University of Stavanger

Faculty of Science and Technology

MASTER'S THESIS

Study program: **Master of Science in Petroleum Technology** Specialization: **Petroleum Engineering/Drilling Technology**	Spring semester, 2016 **Restricted access**
Writer: Ivan Vojteski	…………………………………………… (Writer's signature)

Faculty supervisor:
Helge Hodne, University of Stavanger

External supervisor(s):
Lei Fang, Baker Hughes Incorporated

Thesis title:

Improving Reliability of Under Reaming While Drilling Operations by Advancing Understanding in Drilling Dynamics

Credits (ECTS): 30

Key words: Vibration Under reamer BHA Rat hole Dual reamer	Pages: 111 + enclosure: 10 (in total 121) Stavanger, June 15, 2016

Acknowledgement

Master thesis is a result of the cooperation between University of Stavanger and Baker Hughes Incorporated. First, I want to thank Rune Sele, Drilling Services Manager for allowing me to be a part of drilling dynamics group and work on this subject. I want to express my gratitude to my supervisor at the University of Stavanger, Helge Hodne. His help was very useful, while giving me confidence and support in my work. I would also like to thank Ph.D Lei Fang a Global Product Line Manager at Baker Hughes for providing me with the most interesting topic for my thesis. His guidance was essential in generating this thesis. I enjoyed having discussions, and having him as a mentor was a real privilege.

While working on my thesis I have been involved with multiple people from the Baker Hughes. I would like to thank you all for your patience and contribution. Special thanks go to End of Well Supervisor, Angela Makohl for content review, Eline Duhs Nilsen and Borsheim Ragnvald who provided me relevant information and excellent advices.

Table of Contents

Table of Figures

List of Abbreviations

AST	Anti Stick – slip Technology
BHA	Bottom Hole Assembly
CoF	Coefficient of Friction
DLS	Dog Leg Severity
ECD	Equivalent Circulation Density
HWDP	Heavy Weight Drill Pipe
LWD	Logging While Drilling
MSE	Mechanical Specific Energy
MWD	Measurement While Drilling
OBM	Oil Based Mud
PDC	Polycrystalline Diamond Compact
RPM	Revolutions Per Minute
RSS	Rotary Steerable System
TD	Total Depth
WBM	Water Based Mud
WOB	Weight on Bit
WOR	Weight on Reamer

Abstract

Under reaming while drilling operations have become a common practice in the oil and gas industry to address drilling challenges, e.g. reactive and swelling formation, equivalent circulation density (ECD), tight casing tolerance, and to increase production. However, the reliability of the operation remains a challenge, largely due to the lack of understanding in drilling dynamics brought by the additional active cutting element, i.e. the under reamer blades, in the bottom-hole-assembly (BHA) and the lack of standard work practices to mitigate the associated risks.

This thesis aims to improve the reliability of under reaming while drilling operations by advancing understanding in drilling dynamics associated with under reamer BHA. Focuses were put on better predicting lateral vibration of the BHA and optimizing BHA design to minimize vibration related failures. These learnings were then applied to evaluate and improve the recently developed Baker Hughes pre-job planning procedures and best drilling practices guideline for under reaming while drilling operations. It is expected that the improved procedures will incorporate better understanding of downhole drilling dynamics and improve quality of service delivery for under reaming while drilling operations.

Through detailed static and dynamic analyses performed with a Baker Hughes proprietary Finite Element Analysis software program, this thesis specifically examined how the well path and mud property impact drilling dynamics with an under reamer BHA, identified ways to optimize BHA design to mitigate risks associated with drilling dynamics induced failures for under reaming while drilling applications, and explored the recommended work flow to eliminate rat hole with a dual reamer BHA.

1 Introduction

Under Reaming while drilling is a common practice used by operators to extend the size of the borehole and to address drilling challenges, such as ECD reduction, swelling and reactive formation and tight casing tolerances, and to increase production (Fang, Schwartze, Grindhaug, & Kanzler, 2016). As of today, however, the operation is still demonstrated to be a high risk operation. Two main challenges existing for this operation are improving the reliability and maximizing the efficiency of the operation. In particular, understanding downhole dynamics during reaming while drilling operations and mitigating the risks associated with the induced drilling dysfunctions are critical to overcome these challenges. It is documented that vibration related failures cost the industry approximately 300M USD per year (S. R. Radford, Hafle, Ubaru, Thomson, & Morel, 2010).

The thesis aims at improving reliability and reducing failures related to under reaming operations by advancing understanding in under reaming dynamics. Particular emphasis was placed on better predicting lateral vibration of the BHA and optimizing BHA design to minimize vibration related failures with the Baker Hughes GaugePro Echo reamer. While researching, the focus was on answering the following questions:

- How does the well path and mud property impact drilling dynamics with an under reamer BHA?
- How to optimize BHA design to mitigate risks associated with drilling dynamics induced failures for under reaming while drilling applications?
- What is the preferred BHA stabilization method for the special rat hole elimination application?

To do so, the current status of Baker Hughes knowledge on under reamers needs to be determined by examining the Baker Hughes procedures for pre-planning and drilling operations. The research was geared toward enhancing the understanding of downhole tool dynamics and toward improving the procedures used. By improving the procedures, it is expected to improve the quality of execution for all reaming operations.

The rest of the thesis is organized as follows. Chapter 2 provides background information that might be needed to understand the content of this thesis including topics such as different types of downhole vibrations, why they occur, their consequences for the operation, how to mitigate the consequences from vibration. Chapter 3 provides important information of the theory and actual procedures used while conducting research trough Baker Hughes propriety FEA software, it also describes the tools used in the BHA when performing static and dynamic simulations. Chapter 4 is focused on researching the impact of various well paths and mud weights on lateral vibration. Chapter 5 has focused on BHA optimization, and answering the most important questions in reaming operations. Chapter 6 provides important information in special applications that can be included in best drilling guideline for under reaming while drilling operations. In chapter 7 the most important findings will be discussed.

2 Background

This chapter provides important information that is needed to properly understand the rest of the thesis. The topics included in this chapter are historical progress of under reamer technology, all types of vibrations generated in under reaming operation, sources initiating and/or amplifying drill string vibrations, consequences of bad pre-planning and reaming while drilling procedures. In addition to bit and reamer synchronization, vibration mitigation strategies are covered.

2.1 Under Reamer Technology

Historically, hole opening technology has been considered unreliable. During the last two decades, significant improvements have been seen within the under reamer technology, such as concentric expendable. Concentric expendable under reamers are widely accepted in the industry. Conventional under reamers are activated by dropping a steel ball, which creates a bore restriction and thus a differential pressure develops to extend the blades. The effectiveness of dropping a ball can be limited by the environment, for example in highly deviated wells, where the process of activation can take 20-60 minutes. Another limitation of conventional under reamers is their placement in the BHA. Due to a ball drop process of activation, reamers have to be placed at the top of the BHA, which will result in leaving the rat hole 30m to 70m short to total depth. These ball drop reamers have also limited activation and deactivation because they are activated with a single ball drop and deactivated in the same manner. Under reamers will remain open as long as the flow rate is above a certain threshold. This limits the ability to circulate and clean the hole while the reamer is inside the casing. The activation status of these reamers cannot be directly confirmed, but indirectly by a secondary indicator such as a drop in differential pressure at the stand pipe. Because of this unclear activation status, the industry has progressed by developing different methods of activation such as multi-cycle ball drop under reamers, hydraulically actuated under reamers or reamers based on radio-frequency identification signals. All of these under reamers still have limitations; none of them is capable of storing the real time data and capturing the vibrations.(Fang, Schwartze, et al., 2016)

Baker Hughes was the most progressive company in the industry when it introduced the integrated under reamer, named Gauge Pro Echo in 2008. The Gauge Pro Echo remains unique among the tools available on the market today. It is a concentric expendable under reamer, activated and deactivated via downlink command. Activation and deactivation can be performed an unlimited number of times. The Gauge Pro Echo needs to be connected with an electrical power bus connection to operate; it needs to be in a BHA together with tools such as AutoTrak and OnTrak. During under reaming operations, the tool is able to provide information about its status, diameter of the opened blades, and vibrations. Data gathered is transmitted in real-time and also stored inside the tools memory. In the case of failures in which the real-time connection is lost, the tool memory can be downloaded and analyzed. The body of the tool is made of non-magnetic materials, which allow the tool to be positioned anywhere in the BHA without the chance of interfering with MWD/LWD tools. A maximum of three Gauge Pro Echo under reamers can be placed in the BHA.

The tool is capable of covering all conventional and unconventional reamer applications such as opening holes in salt formations, unstable formations, swelling formations. It can selectively ream critical hole sections, perform back reaming and even up-drilling (Fang, Manseth, Stue, Johansen, & Skappel, 2016). Present technology, despite all advancement, has its limitations to ensure execution of successful under reaming operation. Advancing the understanding in drilling dynamics associated with under reamer BHA is necessary in order to have a higher control of the under reaming operation. The key step in achieving better control of the under reaming operation is the proper understanding of vibrations.

2.2 Vibration

To properly understand the implications faced by the industry today, theoretical background must be provided. When researching drilling and under reaming dynamics, vibration is the origin of drill string failures. The following chapter informs on drill string vibration, how vibration is initiated, its different types, and the consequences of vibration for the operation.

2.2.1 Vibration Types

Drill string vibrations are defined based on their characteristic modes:

- axial,
- torsional,
- and lateral or transverse vibration.

Each type of vibration has different destructive impacts on the operation. Vibrations at low levels are harmless. The three types of vibration have individual vibrational patterns with varying severity and are generated by different sources. These types can occur in combination, which increases their unpredictability.

Severe vibration is destructive and causes bottom hole assembly failure, pre-mature bit wear/failure, hole opener damage, external drill string component damage, reduced rate of penetration, hole enlargement or poor hole quality, increased number of trips, and non-productive time. To improve under reaming and drilling performance, the drill string's response to dynamic physical conditions during the drilling process has to be understood.

The different types of vibration are as follows:

- Drill String Vibration
- Axial Vibration
    - Axial Oscillation
- Torsional Vibration
    - Torsional Oscillation
- Lateral/Transverse Vibration
- Snaking (Buckling)
- Whirl
    - Backward Whirl
    - Forward Whirl
    - Chaotic Whirl and Distinction to Lateral Acceleration

2.2.1.1 Axial Vibrations

Axial vibrations have been studied thoroughly over the years. Axial vibration refers to vibration created in the direction of the drill string's axis, i.e. in the wellbore direction. Axial vibrations are the result of the drill string moving upwards and downwards, and it can generate bit bounce as shown in Figure 1:

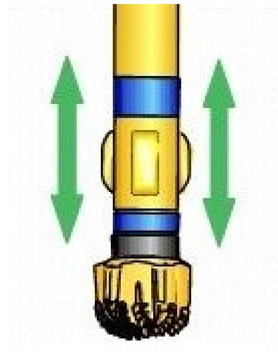


Figure 1: Axial vibration motion (*Internal Document - Dynamic Analysis*, 2016)

Bit bounce is a drilling dysfunction usually associated with roller-cone bits developing a tri-lobe bottom hole pattern after poor connection procedures or during on-bottom drilling. It occurs due to fluctuations in weight on bit (WOB).

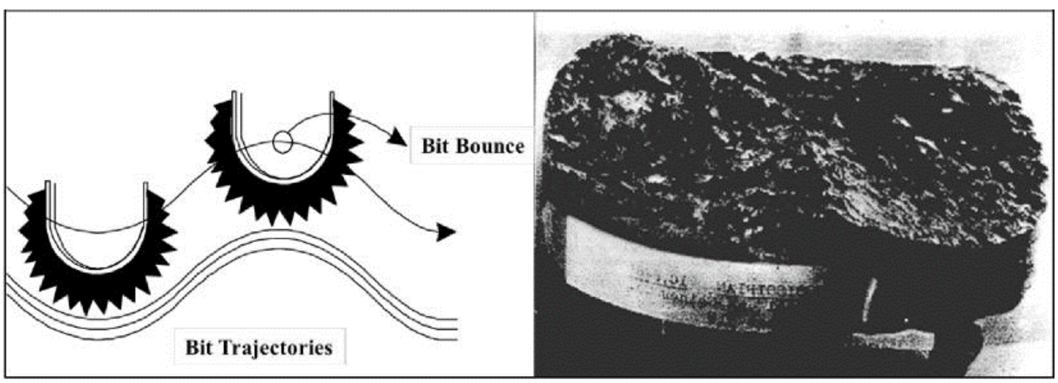


Figure 2: Left: Schematic of a roller cone lifting off bottom after reaching the crest of a tri-lobe pattern. Right: Example of a bottom hole tri-lobe pattern (*Internal Document - Dynamic Analysis*, 2016)

In the tricone/tri-lobe pattern example of Figure 2, three valleys and three crests have developed on the circumference of the well face. The dynamic energy is so high that the three cones, which will simultaneously reach the three crests, have so much upward inertia that they temporarily lift off the surface of the bottom of the hole despite the (low) weight of the BHA and the string that is pushing the bit down. The cones then again impact on the trailing slopes, applying instantaneous WOB in the valley of the tri-lobe pattern, which in turn causes better cutting of the formation (*Internal Document - Dynamic Analysis*, 2016).

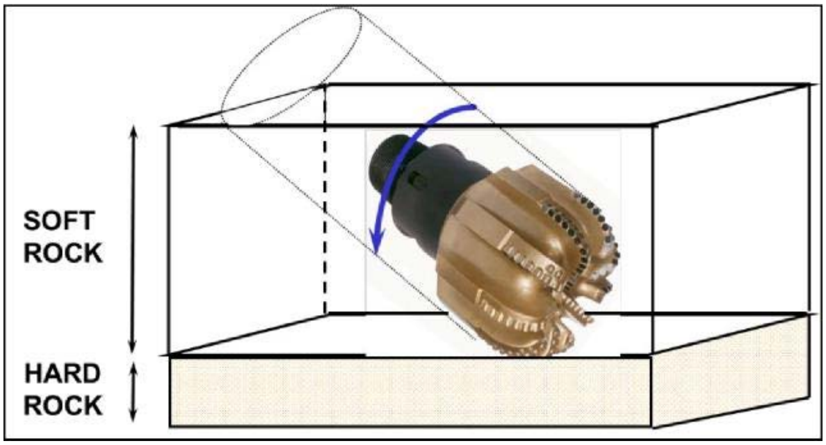


Figure 3: PDC bit hitting a soft rock/hard rock interface at an angle (*Internal Document - Dynamic Analysis*, 2016)

Although bit bounce as shown in Figure 3 is not too common and requires a formation interface of different unconfined rock strength, it is possible for the polycrystalline diamond compact bit to develop bit bounce. The individual blades bounce on the surface of the hard formation as illustrated in Figure 3.

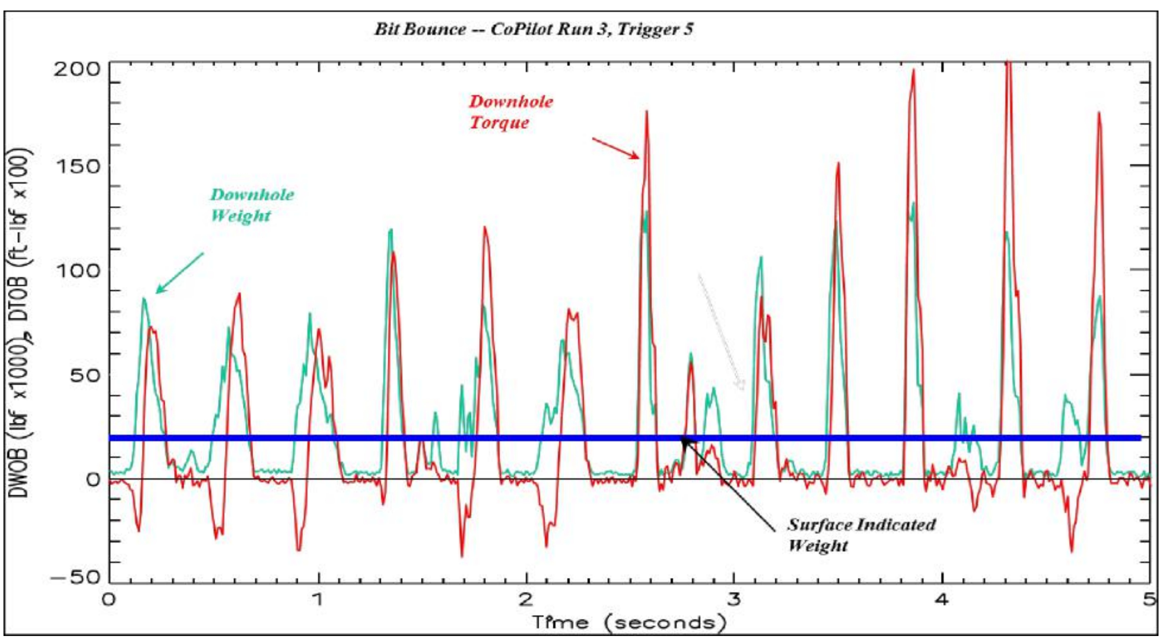


Figure 4: Weight and Torque data gathered by the downhole measurement tool (*Internal Document - Dynamic Analysis*, 2016)

Figure 4 shows torque and weight data gathered by the downhole measurement tool called CoPilot. In this case, extreme bit bounce is observed. It can be seen that the bit was off bottom for about half of the time when the green curve falls to zero and it reaches its maximum at 120klbf as compared to the 20klbf (blue line) seen on the surface. It is known that bit bounce is occurring when it moves upwards regularly and hits the bottom again (*Internal Document - Dynamic Analysis*, 2016).

Real-time corrective actions include optimizing the parameters, i.e. by reducing the revolutions per minute (RPM) and increasing the weight on bit (WOB). If this option does not work, it is advised to pick the drill string off bottom and to stop rotating. Then the original operation has to be restarted with half of the original RPM and the WOB has to increase slightly (*Internal Document - Drilling Dynamics*).

2.2.1.2 Difference between Bit Bounce and Axial Oscillations

WOB fluctuations, hook load fluctuations, axial oscillations can occur in the BHA, the string, the bit, or on surface (e.g. due to heave), and even at the bit following a tri-lobe pattern. As long as the bit does not lift off and impact the bottom of the hole, this is called axial oscillation, not bit bounce (*Internal Document - Dynamic Analysis*, 2016).

2.2.1.3 Axial Oscillations

Axial oscillation refers to the motion of the string when the bit does not lift off and affect the bottom of the hole. This phenomenon is experienced during drilling operation such as hook load variation, weight on bit fluctuation, oscillation due to heave etc. Low axial oscillations with low amplitudes are not destructive if they occur at a frequency range similar to that of bit bounce. High frequency axial oscillations are triggered by the agitator in the BHA. Axial accelerations can affect the cutting process and damage electronic components in the drill string. Axial oscillations can occur in a frequency range similar to that of bit bounce (e.g. due to tri-lobe pattern, heave, or drilling on a hard interface). This is then called low frequency axial oscillation. They are usually harmless as long as the amplitude is not excessive and they are not associated with strong accelerations or impact shocks. They are usually slow and have comparatively high axial deflection amplitudes.

Some components such as agitator tools excite much faster axially, resulting in high frequency axial oscillation. In this scenario, axial accelerations can become detrimental to not the cutting process and to electronic components (*Internal Document - Dynamic Analysis*, 2016).

2.2.1.4 Torsional Vibration

As illustrated in Figure 5, torsional vibrations are observed as twisting motions in the drill string. Torsional vibrations are usually generated by stick-slip.

Figure 5: Torsional vibration motion (*Internal Document - Dynamic Analysis*, 2016)

The vibrations occur when the bit and drill string are periodically accelerated or decelerated, due to frictional torque on the bit and the BHA. Due to low torsional stiffness of the drill string, the rotary speed at the bit oscillates around the surface speed. . A stick-slip is phenomena where the bit can periodically stop (stick) and start again (slip), accelerating to speeds two or three times the surface rotary speed (Bybee, 1999). Torsional vibration is a different term for RPM variations (or possibly dynamic torque fluctuations).

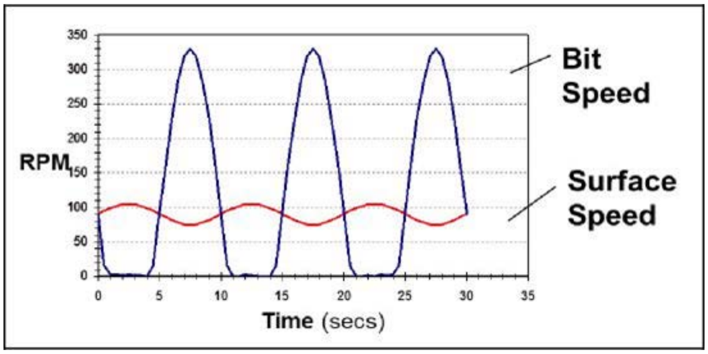


Figure 6: Fully developed stick-slip on bottom, causing rpm (and torque) oscillations on surface (*Internal Document - Dynamic Analysis*, 2016)

Low rotating speed of the bit, aggressive bits, high friction factors along the drill string, and limber (long, small diameter) drill strings plus certain drilling practices and environments encourage the development of stick-slip (*Internal Document - Dynamic Analysis*, 2016).

Stability maps can vary significantly in values and character depending on the application. Bit whirl develops at low WOB and high RPM while stick-slip is a dysfunction expected at high WOB and low RPM as illustrated in Figure 6. Stick-slip is a relatively slow dysfunction with frequencies typically below 1 Hz. It does not only depend on the BHA but also on the drill string (*Internal Document - Dynamic Analysis*, 2016).

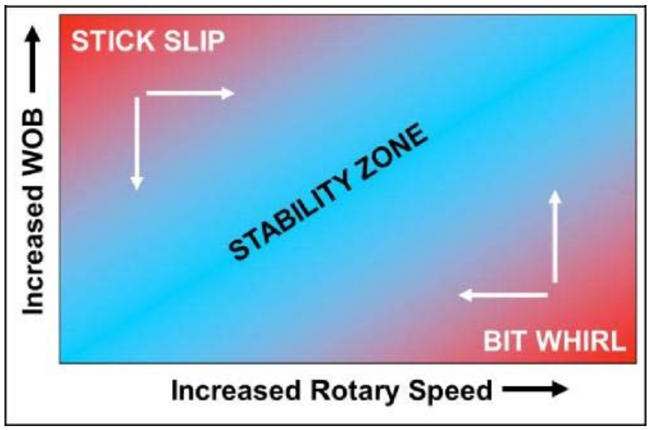


Figure 7: Stability map (*Internal Document - Dynamic Analysis*, 2016)

The time the bit stays in stationary mode depends on stick slip severity and rotational acceleration speed at the end. Due to rotational acceleration, the bit RPM can become several times higher than the surface RPM.

Damage due to torsional vibrations is identified as one of the main causes of drill string fatigue and bit wear. In severe cases, torsional vibration can lead to destructive fluctuating torques in the drill string and the BHA itself. Once the vibration is out of control, it will damage the bit and/or the drill string (Robnett, Hood, Heisig, & Macpherson, 1999).

The interaction between the drill string and the borehole wall and rock-bit contact create stick slip. It mostly occurs in high angle wells with long laterals and deep wells. Stick slip can also be triggered by aggressive polycrystalline diamond compact (PDC) bits with high WOB, and hard formations or salt (*Internal Document - Dynamic Analysis*, 2016).

The drill string is continuously exposed to torsional vibrations, as the bit and drill string are subjected to friction. Torsional vibrations are damped by the torsional stiffness of the drill string and by the friction against the wellbore wall. The stiffness in torsional direction is not as significant as the stiffness in the length direction and hence the dampening is less pronounced than in axial vibrations. Due to the elasticity of the drill string, the rotations will most often be irregular. A stiffer drill string could potentially dampen the stick-slip indices. The vibration mode is observed at surface as large variations in torque values. Even in deviated wells, torsional vibrations can be detected by surface measurements and reduced by the driller (*Internal Document - Drilling Dynamics*)

The bit, under reamer, and drill string are constantly exposed to friction, resulting in torsional vibrations. It can be minimized by stiffing the drill string as well as reducing wellbore friction. Stick slip indices could be decreased by a stiffer drill string. Large fluctuations in surface torque values reflect vibration; therefore, the driller always has a chance to reduce it (Robnett et al., 1999).

At high WOB, the cutter penetrates deeper into the formation, which results in higher torque and side forces. It will increase the possibility of torsional vibration. When looking at RPM, torsional vibrations can occur at an ideal value that varies from well to well depending on its condition. Therefore, it is always recommended to have low WOB and high RPM in order to avoid stick slip.

2.2.1.5 Torsional Oscillations

Torsional oscillation refers to changes in RPM when the string does not come to a full rotational stop. This is quite normal in drilling and under reaming. Torsional oscillations are not detrimental to the mechanical integrity of drill string, the BHA, specifically the bit and the under reamer if they occur at a low frequency. They only affect the quality of LWD data for example image logs. Similar to stick slip, torsional oscillation also depend on both the BHA and the drill string design (*Internal Document - Dynamic Analysis*, 2016).

2.2.1.6 Lateral/Transverse Vibrations

Lateral vibrations are observed as side-to-side motion in a lateral/transverse direction relative to the drill string as illustrated in Figure 8. The vibration mode is primarily generated by whirl. The BHA must have sufficient lateral movement to bend and touch the borehole wall. The result of this scenario is that lateral vibration occurs. It is also related to mode coupling, a phenomenon where the lateral vibration initiates both axial and torsional vibrations. (Christoforou & Yigit, 2003)

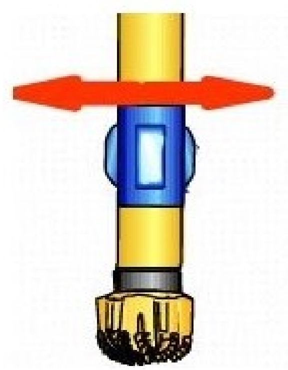


Figure 8: Lateral vibration motion (*Internal Document - Dynamic Analysis*, 2016)

In inclined wellbores, lateral vibrations lead BHA components to have a steady state low wall contact while rotating. Thus, due to friction, the BHA climbs up the wall and then falls back with fluctuating or no wall contact. The vibration amplitude is function of well bore friction and the system energy.

The damage done to BHA components and wellbore is very high; therefore, it is considered as one of the most destructive influences. Due to the interaction of bit/BHA and under reamer with the borehole wall, while experiencing lateral vibrations, several drilling problems arise which include equipment damage, loss of control of the well trajectory, fatigue in the drill string, and hole erosion.

Detecting these vibrations on surface is difficult compared to torsional vibration because transverse vibrations have a tendency to dampen out along the drill string in upward direction. That is why, the directional driller is limited in taking preventive measures

(Bybee, 2009). During drilling, reduction in RPM and increased WOB can minimize the level of vibration if lateral vibrations get recorded. Optimizing the drilling parameters after lifting off the assembly to unwind the torque also leads to reduction in vibrations. The use of a shorter and stiffer BHA in lateral direction can also prevent these vibrations (*Internal Document - Dynamic Analysis,* 2016).

2.2.1.7 Snaking

Snaking is a lateral movement along the low side of an inclined wellbore. This is much like the movement of a snake. Snaking is friction induced and typically applies to long sections of string with few and small upsets only, especially in long horizontal tangents (Heisig & Neubert, 2000). Snaking only causes moderate bending loads, which is why it is deemed harmless. There is no impact on the wall, as the tools always stay on the low side of the borehole wall. The greatest danger posed by snaking is that it can develop into other forms of vibration: lateral acceleration and whirl (*Internal Document - Dynamic Analysis*, 2016).

2.2.1.8 Whirl

Whirl is a lateral vibration with almost circular lateral movement around the borehole axis (not around the steady state lateral deflection of a certain BHA position). Unlike to torsional vibration, whirl does not have any RPM fluctuation associated to it. Whirl depends on following two types of motion:

1. The near circular lateral movement around the borehole axis.
2. The direction of the rotation of the drill string around its own axis.

Whirl can be classified into backward and forward whirl (and chaotic whirl, which is a frequent switch between the backward and forward whirl). Therefore, a similar BHA can have forward whirl in one section; backward whirl in another, and lateral acceleration in a third.

High energy whirls are more severe than low energy whirls as they have high deflection and bending load amplitudes. This results in many wall contacts and makes high energy whirl harmful. Low energy whirls have fewer or no wall contact and are not damaging. When high energy whirls are compared with low ones during lateral acceleration diagnostics in a downhole measurement tool/CoPilot diagnostic system, high energy shows large values. On whirl diagnostics, both of them give high values (*Internal Document - Dynamic Analysis*, 2016).

2.2.1.9 Backward Whirl

As shown in Figure 9, lateral movement around the borehole axis and drill string rotation direction are opposite (the one rotates clockwise, and the other one is anticlockwise).

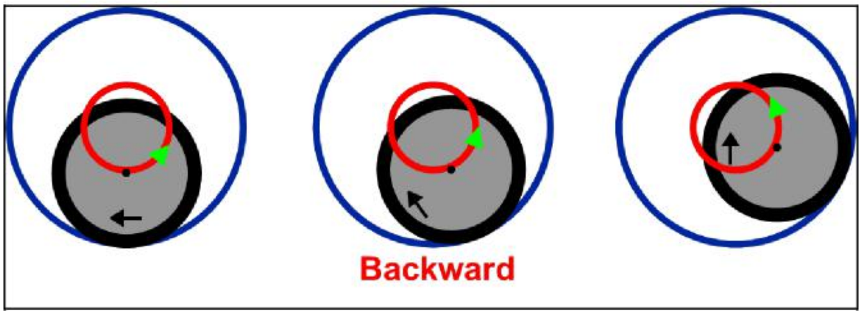


Figure 9: Backward Whirl (*Internal Document - Dynamic Analysis*, 2016)

Backward whirl is similar to BHA components rolling rather than scraping along the wellbore wall (full circle, not just on the low side).

Detrimental consequences of backward whirl are

- high dynamic bending loads particularly in large holes.
- and, the frequency at which the BHA is bent can be multiple of the respective strings RPM.

Figure 10: Crack that developed due to bending fatigue from backward whirl (*Internal Document - Dynamic*

***Analysis*, 2016)**

Circular rolling movement becomes faster in smaller clearance (between tool and borehole). As a result, the associated bending loads are smaller but on the other side fatigue cycles sum up faster.

Different clearances are present at BHA wall contact. It makes the whirl frequency different for different components. Therefore, most BHAs have no well-defined whirl frequency. Normally, the components settle at a “mean” frequency with small intervals of lateral acceleration. There is also wall contact each stabilizer (*Internal Document - Dynamic Analysis*, 2016).

2.2.1.10 Forward Whirl

Forward whirl is defined as whirl in which both the circular lateral movement of the BHA around the borehole axis and the rotation of the string around its own axis are clockwise as illustrated in Figure 11.

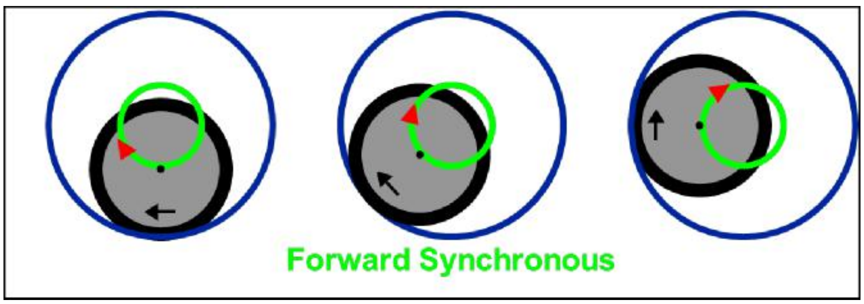


Figure 11: Forward Synchronous whirl (*Internal Document - Dynamic Analysis*, 2016)

In this whirl, the rotational speed of two movements is identical. All the wall contacts have the same tool face position of the BHA. Accelerated wear can be experienced due to forward whirl.

The effect of forward synchronous whirl on the BHA is accelerated wear. The following are typical features associated with forward whirl:

- the wear is non-uniform around the circumference of the tool
- contact forces are higher

- additional wall contacts, which are often present at low wear protection areas of tools

As opposed to backward whirl, bending fatigue is not associated with forward whirl because the tools are bending in the same direction as its own field of reference. In well-rounded boreholes, the impact damage is very small in forward whirl.

Figure 12: Wear flat that was caused by forward synchronous whirl (*Internal Document - Dynamic Analysis*, 2016)

When tools are subjected to high bending loads with mass imbalance, they always tend to show forward whirl, a preferred bending direction, or pre-bent in the BHA. Due to mass imbalance or smallest moment of inertia, flat wear usually occurs on the tool face (*Internal Document - Dynamic Analysis*, 2016).

2.2.1.11 Chaotic Whirl and Distinction to Lateral Acceleration

Chaotic whirl occurs when the circular movement of the BHA alternates rapidly between clockwise and anticlockwise rotation. Chaotic whirl and lateral acceleration normally alternate. Because the whirl occurs in a circular manner around the borehole wall and lateral acceleration in the center of the well, the lateral vibrations are usually in between. In chaotic whirl, wall contacts occur at different positions in particular depths. This is contrary to forward and backward whirl where continuous wall contact exists (*Internal Document - Dynamic Analysis*, 2016).

2.2.1.12 Bit Whirl and Under Reamer Whirl

Forward, backward, and chaotic whirls impact the bit or hole opener/under reamer. It can cause chipped teeth, rounded blade gauges, and polygon-shaped holes. Bit whirl is typically a type of backward whirl along with chaotic elements. Due to bit whirl, polygon shaped holes are drilled with one lobe more than the bit. Bit whirl often occurs with unstable bits resulting in cuts with lower depth. Whirl of the main under reamer in the BHA can destroy the cutter and LWD/MWD components below it (*Internal Document - Dynamic Analysis*, 2016).

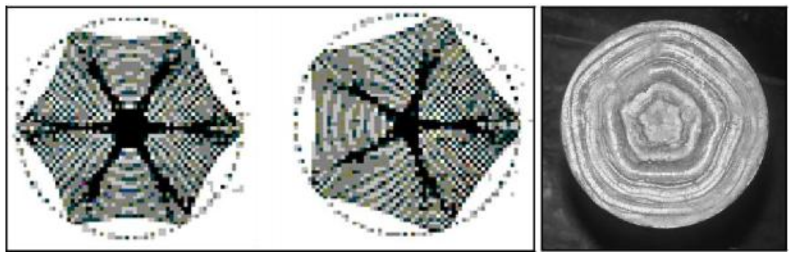


Figure 13: Bottom hole patterns of bits drilling with fully developed backward whirl. Left: 5-bladed bit causing 6-lobe pattern, Middle: 4-bladed bit causing 5-lobe pattern; Right: face of a field core (*Internal Document - Dynamic Analysis*, 2016)

Figure 14: Shoulder cutters of a whirling PDC bit can experience intermittent backward motion. Middle: Chipped or broken cutters due to backward whirl. Right: Rounded blades are also a common consequence (*Internal Document - Dynamic Analysis*, 2016)

2.2.2 Sources Initiating and/or Amplifying Drill String Vibrations

This section examines the sources of vibration in the drill string and also highlights their origin and mechanism of excitation. Particular drilling conditions usually trigger drill string vibrations when bit, under reamer, and drill string components interact with the formation. Several factors can be a source of bottom hole vibrations. These sources can also create resonance in the drill string and can be the cause for activating other vibration mechanisms. Resonance is a phenomenon which occurs when there is a match between excitation source frequency and natural frequency of vibration. Critical speed is a term associated with the speed at which the resonance occurs. Vibration may exist in both presence and absence of resonance. The drill string vibrates significantly if high levels of excitation exist. It is an example of vibration independent from resonance. Accelerated fatigue is generated in large amplitude vibrations.

2.2.2.1 Drilling Parameters (RPM, WOB, WOR and Mud Lubricity)

Drilling parameters such as revolutions per minute (RPM), weight on bit (WOB), weight on reamer (WOR), flow rate and mud lubricity impact vibration. Selecting optimal parameters during drilling can significantly reduce the vibration level. Therefore, the role of the directional driller is critical.

Different studies have shown that high RPM contribute to drill string vibration. But, it depends on both RPM and WOB and WOR. Selecting a suitable range of drilling/reaming parameters is necessary to make drilling efficient. This range can be based on the type of vibration to be avoided, wellbore condition, BHA design etc.; the range can be designed using commercial software for BHA design and can be optimized while drilling/reaming.

Mud lubricity is an important characteristic of drilling mud, which can help to reduce drilling vibration. In particular, stick-slip can be reduced by reducing friction at the bit and BHA. Water-based muds use lubricants as an additive for friction reduction while oil-based muds have an already reduced co-efficient of friction against the wall. This is the best mud system with regard to stick-slip as explained in chapter 2.2.2.1.4.

2.2.2.2 Significance of WOB, WOR, RPM and Flow Rate

Drill string vibrations depend directly on drilling parameters such as WOB, RPM and flow rate. These vibrations can also be affected if hole opening is done simultaneously because of the weight and torque distributions. Parameters used with an eccentric reamer or a concentric under reamer can be an additional source of vibrations. The weight from surface is distributed both on reamer and bit.

The evaluation of parameters both on bit and reamer is very important to avoid conditions suitable for vibration. Normally, bit aggressiveness is used to analyse WOB required to generate specific torque on bit when drilling without reamers. Another approach is to use blade aggressiveness in designing parameters. The theory behind blade aggressiveness considers bit and reamer as discrete number of spokes. (Meyer-Heye, Reckmann, & Ostermeyer, 2010)

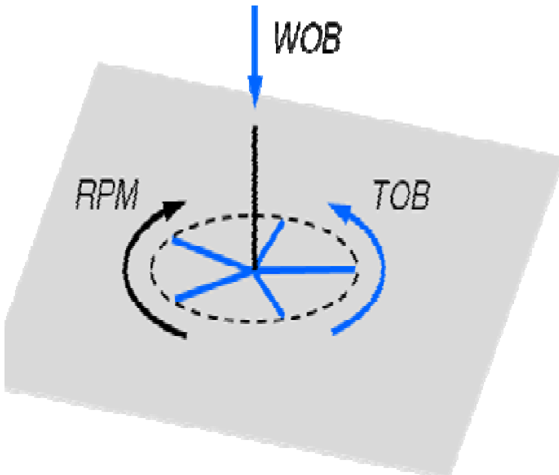


Figure 15: Bit with 5 blue blades

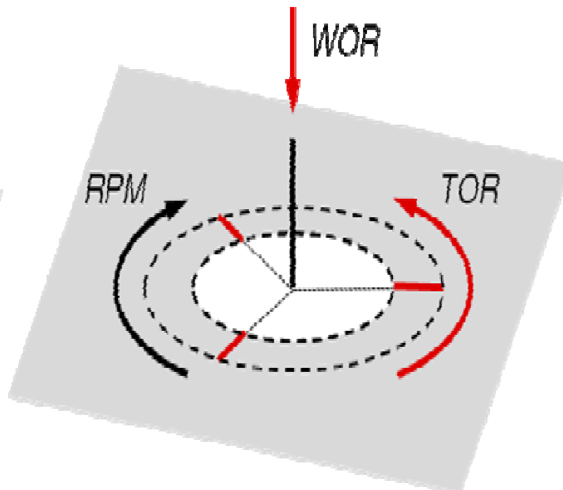


Figure 16: Under reamer with 3 red blades

2.2.2.3 Calculating Parameters

The parameters are given as

$$\mathrm{WOB} = \frac{\mathrm{TOB}}{a_B * \mu_B} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

Where: TOB – Torque on bit;

WOB – Weight on bit;

a_B – Arm of the torque of the bit;

μ_B – Aggressiveness of the bit;

$$\mathrm{WOR} = \frac{\mathrm{TOR}}{a_R * \mu_R} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Where: TOR – Torque on under reamer;

a_R – Arm of the torque of the under reamer;

μ_R – Aggressiveness of the under reamer;

WOR – Weight on under reamer;

Drill string Vibration and Mechanical specific Energy (MSE) depend on each other. Mechanical-specific energy describes energy required to remove a unit volume of rock. The relationship between MSE and vibration can be understood from the fact the total energy being transferred from the surface is utilized to drill and under ream the formation. Parts of it can dissipate and can contribute to increased vibrations. High MSE is a result of levels of vibration which as a result reduce drilling efficiency as it takes away energy that could be utilized for drilling and under reaming (S. R. Radford et al., 2010).

$$\mathrm{MSE_B} = \frac{\mathrm{WOB}}{\mathrm{A_B}} + \frac{120\pi * \mathrm{RPM} * \mathrm{TOB}}{\mathrm{A_B}} \quad \ldots \ldots (3)$$

Where: $\mathrm{MSE_B}$ – Mechanical specific energy of the bit;

$\mathrm{A_B}$ – Cross sectional area of the bit;

RPM – Revolutions per minute;

$$\mathrm{MSE_R} = \frac{\mathrm{WOR}}{\mathrm{A_R}} + \frac{120\pi * \mathrm{RPM} * \mathrm{TOR}}{\mathrm{A_R}} \quad \ldots \ldots (4)$$

Where: $\mathrm{MSE_R}$ – Mechanical specific energy of the under reamer;

$\mathrm{A_R}$ – Cross-sectional area of the under reamer;

(Meyer-Heye et al., 2010)

2.2.2.4 Significance of Mud Lubricity

Mud lubricity also has an impact on drill string vibrations. Both water-based muds and oil-based mud offer different lubricity solution in their applications areas. WBMs have less lubricity than OBMs although it has several other benefits such as low cost, less environmental concerns, ease in storage and disposal. Lubricity can be quantitatively analyzed using co-efficient of friction (CoF) between the tools and wellbore wall. WBMs offer high CoF typically between 0.2-0.5 where as in OBMs, CoF can be reduced up to 0.1 (Schuh et al., 2014).

Table 1: Types of drilling fluids and their corresponding friction factors (Samuel, 2010)

Drilling Fluid Type	Friction Factors	
	Cased Hole	Open Hole
Oil Based	0.16-0.20	0.17-0.25
Water Based	0.25-0.35	0.25-0.40
Brine	0.30-0.40	0.30-0.40
Polymer Based	0.15-0.22	0.20-0.30
Synthetic Based	0.12-0.18	0.15-0.25
Foam	0.30-0.40	0.35-0.55
Air	0.35-0.55	0.40-0.60

Recent techniques for reducing wellbore friction in deviated wells involve adding new additives to the mud, especially in WBMs; refer to the biotechnology method for encapsulating oil in polysaccharide-based polymers systems (Schuh et al., 2014).

2.2.2.5 Mass Imbalance

Several components in drill string introduce an “unbalanced” condition. This unbalanced condition is called a mass imbalance which can happen due to misalignment of borehole, initial bending and curvature, and wear during operation. A balanced string has center of gravity coinciding with axis of rotation. If this condition does not exist, an unbalanced situation arises which creates centrifugal forces during rotation which in the end makes string to vibrate. Figure 19 shows how bending or whirl occurs- due to imbalance force. When the natural frequency of such string matches with the rotary speed, shocks occur and strings can have impact with wellbore wall. Therefore lateral vibrations occur very much due to mass imbalance of drill string components (Dykstra, Chen, Warren, & Azar, 1996).

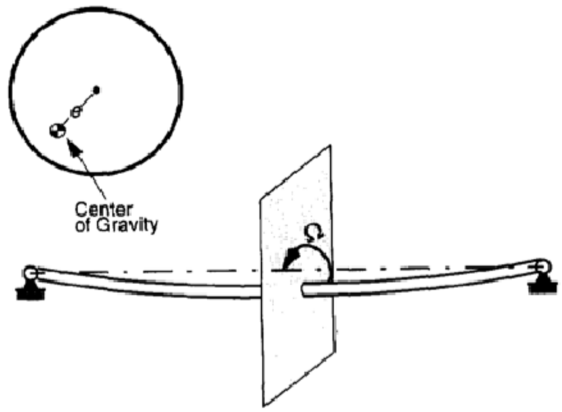


Figure 17: Imbalance force acts on a rotating shaft causing it to bend (Dykstra et al., 1996)

2.3 Consequences of Bad Pre-Planning and RWD Procedures

The section will cover the consequences of high severity vibrations, which result in poor reaming and drilling performance, NPT, damage to the tools and overall increase of costs.

2.3.1 Wellbore Instability

Chemical interaction between drilling fluid and formation has been considered as main concern for wellbore instability. Drill string vibrations are only brought under study to analyse drill string fatigue issues not for wellbore instability. As a result, downhole vibrations have never been looked into from wellbore instability perspective. To make operation an economical one, the identification of source for wellbore instability is very critical and it can save a lot of time if done at initial stages.

When significant lateral amplitudes of drill string hit the wall, an irreparable loss to the borehole occurs. As a result of this, a situation of unstable hole may arise and blocks of rock begin to fall into the well. In hard formation, the drilling fluid and rock have minimal chemical interactions, therefore, drill string vibrations should be carefully studied in these situation as a potential source of wellbore instability (Santos, Placido, & Wolter, 1999).

Near bit sensors are used to measure drill string lateral acceleration. In better drilling environment, the values normally lies between 20 and 30g, where g = 9.81 m/s2. The value can go up to 80g in harsh environment and even up to 200g with extremely high lateral acceleration.

In order to check the significance of acceleration on wellbore condition, let us consider an example of 80g for 12.25in hole having drill collar of 8in. (150lbm/ft). The lateral force exerted by 1 ft of drill collar on the wall is given by,

$$F = ma \ldots \ldots \ldots \ldots \ldots \ldots (5)$$

$$F = \frac{150lbm}{ft} * 1ft * \frac{0.454kg}{lbm} * 80 * \frac{9.81m}{s^2} \ldots \ldots \ldots \ldots (6)$$

$$F = 53390N = 5.41\ tons \ldots \ldots \ldots \ldots (7)$$

When the lateral force of 5 tons is exerted the on wall, it will cause a considerable damage or at least create a large fractured area. If the vibration scenario is added to this, the effect can becomes significantly devastating for hole stability because of repetitive exposure of such a high lateral force (Santos et al., 1999).

The amount of kinetic energy transferred from drill string to wellbore wall can be calculated by continuing with the above example of drill collar and taking impulse into consideration. The impulse on 1ft of drill collar for 0.02s duration is given by,

$$I = \int F\ dt \ldots \ldots \ldots \ldots (8)$$

$$I = F\Delta t \ldots \ldots \ldots \ldots (9)$$

$$I = 53390 * 0.02 = 1067.8Ns \ldots \ldots \ldots \ldots (10)$$

Considering plastic shock (zero final velocity), drill string velocity prior to impact can be calculated as,

$$I = mV_o \ldots \ldots \ldots \ldots (11)$$

$$V_o = \frac{I}{m} = \frac{1067.8}{150 * 0.454} = 15.7\frac{m}{s} \ldots \ldots \ldots \ldots (12)$$

The mechanical energy (kinetic energy) is given by,

$$K.E = \frac{1}{2}mV^2 \ldots \ldots \ldots \ldots (13)$$

$$K.E = \frac{1}{2} * 150 * 0.454 * 15.7^2 \ldots \ldots \ldots \ldots (14)$$

$$K.E = 8392.98\ \mathrm{J} \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (15)$$

(Santos et al., 1999)

2.3.2 Damaged Downhole Components

The vibrations in drill string can be destructive and can lead to both minor and major failures. Following points through light on the failures and damages that vibration can result in.

- Unplanned trips due to damages in rotary steerable system
- ROP reduction due to bit and reamer damage.
- Drill string fatigue which can further lead to twist off and fishing. Sidetracks would be the last resort if stuck tools would not get retrieved.
- Failure of downhole tools as their electronics get affected by the vibrations.
- Disturbing the mud pulse telemetry of downhole tools.
- Inefficiency of rig equipment, sometimes catastrophic damages.(S. R. Radford et al., 2010)

2.3.3 Increased Costs

Petroleum industry always focuses on profit maximization and cost reduction. Therefore, consideration of economic effect of drill string vibrations is very vital and influential. Minimizing the vibrations can lead to an efficient drilling system, low cost of maintenance, less number of trips and fishing jobs. It will benefit both the operator and service companies. Neglecting the importance of vibration mitigation in early stages will result in wasting of precious energy, ROP reduction and high NPT. Statements have been made that both operators and service companies in total lose 300 million $ annually due to vibrations (S. R. Radford et al., 2010).

2.4 Vibration Mitigation Strategies when Planning an Application

Common vibration analysis with software that utilize finite element analysis focus more on eliminating the vibration related issues and try to grasp the particular dynamic behaviour behind it. This is done by eliminating the vibrations as a whole, reducing amplitudes of vibration or by keeping the vibration zones in the strong components. A lot of vibration analyses have aim to optimize but they all are limited to particular applications which sometimes go against each other. But it always allows us to plan the vibrations mitigation in early stages so that the cases can be compared with each other.

2.4.1 Reduce the Friction Factor

High friction factor between well and tools create almost all types of vibrations. Torsional and lateral vibrations occur in rotating mode whereas axial direction vibration can be present in sliding application because of the pipe movement in axial direction.

Mud lubricity plays a vital role in reducing friction factor. Use of lubricants, controlling the solid and alteration in mud system can have a great impact in vibration mitigation. Non-rotating stabilizers especially for RSS tools like AutoTrak steering units can be influential. On the drill pipe, use of protectors around tool joints can offer less friction factor. Torque and drag values for any interval depends on the quality of hole. Variations can occur due to changes in lubricity, hole cleaning efficiency, drill string dynamics, swab and surge effects, use of torque reducing tools (Payne & Abbassian, 1997). Minimizing friction will enable to drill extended reach well with less vibration and low torque and drag losses.

Minimizing the friction factor actually makes the contact forces low which in turn reduce sticking and slipping of tools. Due to this, number of stabilizers and other wall contacts are also reduced. The potential of whirl is also minimized as there is low chance that the string can climb to the higher side.

Adding oil and other lubricants can have detrimental effect on mud properties, increase the cost and make it difficult to treat. Use of protector around drill pipe is time consuming process as well as expensive operation. Under reamer BHA features helps in under reaming and drilling good quality well but sometimes the steering ability becomes limited (*Internal Document - Dynamic Analysis*, 2016).

2.4.2 Stiffen the Drill Pipe

Stiffen the drill pipe means that it will be less prone to vibrations specially stick slip and torsional oscillation.

The stiffness of drill pipe depends on its length, diameter and wall thickness. Taper string is a common method to reduce stick-slip with smaller diameter followed by larger diameter drill pipe.

The natural frequency of drill string is a function of length and cross sectional area of drill pipe, DC, derrick and cable line (Li, 1986). The natural frequency of the whole system increases when stiffening the drill pipe. Less stiff pipe twists more as compared to a stiffer pipe. Although the torsional inertia increases with stiffness, fluctuation in rotational speed will be less at same rotational acceleration.

The drawback of stiffing pipe is that it adds more friction to the system especially in inclined wells.

Low inclination wells provide fewer vibrations with stiffer and larger pipes. For highly deviated wells, stiffer pipes can produce more torsional vibrations (*Internal Document - Dynamic Analysis*, 2016).

2.4.3 Reduce the BHA Weight

Reduction in BHA weight helps to minimize stick-slip and torsional oscillations.

Weight of BHA depends on number of drill collars (DCs) and heavy weight dill pipes (HWDPs) along with their dimensions especially outer diameters and thickness. Reducing number of DCs and HWDPs make the BHA weight lesser.

BHA with less weight has less inertia. As a result, torsional natural frequencies become higher. Frictional torque also reduces as the BHA weighs lesser.

High risk of buckling and less available WOB are the demerits of reducing BHA weight to minimize the vibrations.

The concept of reducing BHA weight and stiffening the drill pipe seems to be contradictory as stiffness and inertia are proportional to each other. But in order to increase stiffness of pipe, the focus should be on making it larger in diameter by maintaining the same weight. Normally BHA is stiff, heavy and short whereas drill pipe are kept as long, limber and light. This strategy must be altered once the operation faces vibration and real time decisions should be taken.

High inclination wells and high friction wells are typically the application areas for BHA weight reduction for minimizing vibrations (*Internal Document - Dynamic Analysis,* 2016).

2.4.4 Apply Torsional Damping

As the name implies, torsional damping applies tools and techniques to minimize torsional vibration issues.

Different tools and special features are used to dampen the torsional fluctuations such as power section of mud motor, downhole tools such as AST (Selnes, Clemmensen, & Reimers, 2008). This tool makes sure that the torque level remains constant so that the fluctuations get minimized. Some top drive control system employs built-in system to control and monitor surface RPM oscillation at surface rather than downhole. Low frequency torsional oscillations can only be countered using these systems. Drilling fluid does not contribute in torsional damping greatly.

The idea behind damping is to dissipate the energy taken out from torsional oscillation and reduce the amplitude of vibrations.

Adding more tools to the BHA increase complexity to the system and elevates the failure risks. Capability of steering also is affected.

The mentioned tools should be used if there is stick-slip history in that area. Top drive control system can be already present or it can be asked for (*Internal Document - Dynamic Analysis,* 2016).

2.4.5 Increase BHA Stiffness

Lateral vibrations can be reduced by making BHA stiffer. It is also researched in chapter 5 that

BHA can be made laterally stiffer either by increasing tool diameter (DCs instead of HWDPs) or by reducing the spacing between stabilizers (BHA section stiffness).

Stiffer BHA adds higher natural frequency to the system and, thereby, avoiding the resonance that would have been created by mass imbalance excitation.

Limited dog leg severity (DLS) is the consequence of stiffer BHA offering high bending loads. Higher mass and inertia of stiffer BHA also put negative effects with respect to whirl stability and intensity of wall impact.

In low inclination wells, making the BHA stiffer will reduce lateral vibrations (*Internal*

Document - Dynamic Analysis, 2016).

2.4.6 Using Heavy DC in Upper BHA

Lateral acceleration and whirl can be avoided by making the upper BHA heavier so that it can lie on the low side of wellbore in inclined sections. The upper part of BHA can be made heavier by using only DCs along with other tools of exactly similar diameter. The use of HWDP and stabilizers above lower BHA should be avoided.

Having a continuous contact on the low side of wellbore enables the BHA to prevent lifting itself from lower side of well if the intensity of excitation is low to medium. BHA, kept at lower side, cannot be tilted. Therefore, it does not experience lateral acceleration. Due to sufficient cumulative contact forces, drill pipe section above and BHA below the upper BHA are not subjected to lateral acceleration.

Limitations of this method of reducing vibrations are high risk of differential sticking and more friction due to increased contact forces.

Low DLS wells with high inclination (or horizontal) utilize this technique. Due to sagging in large diameter holes, heavier upper BHA create high bending loads which make the situation even worse (*Internal Document - Dynamic Analysis*, 2016).

2.4.7 Large Amounts of Stabilizers

Using more stabilizers in lower BHA can prevent lateral acceleration and whirl.

Reducing the distance between stabilizers adds stiffness to the system and makes them harder to deflect. This geometry also constrains the lateral vibrations at the positions of stabilizers. If the BHA contains more stabilizers, the natural frequency will be higher. It will minimize the probability of resonance at string RPM (*Internal Document - Dynamic Analysis*, 2016).

Efficiency of hole cleaning is affected negatively by adding more stabilizers to the BHA. Large drag is usually experienced even in the rotary applications at the ledges in the hole. Use of two spiral near gauge stabilizers with the spacing less than 3m, enhances the chances of hang-up as the geometry does not allow them to move with respect to lateral deflection of hole. This will increase the friction drastically due to strong forces of lateral contact (*Internal Document - Dynamic Analysis*, 2016).

The benefits of large number of stabilizers are promising in low friction wells, less inclination wells if hole cleaning and the hole quality are maintained properly. In high inclination wells, stabilized section of BHA can result in large side forces which might influence the directional behaviour of BHA (Ishak, Daily, Miska, & Mitchell, 2012).

2.4.8 Uneven Stabilizer Distribution

All kinds of lateral vibrations can be addressed through this mitigation technique. The spacing between stabilizers along the BHA should neither be kept equal nor the multiples of each other. It should be made changed along the whole BHA. The impact of stabilizer clearance in pilot BHA and at the reamer defines the difference in lateral frequencies. (Meyer-heye, Reckmann, & Ostermeyer, 2011)

Using varying spacing between stabilizers allow to dampen the lateral movement by neighbouring sections, which was created in one section. The spiral nature of hole can also be minimized.

The steering behaviour of the BHA gets affected negatively by using uneven stabilizer distribution. Stabilizers may experience very high lateral contact forces if they are not placed at the multiple distance of the pitch which is equivalent to the distance between first wall contact and bit when a spiral hole is being drilled. It will eventually result in hang-up and blade damages.

This technique of vibration mitigation can be applied in any kind of well (*Internal Document - Dynamic Analysis*, 2016).

2.4.9 Stabilizer above Hole Opener/Under Reamer

Reduction in torsional and lateral vibrations can be observed by using stabilizer above under reamer/hole opener.

In this technique, a maximum blade outer diameter stabilizer is positioned at 30 to 60 ft above an under reamer/hole opener.

As the BHA above under reamer lies in the opened hole, drill collar section experience lateral movement which can be reduced by placing an stabilizer in that section. It will also minimize the bending loads and reduce the impact energy as well. Lateral stability of under reamer can also be improved when the lateral movements are reduced above the under reamer.

Steady state bending loads in particular hole trajectories can become higher due to the addition of stabilizer above the under reamer.

All types of under reamer applications can be benefitted with this technique (*Internal Document - Dynamic Analysis*, 2016).

2.4.10 Long Pilot BHA below a Hole Opener/Under reamer

Vibration directions and drilling dysfunctions is a technique that is advisable to reduce lateral movements.

The spacing between lowermost stabilizer and the under reamer should be more than a stand of stiff drill collar (not with heavy weight DP).

In the event of short reaming, this spacing between under reamer and lowermost stabilizer will ensure that the pilot BHA will not come out of the pilot hole, eliminating the risk of high dynamic pressure at RSS tools as well as high lateral forces at the bit. Pulling the pilot BHA out of pilot hole would result in increased lateral vibrations at the bottom. Another feature of this technique is to maximize the distance between the potential source of vibration (under reamer) and the LWD tools that might be damaged due to vibrations.

The drawback is to have long pilot hole which limits the casing/liner running operations.

Although long spacing is used to mitigate the vibrations, the operators usually are interested in keeping the pilot hole length as small as possible.

Therefore, the spacing selection depends on the field experience, the operators preference and vendors recommendations (Internal Document - Dynamic Analysis, 2016).

2.5 Bit Selection and Under Reamer Synchronization

Bit selection and under reamer synchronization is a typical challenge and in every under reaming operation, since the major source of vibrations is improper synchronization of the bit and under reamer. This section will cover important aspects and criticalities how to properly balance the cutting action of the bit and under reamer. Information is provided, how the under reamer BHA will behave in different formations, what is the impact of hole angle and hole size, and the importance of the BHA design. This information is necessary to be properly understood prior to executing the under reaming operation.

The interaction between bit/reamer and formation is potential source for a lateral vibration. Each bit and under reamer drill the rock with a specific pattern on the rock face. If the pattern is disturbed, the cutting elements begin to jump over the ridges made by the cutter. It creates differential loading because some cutters are free while others are involved in cutting the formation.

Different bit types have different types of vibration associated with it. Drilling with PDC bit mostly offer torsional and lateral vibrations while axial vibrations are experienced in drilling with roller cones. Different bit features are responsible to make it more or less aggressive for example number of cutters, back rake angle, cutter size, and depth of cut control. Aggressive bits have higher vibration levels than less aggressive one.

In the past, under reamers and bits were chosen independently and not as component of whole drilling system. The presence of under reamer was not taken into consideration while selecting the bit. The only factors involved in bit selection were rock strength and formation type to be drilled. This approach introduced drilling problems specially downhole vibration and failure in downhole tools (S. R. Radford et al., 2010).

Now, the aggressiveness of an under reamer has a direct influence on bit selection in order to avoid vibrations especially in interbedded formations. Homogenous formations do not generate too much vibration as the bit and under reamer are in the same formation. Therefore, at the formation interfaces, the bit and under reamer aggressiveness can introduce vibration especially lateral vibration and whirl. These challenges became the basis for the bit and under reamer manufacturers to come up with depth of cut control. Depth of cut control is related to the penetration of cutters in the formation which further defines the amount of energy spent to remove the rock. DOCC utilizes a bearing surface that would engage the bottom hole once a prescribed penetration is achieved (Thomson, Radford, Powers, Shale, & Jenkins, 2008). It requires very high WOB to penetrate further and therefore the aggressiveness of the bit is controlled. Under reamers are always made more aggressive than the bit so that they are not being out drilled by the bit.

2.5.1 Minimum Required Weight on Bit

The neutral point should be at least one stand above the main reamer during under reaming while drilling. Keeping the neutral point above the reamer defines the loading conditions within the BHA. The BHA and string have to be designed accordingly considering the planned WOB. The following scenarios have to be calculated (*Internal Document - Dynamic Analysis*, 2016):

- Minimum estimated WOB at section start
- Minimum estimated WOB at section TD

2.5.2 Hole Angle and Hole Size

Both vertical and horizontal wells have shown drill string vibrations depending upon the inclination and size of the hole. Severe axial and lateral vibrations are recorded in vertical and near-vertical wells which lead to BHA tools failure. Bit bounce may come up as severe problem in vertical wells as WOB has larger impact in these wells. In highly deviated well, the rotating assembly is pushed more towards the low side of wellbore therefore, its sideways motion is reduced due to gravity pull. For the angle greater than 30 degrees, it is observed that the lateral vibrations get reduced as BHA becomes more stabilized in the wellbore. This is further researched in chapter 4. Frictional torque between tools and wellbore is the source to trigger torsional vibration in deviated wells. Due to frictional torque, bit and under reamer experience less energy which increases stick slip on the string. Torsional vibrations also increase with tortuosity of the well. High DLS and sudden changes in the borehole inclination create more frictional torque.

The size of the borehole and outer diameter of BHA components defines the level and type of vibration in that well. Lateral vibration occur more in over-gauge holes as these holes have more clearance between BHA and borehole wall therefore sideways movement is generated. Contrary to over-gauge sections, under-gauge holes are more prone to torsional vibrations due to high frictional torque.

2.5.3 BHA Design

The design of BHA is one of the most important factors that can alter drill string vibrations. Efficient pre-planning of BHA components such as bit, reamers and stabilizers can reduce the vibration to a greater extent. This is due to the fact the function of those critical components and their interaction with surroundings are taken into consideration. Once the tools are designed and their positions in BHA are analysed with respect to vibrations, changes should not be made unless required because these alterations can lead to high vibration level and more non-productive time.

The type of formation, rig capabilities and limitation, measured depth of well, desired inclination and degree of stabilization should be thoroughly studied before designing BHA. Although the designs are usually based on offset well analysis and common assumptions, the variations of the new wells should be included in the design. From vibrations perspective, BHA design typically includes selecting bit/reamer size, choosing blade count and cutter size, profiling of bit and under reamer, under reamer stabilization, clearance and available contact area of BHA with formation during drilling (Thomson & Mathur, 2010). Weight distribution between reamer and bit should be analysed as a part of design process in order to avoid severe lateral vibrations when most of weight is put on the reamer instead of the bit (Thomson et al., 2008).

2.5.4 Formation Type

The type of formation and its hardness directly influence the nature and frequency of vibrations in drill string. High vibrations are often associated with more a harder rocks. But softer rocks can also generate high level of vibration. Drill string move sideways very easily in washouts and unconsolidated formations and create lateral vibrations.

Interbedded formations have varying hardness which act as source of vibration specially with reaming while drilling. High and low compressive strengths generate detrimental vibrations when bit and under reamers are in different types of hardness.

Figure 17 shows four scenarios where the bit and under reamers have drilled through formations of different unconfined compressive strengths. First and third scenario generate relatively less vibrations as both the bit and under reamer are in the same type of formation. In scenario 2, bit is in relatively softer formation than the under reamer. This scenario is the most critical one among all. There is a high probability that bit will out drill the reamer if improper reaming and drilling parameters are applied, BHA will experience lateral vibrations as there will be not enough WOB to support the bit. The rate of penetration will be governed by the aggressiveness of reamer therefore it is worst for the durability of reamer.

In scenario 4, the degree of stabilization is better as compared to scenario 2 but there are still lateral vibrations as most of the downhole weight is stacked on the bit. This scenario is worst from bit point of view (Centrala et al., 2011).

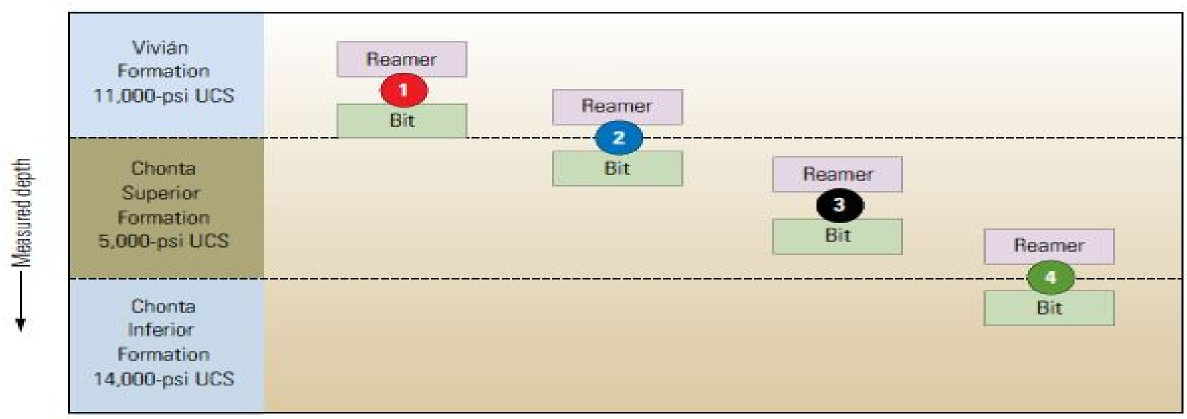


Figure 18: Four critical scenarios (Centrala et al., 2011)

The scenarios in Figure 18 can also be illustrated on WOB-torque diagram illustrated in Figure 19. The first diagram in Figure 19 is a typical behaviour of bit and under reamer in homogenous formation as shown in scenarios 1 and 3 in Figure 18. Most of downhole weight is stacked on the bit. When the bit is in hard and under reamer is in soft formation (Scenario 4, Figure 18), the downhole WOB will be shifted more to the bit and torque level on under reamer increases as shown in middle diagram in Figure 19. There will be an increase in lateral vibrations compared to the first case in where both bit and under reamer are in the same formation. The last diagram in Figure 19 depicts WOB-torque behaviour in worst scenario when bit is in soft formation whereas the reamer is in hard formation (Scenario 2, Figure 18). As compared to first diagram in Figure 19, the downhole weight is now shifted more on the reamer, making BHA under it in severe lateral vibration mode (S. R. Radford et al., 2010).

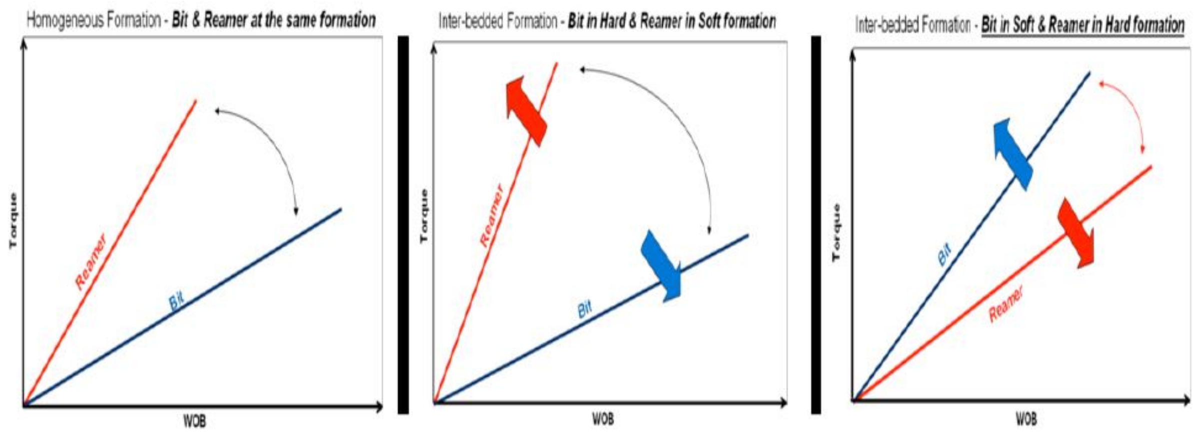


Figure 19: The schematic shows three different scenarios when drilling and under reaming and how these scenarios affect under reamer's and bit's apparent aggressiveness (S. R. Radford et al., 2010)

3 Methodology

Methodology will provide inside information of techniques and tools utilized in the research. In previous sections the theoretical background was provided and its proper understanding is necessary for the upcoming research. The methodology will cover important aspects of static and dynamic analysis utilized in the research, the tools used when designing the BHA and the software used for computing the simulations. It will also provide the theoretical background of static and dynamic analysis.

3.1 BHASYS Pro Introduction

This software utilizes a highly discretized linear finite element methodology in the analysis of complex BHAs in curved and three dimensional wellbores (Internal Document - Drilling Dynamics).

It provides answers related to the integrity and BHA optimization.

Static load distributions:

- Bending load curves for fatigue analysis and comparison with real time CoPilot bending moments measurements.
- Axial loading for neutral point definition
- Wall contact force analysis for wear issues.

The frequency domain of the software covers:

- Natural vibration frequencies and modes shapes.
- Forced vibration response to excitation mechanisms.

It is fully applicable to for multiple hole diameters and under reamer behaviour

Simulated various downhole conditions (i.e. reaming while drilling and reaming off bottom)

In addition it provides:

Answers of build rate predictions

- Models parameter variation
- Generates sag calculations

- Advanced buckling analysis

BHASys Pro has the capability for analysis of customizable parameters such as boundary conditions, mass imbalance, frictional effects, and drilling fluid properties.

Through this approach it is used to provide answers for the entire modelling procedure (Internal Document - Drilling Dynamics).

3.2 Drilling and Under Reaming Tools

In addition to the GaugePro Echo described in the Under Reamer Technology (Chapter 2), additional tools typically used in under reaming operation are presented. These tools will be used in research covered in this thesis excluding the CoPilot in the simulations.

3.2.1 AutoTrak Rotary Closed Loop System introduction

AutoTrak a rotary steerable tool, was initially introduced to the market in early 2000 where it has shown great steering capabilities, high ROP, and the capability of placing the well in desired location (Fiksdal, Rayton, & Djerfi, 2000). It has had great successes in the Troll field. Today's version of AutoTrak G3 is the most technologically advanced RCLS on the market. AutoTrak G3 gives new advantages in reaming and drilling operations, including geosteering and extended reach applications. The system itself is extremely precise with ability to steer to the target utilizing continuous closed loop control. Adjustments in the well path trajectories are effectively communicated from the Rig floor to the tools without any interruption in reaming and drilling process. Near to Bit Directional, Azimuthal Gamma Ray, Multiple Propagation Resistivity, real time pressures for ECD control and vibrations are included in the AutoTrak G3 (*Internal Document - AutoTrak G3 - Introduction and History*, 2014). The AutoTrak provides several benefits:

- Continuous steering while drilling and reaming
- Automated directional control
- Formation evaluation
- Two way communication
- Optional measurements

3.2.2 OnTrak Introduction

OnTrak is an integrated MWD and LWD tool which provides real-time directional surveys, gamma ray, multiple propagation resistivity, annular pressure for ECD control, drilling dynamics measurements. This specific design of the tool takes into account all challenges that might be faced during drilling and under reaming operations. The Baker Hughes propriety multiple propagation sensor design gives accurate measurements in the industry. It provides information for formation evaluation thus enabling accurate well placement in the pay zone, in addition it provides directional control. OnTrak is the master mind in the BHA and controls the tools, such as: AutoTrak G3 assembly, LithoTrak, SoundTrak, GyroTrak, CoPilot, and the Gauge Pro Echo (*Internal Document - OnTrak Technical Data Summary*, 2016).

3.2.3 CoPilot Introduction

The CoPilot is a unique MWD-based tool. Service of the tool will provide a high level of reaming and drilling process control to the driller by establishing a real-time feedback between the downhole reaming and drilling system and the crew on the rig floor. It provides values of downhole mechanical measurements, the CoPilot tool can diagnose the occurrence and the severity of reaming and drilling dynamics-related issues, this information is transmitted to the surface via mud pulse telemetry. Transmitted data is displayed on the rig floor screen along with other surface data thus enabling the driller to take immediate action in the case of reaming and drilling related issues occurring, presented in Figure 23.

The vibrations are inevitable, but they can be measured and controlled. The severity of vibrations will depend on various factors. Factors include bit, under reamer, BHA design, formations being drilled and parameters used such as WOB and RPM as explained in chapter 2.5.4. Before the introduction of the CoPilot service, a driller was forced to rely only on his experience and senses to adjust the parameters (Hood, Leidland, Haldorsen, & Heisig, 2001).

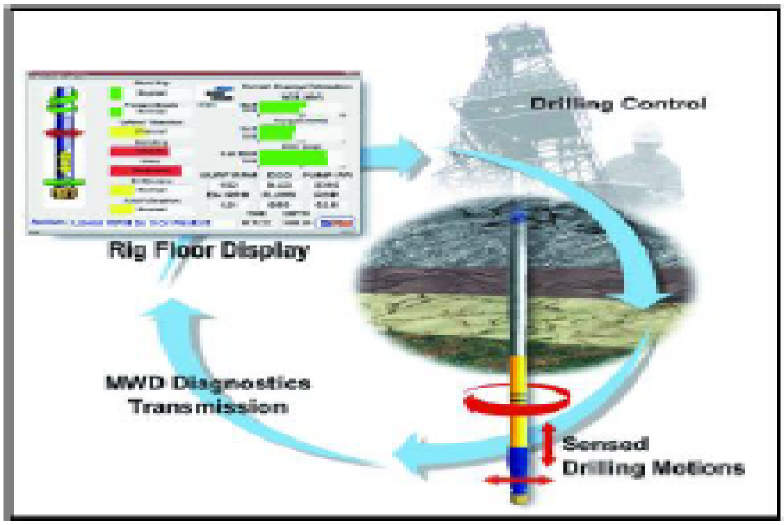


Figure 20: Real-time Vibration management (*Internal Document - CoPilot Introduction*, 2015)

The innovative design of the tool provides a downhole diagnostics system which contains various process sensors and high-speed data acquisition system. There are several benefits of using CoPilot service:

- Detecting the vibrations
- Determining the weight distribution problems
- Defining the hydraulics issues
- High ROP and efficient reaming and drilling can be achieved
- Improved borehole quality
- Data set for post job analysis

Some examples are presented how a downhole measurement tool captures drilling and reaming related issues such as backwards whirl and stick-slip (Hood et al., 2001).These examples of how CoPilot captures the energy of different vibration modes, are presented in Figures 21 and 22.

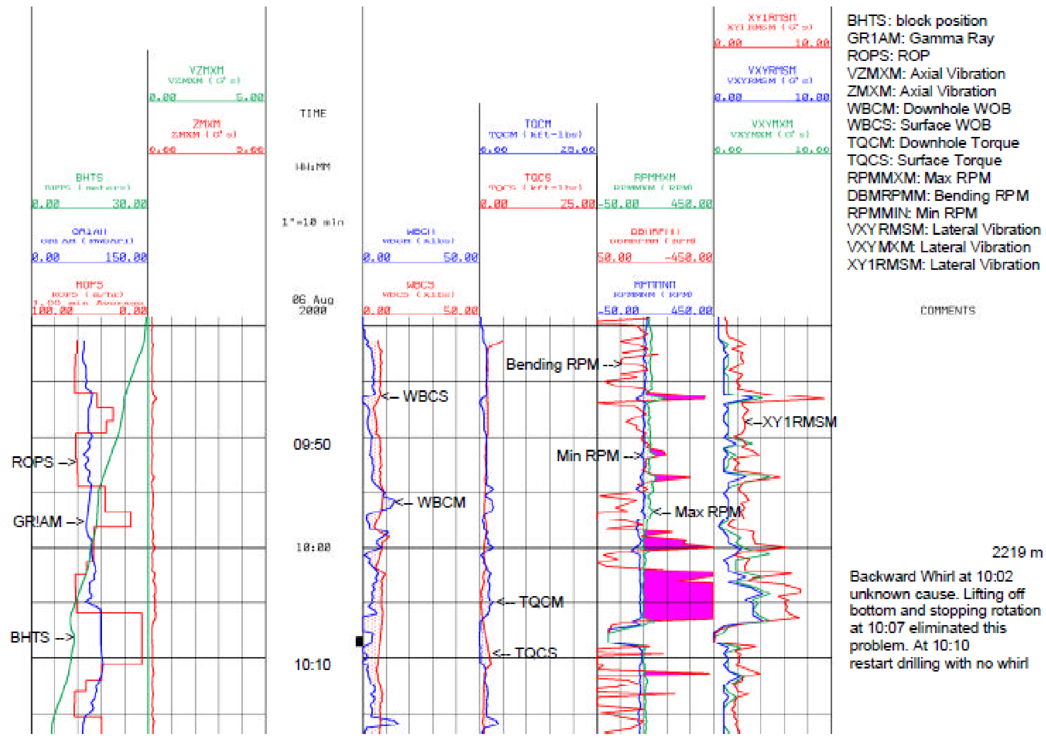


Figure 21: Backwards whirl (Hood et al., 2001)

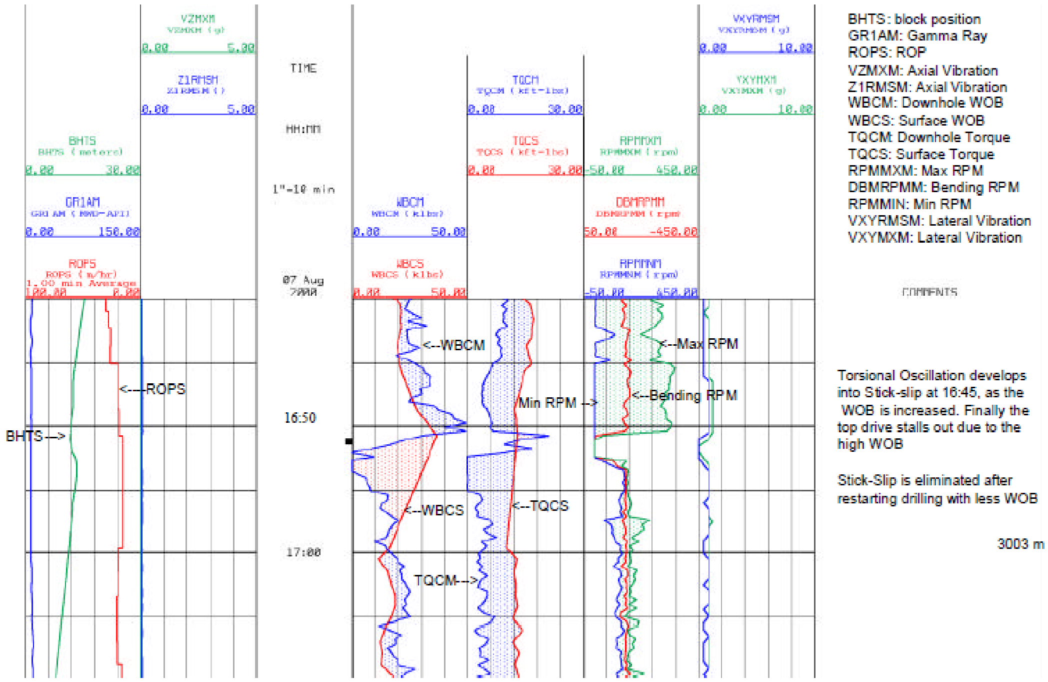


Figure 22: Stick-slip (Hood et al., 2001)

3.3 Statics Analysis Procedure

A system experiencing loads will be subjected to static analysis, the goal of static analysis is to find the equilibrium of the system under the external forces. In static analysis initially the axial forces in all tools which are incorporated in the drill string are required to be found. After determining the axial forces, objective is to find the total stiffness matrix (K), from the whole system. The stiffness matrix [K] is composed of the individual stiffness of all the constituent elements of the finite element model, provided by relation (Stokvik, 2010).

In static analysis the basic formula which is used in the background by the software is;

$$\boldsymbol{Ku} = \boldsymbol{R} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (16)$$

Where,

$\boldsymbol{K}$ - is the stiffness of the components of the drill string;

$\boldsymbol{u}$ - is the local displacement;

$\boldsymbol{R}$ - is the external force;

The load matrix R will only contain the external forces, i.e. hydrodynamic flow and offset at the upper end, as the effect of top tension and structural weight are included in the geometrical stiffness.

In the finite element analysis, the model is divided into many small parts and this calculation is performed at the nodes which are created as a result of the divisions. Stiffness of the elements of the finite element model is affected by the stiffness of the surrounding elements and also the change in stiffness which takes place due to the deformation caused by the external loading. $[\boldsymbol{K}]$ is then the stiffness matrix, $[\boldsymbol{u}]$ is the displacement matrix and $[\boldsymbol{R}]$ is the matrix of the external force components (Stokvik, 2010).

The static analysis is an iterative process, where load is applied into small steps and the stiffness matrix is updated with every iteration of the static analysis. In every iteration, displacement is calculated on the basis of the basic equation using the stiffness and the external loads.

Stiffness $(\boldsymbol{K})$ is calculated using the material properties and the dimensions of the individual components of the drill string. This data is inserted as an input in the software.

Also, external load which is used for calculating the displacement $(\boldsymbol{u})$ is given as an input to the software.

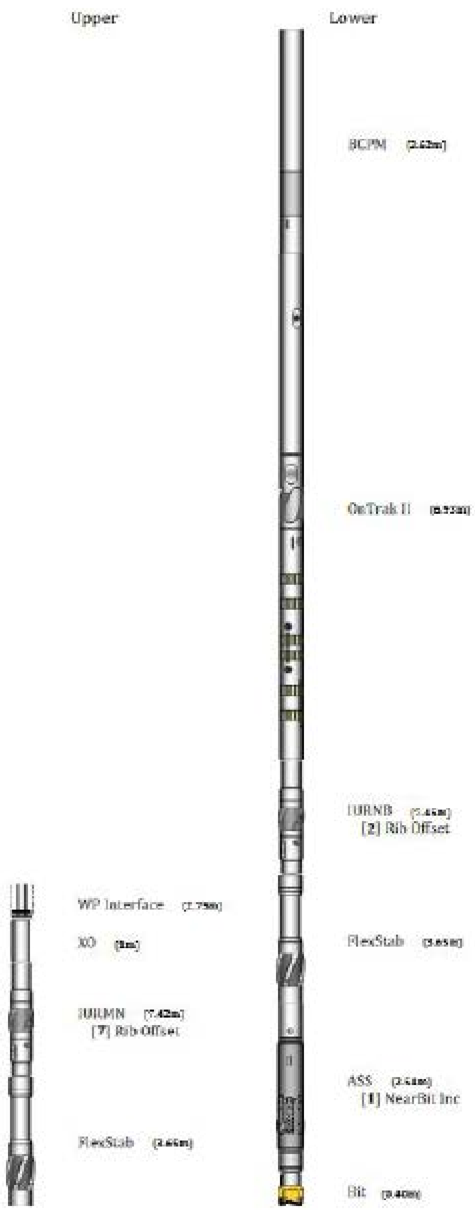
This displacement is used for updating the stiffness of the elements. In the next step the load is incremented and is again used to find out new displacements for the elements.

These iterations are continued until complete load is applied.

The output of the static analysis provides the final displacement and stresses generated in the components as a result of the external forces applied to the system. |If the system is constrained due to the presence of bodies around (Borehole in this case) the main component (Drill string in this case) contact forces are also expected to be generated. Static analysis also provides the contact forces and contact pressures generated at the points of contact. These are the stress hot spots and provide vital input information for performing the dynamic analysis.

The purpose of static analysis is to evaluate all the criteria before proceeding towards dynamic analysis. These criteria such as bending moments should be within the specified limit of the BHA which is comprised out of multiple tools.

In order to perform static analysis in BHASYS Pro, several parameters need to be placed as an input. These include BHA components, survey data, mud properties and constraints. In the BHA section, the data for each tool is either picked from the tool library or entered manually. BHA is designed in pre-planning stage and Figure 23 shows the schematic of the BHA, Figure 24 provides additional information.



1	1/4" PDC bit, 4" gage
2	9-5 ATK Steerable Stab 12-25
3	9-5 Modular Flex Stab 12-125
4	GaugePro_Echo_1225x1475
5	9-5 OnTrak V2 stab
6	9-5 BCPM
7	9-5 Modular Flex Stab 12-125
8	GaugePro_Echo_1225x1475
9	9-5 Mod X/O sub
10	Interface Sub
11	Float Sub
12	Float Sub
13	String Stab 12 1/8
14	Spiral - 8"
15	Spiral - 8"
16	Spiral - 8"
17	Hydra-Jars by Smith
18	Spiral - 8"
19	Spiral - 8"
20	Accelerator
21	Spiral - 8"
22	X/O Sub

Figure 23: BHA with two MOD-FLEX stabilizers

Figure 24: Additional information of the BHA

Figure 25 represents changed design of the BHA, where Flex stabilizers are replaced with modular stabilizers. This BHA is also used in the research.

1	12 1/4" PDC bit, 4" gage
2	9-5 ATK Steerable Stab 12-25
3	9-5 Modular Stabilizer 12-125
4	GaugePro_Echo_1225x1475
5	9-5 OnTrak V2 stab
6	9-5 BCPM
7	9-5 Modular Stabilizer 12-125
8	GaugePro_Echo_1225x1475
9	9-5 Mod X/O sub
10	Interface Sub
11	Float Sub
12	Float Sub
13	String Stab 12 1/8
14	Spiral - 8"
15	Spiral - 8"
16	Spiral - 8"
17	Hydra-Jars by Smith
18	Spiral - 8"
19	Spiral - 8"
20	Accelerator
21	Spiral - 8"
22	X/O Sub

Figure 25: Information of the secondary BHA, where modular flex stabilizers are replaced with modular stabilizers

Both static and dynamic analyses are repeated for another BHA in which flex stabilizers are replaced with modular stabilizers. Survey data consist of inclination and azimuth thus BHASYS Pro generates the dog leg severity. Density of drilling fluid and its viscosity are placed in the section of mud. Constraints are the limits that the static analysis will be subjected to during its computation. It includes WOB and bit torque.

The bending moments for different sections such as build, drop and turn are calculated for both BHAs and maximum build up rates are determined at which the load is near to bending moment limits of each tool joints. If the build-up rate exceeds the maximum calculated value, bending moment might go above the tool limit at the connection which can be detrimental and lead to flooding of the tool or, in more severe case, fishing of the tools. This build-up rate serves as foundation for the dynamic analysis.

3.4 BHA Reactive Torque Calculation

The reactive torque calculation is a part of static analysis when analysing the BHA behaviour in off-bottom operations.

Reactive torque is a left-hand torque that occurs after a BHA becomes stuck. Theoretically it can occur at any point in the BHA. Regarding this event reactive torque calculations become important when off-bottom rotation, operation is performed where the BHA contains under reamers or hole openers.

Historically speaking, several connections backed off after unexpected sticking while rotating off bottom. The most critical connection in the BHA that can back off is below the under reamer. When the under reamer stalls, the BHA below it will continues to rotate, generating a reactive torque, simply because of mass moment of inertia below the stuck point. The reactive torque has capacity to unscrew a connection if the operational RPM is high enough. Backing off connection can be avoided if operators control the RPM in off bottom operations. The reactive torque equation is linearly dependent on operational RPM (e.g., tripling the RPM will triple the reactive torque). In the event where the reactive torque values exceed the make-up torque of the connection, the BHA can get flooded, and/or twist off can occur in which fishing operation will be necessary to retrieve the tools (*Internal Document - BHA Reactive Torque Calculation*, 2015).

3.4.1 Procedure

The BHA needs to be adjusted for the following calculation by removing the under reamer and all tools above it. Next step is to place the adapter sub at position of removed under reamer in the BHA, presented in Figure 26;

Row	Name	Length m	Distance m
1	12 1/4" PDC bit, 4" gage (standa	0,42	0,42
2	9-5 ATK Steerable Stab 12-25	2,54	2,96
3	9-5 Modular Flex Stab 12-125	3,63	6,59
4	GaugePro_Echo_1225x1475	7,46	14,04
5	9-5 OnTrak V2 stab	7,01	21,06
6	9-5 BCPM	3,69	24,75
7	9-5 Modular Flex Stab 12-125	3,63	28,38
8	9-5 Adapter Sub	0,59	28,97

Figure 26: Example of the BHA components

Make-up torque used in the calculation, MUT=73000Nm, this value is calculated using the safety factor of 0.3;

Make up torque used in the calculation is at the connection between the Adapter Sub and modular flex stabilizer;

The mass moment of inertia is obtained from the BHASYS Pro;

Calculate torsional deflection in [rad] (φ) under static results;

$Torque[Nm]$- is applied in constraints section in the BHASYS Pro. For all calculations, value of 5 [kNm] is applied;

Calculate the rotational stiffness as in equation

$$C = \frac{Torque[Nm]}{\varphi[rad]} \quad (17)$$

Calculate the angular frequency Ω. Operational off bottom RPM needs to be converted into angular frequency by applying equation 28:

$$\Omega = 2 * \pi * rpm * \frac{1}{60} \quad (18)$$

The value of $RPM_{inertia} = 120rpm$ will be used for the following calculations;

$$T(\Omega) = \Omega\sqrt{J * C} \quad (19)$$

$T(\Omega)$: Reactive torque [Nm];

Ω- Angular frequency [rad/s];

J- Cumulative mass moment of inertia below the stuck point [kgm^2];

C- Rotational stiffness [Nm/rad];

Calculate the $T(\Omega) = T_{inertia}$;

Calculate the maximum allowed off bottom operational $RPM_{allowed}$ as presented in equation 30.

$$RPM_{allowed} = \frac{RPM_{inertia} * T_{allowed}}{T_{inertia}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots \quad (20)$$

Where:

Rotation speed used for moment of inertia calculation - $RPM_{inertia}$;

Left hand torque from cumulative mass moment of inertia - $T_{inertia}$;

Allowable left hand torque on Adapter Sub - $T_{allowed}$;

Allowable rotation speed - $RPM_{allowed}$.

(*Internal Document - BHA Reactive Torque Calculation*, 2015)

3.5 Dynamic Analysis Procedure

Static analysis can ensure that the designed drill string will withstand the loading conditions where the loads are not varying over time also known as steady-state conditions, but this may not be sufficient. The objective of the dynamic analysis is to gather information how a structure will respond and behave when it becomes exposed to loads varying over time. As mentioned previously, static loads are constant. In the dynamic analysis, the non-linear equation is represented as

$$[\boldsymbol{M}]\ddot{\boldsymbol{u}} + [\boldsymbol{F_f}]\dot{\boldsymbol{u}} + \boldsymbol{F_G}\,\boldsymbol{u} + \boldsymbol{F_W}\,\boldsymbol{u} = \boldsymbol{R} + \boldsymbol{F_{Exc}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots \quad (21)$$

Where: $\boldsymbol{M}$ is the mass Matrix;

$\boldsymbol{F_f}$ - Is the damping Matrix;

$\boldsymbol{F_W}$ - Is the external wall contact force vector;

$\boldsymbol{F_G}$ - Vector represents the nonlinear elastic forces;

$\boldsymbol{R}$ - Vector represents static forces such as weight, buoyancy, WOB;

$\boldsymbol{F_{Exc}}$ - Represent the excitation force vectors such as mass imbalance;

$\ddot{\boldsymbol{u}}$ - Variable is the displacement vector-acceleration;

$\dot{\boldsymbol{u}}$ – Variable is the displacement vector-velocity;

Mass matrix $\boldsymbol{M}$ includes both drill string's own mass and the mass added from the surrounding water or mud in the borehole. Added mass is typically inspected by checking the changes in inertia forces where the drill string is in accelerated fluid. The biggest contribution in the damping in the drill string is hydrodynamic damping. Hydrodynamic damping is generated by the relative velocity and surrounding mud. The damping included in $\boldsymbol{F_f}$ matrix is the drill string damping, which can be in some cases proportional to mass matrix and the stiffness matrix. Stiffness matrix is calculated from the static analysis and in the dynamic analysis it is represented as $\boldsymbol{F_G}\,\boldsymbol{u}$ $+\boldsymbol{F_W}\,\boldsymbol{u}$. There are several ways to how the dynamic equation can be solved, through time domain and frequency domain (Schmalhorst & Neubert, 2003).

Advanced frequency domain is used in the BHASYS Pro. ”The drill string is modelled with geometrically nonlinear beam elements”. Deformations of the drill string are measured by three nodal displacements and three rotations”(Schmalhorst & Neubert, 2003).

Lateral displacements: $\boldsymbol{u_1}$, $\boldsymbol{u_2}$

Lateral rotations: $\boldsymbol{\theta_1}$, $\boldsymbol{\theta_2}$

Axial displacement: $\boldsymbol{u_3}$

Axial rotation: $\boldsymbol{\theta_3}$

Finite element nodes are bounded though the approach of the penalty function in the wellbore.

In the scenario where the component of the drill string gets into contact with the borehole wall, a reactive constraining force will act on that element. This type of model allows pre-deformations to be observed in the drill string, when the drill string is in the inclined 3D wellbores. Formulation of this sort combined with geometrical nonlinearity provides an analysis of coupled vibrations such as axial, torsional and lateral in the frequency domain, in addition it also provides the calculation of loads that occur due to buckling (Schmalhorst & Neubert, 2003).

Dynamic analysis is generated in three steps. The first step is to calculate the statics results through the Newton's scheme:

$$\boldsymbol{F_w}(\boldsymbol{u}) + \boldsymbol{F_g}(\boldsymbol{u}) = \boldsymbol{R} \quad (22)$$

$$\left(\frac{\partial \boldsymbol{F_w}}{\partial \boldsymbol{u}}(\boldsymbol{u_n}) + \frac{\partial \boldsymbol{F_g}}{\partial \boldsymbol{u}}(\boldsymbol{u_n})\right)\Delta\boldsymbol{u_{n+1}} = \boldsymbol{F_w}(\boldsymbol{u_n}) + \boldsymbol{R} \quad (23)$$

$$\boldsymbol{u_{n+1}} = \boldsymbol{u_n} + \Delta\boldsymbol{u_{n+1}} \quad (24)$$

Natural vibration analysis presented in equation 21 is linearized with/about steady state displacements $\boldsymbol{u}$ obtained from the equations 22, 23, 24. Small deviations have been assumed $\boldsymbol{\xi}$ from the steady state solutions, thus natural frequencies and mode shapes are calculated from:

$$\boldsymbol{M}\ddot{\boldsymbol{\xi}} + \boldsymbol{K}\boldsymbol{\xi} = \boldsymbol{0} \quad (25)$$

$$\boldsymbol{K} = \frac{\partial \boldsymbol{F_g}}{\partial \boldsymbol{u}}(\boldsymbol{u_{static}}) + \frac{\partial \boldsymbol{F_w}}{\partial \boldsymbol{u}}(\boldsymbol{u_{static}}) \quad (26)$$

$$\boldsymbol{\xi} = \hat{\boldsymbol{\xi}}\boldsymbol{e^{j\omega t}} \quad (27)$$

$$\left(\boldsymbol{K} - \boldsymbol{\omega^2 M}\right)\hat{\boldsymbol{\xi}} = \boldsymbol{0} \quad (28)$$

Where:

$\boldsymbol{\xi}$ - is the amplitude of the small deviations;

$\boldsymbol{\omega}$ - is the frequency at which the Eigen frequency occurs;

The third and the final step is comprised of the analysis of forced vibrations. For forced vibration analysis two excitation sources are included in the model: axial bit excitation and mass imbalance excitation. A set of differential equations describe the issue of forced vibrations with harmonic excitations with frequency(Ω) (Schmalhorst & Neubert, 2003).

$$\boldsymbol{M}\Delta\ddot{\boldsymbol{u}} + \boldsymbol{B}\Delta\dot{\boldsymbol{u}} + \boldsymbol{K}\Delta\boldsymbol{u} = \boldsymbol{P_{0C}}\,\mathbf{cos}(\Omega\boldsymbol{t}) + \Delta\boldsymbol{U_s}\,\mathbf{sin}(\Omega\boldsymbol{t}) \quad (29)$$

The equation 29 is solved by equation 30.

$$\Delta\boldsymbol{u}(\boldsymbol{t}) = \Delta\boldsymbol{U_c}\,\mathbf{cos}(\Omega\boldsymbol{t}) + \Delta\boldsymbol{U_s}\,\mathbf{sin}(\Omega\boldsymbol{t}) \quad (30)$$

From the displacements - $\Delta\boldsymbol{u}(\boldsymbol{t})$, dynamic axial loads, dynamic torsional and dynamic bending moments can be found.

The program is fully applicable for multiple hole diameters and under reamer behaviour, thus it will be utilized in the following research.

Steady- state bending moment analysis using the BHASYS Pro is performed to identify the severity of the risk of damaging or destroying downhole tools. This can be performed either by pure bending fatigue (for tools that are rotating) or by plastically deforming or sudden cracking (for sliding and rotating tools) due to hole curvature, sagging from gravity, eccentricities, and – with limitations – from buckling (Schmalhorst & Neubert, 2003).

Further, the analysis can give clues as to which tools in the BHA are most likely to fail in this way first, and which borehole sections and drilling parameters may be most detrimental to the life of the tools.

After preforming static analysis several adjustments need to be made solely dedicated to modelling dynamic results. Various changes are made to the model and thus making it no longer suitable for other calculations (Schulte & John, 2016).

- The string length is double the length of the BHA;
- For reaming while drilling case, the bit Is placed on bottom;
- For back reaming and rat hole reaming, the bit is placed off bottom;
- WOB and torque should reflect average drilling and reaming values;
- Planned well path needs to be shown in the survey;
- The survey is adjusted to pilot hole size and reamer blades size diameter;
- No eccentricity is assumed at the steering ribs;
- Mass imbalance is defined trough adjustment made on the under reamer blade component to 1 inch eccentric, in order to simulate the excitation from the under reamer;

After making adjustments static results are recalculated in order to activate the wall contact option;

Additionally identification of the frequency range is necessary to be analysed in the forced vibration calculation, this is done by converting the minimum and maximum rotations speeds into frequencies by dividing the RPM with 60 ($\Omega = RPM/60$). Multiplying the rotational frequency with number of under reamer blades, in order to consider each under reamer blade specifically.

Example is presented:

Number of under reamer blades: 3

Minimum operational $RPM = 20\ rpm$

$$20\frac{rpm}{60} = 0.333Hz \quad (31)$$

$$0.333Hz * 3 = 1Hz \quad (32)$$

Maximum operational $RPM = 180\ rpm$;

$$180\frac{rpm}{60} = 3\ Hz \quad (33)$$

$$3\ Hz * 3 = 9Hz \quad (34)$$

In this example the frequency of the forced vibration analysis is from $1Hz$ to $9Hz$. These values will represent the frequency range for the calculation which is performed in five hundred steps. In addition the source of excitation is stated to be the mass imbalance. After completing the previous steps, wall contact point at the reamer blades is released, thus process of making the under reamer as the source of excitation is completed. The last step is to perform the forced vibration calculation (Schulte & John, 2016).

4 Impact of well path and mud density on drilling dynamics

When planning an under reaming operation it is critical to understand the impacts of well path trajectories and mud densities, as they will provide us the relevant information of their impact in under reaming dynamics, thus providing a better understanding of downhole vibrations.

4.1 Impact of well path on Reaming While Drilling Dynamics and Discussion

In this section the results are presented after performing multiple dynamic simulations in the BHASYS Pro which are combined and processed by the Mat Lab for which a specific code is written. For each dog leg severity a corresponding well path trajectory is shown. This section consists of 8 cases generated of 87 BHASYS Pro simulations:$1{,}5°/30m$; $2°/30m$; $2{,}5°/30m$; $3°/30m$; $3{,}5°/30m$; $4°/30m$; $4{,}5°/30m$;$5°/30m$. It is noted that operational window is between 1Hz and 9Hz corresponding to 20rpm and 180rpm..The well path represents a build case from a full vertical well to a full horizontal. The following results are presented:

- Forced Vibration Lateral Deflection[m]
- Forced Vibration Frequency[Hz]
- Forced Vibration Bending Moments[Nm]

The BHA used in calculation is presented in Figure 27 as this BHA design is used in all eight cases.

1	1/4" PDC bit, 4" gage
2	9-5 ATK Steerable Stab 12-25
3	9-5 Modular Flex Stab 12-125
4	GaugePro_Echo_1225x1475
5	9-5 OnTrak V2 stab
6	9-5 BCPM
7	9-5 Modular Flex Stab 12-125
8	GaugePro_Echo_1225x1475
9	9-5 Mod X/O sub
10	Interface Sub
11	Float Sub
12	Float Sub
13	String Stab 12 1/8
14	Spiral - 8"
15	Spiral - 8"
16	Spiral - 8"
17	Hydra-Jars by Smith
18	Spiral - 8"
19	Spiral - 8"
20	Accelerator
21	Spiral - 8"
22	X/O Sub

Figure 27: BHA used in all eight cases

Case one: $1,5°/30m$;

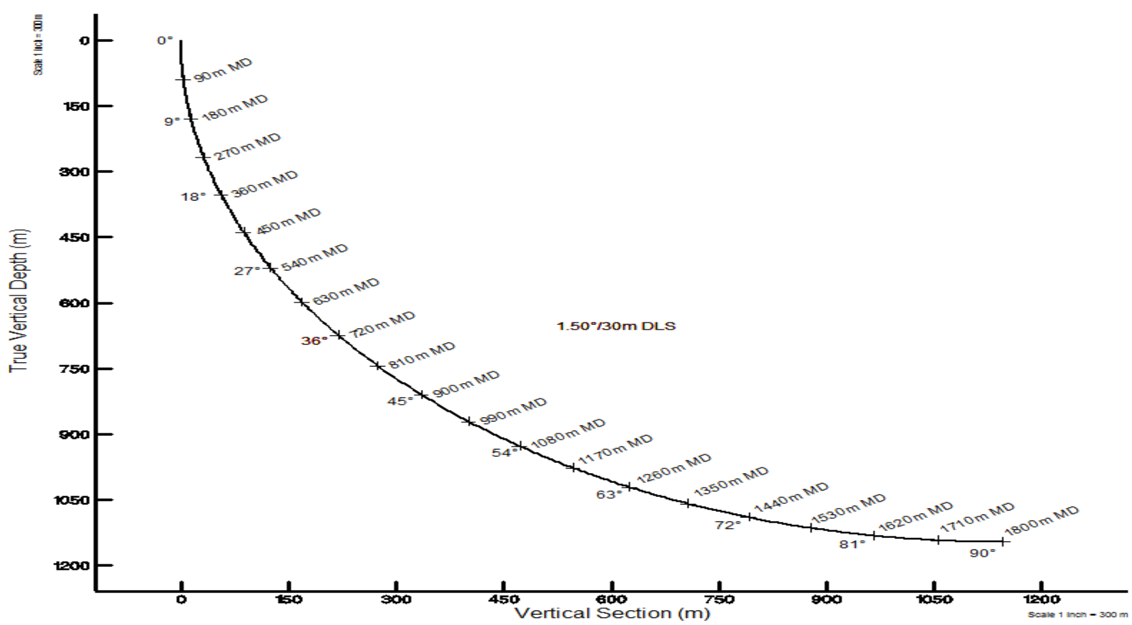


Figure 28: Well path

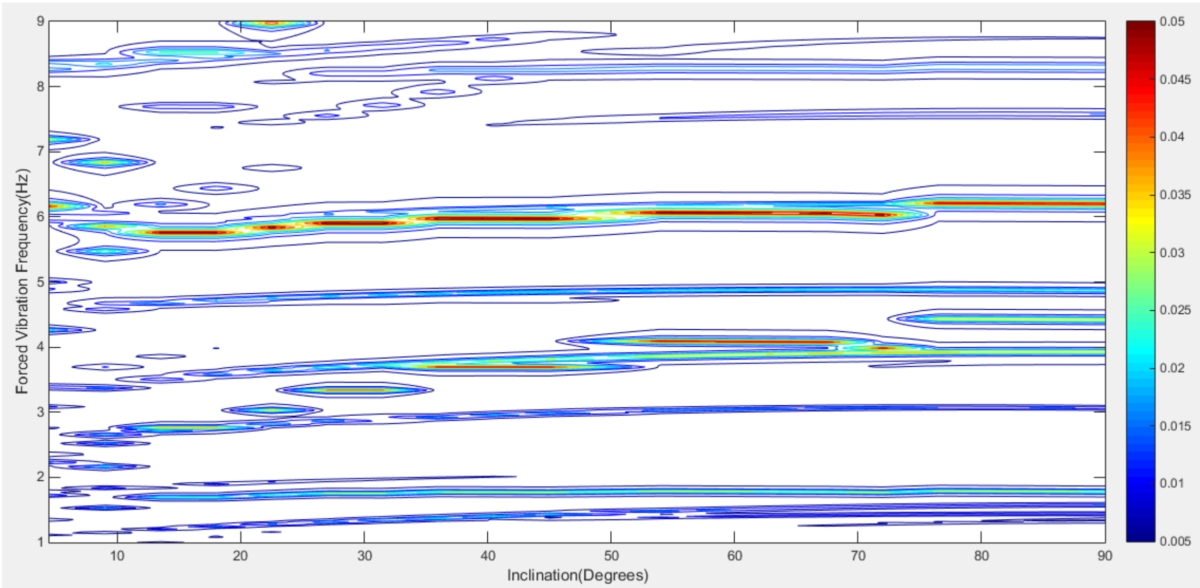


Figure 29: Calculated Forced Vibration Later Deflection

In Figure 29 the under reaming dynamics are the worst compared to all other cases. Forced Vibration Frequencies are covering most of the operational window.

Critical speeds are noted from 30-90 degrees of Inclination:

- 1,66Hz (33rpm). It possesses low magnate of lateral deflection due to insufficient energy being provided from the top drive;
- 2,7Hz (54rpm). Low values of lateral deflection throughout the whole section(30-90degrees of inclination)
- 3,33hz (66rpm). This critical frequency is shifting upwards with an increase of inclination, it contains sufficient magnitude to potentially damage the BHA;
- 4,6Hz (92rpm). Low magnitude od of lateral deflection is observed;
- Critical speed at approximately 6Hz (120rpm) is the most critical of all forced vibration frequencies. If this frequency is matched with operational RPM the resonance will occur, which will generate a severe damage to the BHA components.
- At 7,3Hz (146rpm) the forced vibration frequency is observed with low magnitude of lateral deflection.

This information provides the actual, operational RPM windows, which can be used during under reaming operation. Safety factor is taken into calculation (±5RPM) equivalent to (±0,25Hz).

Operational RPM windows from 30-90degrees of inclination:

- Between 33rpm and 54rpm. Resulting operational window is from 38rpm-49rpm.
- Between 54rpm and 66rpm. Resulting operational window is from 59-61rpm.
- Between 66rpm and 92rpm. Resulting operational window is from 71rpm and 87rpm with a decrease in sections from 48 to 70 degrees of inclination and 73 to 90 degrees.
- Between 92rpm and 120rpm. Resulting operational window is from 97rpm to 115rpm, with a slight increase in the operational window due to upwards shifting of corresponding frequency at 6Hz.

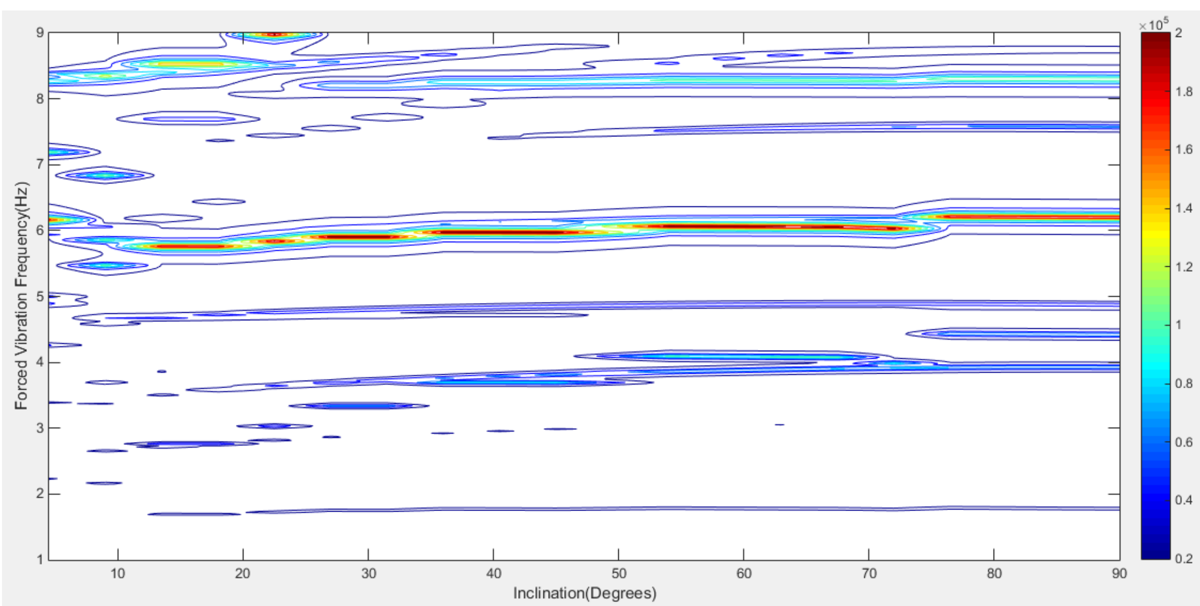


Figure 30: Calculated Forced Vibration Bending Moments

Figure 30 represents the results of calculated dynamic bending moments. The dynamic bending moments will typically follow the forced vibration frequencies that contain the highest magnitude of lateral deflection. This is observed in frequency at 6Hz which contains the highest values of bending moments throughout this frequency. These bending moments are sufficient to generate a catastrophic damage to the BHA. In other critical frequencies also observed in Figure 29 the dynamic bending moments are low.

Case two: $2°/30m$

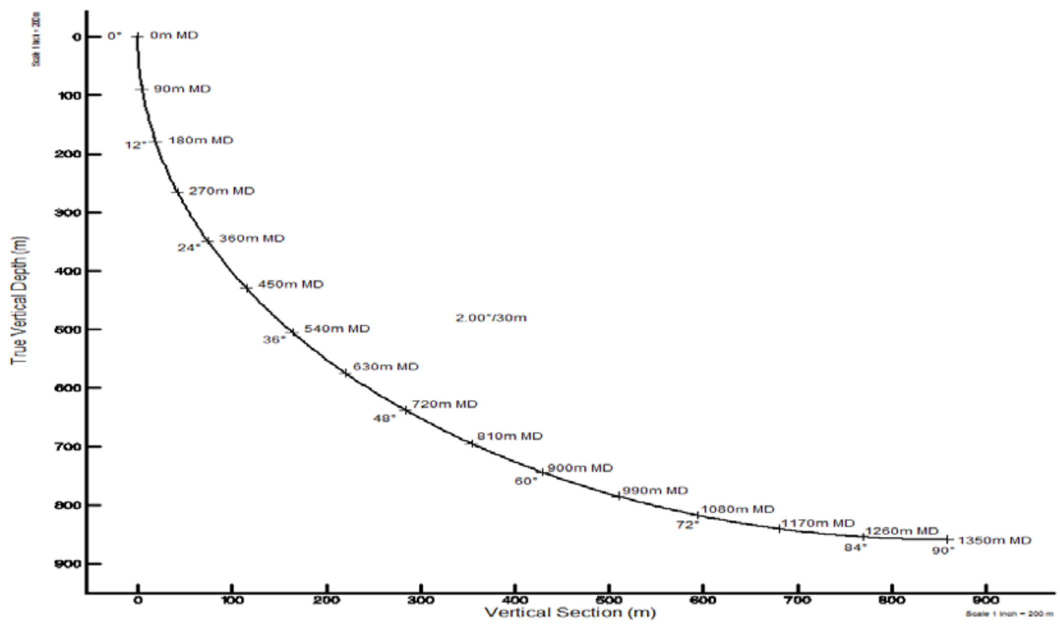


Figure 31: Well path

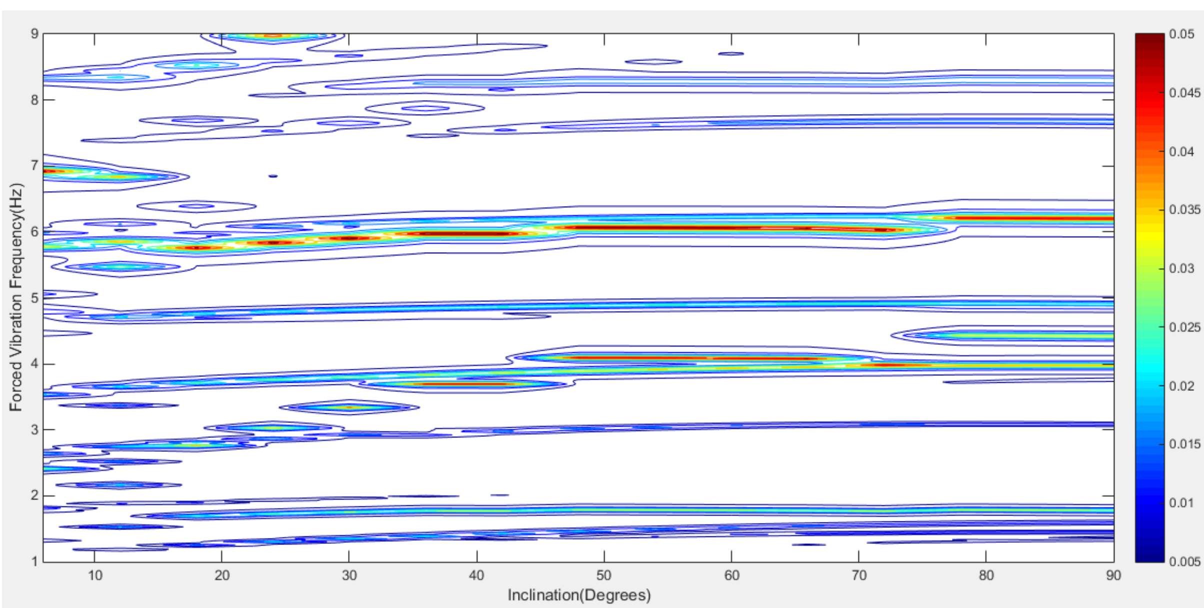


Figure 32: Calculated Forced Vibration Later Deflection

Figure 32 can be compared to Figure 29 in the first case. With an increase of dog leg severity from $1{,}5°/30m$ to $2°/30m$ a minor improvements in resulting dynamics are observed. The operational window has slightly increased, additionally the most critical frequency at 6Hz is shifted upwards.

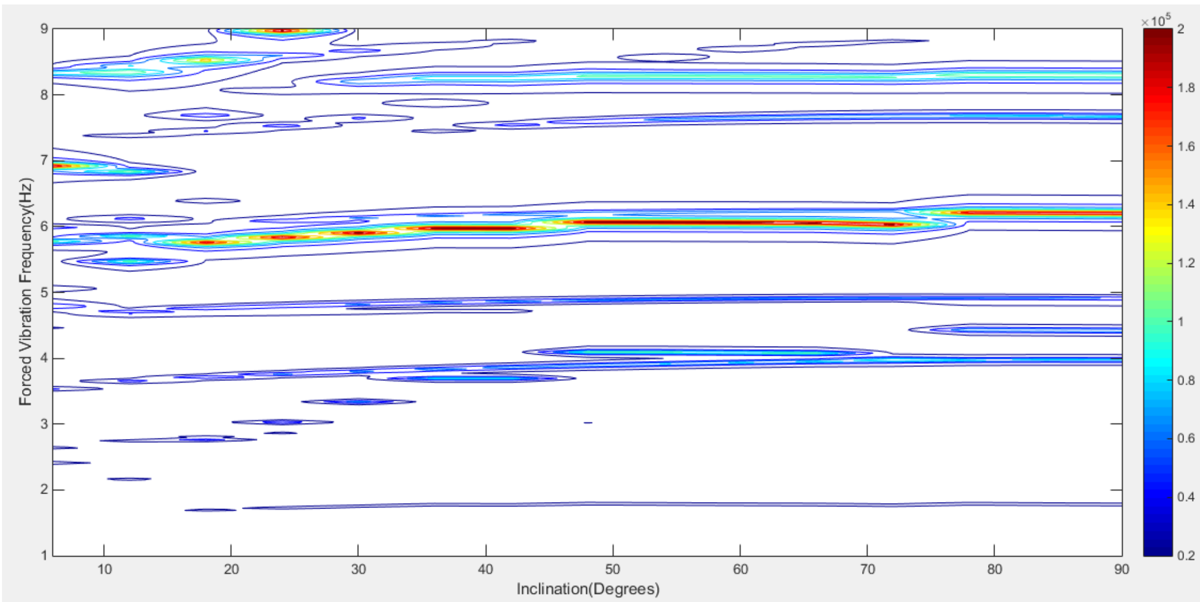


Figure 33: Calculated Forced Vibration Bending Moments

Results in Figure 33 are similar to the results in Figure 30, where the highest values of dynamic moments are observed at critical speed at 6Hz. As the operational window increases the risk of BHA experiencing high values of dynamic bending moments are decreased.

In the following cases of: $2{,}5°/30m$; $3°/30m$; $3{,}5°/30m$; $4°/30m$; $4{,}5°/30m$; $5°/30m$ same pattern can be observed for calculated forced vibration lateral deflections and for forced vibration bending moments.

In the last case, where dog leg of $5°/30m$ is computed for the simulations, is explained in detail.

Case three: $2,5°/30m$

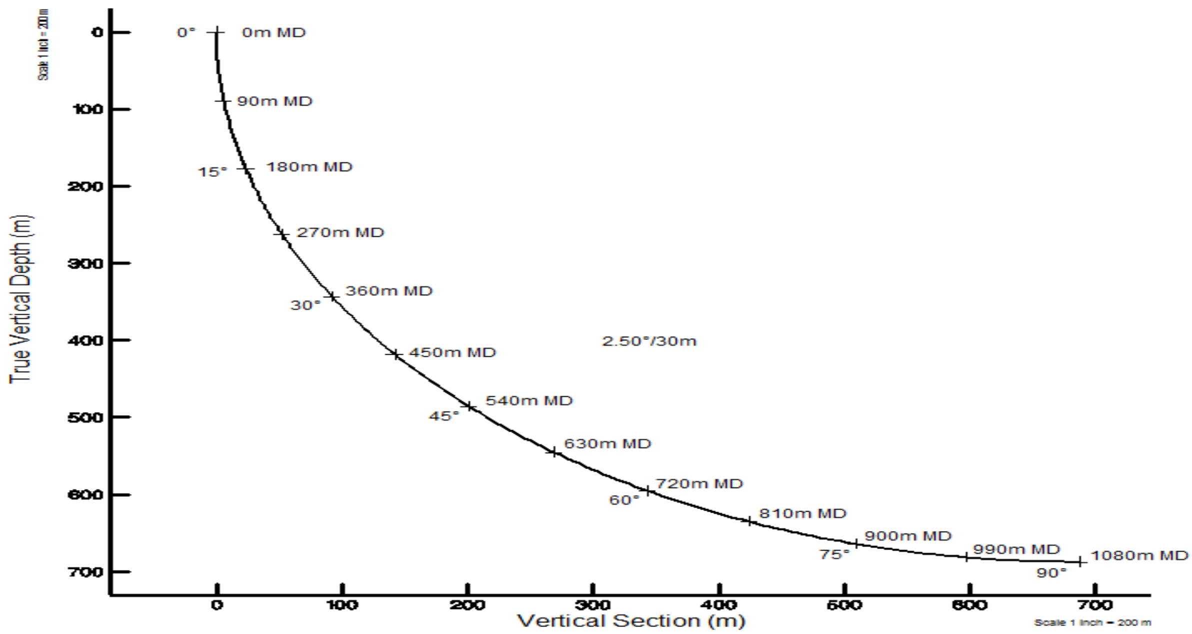


Figure 34: Well path

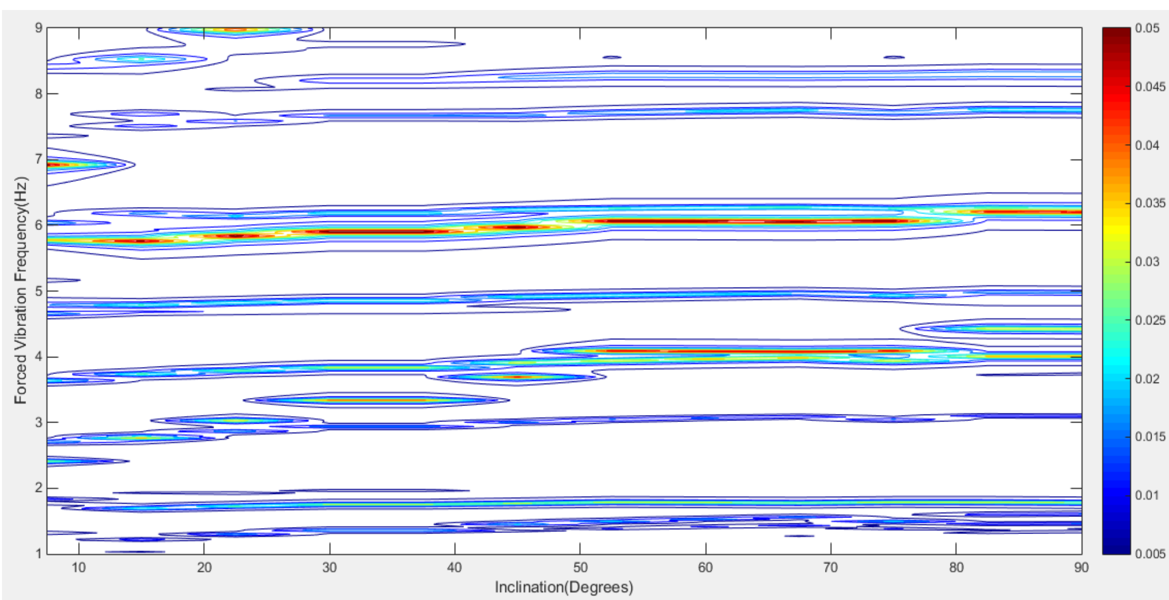


Figure 35: Calculated Forced Vibration Later Deflection

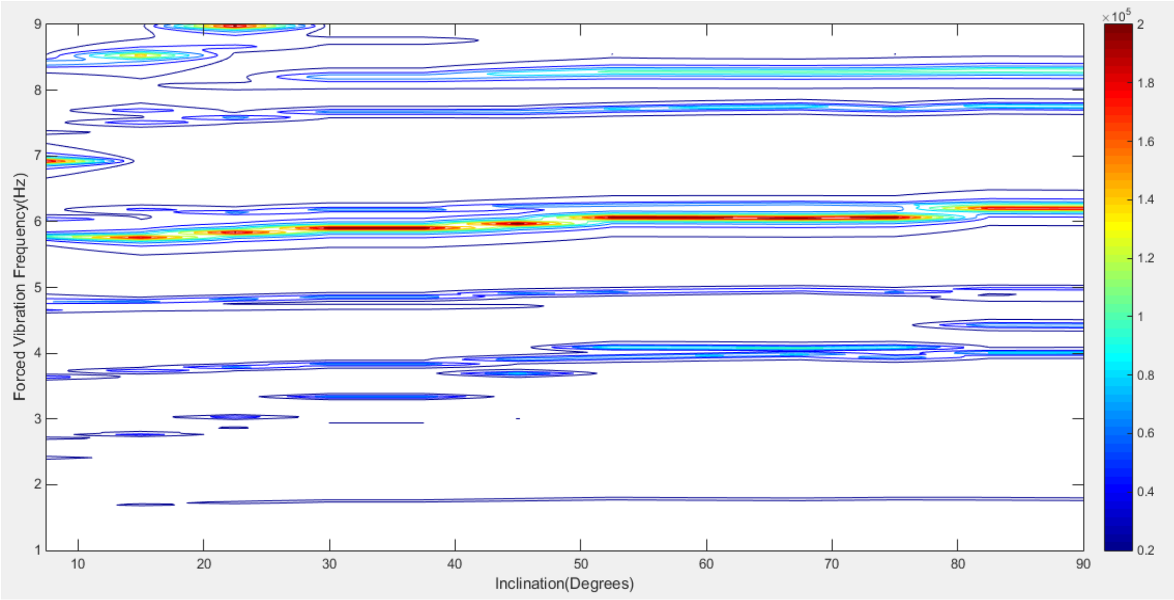


Figure 36: Calculated Forced Vibration Bending Moments

Case four: $3°/30m$.

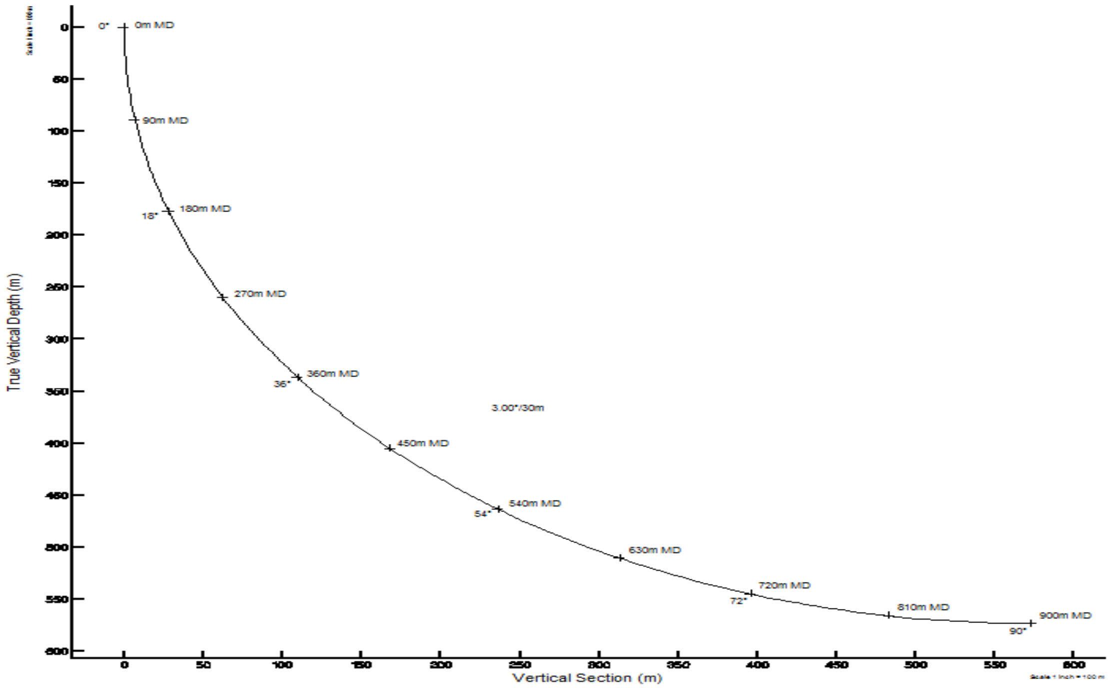


Figure 37: Well path

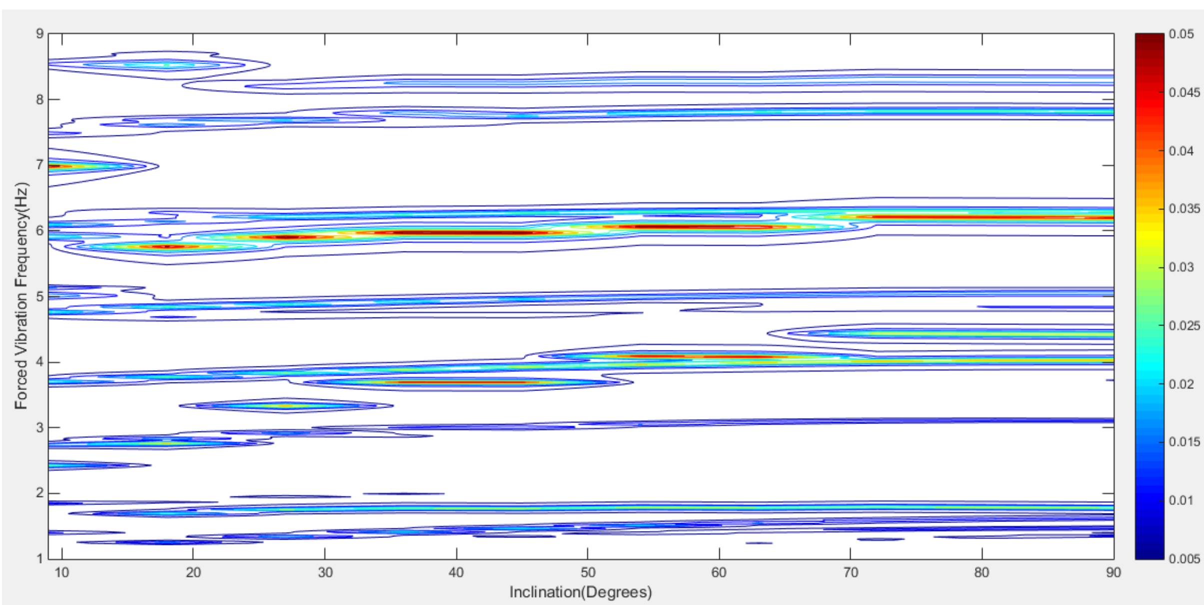


Figure 38: Calculated Forced Vibration Later Deflection

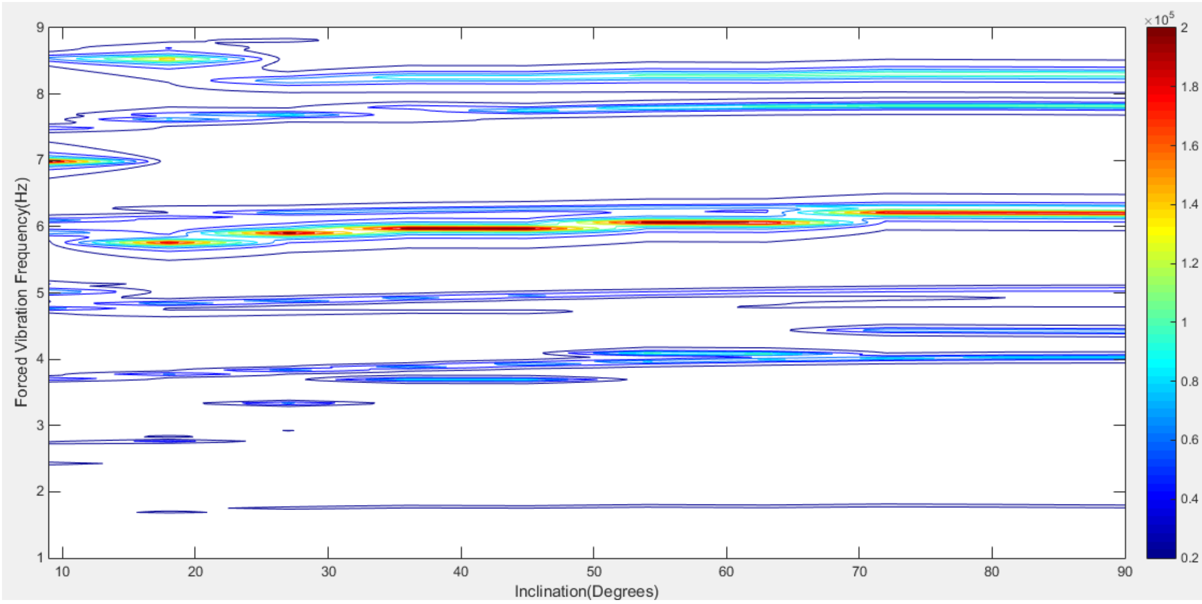


Figure 39: Calculated Forced Vibration Bending Moments

Case five: $3,5°/30m$

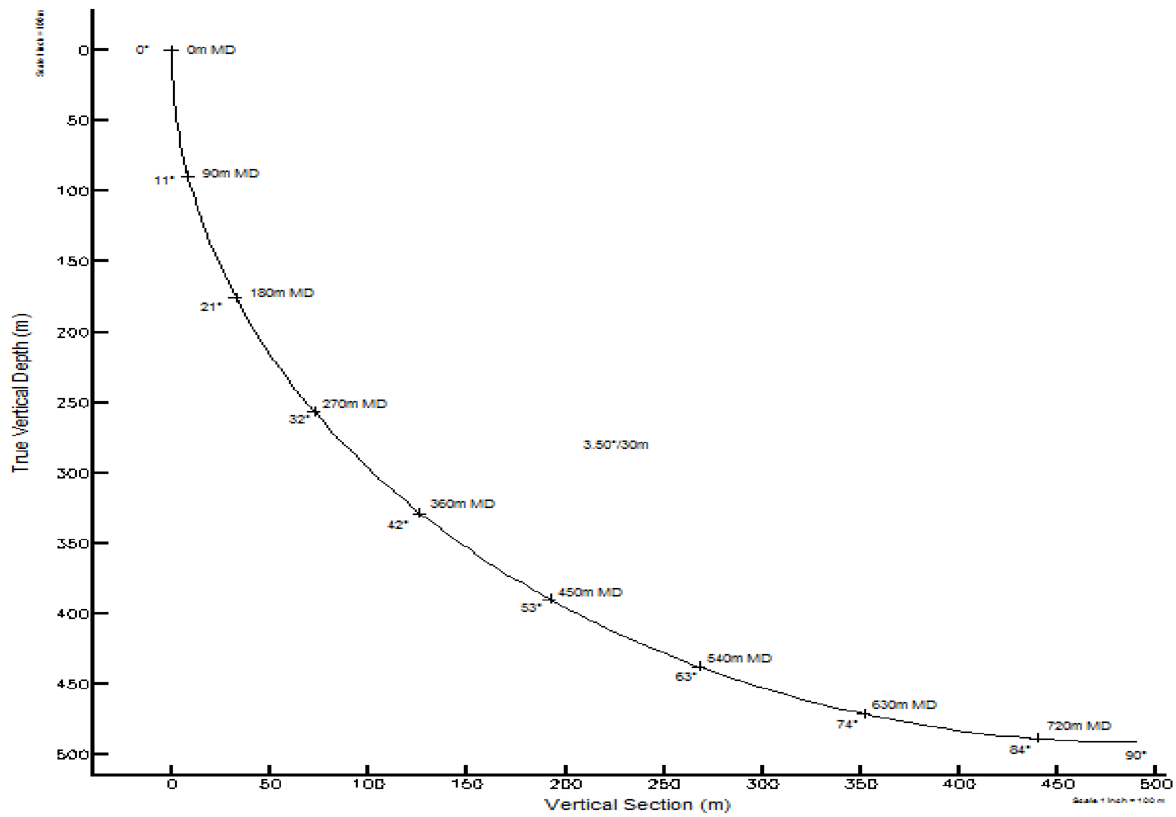


Figure 40: Well path

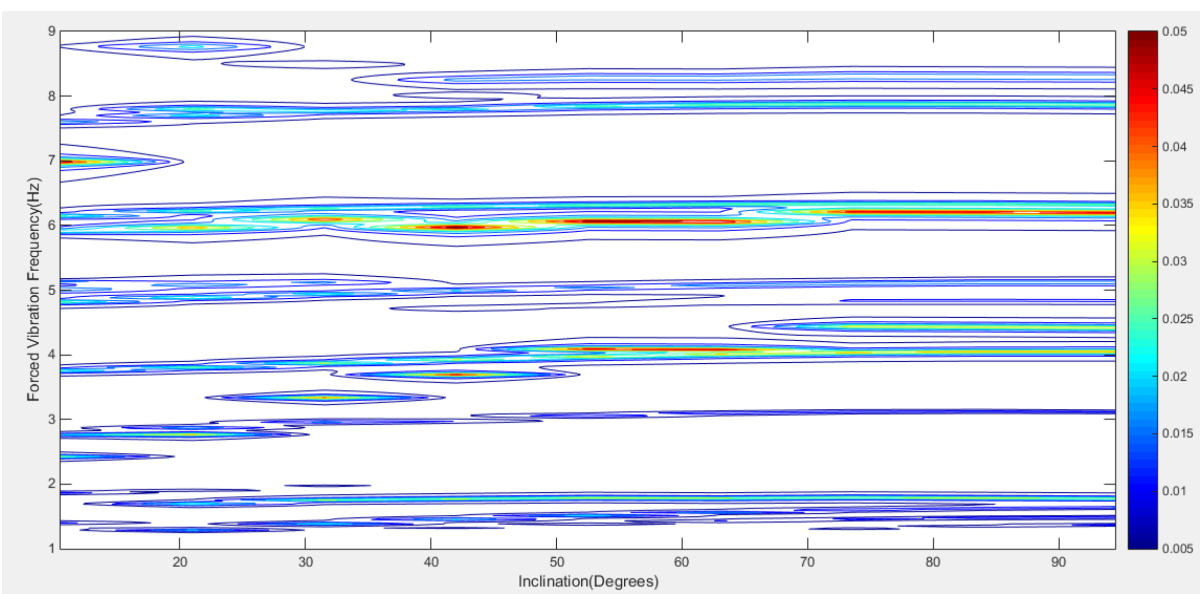


Figure 41: Calculated Forced Vibration Later Deflection

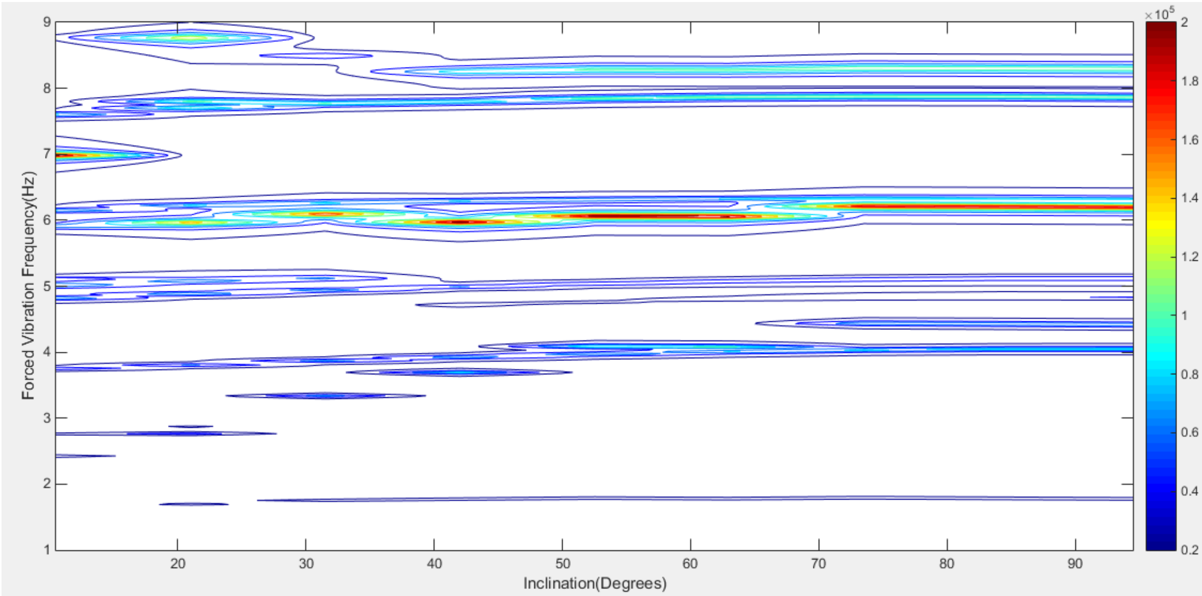


Figure 42: Calculated Forced Vibration Bending Moments

Case six: $4°/30m$

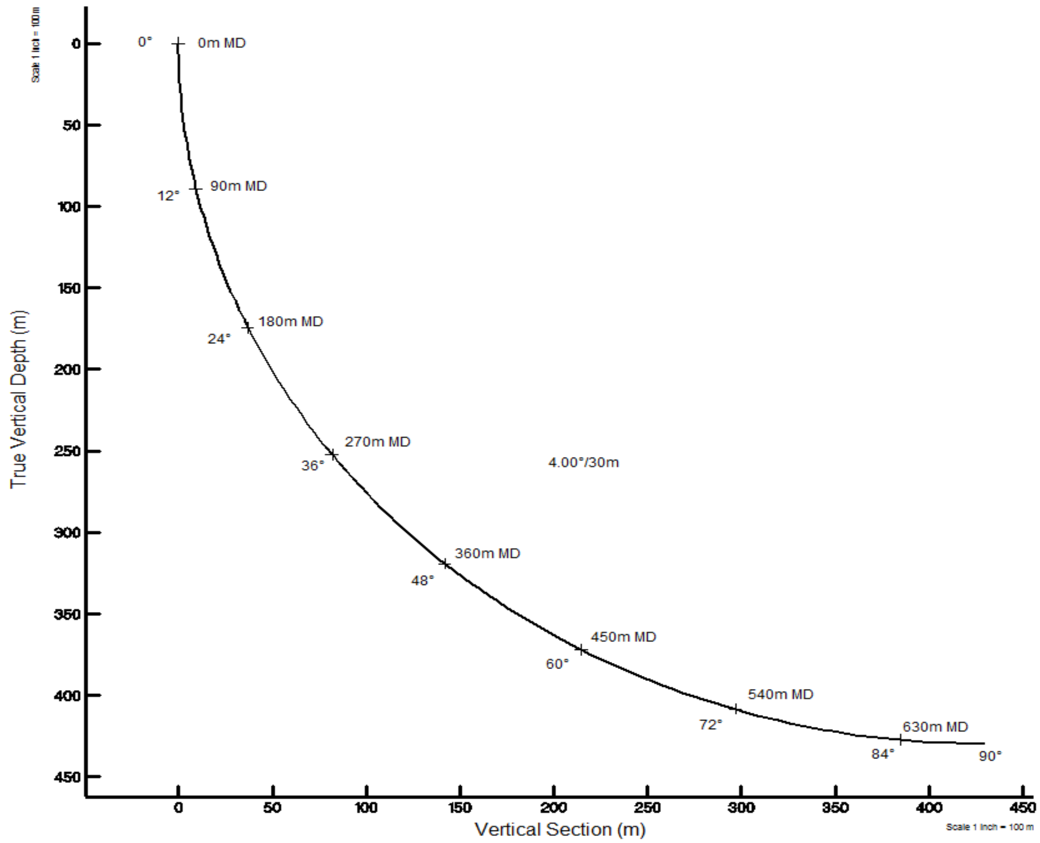


Figure 43: Well path

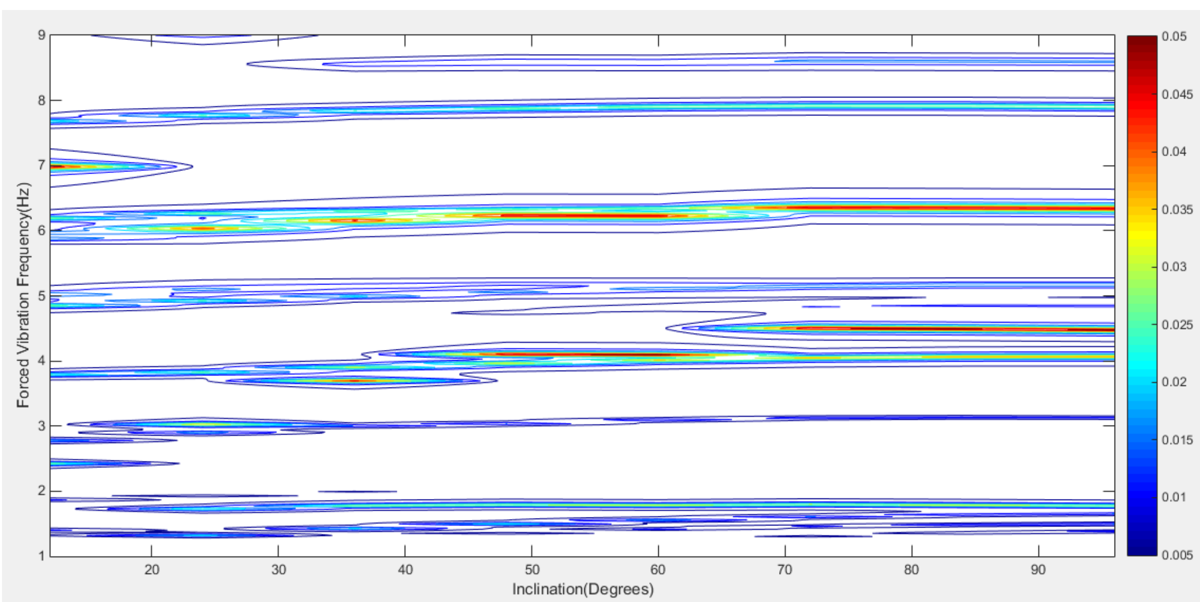


Figure 44: Calculated Forced Vibration Lateral Deflection

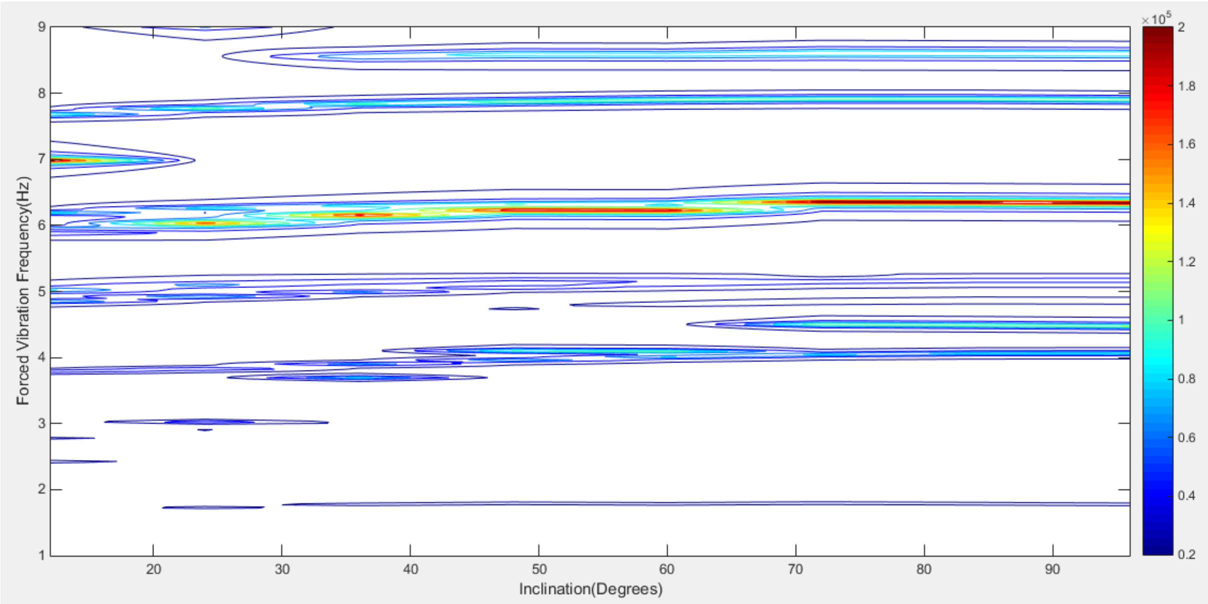


Figure 45: Calculated Forced Vibration Bending Moments

Case seven: $4,5°/30m$

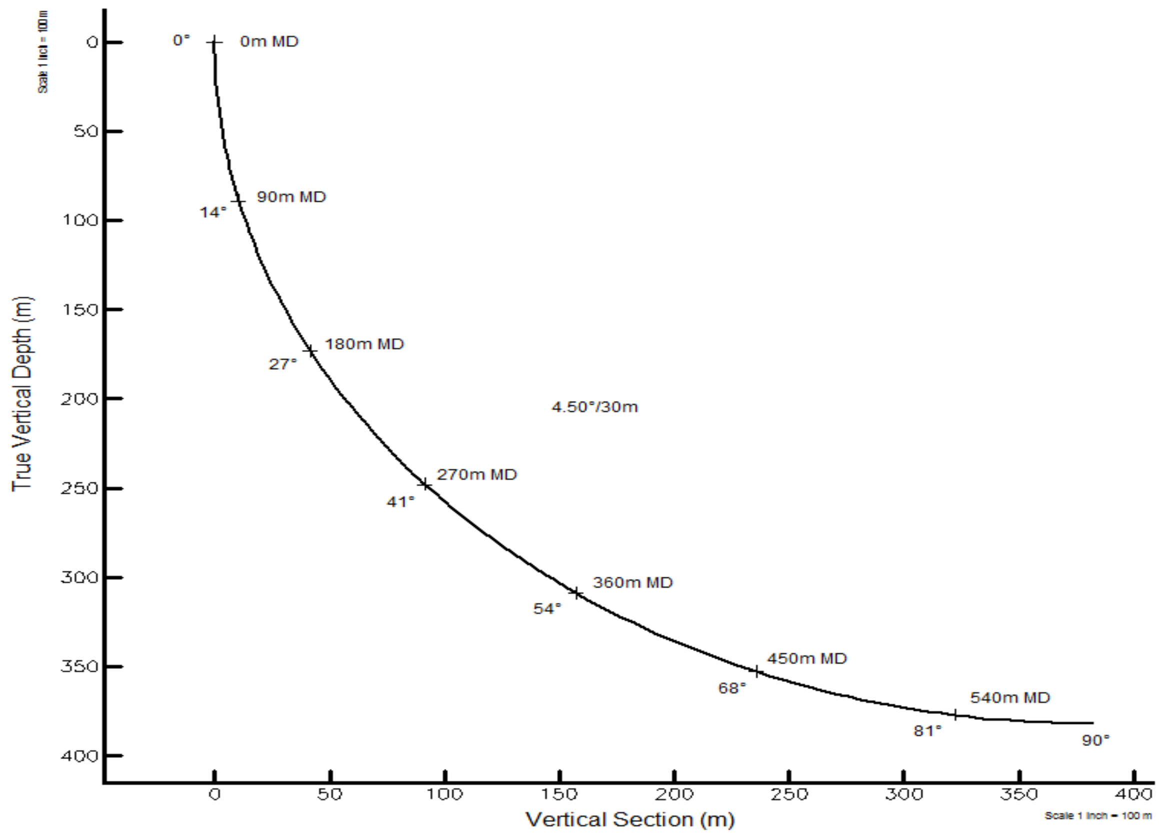


Figure 46: Well path

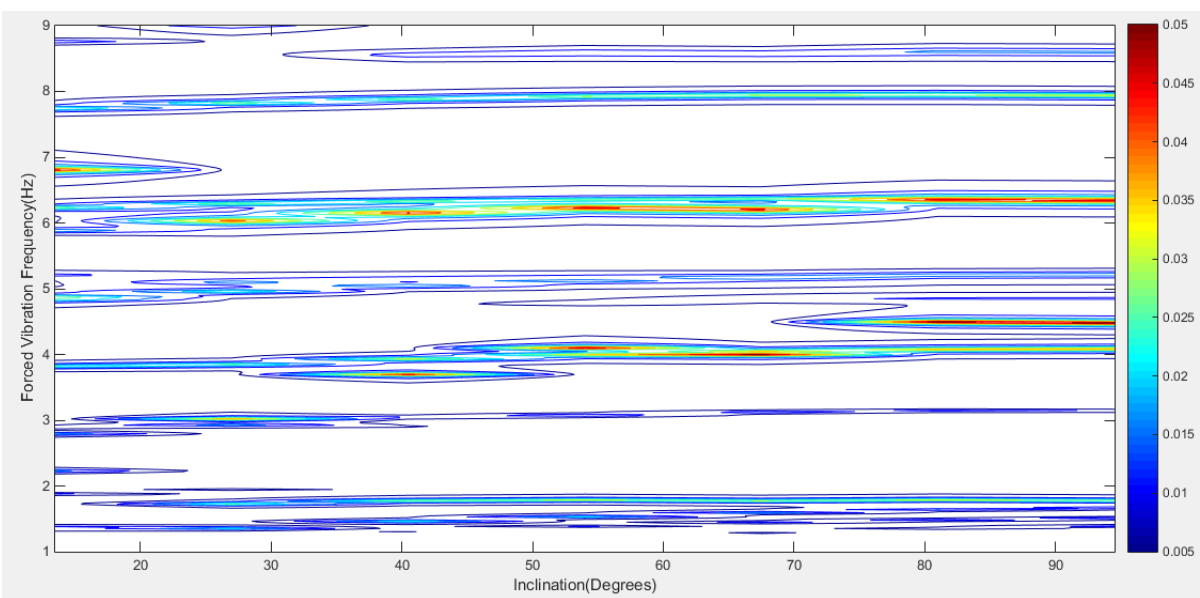


Figure 47: Calculated Forced Vibration Lateral Deflection

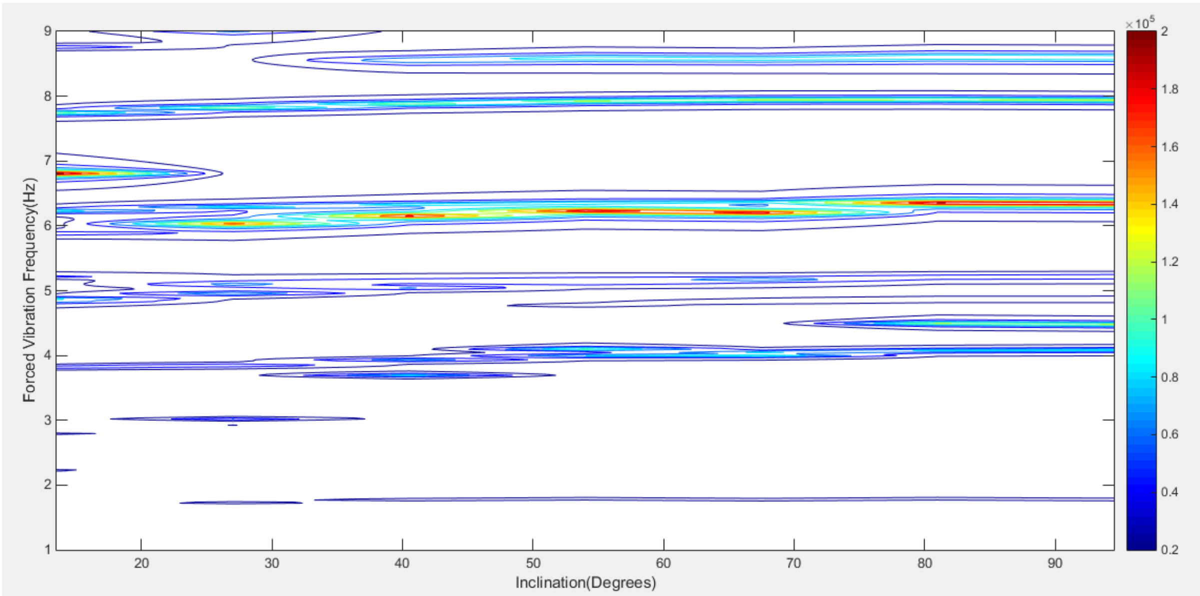


Figure 48: Calculated Forced Vibration Bending Moments

Case eight: $5°/30m$

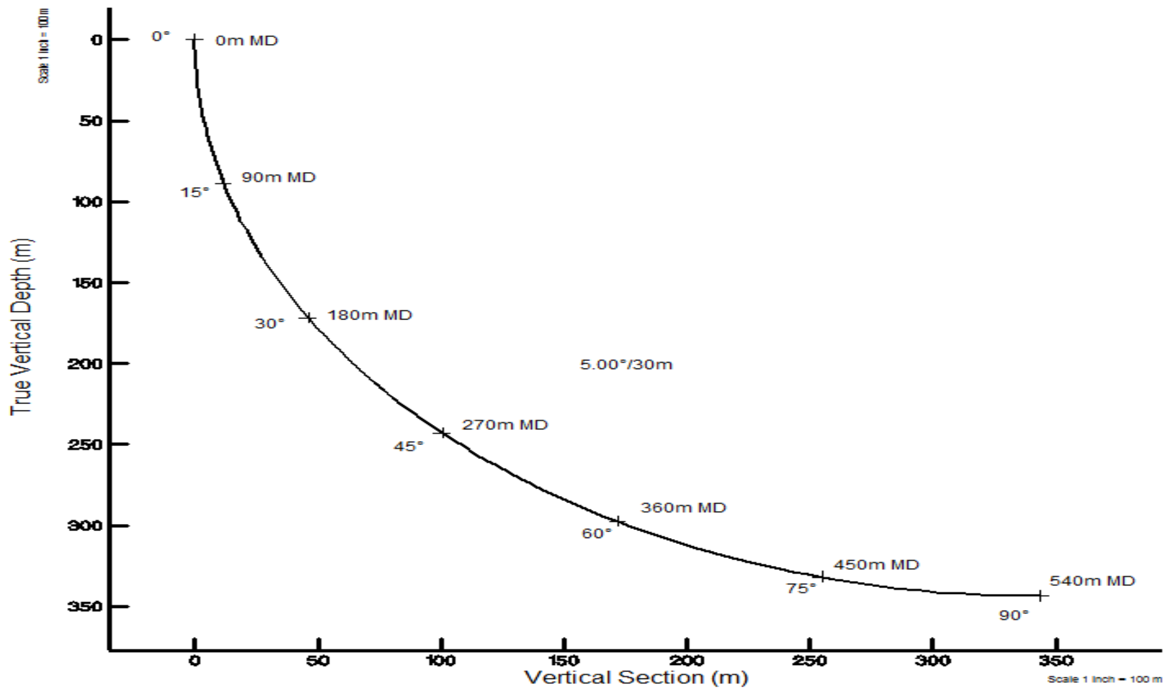


Figure 49: Well path

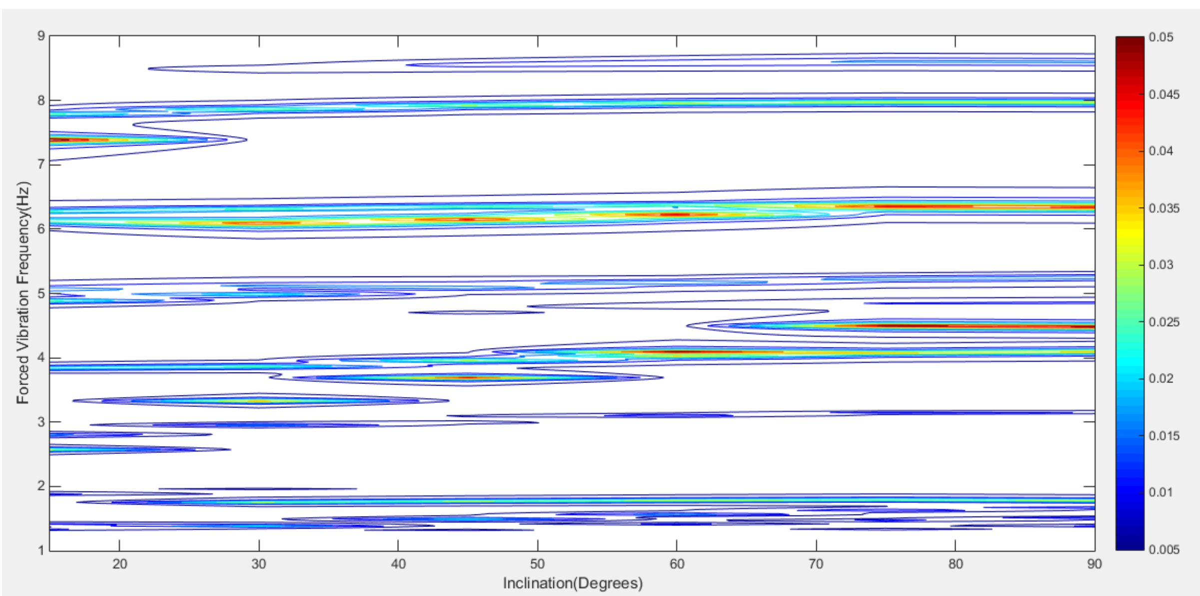


Figure 50: Calculated Forced Vibration Lateral Deflection

Critical frequencies observed in Figure 50 show great stability throughout the whole well path. Significant improvement has been observed compared to the first case and the results presented in Figure 29. In eighth case, the operational window can be determined in the first thirty degrees of inclination which was not possible in the first case.

Operational window is noted:

- Between 40rpm and 60rpm. Above 30degrees of inclination the resulting operational window is from 45-55rpm;
- Between 80rpm and 100rpm. Resulting operational window is from 85rpm-95rpm. This window is valid until the bit reaches 65 degrees of inclination where the window decreases significantly;
- Between 100rpm and 120rpm. Resulting operational window is from 105rpm-115rpm. This operational window is constant throughout the hole well path trajectory;
- Between 128rpm and 140rpm. Resulting operational window is 133rpm-135rpm. This operational window is also stable and it increases from 30-90 degrees if inclination, where the operational window is from 133rpm-147rpm.

  The most important operational RPM windows are between 100RPM and 160RPM since these values are typically used in under reaming operation in Norway.

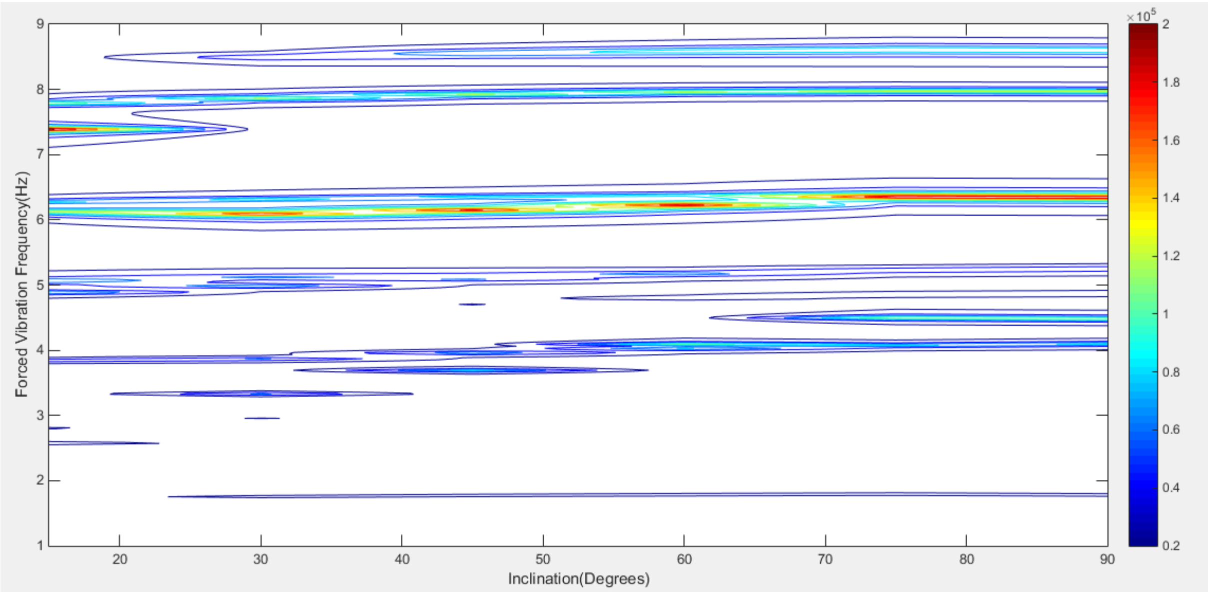


Figure 51: Calculated Forced Vibration Bending Moments

The dynamic bending moments in Figure 51 are now shifted compared to previous cases, the maximum values of dynamic bending moments are observed between 70 and 90 degrees of inclination. Critical frequencies from 1Hz to 5 Hz will not pose a big threat to the BHA, when considering the magnitude of dynamic bending moments.

4.2 Damping Effect of Different Mud Densities on Lateral Vibration

For these simulations, densities varying from 1 s.g to 3,6 s.g are computed into the calculation. In real drilling and reaming operation the mud weights typically used in Offshore Norway are between 1,03 s.g and 1,8 s.g. In the simulations the most critical frequency is initially at 6,6 Hz corresponding to 132rpm which would create devastating effects on the reaming BHA. Dynamic analysis is performed to see the behaviour of this critical frequency when mud weights are being increased. In addition the magnitudes of lateral deflections corresponding to this critical frequency are observed. Results are presented in Figure 52 and 53.

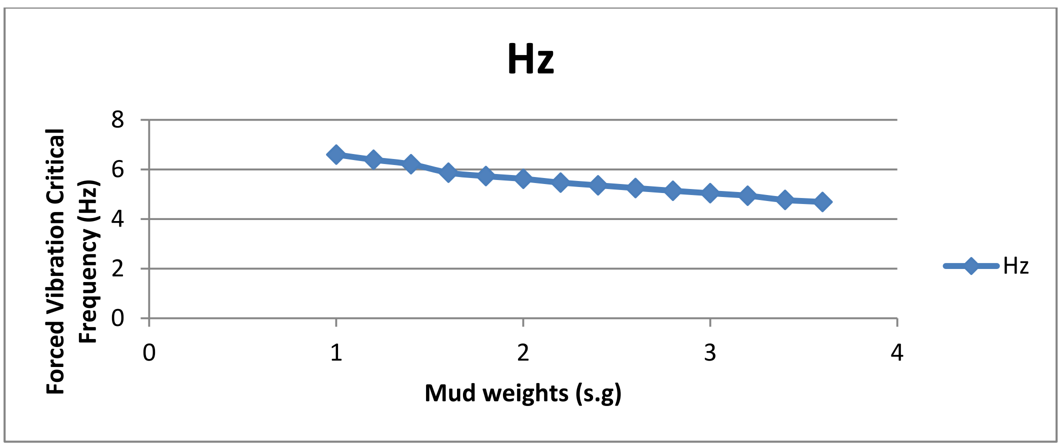


Figure 52: Effects of increased mud weights on critical frequency

Dampening effect on lateral vibration is clearly seen when the initial critical frequency at 6,6 Hz is shifted downwards below 6 Hz at 1,8 s.g. As the mud weight increases, the dampening effect will be higher.

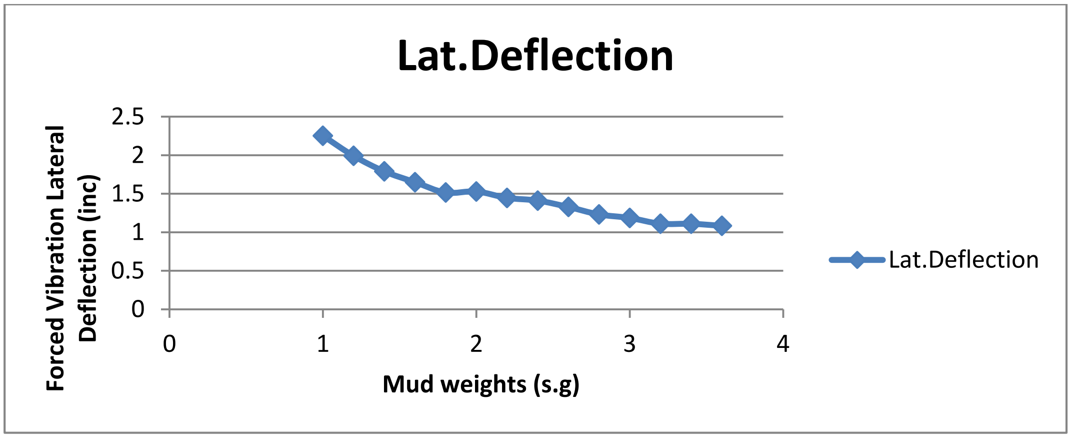


Figure 53: Damping effect of increased mud weights on lateral deflection

Similarly for the same critical frequency the magnitudes of lateral deflection are also decreased from 2,252 inc at 1s.g to 1,515 inc at 1,8 s.g as shown in Figure 53.

5 BHA Optimization for Vibration Mitigation

In this chapter, both static and dynamic analyses are performed on different BHAs. This is done in order to observe how a slight change in the design of BHA can impact the under reaming operation. Optimizing the BHA is a crucial part in pre-planning stage.

5.1 Static Analysis

Static analysis will be performed in order to evaluate build up rate capabilities of different BHAs. This covers the maximum build, drop and turn dog leg severities that the BHA can withstand.

The analysis in the first case is performed on the BHA presented in Figure 54.

1	1/4" PDC bit, 4" gage
2	9-5 ATK Steerable Stab 12-25
3	9-5 Modular Flex Stab 12-125
4	GaugePro_Echo_1225x1475
5	9-5 OnTrak V2 stab
6	9-5 BCPM
7	9-5 Modular Flex Stab 12-125
8	GaugePro_Echo_1225x1475
9	9-5 Mod X/O sub
10	Interface Sub
11	Float Sub
12	Float Sub
13	String Stab 12 1/8
14	Spiral - 8"
15	Spiral - 8"
16	Spiral - 8"
17	Hydra-Jars by Smith
18	Spiral - 8"
19	Spiral - 8"
20	Accelerator
21	Spiral - 8"
22	X/O Sub

Figure 54: BHA containing two modular flex stabilizers

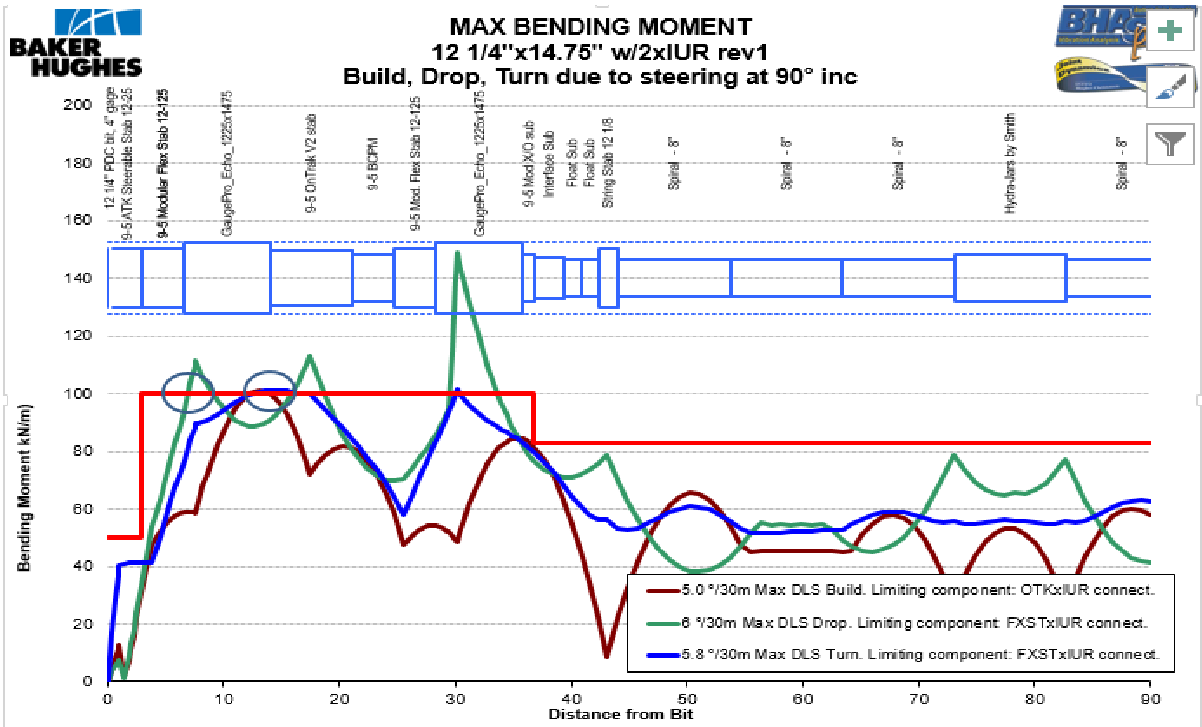


Figure 55: Calculated static results for build, drop, and turn rates

Resulting dog leg capabilities of the BHA presented in Figure 55 are:

- Maximum dog leg severity for Build case- $5°/30m$, where limiting component is connection between OnTrak and Gauge Pro Echo/IUR;
- Maximum dog leg severity for Drop case- $6°/30m$, where limiting component is connection between modular flex stabilizer and Gauge Pro Echo/IUR;
- Maximum dog leg severity for Turn case- $5{,}8°/30m$, where limiting component is connection between modular flex stabilizer and Gauge Pro Echo/IUR.

Second case is also evaluated where the BHA is now changed. Changes are made by replacing both modular flex stabilizers with two modular stabilizers. The BHA is presented in Figure 56.

1	12 1/4" PDC bit, 4" gage
2	9-5 ATK Steerable Stab 12-25
3	9-5 Modular Stabilizer 12-125
4	GaugePro_Echo_1225x1475
5	9-5 OnTrak V2 stab
6	9-5 BCPM
7	9-5 Modular Stabilizer 12-125
8	GaugePro_Echo_1225x1475
9	9-5 Mod X/O sub
10	Interface Sub
11	Float Sub
12	Float Sub
13	String Stab 12 1/8
14	Spiral - 8"
15	Spiral - 8"
16	Spiral - 8"
17	Hydra-Jars by Smith
18	Spiral - 8"
19	Spiral - 8"
20	Accelerator
21	Spiral - 8"
22	X/O Sub

Figure 56: BHA containing two modular stabilizers

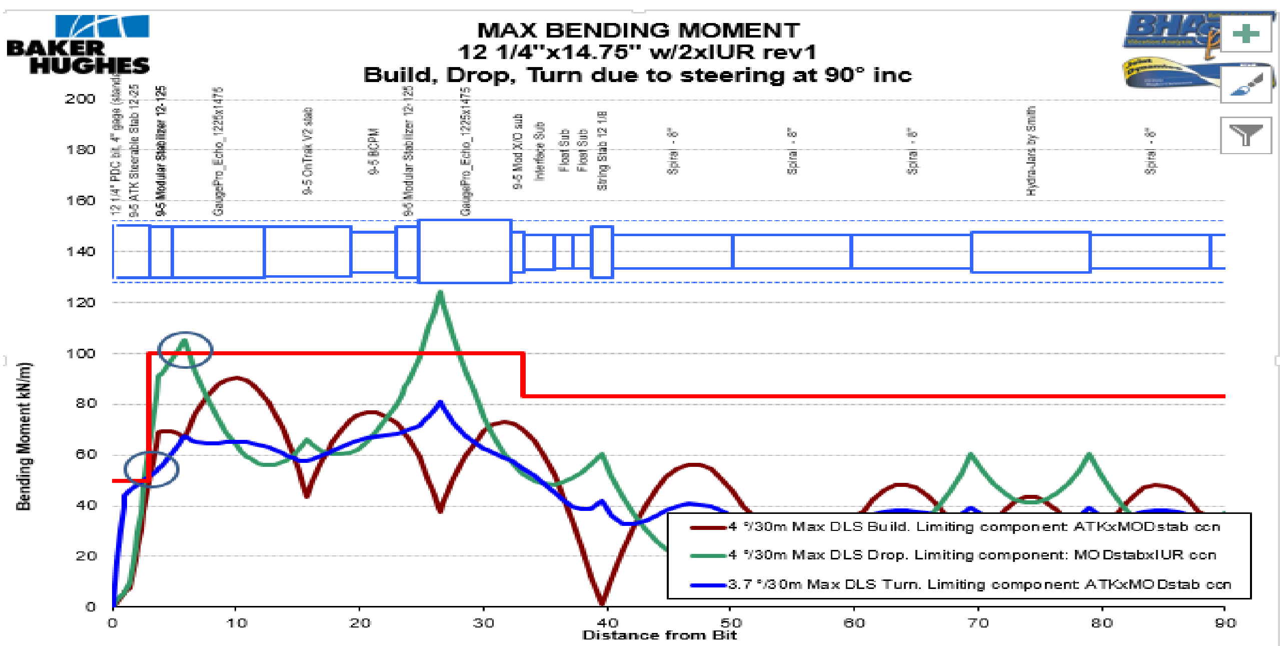


Figure 57: Calculated static results for build, drop, and turn rates

Resulting dog leg capabilities of the BHA presented in Figure 57 are:

- Maximum dog leg severity for Build case- $4°/30m$, where limiting component is connection between AutoTrak and modular stabilizers;
- Maximum dog leg severity for Drop case- $4°/30m$, where limiting component is connection between modular stabilizers and Gauge Pro Echo/IUR;

- Maximum dog leg severity for Turn case- $3{,}7°/30m$, where limiting component is connection between AutoTrak and modular stabilizers.

5.2 Dynamic Analysis of Changed Design of the BHA

In this section dynamic analysis will be performed on a BHA where the modular flex stabilizers are replaced with two modular stabilizers shown in Figure 56. Noticing that 1Hz is equal to 20RPM.

The results consist of 20 dynamics BHASYS Pro simulations. The dog leg severity computed for this case is $3°/30m$. The well path is the shown in Figure 58.

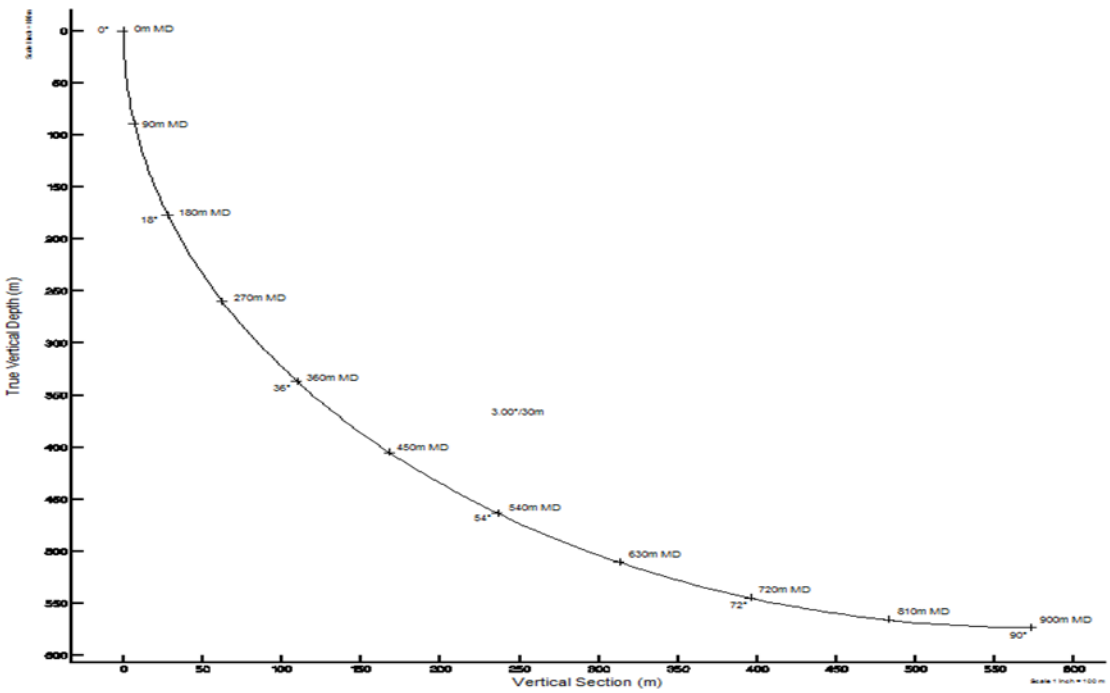


Figure 58: Well path

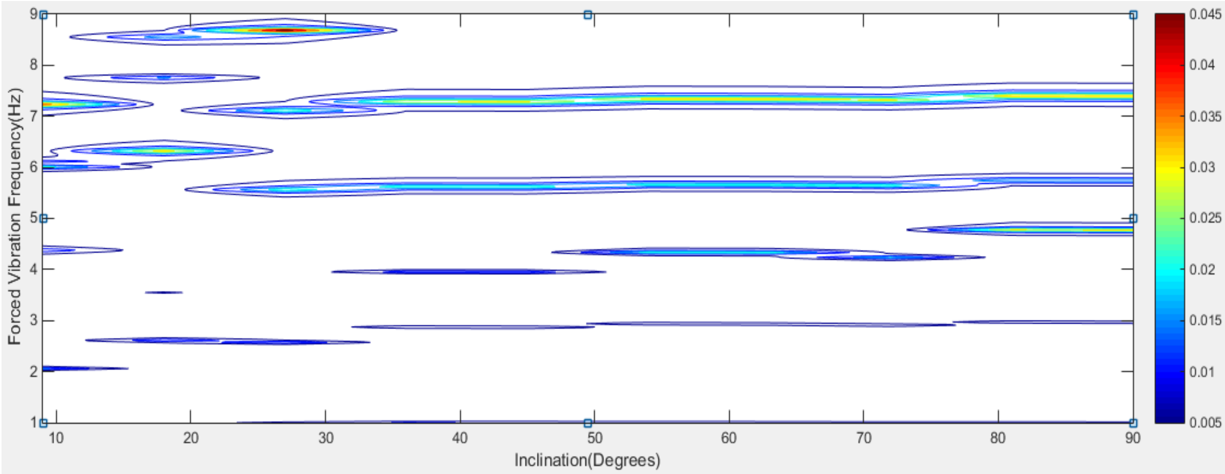


Figure 59: Forced Vibration Lateral Deflection

Operational window noted from Figure 59:

- Between 20rpm and 60rpm. Above 33 degrees of inclination the resulting operational window is from 25rpm-55rpm;
- Between 60rpm and 80 rpm with a slight increase in operational window above 50 degrees of inclination. Resulting operational window is from 65rpm-75rpm.
- Between 40rpm and 60rpm. Above 30degrees of inclination the resulting operational window is from 45rpm-55rpm;
- Between 80rpm and 110rpm. Resulting operational window is 85rpm-105rpm in the section between 30 and 48 degrees of inclination. Above 48 degrees of inclination the operational window will decrease;
- Between 110rpm and 140rpm. Resulting operational window is between 115rpm-135rpm between 30 and 90 degrees of inclination;
- Between 144rpm and 180rpm. Resulting operational window is between 149rpm and 180 since there are no critical frequencies at 9Hz.

Maximum value of the lateral deflection is 0.045m. If the operational RPM is equal to 8,7Hz (174rpm) then the BHA will experience maximum lateral deflection which would result in severe damage to the tools.

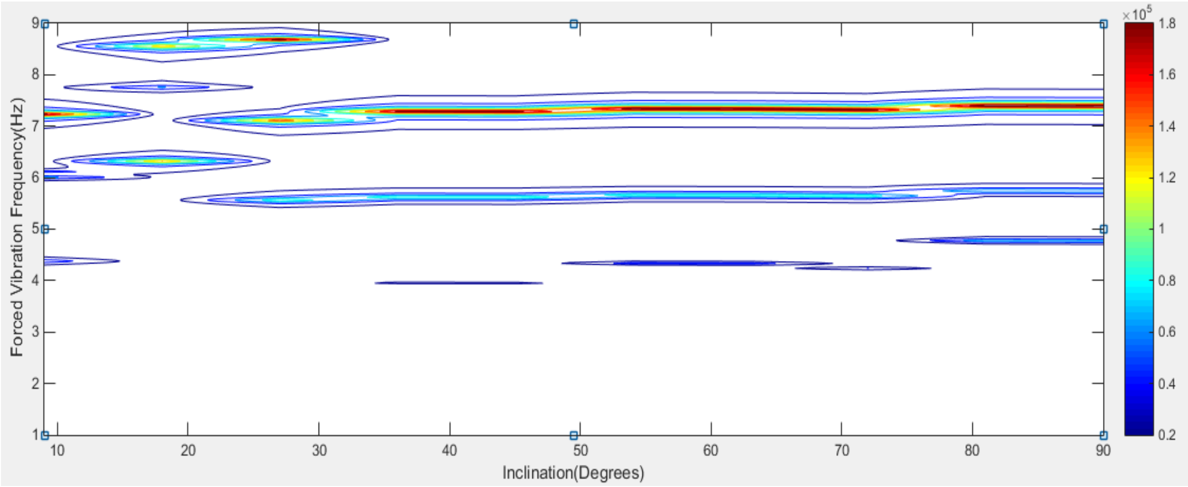


Figure 60: Forced Vibration Bending Moments

Results in Figure 60 are corresponding to the results presented in Figure 59. The highest values of dynamic bending moments are observed throughout critical frequency at 7,2Hz equal to 144RPM. The maximum value of bending moment will be experienced at 8,7Hz.

This BHA with modular stabilizers is stiffer than the BHA with modular flex stabilizers. This will result in better dynamics, the drawback of this BHA are the build, drop and turn capabilities in add

5.3 Reactive Torque Calculations for Backing off Connections

Reactive torque calculations will be performed on different BHAs. Adjustments in the design will be based on changing the various tools in the BHA, such as replacing the stabilizers as presented in chapter 3.4, and adding additional tools to the BHA which will increase its length.

5.3.1 Comparison of Two Different Types of Stabilizers

First case: BHA contains two modular flex stabilizers presented in Figure 61.

1	1/4" PDC bit, 4" gage
2	9-5 ATK Steerable Stab 12-25
3	9-5 Modular Flex Stab 12-125
4	GaugePro_Echo_1225x1475
5	9-5 OnTrak V2 stab
6	9-5 BCPM
7	9-5 Modular Flex Stab 12-125
8	GaugePro_Echo_1225x1475
9	9-5 Mod X/O sub
10	Interface Sub
11	Float Sub
12	Float Sub
13	String Stab 12 1/8
14	Spiral - 8"
15	Spiral - 8"
16	Spiral - 8"
17	Hydra-Jars by Smith
18	Spiral - 8"
19	Spiral - 8"
20	Accelerator
21	Spiral - 8"
22	X/O Sub

Figure 61: BHA containing two modular flex stabilizers

$$Torque = 5kNm \tag{35}$$

$$\varphi = 0{,}0089 \tag{36}$$

$$C = \frac{Torque[Nm]}{\varphi[rad]} \tag{37}$$

$$C = 555555{,}556\frac{Nm}{rad} \tag{38}$$

$$J = 73{,}86kgm^2 \tag{39}$$

$$RPM_{inertia} = 120rpm \tag{40}$$

$$\Omega = 2 * \pi * rpm * \frac{1}{60} \tag{41}$$

$$\Omega = 12{,}56rad \tag{42}$$

$$T_{inertia} = \Omega\sqrt{J * C} \tag{43}$$

$$T_{inertia} = 12{,}56\sqrt{73{,}86kgm^2 * 555555{,}556\frac{Nm}{rad}} \tag{44}$$

$$T_{inertia} = 80455{,}92619Nm \tag{45}$$

$$RPM_{allowed} = \frac{RPM_{inertia} * T_{allowed}}{T_{inertia}} \tag{46}$$

$$RPM_{allowed} = \frac{120rpm * 73000Nm}{80455{,}92619Nm} \tag{47}$$

$$RPM_{allowed} = 109rpm \tag{48}$$

Second case: BHA contains two modular stabilizers, presented in Figure 62.

1	12 1/4" PDC bit, 4" gage
2	9-5 ATK Steerable Stab 12-25
3	9-5 Modular Stabilizer 12-125
4	GaugePro_Echo_1225x1475
5	9-5 OnTrak V2 stab
6	9-5 BCPM
7	9-5 Modular Stabilizer 12-125
8	GaugePro_Echo_1225x1475
9	9-5 Mod X/O sub
10	Interface Sub
11	Float Sub
12	Float Sub
13	String Stab 12 1/8
14	Spiral - 8"
15	Spiral - 8"
16	Spiral - 8"
17	Hydra-Jars by Smith
18	Spiral - 8"
19	Spiral - 8"
20	Accelerator
21	Spiral - 8"
22	X/O Sub

Figure 62: BHA containing two modular stabilizers

$$Torque = 5kNm \tag{49}$$

$$\varphi = 0{,}0064rad \tag{50}$$

$$C = \frac{Torque[Nm]}{\varphi[rad]} \tag{51}$$

$$C = 781250\frac{Nm}{rad} \tag{52}$$

$$J = 70{,}86kgm^2 \tag{53}$$

$$RPM_{inertia} = 120rpm \tag{54}$$

$$\Omega = 2 * \pi * rpm * \frac{1}{60} \tag{55}$$

$$\Omega = 12{,}56rad \tag{56}$$

$$T_{inertia} = \Omega\sqrt{J * C} \tag{57}$$

$$T_{inertia} = 12{,}56\sqrt{70{,}86kgm^2 * 781250\frac{Nm}{rad}} \tag{58}$$

$$T_{inertia} = 93451{,}27447Nm \quad (59)$$

$$RPM_{allowed} = \frac{RPM_{inertia} * T_{allowed}}{T_{inertia}} \quad (60)$$

$$RPM_{allowed} = \frac{120rpm * 73000Nm}{93451{,}27447Nm} \quad (61)$$

$$RPM_{allowed} = 94rpm \quad (62)$$

Comparing these two cases it is observed that replacing the modular flex stabilizers with modular stabilizers in the BHA will have the most severe impact on allowable reactive torque. The initial allowable RPM was 109rpm while with the changed stabilizers it is 94rpm. This drastic decrease in rotational stiffness has made even bigger impact on reactive torque than the increase in cumulative mass moment of inertia calculated in chapter 8.4 for the cases.

5.4 Evaluating the Effect of Adding MWD/LWD Tools to the BHA in Reactive Torque Calculations

In the first case a SeismicTrak/LWD tool is added to the BHA. Figure 63 provides information of the BHA used in the calculation.

Row	Name	Length m	Distance m
1	12 1/4" PDC bit, 4" gage (standa	0,42	0,42
2	9-5 ATK Steerable Stab 12-25	2,54	2,96
3	9-5 Modular Flex Stab 12-125	3,63	6,59
4	GaugePro_Echo_1225x1475	7,46	14,04
5	9-5 OnTrak V2 stab	7,01	21,06
6	9-5 SeismicTrak	3,46	24,52
7	9-5 BCPM	3,69	28,21
8	9-5 Modular Flex Stab 12-125	3,63	31,84
9	9-5 Adapter Sub	0,59	32,43

Figure 63: BHA containing SeismicTrak

$$Torque = 5kNm \quad (63)$$

$$\varphi = 0{,}0098rad \quad (64)$$

$$C = \frac{Torque[Nm]}{\varphi[rad]} \quad (65)$$

$$C = 510204{,}081\frac{Nm}{rad} \quad (66)$$

$$J = 82{,}67kgm^2 \tag{67}$$

$$RPM_{inertia} = 120rpm \tag{68}$$

$$\Omega = 2 * \pi * rpm * \frac{1}{60} \tag{69}$$

$$\Omega = 12{,}56rad \tag{70}$$

$$T_{inertia} = \Omega\sqrt{J * C} \tag{71}$$

$$T_{inertia} = 12{,}56\sqrt{82{,}67kgm^2 * 510204{,}081\frac{Nm}{rad}} \tag{72}$$

$$T_{inertia} = 81570{,}959Nm \tag{73}$$

$$RPM_{allowed} = \frac{RPM_{inertia} * T_{allowed}}{T_{inertia}} \tag{74}$$

$$RPM_{allowed} = \frac{120rpm * 73000Nm}{81570{,}959Nm} \tag{75}$$

$$RPM_{allowed} = 107rpm \tag{76}$$

Second case in addition to SeismicTrak a GyroTrak is added to the BHA. The BHA is represented in Figure 64.

Row	Name	Length m	Distance m
1	12 1/4" PDC bit, 4" gage (standa	0,42	0,42
2	9-5 ATK Steerable Stab 12-25	2,54	2,96
3	9-5 Modular Flex Stab 12-125	3,63	6,59
4	GaugePro_Echo_1225x1475	7,46	14,04
5	9-5 OnTrak V2 stab	7,01	21,06
6	9-5 SeismicTrak	3,46	24,52
7	9-5 GyroTrak	5,84	30,36
8	9-5 BCPM	3,69	34,05
9	9-5 Modular Flex Stab 12-125	3,63	37,68
10	9-5 Adapter Sub	0,75	38,43

Figure 64: BHA containing SeismicTrak, GyroTrak

$$Torque = 5kNm \quad (77)$$

$$\varphi = 0{,}0110rad \quad (78)$$

$$C = \frac{Torque[Nm]}{\varphi[rad]} \quad (79)$$

$$C = 454545{,}45455\frac{Nm}{rad} \quad (80)$$

$$J = 97{,}60kgm^2 \quad (81)$$

$$RPM_{inertia} = 120rpm \quad (82)$$

$$\Omega = 2 * \pi * rpm * \frac{1}{60} \quad (83)$$

$$\Omega = 12{,}56rad \quad (84)$$

$$T_{inertia} = \Omega\sqrt{J * C} \quad (85)$$

$$T_{inertia} = 12{,}56\sqrt{97{,}60kgm^2 * 454545{,}455\frac{Nm}{rad}} \quad (86)$$

$$T_{inertia} = 83657{,}17753Nm \quad (87)$$

$$RPM_{allowed} = \frac{RPM_{inertia} * T_{allowed}}{T_{inertia}} \quad (88)$$

$$RPM_{allowed} = \frac{120rpm * 73000Nm}{83657{,}17753Nm} \quad (89)$$

$$RPM_{allowed} = 105rpm \quad (90)$$

Third case, in addition to SeismicTrak and GyroTrak, a SoundTrak is added to the BHA, presented in Figure 65.

Row	Name	Length m	Distance m
1	12 1/4" PDC bit, 4" gage (standa	0,42	0,42
2	9-5 ATK Steerable Stab 12-25	2,54	2,96
3	9-5 Modular Flex Stab 12-125	3,63	6,59
4	GaugePro_Echo_1225x1475	7,46	14,04
5	9-5 OnTrak V2 stab	7,01	21,06
6	9-5 SeismicTrak	3,46	24,52
7	9-5 SoundTrak APX	9,97	34,49
8	9-5 GyroTrak	5,84	40,33
9	9-5 BCPM	3,69	44,02
10	9-5 Modular Flex Stab 12-125	3,63	47,65
11	9-5 Adapter Sub	0,75	48,40

Figure 65: BHA containing SeismicTrak, GyroTrak and SoundTrak

$$Torque = 5kNm \tag{91}$$

$$\varphi = 0{,}0140rad \tag{92}$$

$$C = \frac{Torque[Nm]}{\varphi[rad]} \tag{93}$$

$$C = 357142{,}8571\frac{Nm}{rad} \tag{94}$$

$$J = 125{,}9kgm^2 \tag{95}$$

$$RPM_{inertia} = 120rpm \tag{96}$$

$$\Omega = 2 * \pi * rpm * \frac{1}{60} \tag{97}$$

$$\Omega = 12{,}56rad \tag{98}$$

$$T_{inertia} = \Omega\sqrt{J * C} \tag{99}$$

$$T_{inertia} = 12{,}56\sqrt{125{,}9kgm^2 * 362318{,}8406\frac{Nm}{rad}} \tag{100}$$

$$T_{inertia} = 83657{,}17753Nm \tag{101}$$

$$RPM_{allowed} = \frac{RPM_{inertia} * T_{allowed}}{T_{inertia}} \tag{102}$$

$$RPM_{allowed} = \frac{120rpm * 73000Nm}{362318{,}8406Nm} \tag{103}$$

$$RPM_{allowed} = 103rpm \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots .104$$

In all three cases adding additional tools will increase the cumulative mass moment of inertia. This will have a negative impact on all off-bottom operations. It is observed that allowable RPM is decreased as the length of the BHA increases.

5.5 Rotating Expandable Under Reamer BHA Across Whipstok

To reduce cost and NPT it is sometimes necessary to commence the under reaming operation in the rat hole, created by milling assembly, as soon as the reamer passes the whipstock. In this situation, it can be potentially necessary to rotate the BHA containing two reamers, where one is placed as the near bit reamer and other as the main reamer, across the whipstock. This poses a threat to under reamer BHA, as the tools can be damaged by the forces acting on the BHA generated by the inclination of the whipstock. The most critical components are recognized to be at under reamers in the BHA and their connections with tools above and below in the string (S. Radford et al., 2013).

This operation needs to be evaluated prior to commencing the operation. For this application both variable and constant parameter calculation in static analysis is performed in the BHASYS Pro. In this example the BHA contains two modular flex stabilizers in addition to other tools, represented in Figure 66.

A whipstock is used in the calculation with $5,8^0$ of inclination and TVD height of 9,66 m.

1	1/4" PDC bit, 4" gage
2	9-5 ATK Steerable Stab 12-25
3	9-5 Modular Flex Stab 12-125
4	GaugePro_Echo_1225x1475
5	9-5 OnTrak V2 stab
6	9-5 BCPM
7	9-5 Modular Flex Stab 12-125
8	GaugePro_Echo_1225x1475
9	9-5 Mod X/O sub
10	Interface Sub
11	Float Sub
12	Float Sub
13	String Stab 12 1/8
14	Spiral - 8"
15	Spiral - 8"
16	Spiral - 8"
17	Hydra-Jars by Smith
18	Spiral - 8"
19	Spiral - 8"
20	Accelerator
21	Spiral - 8"
22	X/O Sub

Figure 66: BHA used in whipstock calculations

Variable parameter calculation is presented in Figure 67, where calculation shows the maximum bending moment limit, which the BHA can handle, is exceeded when rotating the BHA over the whipstock by 7%. However, this does not mean it is not doable. The red zones observed in Figure 67 are actually the highest bending moments acting on the near bit reamer and the main reamer, these red zones are further analysed in order to see whether the highest bending moments are acting on the body of the tool or on the connections.

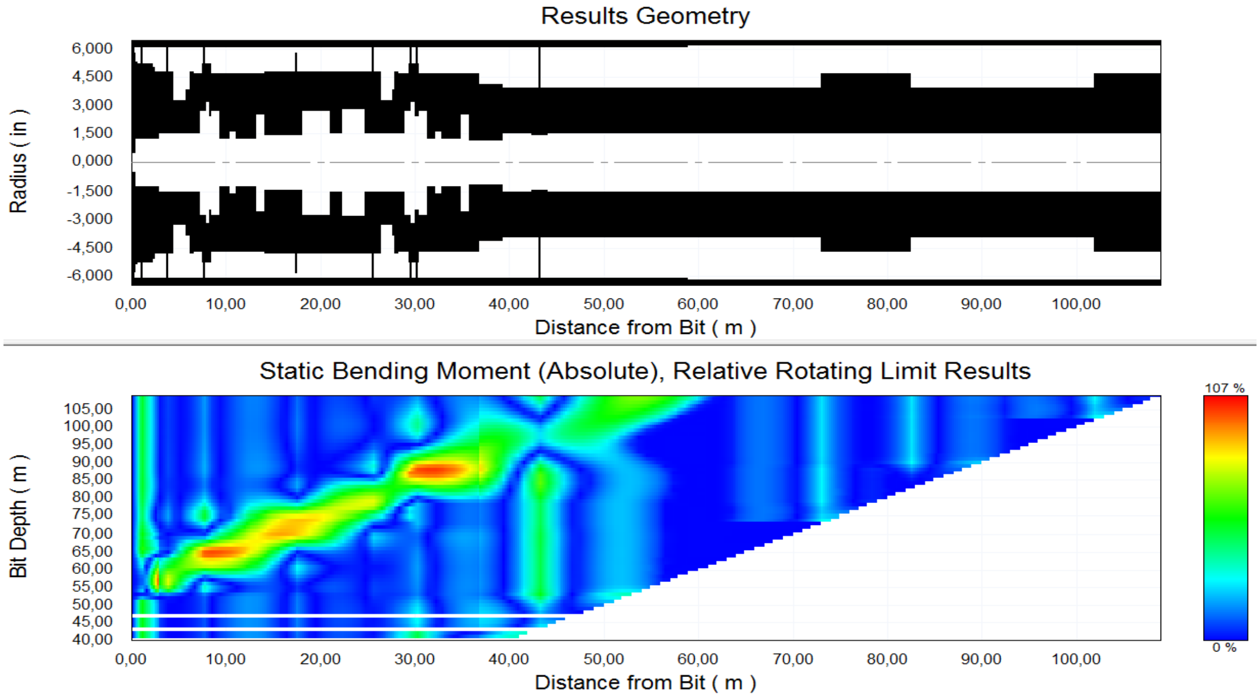


Figure 67: BHA rotating when passing through the whipstock

Figure 68 provides the information exactly where the bending moments are acting on the near bit reamer.

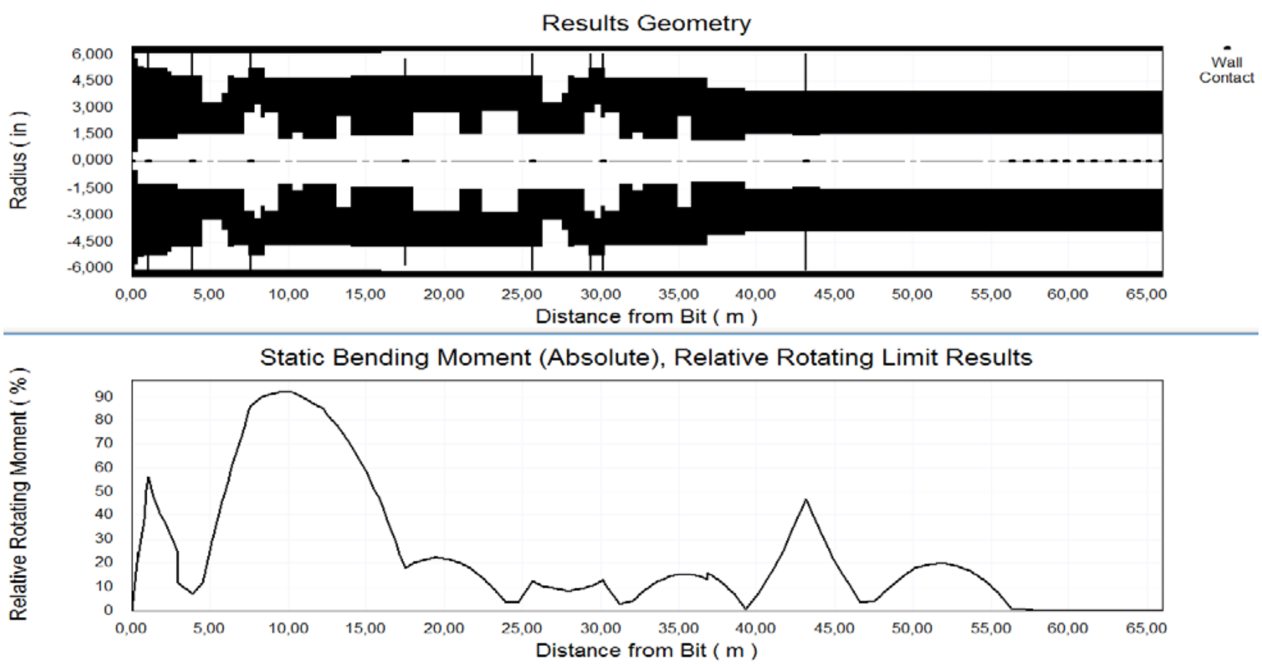


Figure 68: Simulation of the bending moments acting on the near bit reamer, when passing through the whipstock

The highest bending moment are acting on the body of the near bit reamer, which poses no threat to the operation. Similarly the same effect of bending moments is evaluated on the main reamer, presented in Figure 69.

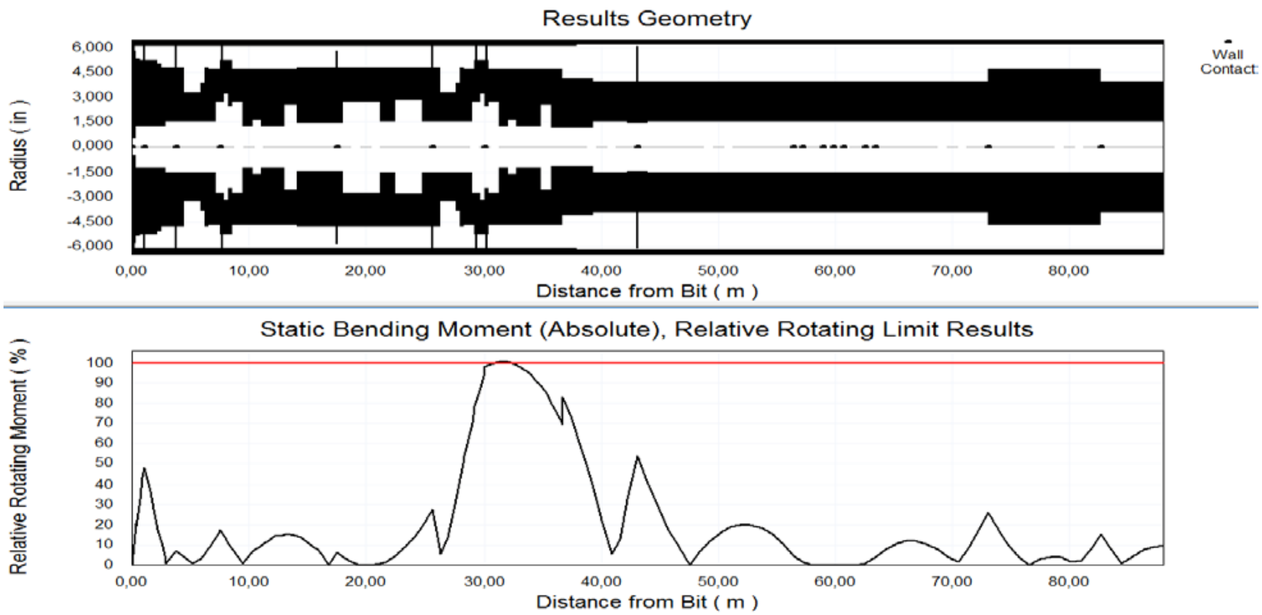


Figure 69: Bending moments acting on the body of the main reamer

In Figure 69, the highest bending moments are acting again on the body of main reamer, where the limit is slightly crossed.

The bending moments simulated when the under reamer BHA passes through the whipstock in rotating mode, does not pose a threat to the operation. In this scenario it is advised to slide the tools through the whipstock, if this is not possible, the BHA can be rotated when passing through the whipstock, as per information provided in these simulations.

6 Special applications – Off bottom reaming and Dual Reaming

Off bottom reaming is the operation where the bit is placed off bottom and the under reamer is activated. Reamer blades are cutting the formation or they are acting as a stabilizer. There are three types of off bottom reaming operations. One type is in the application of the rat hole removal, while the second is the backwards reaming. In addition the third type of the off bottom reaming is the so called up-drilling. In this chapter backwards reaming operation will be analysed in addition to the rat hole removal. In the following analysis the same BHA is used in simulations. Bit size 12,25in; Near Bit Reamer size 13,5in, Main Reamer size 13,5in.

6.1 Conventional Rat hole Reaming

Typically in standard reaming while drilling operations, two Gauge Pro Echo tools are placed in the BHA, where one is positioned above the RSS and the other is on the top the BHA. Making such a configuration of the tools in the BHA will provide the ability of decreasing the rat hole section up to 4m depending on the bit length, therefore the additional run is avoided. The main reamer on the top the BHA will open the hole up to desired size until the bit reaches TD, while the near bit reamer will remains deactivated. After reaching the TD the string is pulled up to the top of the pilot hole, where the near bit reamer will be activated, after which the near bit reamer will ream until the bit reaches TD (Fang, Schwartze, et al., 2016). This procedure today is common among the operators. This scenario is presented in Figure 70.

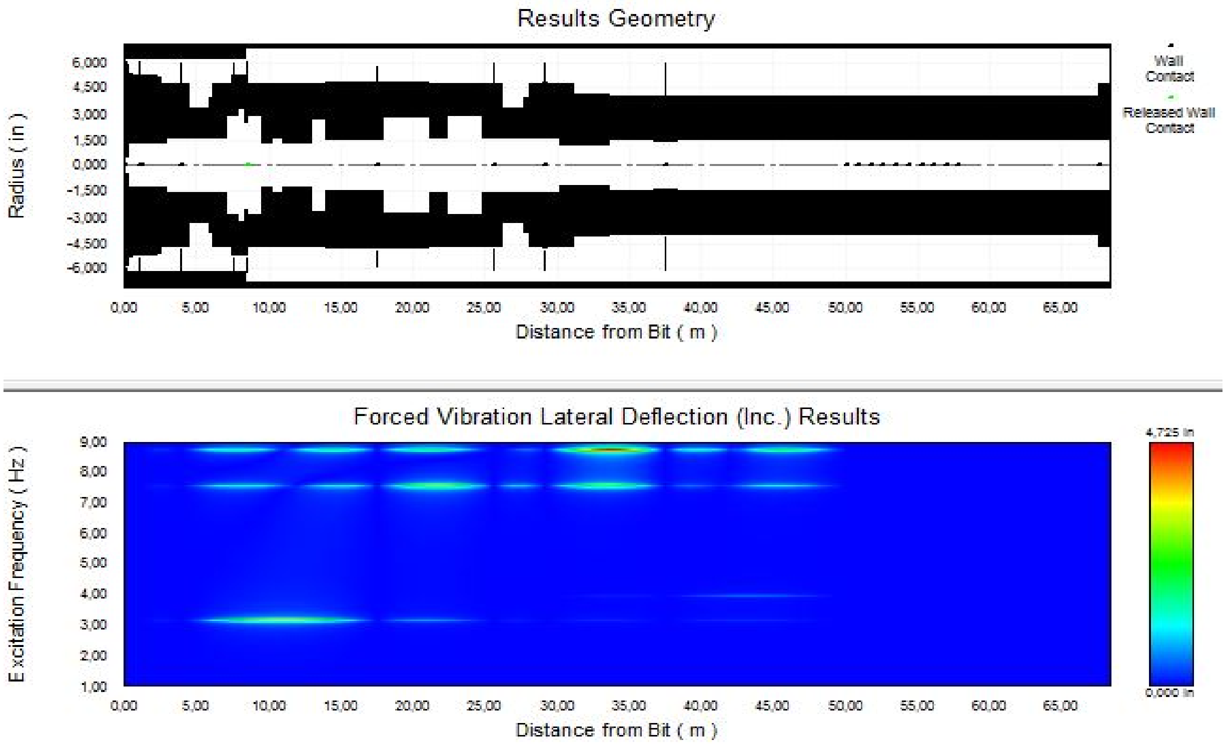


Figure 70: Conventional rat hole reaming

Three critical frequencies are noted with corresponding magnitudes of:

- Forced Vibration Lateral Deflection
- Forced Vibration Bending Moments

First critical frequency is at 3,16 Hz corresponding to 63 rpm.

Where:

- Magnitude of lateral deflection= 3,297 in
- Magnitudes of dynamic bending moments are 188,63 kNm

Second critical frequency is at 7,53 Hz corresponding to 150 rpm

Where:

- Magnitude of lateral deflection is 3,229 in
- Magnitudes of dynamic bending moments are 425,63 kNm

Third critical frequency is at 8,73 Hz corresponding to 175 rpm.

Where:

- Magnitude of lateral deflection is 4,725 in
- Magnitudes of dynamic bending moments are 672,30 kNm

Resulting operational windows are between these three frequencies. Although throughout these windows there are still levels of lateral deflection varying from 0.1 in to 0.25 in. Overall lateral vibrations are experienced.

6.2 Unconventional Rat Hole Reaming

In the first case, slight changes in the operation are evaluated. Instead of activating only the near bit reamer at the top of the rat hole, the main reamer is also activated in addition. The main reamer is now acting as a stabilizer while the near reamer bit is cutting formation.

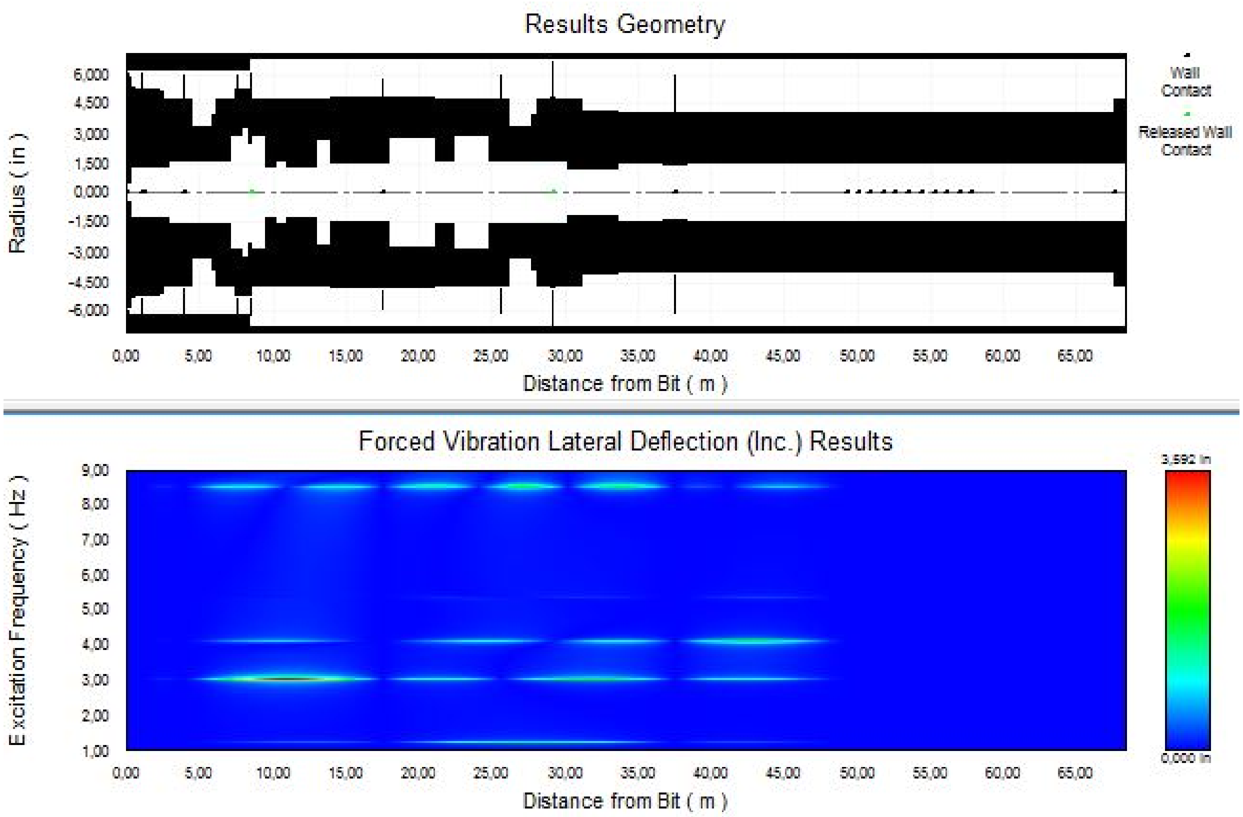


Figure 71: Both under reamers are activated, where the near bit reamer is cutting formation and the main reamer is acting as a stabilizer

In Figure 71, the first critical frequency is at 3,05 Hz corresponding to 61 rpm.

Where:

- Magnitude of lateral deflection= 3,52 in
- Magnitudes of dynamic bending moments are 197,63 kNm

Second critical frequency is at 4,08 Hz corresponding to 81 rpm

Where:

- Magnitude of lateral deflection is 1,395 in
- Magnitudes of dynamic bending moments are 90,01 kNm

Third critical frequency is at 8,54 Hz corresponding to 170 rpm.

Where:

- Magnitude of lateral deflection is 1,964 in
- Magnitudes of dynamic bending moments are 271,99 kNm

This case provides better dynamics compared to the conventional case of removing the rat hole. Overall it is observed that values of lateral deflections and dynamic bending moments are lower at all critical frequencies compared to the case in section in 6.1.

6.3 Dual Reaming

In the case of dual reaming, a new approach of decreasing the rat hole is evaluated based on dynamic analysis. A new type of operation would exclude the option of drilling to TD with main reamer activated and then pulling back the string to the top of the rat hole. Operation is performed prior bit reaching the TD. The bit is on bottom, the near bit and the main reamer are both cutting the formation (Fang, Manseth, et al., 2016).The main reamer will cut the formation for the first 21 meters which is a distance between the near bit and the main reamer. After which the main reamer is acting as a stabilizer until the bit reaches TD. This will make a certain impact on the LWD tools in the enlarged hole, which is not covered in this paper. Slight adjustment is made to the survey and main reamer blade size prior to performing the simulation. Bit size 12,25in; Near Bit Reamer size 13,5in; Main Reamer size 13,625in.

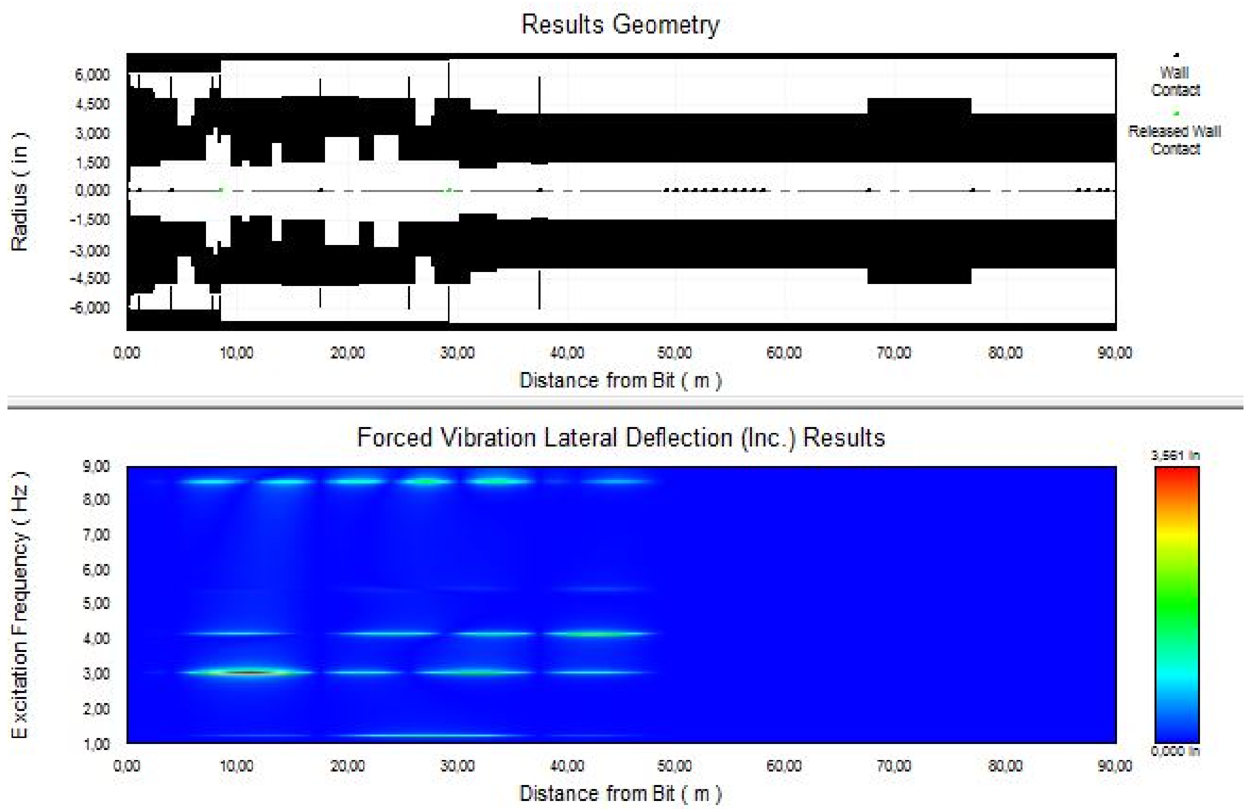


Figure 72: Both under reamers are activated, where the near bit reamer and the main reamer are cutting formation

In Figure 72, the first critical frequency is at 3,05 Hz corresponding to 61 rpm.

Where:

- Magnitude of lateral deflection= 3,5 in
- Magnitudes of dynamic bending moments are 192,92 kNm

Second critical frequency is at 4,14 Hz corresponding to 83 rpm

Where:

- Magnitude of lateral deflection is 1,409 in
- Magnitudes of dynamic bending moments are 98,47 kNm

Third critical frequency is at 8,54 Hz corresponding to 170 rpm.

Where:

- Magnitude of lateral deflection is 1,974 in

- Magnitudes of dynamic bending moments are 279,33 kNm

The dynamics in dual reaming case are very similar to the results presented in 6.2. Improvement in dynamics compared to conventional rat hole reaming is due to increased stabilization of the BHA in the hole, even though both reamers are used in the simulation as the source of excitation. This is operational procedure has been proven in a well on the Norwegian Continental Shelf as time reducing, and more reliable from dynamics point of view.

6.4 Back reaming operation

A back reaming operation is typically performed after the bit reaches the TD and when there is a need to remove the swelling formations prior to running the casing. This is a common practice in Norwegian Sea, but its operation is highly critical and poses a great threat to the BHA due to loss of stability when the part of the BHA below the main reamer is pulled out of the pilot hole. In the following cases, the dynamic analysis is performed.

In the first case the bit is pulled 11m off bottom, represented in Figure 73. The contact point below the main reamer is lost.

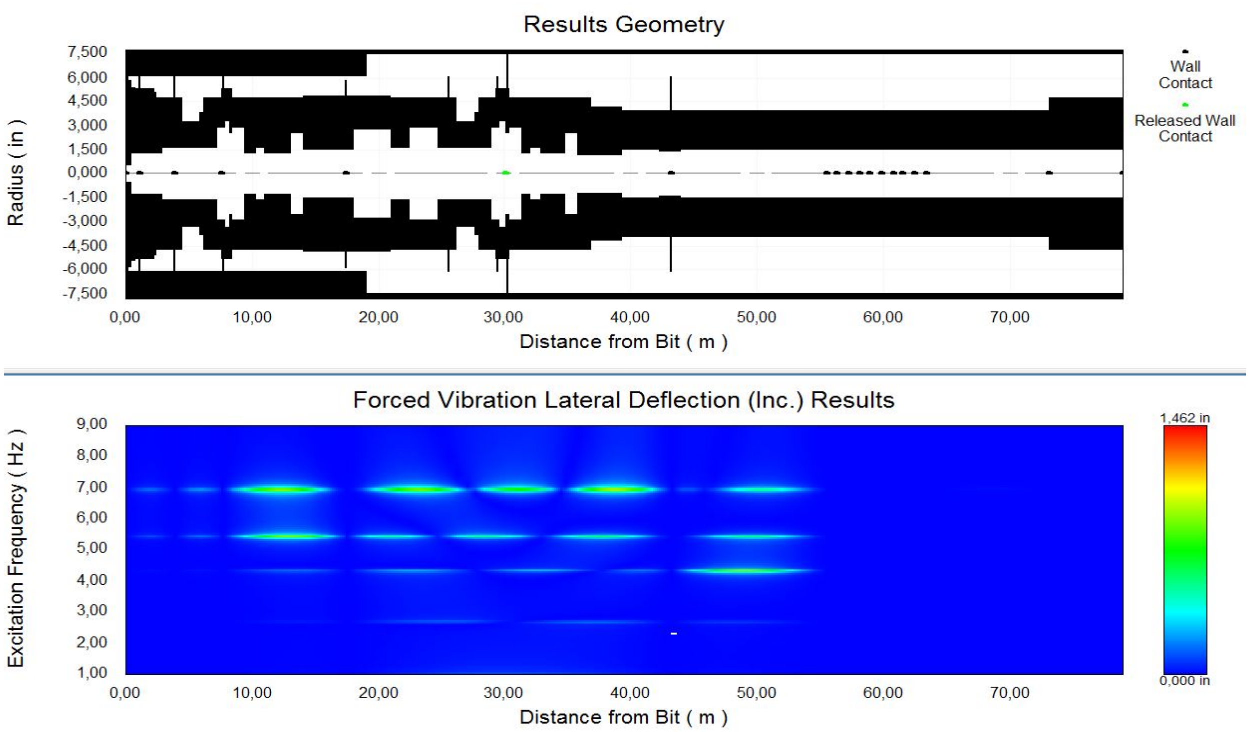


Figure 73: Dynamic analysis, where the bit is pulled 11m off bottom

In Figure 73, the first critical frequency is at 2,67 Hz corresponding to 53 rpm.

Where:

- Magnitude of lateral deflection is 0,231 in
- Magnitudes of dynamic bending moments are 11,7 kNm

Second critical frequency is at 4,30 Hz corresponding to 86 rpm

Where:

- Magnitude of lateral deflection is 0.410 in
- Magnitudes of dynamic bending moments are 36,48 kNm

Third critical frequency is at 5,43 Hz corresponding to 108 rpm.

Where:

- Magnitude of lateral deflection is 0,915 in
- Magnitudes of dynamic bending moments are 85,40 kNm

Fourth critical frequency is at 6,93 Hz corresponding to 138 rpm.

Where:

- Magnitude of lateral deflection is 1,462 in
- Magnitudes of dynamic bending moments are 232,75 kNm

In the second case the bit is pulled off bottom 21m, represented in Figure 77.

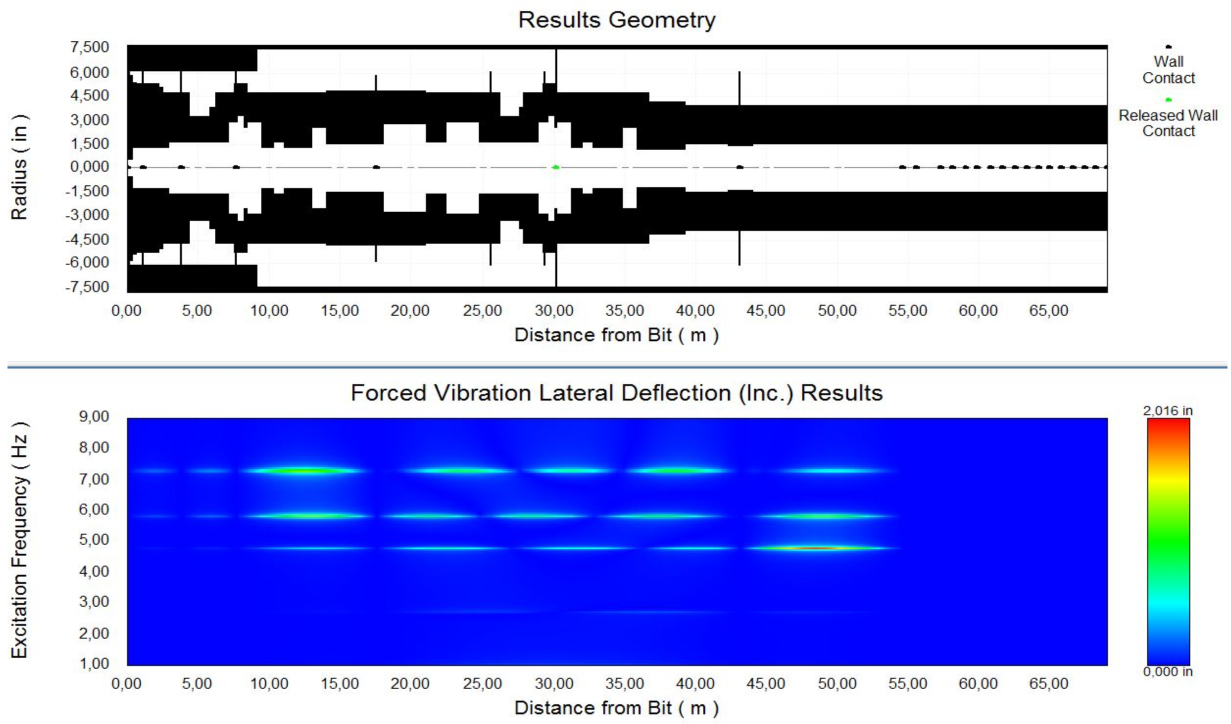


Figure 74: Dynamic analysis, where the bit is pulled 21m off bottom

In Figure 74, the first critical frequency is at 2,72 Hz corresponding to 54 rpm.

Where:

- Magnitude of lateral deflection is 0,247 in;
- Magnitudes of dynamic bending moments are 14,4 kNm.

Second critical frequency is at 4,78 Hz corresponding to 95 rpm

Where:

- Magnitude of lateral deflection is 0,609 in
- Magnitudes of dynamic bending moments are 59,96 kNm

Third critical frequency is at 5,83 Hz corresponding to 116 rpm.

Where:

- Magnitude of lateral deflection is 1,523 in;
- Magnitudes of dynamic bending moments are 159,22 kNm.

Fourth critical frequency is at 7,28 Hz corresponding to 145,6 rpm.

Where:

- Magnitude of lateral deflection is 2,016 in
- Magnitudes of dynamic bending moments are 287,41 kNm

In the third case the bit is pulled 24m off bottom, where the near bit reamer is pulled out of pilot hole, represented in Figure 78.

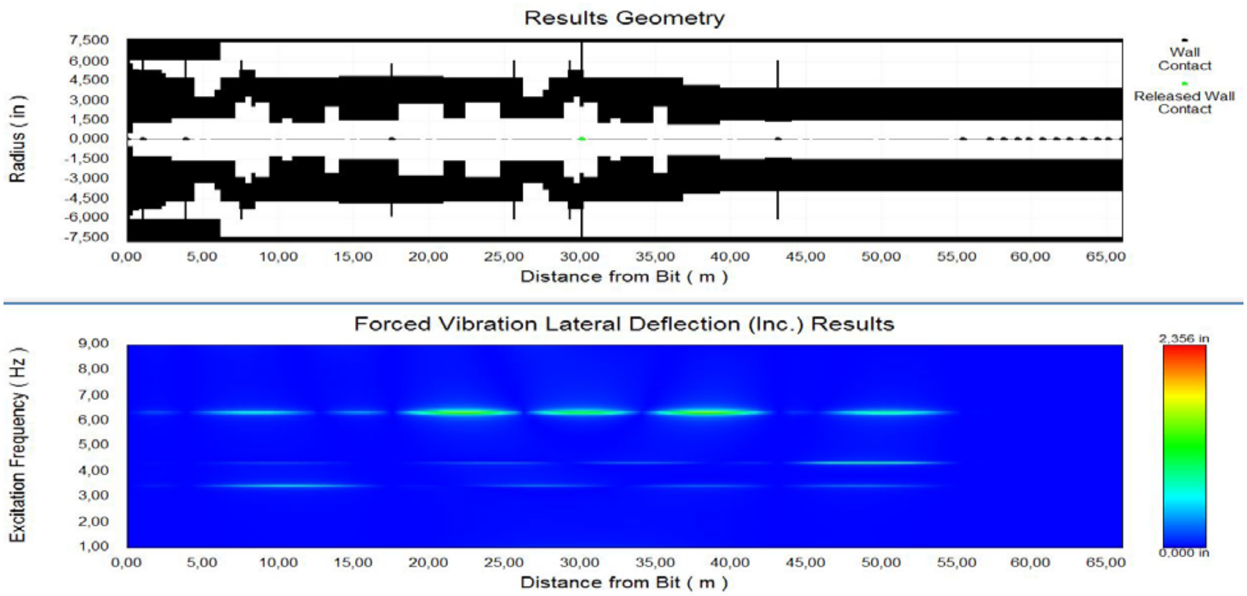


Figure 75: Dynamic analysis, where the bit is pulled 24m off bottom

In Figure 75, the first critical frequency is at 3,44 Hz corresponding to 69 rpm.

Where:

- Magnitude of lateral deflection is 0,881 in;
- Magnitudes of dynamic bending moments are 48,01 kNm.

Second critical frequency is at 4,32 Hz corresponding to 86 rpm

Where:

- Magnitude of lateral deflection is 0,288 in
- Magnitudes of dynamic bending moments are 65,79 kNm

Third critical frequency is at 6,32 Hz corresponding to 126 rpm.

Where:

- Magnitude of lateral deflection is 2,356 in;
- Magnitudes of dynamic bending moments are 247,15 kNm.

In the fourth case, the bit is pulled 40 off bottom, completely out the pilot hole, represented in Figure 76.

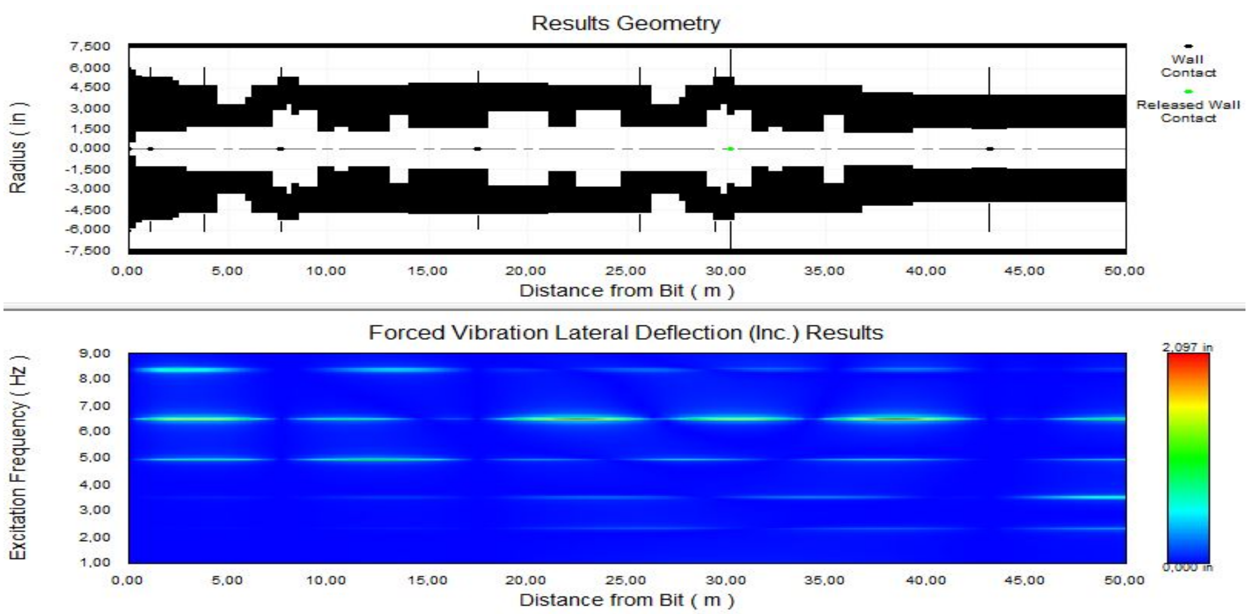


Figure 76: Dynamic analysis, where the bit is pulled 40m off bottom

In Figure 76, the first critical frequency is at 2,22 Hz corresponding to 44 rpm is neglected due to low magnitudes of dynamic bending moments and lateral deflection.

Second critical frequency is at 3,50 Hz corresponding to 70 rpm.

Where:

- Magnitude of lateral deflection is 0,422 in
- Magnitudes of dynamic bending moments are 27,73 kNm

Third critical frequency is at 4,96 Hz corresponding to 99 rpm.

Where:

- Magnitude of lateral deflection is 1,056 in;
- Magnitudes of dynamic bending moments are 91,34 kNm.

Fourth critical frequency is at 6,52 Hz corresponding to 130 rpm.

Where:

- Magnitude of lateral deflection is 2,097 in
- Magnitudes of dynamic bending moments are 226,72 kNm

Fifth critical frequency is at 8,37 Hz corresponding to 167,4 rpm.

Where:

- Magnitude of lateral deflection is 1,026 in
- Magnitudes of dynamic bending moments are 164,74 kNm

In the fifth case the BHA is pulled 49m off bottom, the BHA is completely destabilized as observed in Figure 77. This simulation is placed in order to see if there are changes in critical speeds compared to the fourth case, since in both cases the BHA is in the reamed hole.

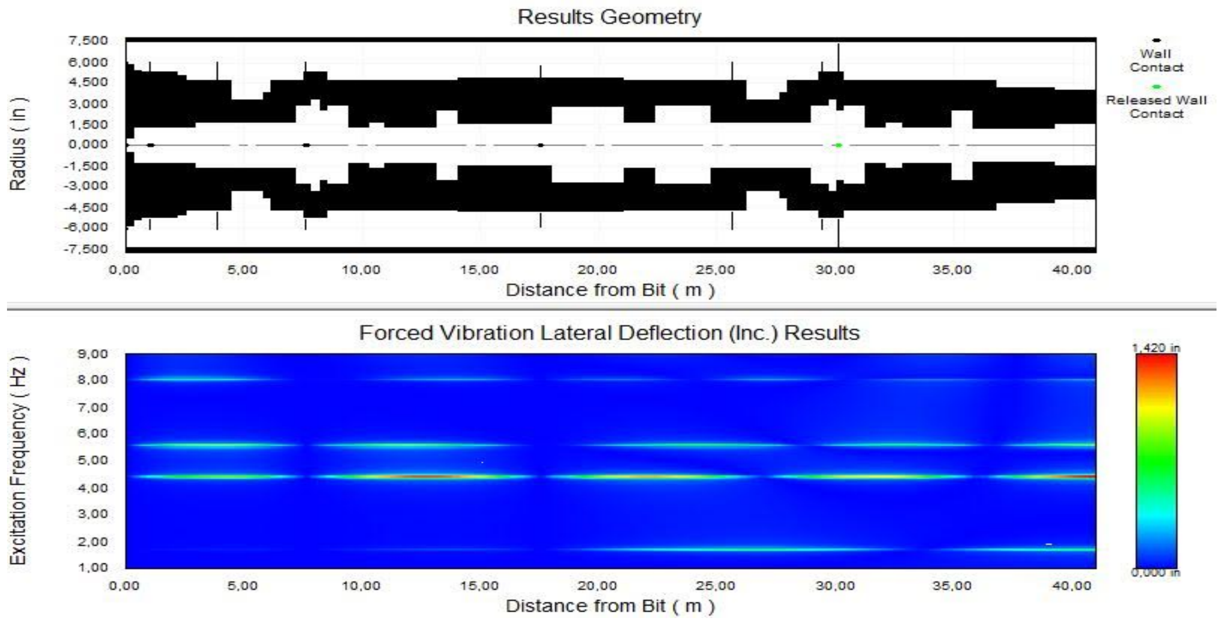


Figure 77: Dynamic analysis, where the bit is pulled 49m off bottom

In Figure 77, the first critical frequency is at 1,71 Hz corresponding to 35 rpm.

Where:

- Magnitude of lateral deflection is 1,298 in;
- Magnitudes of dynamic bending moments are 27,25 kNm.

Second critical frequency is at 4,41 Hz corresponding to 88 rpm

Where:

- Magnitude of lateral deflection is 1,420 in
- Magnitudes of dynamic bending moments are 102,58 kNm

Third critical frequency is at 5,59 Hz corresponding to 112 rpm.

Where:

- Magnitude of lateral deflection is 0,950 in;
- Magnitudes of dynamic bending moments are 111,40 kNm.

Third critical frequency is at 8,05 Hz corresponding to 161 rpm.

Where:

- Magnitude of lateral deflection is 0,501 in;
- Magnitudes of dynamic bending moments are 72,13 kNm.

Comparing these five cases it observed that operational RPM windows are frequently changing due high changes in the stabilization of the BHA inside the borehole. Stiffness matrix behind the calculation is being changed each time a BHA is moved. Overall the back reaming operation poses a great threat with high potential to damage the BHA.

7 Discussions

Through proper understanding of vibrations and BHA optimization, improved reliability can be achieved. Downhole vibrations are classified into three modes, i.e. axial, lateral and torsional. The most severe one is lateral vibration which has detrimental effects on downhole tools. Vibrations cannot be avoided but they can be controlled. Part of the energy is transformed into vibrations. The severity of vibrations experienced by the BHA are not only dependent on the under reaming and drilling parameters utilized in the operation, but also on the specific BHA design and bit to reamer synchronization selected for the operation.

Full knowledge must be gained of how vibrations occur, what the initiating sources are, and how they are controlled and/or experienced in real time operation.

The research has been limited to BHASYS Pro simulations and literature review.

BHASYS Pro Limitations

- BHASYS Pro has some limitations when performing dynamic simulations. The results generated for well path analysis cannot be taken to be a hundred percent accurate in the first 30^o of inclination as the operational windows are covered in critical frequencies.
- Critical frequencies provided from all dynamic simulations can be in real operation shifted up or down by 0.25 Hz.
- Simulating a dynamic case where two reamers are activated in the BHA, cannot be properly done. Instead of main reamer a modular stabilizer is placed. This will provide better results when simulating the dual reaming case. When simulation this scenario blade diameters on the reamers have to of different size in order for simulation to be properly computed.
- The actual magnitudes of dynamic bending moments and lateral deflection do not necessarily represent the actual real time values; instead they should be used as a guideline.

Impact of Well Path Trajectory

In section 4.1, one of the aspects in the pre-planning stage is the well path trajectory. The dog leg severity and inclination will define optimal operational parameters. In-depth study has shown of how dog leg severity and inclination affect downhole vibrations.

In this research an under reamer BHA with two modular flex stabilizers were utilized. Simulations provide answers of the BHA's dynamic response to different well paths. When the dog log severity and inclination are low, the BHA is highly prone to lateral vibration, and the corresponding operational windows of RPM are small.

As the inclination increases for the same dog leg severity, higher stability of the BHA in the borehole is observed due to increased contact points between BHA and borehole wall. This provides bigger operational windows of RPM.

In a scenario in which both the dog leg severity and the inclination are highest, the best operational RPM windows are obtained. In addition to largest RPM windows, critical frequencies have high stability and are not prone to changes. A field validation is necessary to confirm these results.

Impact of Mud Weight

In section 4.2 the dampening effect of mud weights on lateral vibration is carefully studied. Results provide information of how the increased mud weight will decrease the magnitudes of lateral vibration and the critical speeds where the vibration is occurring in terms of operational RPM. Mud weight is designed based on accurate information of the pore pressure and fracture pressure gradients.

These mud weights must be taken into account when computing the dynamic simulations.

BHA Optimization

In section 5 .1 static analysis was performed to evaluate maximum build up rate capabilities of two different BHAs. One BHA contains two modular flex stabilizers while the other contains two modular stabilizers. A flexible BHA has higher build up rate capabilities compared to the BHA with modular stabilizers where build up rates are decreased due to its increased stiffness. This is one of the important aspects in the pre-planning procedures as it is foundation for dynamic analysis.

After comparing the statics of these cases the dynamic analysis was performed on these two

BHAs with same well path trajectories in section 5.2.

This is the point where different BHA setups are compared to find the best solution. The BHA setup shall be chosen to be the most resilient to vibrations, keeping in mind of which build up rates it needs to achieve. If the BHA is used in building a highly inclined well, the modular flex stabilizers are utilized in the BHA. It is known that flexible under reamer BHA is more prone to vibrations compared to a stiffer BHA. Stiffness of the BHA is optimized through the placement of either modular flex stabilizers or modular stabilizers and it also depends on their number and the total length of the BHA.

The simulations performed in section 5.2 provide valuable information of dynamic behaviour of these two BHAs. BHA with modular stabilizers provides larger operational RPM windows compared to the BHA with modular flex stabilizers. In addition the magnitudes of lateral vibration will be decreased if stiffer BHA is used.

Operators are typically interested in using a stiff under reamer BHA for the operations as they are less prone to lateral vibration. This poses limitation for achieving specific dog leg severity and poses a threat when backing reaming operation is performed.

Analyses in section 5.3 provide information of how severe this impact can be. Reactive torque calculations are performed on the same BHA setups as in section 5.2. Flexible under reamer BHA containing modular flex stabilizers is less prone for backing off connections compared to the BHA where modular flex stabilizers are replaced with modular stabilizers. The impact of changed stabilizers in the BHA is extremely severe, reducing the allowable RPM for backing off connections by 14 %. This impact is generated by the increase in rotational stiffness of the BHA.

Having a long BHA where the main reamer is placed at the top, might provide better dynamic behaviour. Section 5.4 examines the impact of increased BHA length on reactive torque calculations by adding various LWD/MWD tools. Answers provided show how the allowable RPM for backing off connection decreases as the BHA length is increased. The impacts of increasing the BHA lengths on reactive torque calculations are much less severe compared to results obtained in section 5.3 where the impact of changed stabilizers plays a major role in these calculations.

Section 5.5 covers a specific case in under reaming operations. It is sometimes necessary to create a sidetrack. In this situation whipstock is placed in the casing and the window in the casing is milled. After performing this operation the under reamer BHA is run in the hole. Important question is placed whether the under reamer BHA can cross over the whipstock in sliding or rotating mode. If not properly analyses under reamers can be damaged which will result in multiple trips and NPT. Variable parameter calculation in BHASYS Pro has provided a full pseudo - static analysis when the entire BHA slides or rotates over the whipstock. Results presented state that this specific BHA can be actually rotated over the whipstock, since the stress hot spots are on the body of the near bit and main under reamer. This needs to be confirmed in the real time operation since the stresses are at their limit and the BHA can potentially fail.

Special Applications

In Chapter 6, off-bottom reaming is analyzed. Three types of this operation are covered and classified as: conventional rat hole reaming, unconventional rat hole reaming, back reaming.

A special case covered in this chapter is dual reaming.

In section 6.1 conventional rat hole reaming operation has been analyzed and shows that it is very critical from dynamics point of view. In this situation the BHA above the near bit reamer is not stabilized hence the lateral vibration is present with very high magnitudes of deflection if the operational RPM is matched with forced vibration frequency. This is occurring due to overall low stabilization of the BHA,

In section 6.2 unconventional rat hole is introduced as it can potentially improve the rat hole reaming dynamics. Only difference between the conventional and unconventional rat hole reaming is that in this case the main reamer in addition to near bit reamer is activated and is acting as a stabilizer. After analyzing this scenario, significant improvement is seen. The size of operational RPM window is improved and the magnitudes of lateral deflections are decreased.

This type of operation shall be considered in all future rat hole reaming operations.

Third case in rat hole reaming is covered in section 6.3 and is classified as dual reaming. Dual reaming is a totally new approach in decreasing the rat hole. Prior to bit reaching TD in reaming while drilling operation, the BHA is stopped. At this point both reamers are activated and are cutting formation along with the bit to TD. Main reamer will only cut the formation for 21m which the distance between two reamers after the main reamer will act as a stabilizer. The rat hole reaming is performed while the bit is on bottom. Having to reamers in cutting action has considered being unsecure and that the vibrations observed would be high. Results obtained through dynamic analysis provide excellent results. The operational RPM windows are bigger compared to conventional rat hole reaming, the magnitudes of lateral deflection, and dynamic bending moments are lower. Utilizing this operation in rat hole reaming can only have a negative impact on logging tools, this is not covered in the thesis. Applying the dual reaming operation can save the operators several rig days.

Section 6.4 describes the back reaming operation commonly utilized when facing the swelling formations. Though in depth study dynamic simulations are computed for five cases, where the bit is pulled off bottom 11m, 21m, 24m, 40m 49m, In each of these cases the stability of the BHA inside the borehole is constantly changing. When the pilot BHA is completely pulled out of the rat hole, it becomes completely destabilized. Lateral vibrations observed are high and destructive. In back reaming operations the critical speeds cannot be properly determined due constant change in contact points. Back reaming operation should be utilized with low operational RPM thus preventing the BHA to experience the full magnitudes of vibrations.

8 Conclusions

This thesis aims at improving the reliability of under reaming while drilling operations by advancing understanding in drilling dynamics associated with under reamer BHA. Focuses were put on better predicting lateral vibration of under reamer BHA at different operating environment and on optimizing BHA design to minimize vibration related failures. This research advanced understanding in impacts of well path and mud density on drilling dynamics of under reamer BHA, identified several recommended practices in optimizing BHA to best mitigate risks associated with vibration and compared the different methods of using dual reamer BHA to eliminate rat hole. The results from this research have been incorporated to improve the newly published Baker Hughes pre-job planning procedures and best drilling guideline for under reaming while drilling operations. It has the potential to improve the field reliability of under reaming while drilling operations.

Detailed static and dynamic analyses have been performed with a Baker Hughes proprietary FEA program, BHASYS Pro. Specific conclusions and recommendations from the research are summarized below:

Impact of Well Path and Mud Density on Under Reaming Dynamics

- The wells with low DLS have high risk of experiencing severe lateral vibrations, especially in low inclinations after KOP.
- As the well inclination increases, the critical frequencies become more stable and operational RPM windows become bigger. This will provide greater flexibility when optimizing the drilling and reaming parameters.
- In wells with low inclinations, BHA with modular stabilizers should be utilized.
- In wells with high dog legs and high inclinations, BHA with modular flex stabilizers should be utilized.
- Dampening of vibration is experienced in drilling and reaming with higher mud weights as critical frequencies become reduced. In addition to this, levels of lateral deflections and dynamic bending moments are also decreased.

BHA Optimization

- Stiffer BHA has high natural frequencies. This is sufficient to push lowermost lateral frequency above the string RPM range, avoiding resonance with mass imbalance excitation. A stiffer BHA is less prone to lateral vibration, compared to BHA with low stiffness, and offers bigger and more stable operational RPM windows when performing the under reaming operation.
- Stiffer BHA , which usually results in high bending loads, can limit dog leg severity of the well path. Increasing the stiffness of the BHA through placement of modular stabilizers will also negatively impact reactive torque for backing off-connections when off-bottom operations are performed due to increased rotational stiffness.
- The impact of having a long BHA on reactive torque calculations is not so prominent compared to the BHA with high rotational stiffness. Rotational stiffness in controlled trough the placement of stabilizers in the BHA.
- CoPilot provides valuable real-time downhole feedback, including weight distribution between bit and under reamer, downhole weight and torque and levels of all vibration modes, and is highly recommended to be included in under reamer BHA.

Special Applications

Three different methods that can be potentially applied to eliminate rat hole are investigated. Conventional rat hole reaming refers to section 6.1. Unconventional rat hole reaming refers to 6.2. Dual reaming refers to 6.3

- BHA is subjected to high level of lateral vibrations due to loss of stabilization (bit off-bottom operation) if conventional rat hole reaming is applied. Conventional rat hole reaming is a scenario where the near bit reamer is activated and placed at the top of the rat hole, in order to decrease it.
- Applying unconventional rat hole reaming provides better BHA stabilization and is less prone to lateral vibrations as compared to conventional rat hole reaming. Unconventional rat hole reaming is similar operation to conventional rat hole reaming. Only difference is that in unconventional rat hole reaming, main reamer is activated in

addition to near bit reamer, and is acting as a stabilizer.

- Dual reaming has the ability to ream rat hole faster with high degree of stability which enables saving of rig time that were previously spent while adopting conventional rat hole reaming procedure.
- In back reaming applications, lateral vibrations observed are high and destructive. Critical speeds cannot be properly determined with BHASYS Pro due to constant changes in the contact points. Back reaming operation should be utilized with low operational RPM to prevent the BHA from experiencing high level of vibrations.

9 References

Bybee, K. (1999). Analysis of stick/slip by use of downhole rotation data. *Journal of Petroleum Technology, 51*(12), 34-35.

Bybee, K. (2009). MWD Vibration Measurements: A Time for Standardization. *Journal of Petroleum Technology, 61*(12), 62-64.

Centrala, P., Chaila, V., Durairajan, B., Meehan, R., Uyen, L. P., & Sigral, P. S. (2011). Bit Design—Top to Bottom. *OilField Review*. Retrieved from

Christoforou, A., & Yigit, A. (2003). Fully coupled vibrations of actively controlled drillstrings. *Journal of sound and vibration, 267*(5), 1029-1045.

Dykstra, M., Chen, D.-K., Warren, T., & Azar, J. (1996). Drillstring component mass imbalance: a major source of downhole vibrations. *SPE drilling & completion, 11*(04), 234-241.

Fang, L., Manseth, A., Stue, K., Johansen, B.-S., & Skappel, J. A. (2016). *A New On-Command Integrated Expandable Under Reamer Technology Enabled Unique Drilling Solutions: Lowering Operational Risks and Saving Operational Cost.* Paper presented at the IADC/SPE Drilling Conference and Exhibition.

Fang, L., Schwartze, S., Grindhaug, G., & Kanzler, C. (2016). *New On-Command Integrated Expandable Under Reamer Technology Reduces Operational Uncertainties and Saves Operational Cost.* Paper presented at the Offshore Technology Conference Asia.

Fiksdal, H., Rayton, C., & Djerfi, Z. (2000). *Application of Rotary Steerable System/PDC Bits in Hard Interbedded Formations: A Multidisciplinary Team Approach to Performance Improvement.* Paper presented at the IADC/SPE Drilling Conference.

Heisig, G., & Neubert, M. (2000). *Lateral drillstring vibrations in extended-reach wells.* Paper presented at the IADC/SPE drilling conference.

Hood, J., Leidland, B., Haldorsen, H., & Heisig, G. (2001). *Aggressive drilling parameter management based on downhole vibration diagnostics boosts drilling performance in difficult formation.* Paper presented at the SPE Annual Technical Conference and Exhibition.

Internal Document - AutoTrak G3 - Introduction and History. (2014).

Internal Document - BHA Reactive Torque Calculation. (2015).

Internal Document - CoPilot Introduction. (2015).

Internal Document - Drilling Dynamics. Baker Hughes.

Internal Document - Dynamic Analysis. (2016). Baker Hughes.

Internal Document - OnTrak Technical Data Summary. (2016). Baker Hughes.

Ishak, G., Daily, J. S., Miska, S. Z., & Mitchell, R. F. (2012). *Modeling Dynamic Behavior of Bottom Hole Assemblies Containing an Underreamer*. Paper presented at the SPE Deepwater Drilling and Completions Conference.

Li, C. (1986). An analytical study of drill string vibrations.

Meyer-Heye, B., Reckmann, H., & Ostermeyer, G.-P. (2010). *Weight distribution in reaming while drilling BHAs.* Paper presented at the IADC/SPE Drilling Conference and Exhibition.

Meyer-heye, B., Reckmann, H., & Ostermeyer, G.-P. (2011). *Underreamer dynamics.* Paper presented at the SPE/IADC Drilling Conference and Exhibition.

Payne, M., & Abbassian, F. (1997). Advanced torque and drag considerations in extended-reach wells. *SPE drilling & completion, 12*(01), 55-62.

Radford, S., Desselle, S., Enterline, J., Allain, M., Oliveire, J., Pearl, B., & Palmer, J. (2013). *Case Study of Rotating Expandable Reamer Across Whipstock and Drilling Ahead over 8,000 ft Reduces Drilling Time.* Paper presented at the SPE/IADC Drilling Conference.

Radford, S. R., Hafle, M. E., Ubaru, C. C., Thomson, I. J., & Morel, B. (2010). *Proper Bit and Underreamer Synchronization Concept Delivered Excellent Performance in GoM Deepwater Well.* Paper presented at the Offshore Technology Conference.

Robnett, E., Hood, J., Heisig, G., & Macpherson, J. (1999). *Analysis of the stick-slip phenomenon using downhole drillstring rotation data.* Paper presented at the SPE/IADC drilling conference.

Samuel, R. (2010). Friction factors: What are they for torque, drag, vibration, bottom hole assembly and transient surge/swab analyses? *Journal of Petroleum Science and Engineering, 73*(3), 258-266.

Santos, H., Placido, J., & Wolter, C. (1999). *Consequences and relevance of drillstring vibration on wellbore stability.* Paper presented at the SPE/IADC drilling conference.

Schmalhorst, B., & Neubert, M. (2003). *Dynamic Modeling Software*. Paper presented at the American Association of Drilling Engineers Conference, AADE-03-NTCE-53, Huoston, Texas.

Schuh, F., Coragliotti, A., DiCicco, C., Nagatani, R., Rea, A., Carlton, T., . . . Noble, R. (2014). *Characterization of Encapsulated Oil as an Additive to Water-Based Drilling Fluids: Operational Improvements in Lubricity, Drag, and ROP*. Paper presented at the SPE Western North American and Rocky Mountain Joint Meeting.

Schulte, C., & John, C. (2016). Internal Document - GaugePro Echo—all Series—Prejob Planning.

Selnes, K., Clemmensen, C., & Reimers, N. (2008). Drilling Difficult Formations Efficiently With the Use of an Antistall Tool. Paper SPE 111874-MS presented at the IADC/SPE Drilling Conference and Exhibition, Orlando, Florida, 4–6 March.

Stokvik, C. (2010). An investigation of forces and moments from drilling risers on wellheads.

Thomson, I. J., & Mathur, R. (2010). *The use of downhole drilling parameters combined with surface and downhole mechanical specific energy data helped identify underreamer dysfunctions in GoM deepwater projects.* Paper presented at the SPE Deepwater Drilling and Completions Conference.

Thomson, I. J., Radford, S. R., Powers, J. R., Shale, L. T., & Jenkins, M. (2008). *A systematic approach to a better understanding of the concentric hole-opening process utilizing drilling mechanics and drilling dynamics measurements recorded above and below the reamer*. Paper presented at the IADC/SPE Drilling Conference.