	47.1
Drill collar	60	4.75	2.25	77.1
HWDP	30.42	3.50	2.06	107.5
Rotary table	3	3.50		110.5

TABLE II
DRILL STRING CONFIGURATION-2

Component	Length (ft)	O.D (in)	I.D (in)	Aggregate length (ft)
Bit	3.25	6.25		3.3
Stabilizer	6.40	4.75	2.25	9.7
Drill collar	30.8	4.75	2.25	40.5
Stabilizer	6.60	4.75	2.25	47.1
Drill collar	60	4.75	2.25	107.1
HWDP	30.42	3.50	2.06	137.5
Rotary table	3	3.50		140.5

TABLE II
DRILL STRING CONFIGURATION-3

Component	Length (ft)	O.D (in)	I.D (in)	Aggregate length (ft)
Bit	3.25	6.25		3.3
Stabilizer	6.40	4.75	2.25	9.7
Drill collar	30.8	4.75	2.25	40.5
Stabilizer	6.60	4.75	2.25	47.1
Drill collar	60	4.75	2.25	137.1
HWDP	30.42	3.50	2.06	167.5
Rotary table	3	3.50		170.5

TABLE IV
PROPERTIES OF DRILL STRING AND DRILLING FLUID

Drill string	Modulus of elasticity (lb/ft^3)	4.32E9
	Density (slug/ft^3)	15.2
	Poisson ratio	0.23
Drilling mud	Density (PPG)	12

B. *Drill Collar Imbalance*

Drill collars have been subject to mass imbalance due to MWD tools or initial curvature or gradual wear due to service. Therefore, it is understood that up to a mass imbalance of 120 Kg is allowed from field experiments. Mass imbalance can be created by either removing or adding mass. In this project to maintain the structural integrity of the drill string, a mass of 30 Kg has been added eccentrically to one of the sides of the drill collar. This is ¼ th of the mass imbalance that can be achieved without compromising the integrity of the drill string. Fig. 3 below depicts how the mass imbalance has been created in the middle of the drill collar.

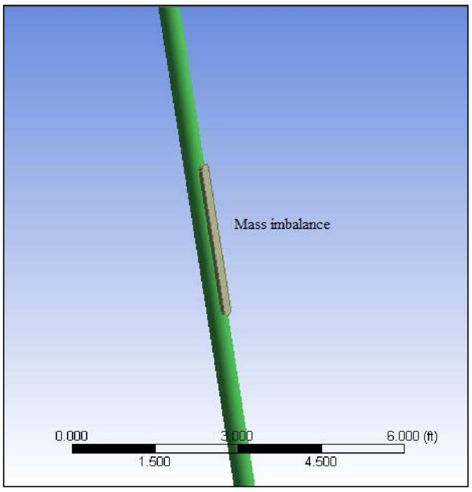

Fig. 3 Mass imbalance of drill string.

C. *Modal Analysis*

As the problems are too complex to be solved analytically especially with the case of mass imbalance, finite element method is used. The types of equations which are observed in solving for modal analysis are those that are see in the eigen systems. Modal analysis is used to determine a structure's vibration characteristics, its natural frequencies and mode shapes. The number of modes extracted from modal analysis is 3 and the maximum frequency encountered for drill string 1 is 4.68 Hz. A general rule of thumb for specifying frequency range in harmonic analysis is to specify 1.5 times the maximum frequency encountered in the solution. Therefore the operating range of frequency for the analysis is from 0 to 7 Hz. The validation of the result is confirmed by running the base case and comparing it with the experimental result. The next step is performing modal analysis on different drill string configurations along with their respective mass imbalance.

D. *Boundary Conditions*

Since this study focuses on the lateral vibrations of the drill string, the boundary conditions are set to observe lateral vibrations specifically. The wellbore effect will be included in the boundary conditions by allocating cylindrical supports at the place of the bit, stabilizers and rotary table. The radial supports are such that they can allow or constraint radial deflection and rotation about the drill string axis. They are specified as follows:

- At the bit position, the drill string is allowed to rotate about its axis and allowed deflection in the radial direction.

- Since the purpose of the stabilizer is to restrict radial motion of the drill string, it is constrained radially but allowed to rotate about its axis.
- At the place of the rotary table, radial deflection of the drill string is constrained, but it is allowed to rotate about its axis.

Figs. 4-6 show the place of supports in the model.

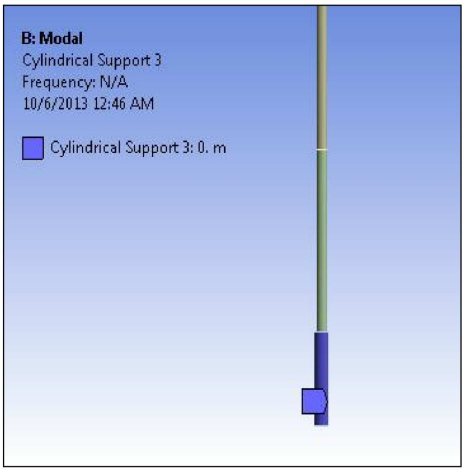

Fig. 4 Cylindrical support at the bit.

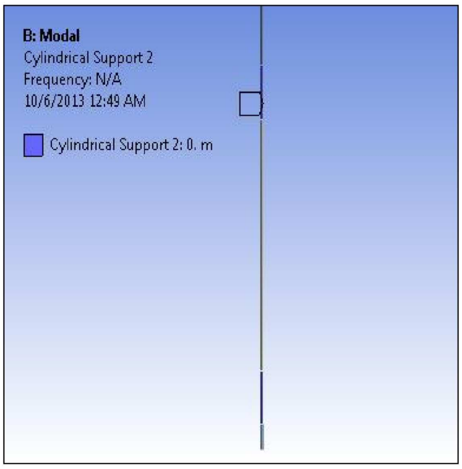

Fig. 5 Cylindrical support at stabilizer.

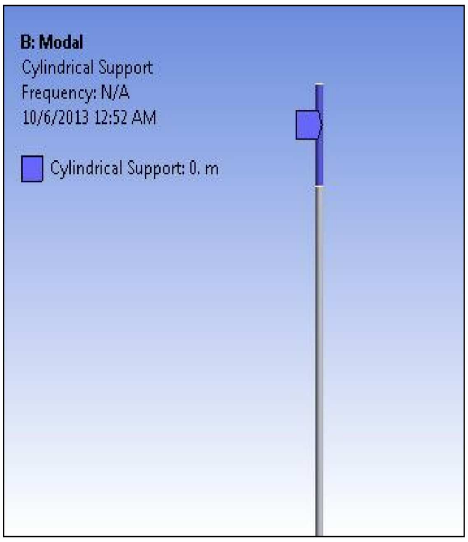

Fig. 6 Cylindrical support at the rotary table.

III. Results and Discussion

The result of the natural frequencies and mode shapes show similarity with that conducted in [10] in the results for the first three critical frequencies, as shown in Table 5. Also, natural frequencies of all the three-drill string design along with their mass imbalance are summarized in Table 6.

TABLE V
COMPARISON OF ANSYS RESULTS WITH THOSE IN [10]

Results	**Length (ft)**	**First natural frequency (Hz)**	**Second natural frequency (Hz)**	**Third natural frequency (Hz)**
ANSYS	110	1.43	1.43	4.38
Ref. [10]	114	1.23	3.15	4.19

TABLE VI
NATURAL FREQUENCIES OF DRILL STRING UNDER DIFFERENT CONFIGURATIONS

Results	**First natural frequency (Hz)**	**Second natural frequency (Hz)**	**Third natural frequency (Hz)**
Drill string 1	1.43	1.43	4.38
Drill string 1 with imbalance	2.03	2.23	4.63
Drill string 2	0.62	0.62	1.85
Drill string 2 with imbalance	0.85	0.90	1.83
Drill string 3	0.36	0.36	1.03
Drill string 3 with imbalance	0.44	0.54	1.06

The reason why 1st and 2nd natural frequency is equal is because of the symmetrical geometry (neglecting well inclination) used in simulation in which case the bending stiffness is equal about the strong and weak axis. If the applied frequency or rotary speed matches with this natural frequency, resonance occurs and the amplitude of lateral vibration exceeds drastically. Thus, the drill string collides with the wellbore and produces huge shocks. To prevent this from happening the operating speed or frequency must be out of these critical frequencies. Since the normal operating speeds of a drill string rotary table is around 100-200 RPM, these critical frequencies are of important value. From the table, for drill string 1 it is clear that without mass imbalance, the resonance of the drill string occurs at 85 and 262 RPM which is clearly out of the operating speeds. But when a mass imbalance is introduced in the drill string, the resonance occurs at 121, 133 and 277 RPM. Thus two operating speeds cause resonance, which can critically damage the drill string. The different length of the drill string has a significant effect on the natural frequency and mode shapes. The natural frequency reduces as the length of the drill collar increases. The mode shapes of drill string 1 for the first three deformations, subject to mass imbalance are shown in Figs. 7-11 below.

From the mode shapes, it is clear which areas of the drill string are subject to displacements. The lower part of the drill string, which consists of the stabilizers and bit are subject to almost no displacements at all. This is because the stabilizers negate the effect of displacements by constraining the drill string to only axial rotations without allowing any radial displacements. It is found that the maximum displacements are found at the place of the second drill collar, which is well above the two stabilizers. The effect of mass imbalance can be seen on figure wherein the amplitude of displacements increases with the same configuration.

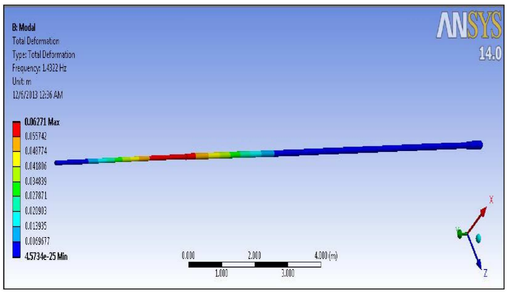

Fig. 7 First deformation of drill string 1.

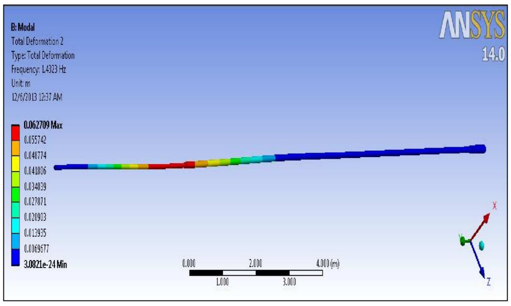

Fig. 8 Second deformation of drill string 1.

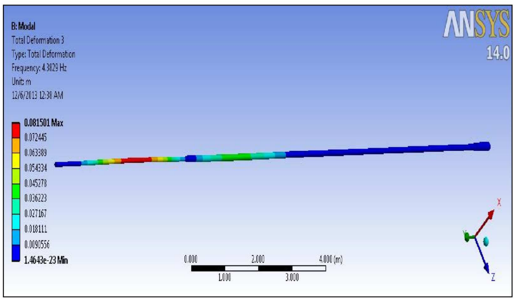

Fig. 9 Third deformation of drill string 1.

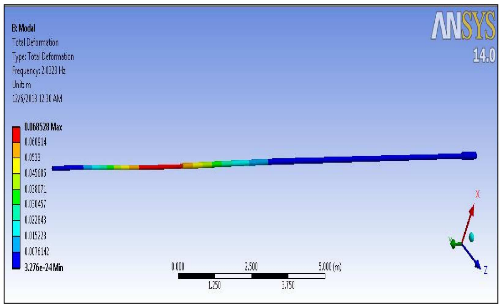

Fig. 10 First deformation of the drill string 1 with mass imbalance.

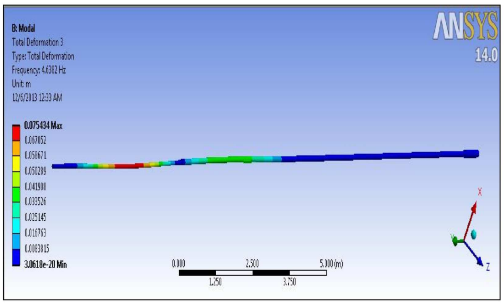

Fig. 11 Third deformation of the drill string 1 with mass imbalance.

Further, harmonic analysis is done mainly to understand the frequency response of a component of drill string when subject to a sinusoidal load. The critical component in this case is the second drill collar which is subject to mass imbalance. Therefore frequency response of the drill collar is plotted concerning lateral deflections in the X direction. It should be noted that while interpreting the normalized amplitude versus the frequency plot, the driving force at the bit is known until down hole measurements of vibration are available. The driving force will not be the same across every component of the drill string BHA. Additionally, the damping effect of the mud is not accounted for in the simulation. Therefore it is best to interpret the peaks of the curve rather than the magnitude of displacements. Figs. 12-14 show the frequency response of the drill collar in terms of lateral displacements.

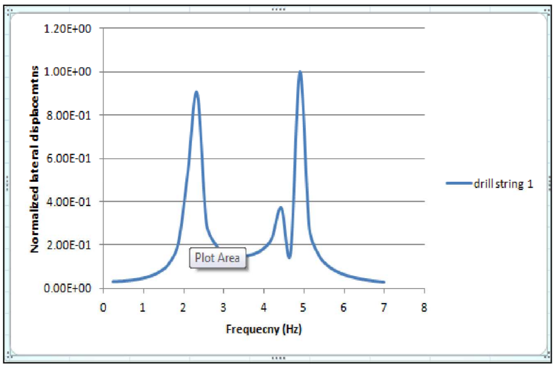

Fig. 12 Harmonic analysis of drill string 1.

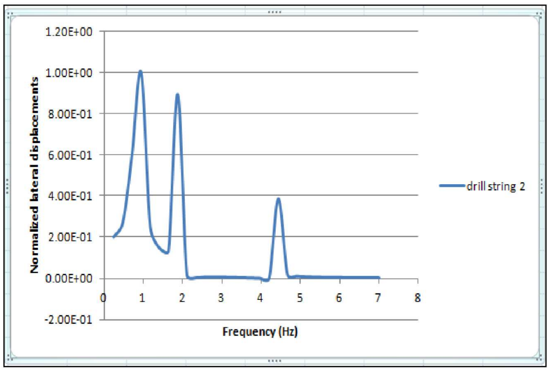

Fig. 13 Harmonic analysis of drill string 2.

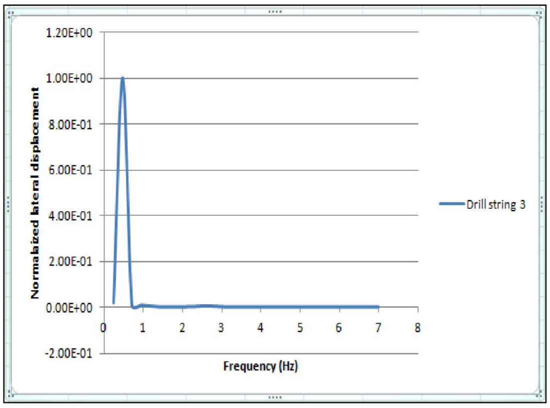

Fig. 14 Harmonic analysis of drill string 3.

The peaks indicate the onset of resonance in the harmonic response plot. This plot can be used to interpret which parts of the drill string is subject to large lateral displacements. In case of drill string 1, the drill collar is subject to large displacements at 4.8 Hz. This indicates that, the drill string is subject to maximum deflection at 4.8 Hz, while the peaks 2.2 and 4.2 Hz account for 90% and 38 % of maximum deflection. Thus, the operator has an idea of which frequency to avoid and which frequency to operate the drill string. Even though drill string experiences resonance at 4.2 Hz, the deflection of the drill string is not as severe as that encountered at 2.2 and 4.9 Hz. Similarly, for drill string 2, the displacement is high at .8 and 1.9 Hz but only 40% of maximum displacement is experienced at 4.5 Hz. However, for drill string 3, maximum displacement occurs at .5 Hz, and very small displacement occurs at 1 Hz. Therefore the drill string is safe to operate at its usual operating conditions.

This study also shows that finite element analysis is very suitable for the use of vibration studies by drilling engineers, in particular when the simulation results are compared with experimental data. In previous studies, finite element analyses are also shown suitable for design and thermal studies of well casings [11]–[13].

IV. Conclusions

Lateral vibrations can cause a significant amount of failures in MWD tools, drilling tools, and drill collars. This vibration effect is further amplified by the mass imbalance created by the MWD tools or the initial drill collar sag. The shocks produced by lateral vibrations can be higher than those which result from torsional or axial vibrations. This is because, under lateral vibrations, the drill string collides with the wellbore wall, creating huge shocks. As a rule of thumb, it can be understood that the more the mass and longer the drill string, the lower is the lateral resonant frequency.

A FEM model has been developed to investigate the problem of vibrations. When compared to field experiment data, the model has produced very close results. Therefore, it is very suitable for use in vibration studies by drilling engineers. The benefits of this study are that identifying the safe operating ranges of rotary speed for the drilling string as well as the lateral displacement of a critical component for a range of frequencies, thus avoiding damage to the string.

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