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Enhanced Drilling Performance Through Controlled Drillstring Vibrations

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Abstract

The purpose of this work is to investigate drillstring vibrations, and to advise on whether it is best to minimize, if not delete these vibrations or if they can be utilized to the benefit of the drilling performance. The work consists of two generic parts. One addressing resonance vibrations when employing roller cone bits, the other dealing with torsional vibrations when drilling using PDC bits.

Introduction

Recently, attention has been paid to improving drilling efficiency by imposing dynamic loading at the bit-rock interface. To date, this has been applied only in the restrictive circumstances of shallow, hard rock drilling where air-based drilling is possible. This work deals specifically with two separate concepts that take advantage of the vibrations of the drill strings to enhance the performance of the bit in formations characterized by low penetration rates. The various modes of vibrations of the drill strings (axial, torsional, and bending which in their most severe forms lead respectively to bit bouncing, stick-slip oscillations, and bit whirling) are generally regarded as detrimental. However, as shown in this work, it appears possible to control some of these vibrations modes in such a way as to enhance drilling performance.

The first technique to be investigated deals with *resonance drilling* which is applicable to "hard" rock drilling using rock bits (roller-cone bits). The key concept of this first step is to either avoid resonance behavior by regulating the operational parameters, in particular rpm, and weight-on-bit (WOB), or by adjusting the resonance frequency of the drillstring through mechanical filters in the bottomhole assembly (BHA) so as to match the loading excitation at the drill bit. Examples of performance improvements are included in the paper.

The second research area explores the benefits of a coupled axial/torsional vibration system, which is aimed at improving drilling rates of PDC bits in "soft" impermeable rocks. The key concept of this second project is to minimize energy dissipation due to friction losses at the wearflats, or chamfers. The generation of parametric maps will help to demonstrate the effect of influencing factors, and eventually should help PDC bit manufacturers optimize bit design, and perfect bit selection.

Two different concepts for hard and soft rocks are needed because of the very different rock destruction mechanisms behind the low penetration rates observed in these formations, as discussed below.

Resonance Drilling

The proposition is to make vibrations work towards rock destruction as opposed to BHA destruction. This paper addresses and solves two issues, one related to hard rock drilling and the other related to the mitigation of vibrations within the BHA that enhances the life of the MWD and LWD tools.

Whereas the approach and methodology for calculating resonance vibrations developed in this paper is not new, and was used by other authors e.g. to describe logging based on dynamic vibrations [1], or to explain drillstring failure [2], the investigation of dynamic loads to improve the rate of penetration (ROP) is different to previously undertaken efforts and is believed to add significant value.

Dynamic and static axial bit loads that occur while drilling affect rock breakdown at the bottom hole. Rock destruction studies analyzing roller cone bits indicated that dynamic forces play a major role in this process, whereas static loads provide continuous contact between the bit and the bottom hole. Dynamic forces result from interaction between the drillstring and the bottom hole through a drill bit that is, simultaneously, a rock destruction mechanism and also, a source of dynamic forces [3].

The resonance drilling approach presented in this paper covers all types of vibrations, i.e. axial, lateral, and torsional. Since it lies outside the scope of this paper to present all of them in detail, the explanations and examples are taken from the axial vibration case only.

When drilling using roller cone bits, a lobe-type structure is formed on the bottom hole. The number of lobes equals the number of cones or sections of the bit. When the bit cones roll over on the bottomhole, ridge-type structures are continuously formed and disintegrated. In other words, a lobe-type bottom hole rotates at a certain speed that depends on the rock breakdown rate [4]. Also, bit cones rolling over on the bottom hole generate quasi-periodical displacements of the lower drillstring. As a first approximation, these displacements can be considered harmonic displacements.

A drillstring is, essentially, a physical system with distributed mechanical parameters: mass, rigidity, and damping. Such systems feature frequency-dependent dynamic parameters, i.e. resonances and anti-resonances [5,6].

When the bit excitation frequency corresponds with the resonance frequency of the drillstring, the bit is easily displaced. In this case, a small applied force is capable of generating intense oscillations in the drillstring. However, when a displacement with an anti-resonance frequency excites the drillstring, a significant force must be applied to displace the bit. It is therefore possible to choose a rotational speed to obtain additional dynamic force at the bottom hole.

Excitation of such system by some displacement generates a force response, the level of which depends on the frequency of the excitant parameter. This means, that the bit's force reaction depends on the frequency of the applied displacement. Varying rotational speed of the drillstring over the periodical lobed bottom hole changes the frequency of the bit displacement, thus varying the force applied to the bit.

The parameter that characterizes resistance of the drillstring to displacement under an applied force is called the mechanical input impedance Z(w) of the drill-string, and calculated by

$$\overline{Z}(\omega) = \frac{\overline{F}(\omega)}{\overline{V}(\omega)} \qquad (1)$$

where F(ω) is the excitation force, and v(ω) is the displacement velocity at bit face, with all parameters being complex quantities, and $\omega = 2\pi f$ the circular frequency.

The diagram shown in the Figure 1 presents the simplified drillstring as a rod that features even cross-section along the full length, with an excitation force applied at the left end, and an input impedance at the right hand side showing the lumped representation of swivel, traveling block and drawworks. For the actual calculations though, an indefinite number of drillstring sections and sizes can be used. Currently, the model is only valid for almost vertical wells, with low borehole wall friction. Although the theoretical incorporation of deviated wells is relatively simple, the estimation of the input parameters is not and therefore set aside for the time being.

The mechanical impedance (or dynamic rigidity) of a drillstring at the drill bit must be high to generate a force high enough to breakdown rock. Such a force is created when the exciting frequency or frequency of displacement at the bit, due to the bit rolling over the lobed bottom hole, is close to anti-resonance of the drillstring. The resistance of the drillstring to displacement is then very high, hence the force applied to the rock is very high. When the force reaches the rock breakdown threshold rock destruction occurs. High mechanical input resistance can be achieved through means of controlling the rotational speed, or changing the bottomhole assembly to accommodate maximum rigidity.

Conversely, when the exciting frequency happens to be close to the resonance of the drillstring, its cross-sections suffer significant oscillations (displacements) without any essential rock breakdown on the bottom hole as the dynamic force at the bit is small.

In case of high-level vibrations, the compression waves travel along the drillstring, converting energy into heat hence the power spent on drillstring swing is either useless or in the worst case harmful damaging downhole instruments. The ultimate goal is to direct this energy back to the bit by reflecting the vibrations to the point of origin: the bottomhole.

Based on the above insights, let's have a closer look at two cases. It is important to distinguish between the input impedance Z(ω), i.e. the ratio between dynamic force and the displacement velocity, the phase shift φ between those two, and the undulating bottomhole. Figure 2a shows the dynamic forces for two different rpm values (60 and 80 rpm) only slightly "out of sync", or phase shifted with the bottomhole structure, whereas Figure 2b depicts an almost 90 degree phase shift between the force and the bottomhole. It is evident, that in the first case the positive dynamic forces (i.e. pointing downhole) are the highest at or close to the top of the bottomhole lobes, thus maximizing the rock destruction process. In the latter case, the dynamic forces are high when the roller cones hit the troughs, thus unfavorably increasing the amplitude of the lobed bottomhole. This will ultimately lead to higher vibration levels and lower penetration rates.

Resonance and anti-resonance are simple physical phenomena. If you excite a rod with the length of a quarter the excitation frequency wave length, the longitudinal wave reflected from the free end comes to the point of excitation with a phase opposite the exciting phase. This is the anti-resonance case, the rod virtually "stiffens up". If the end of the rod is rigidly fixed the reflected wave comes in phase with the exciting frequency. This is the resonance case. The bottomhole assembly (BHA), like any pipe, rod, or arrangement of different pipes, possesses an infinite number of resonance and anti-resonance regions. Of those, only the first few have significance, as the vibration modes of higher order are subject to strong damping.

The above said, the installation point for a wave separator can thus be given by

$$L_{inst} = \frac{v_A}{4f}(2n-1) \qquad (2)$$

with n=1 for rotary drilling.

Hence, the variation of the input impedance of a drillstring allows the control of the dynamic load. In most cases, at the maximum input impedance of a drillstring a higher rate of penetration (ROP), and total metrage per bit is observed. The input impedance may be varied by different methods [7]:

1. Full or partial wave separation of a drillstring. This is the full or partial reflection of an incident longitudinal plane wave in a particular section of a drillstring. Reflection can be achieved by using either special drillstring components, or certain combinations of drill pipes. One of these drillstring separator designs features a jar with a 1.5m rod stroke. The rod moves inside a sealed mandrel, filled up with air or oil, and provides a maximum wave reflection coefficient at the boundary of the two media, i.e. metal/air (fluid).
2. Some type of shock absorbers, installed at a calculated distance from the bit, can also perform the function of a wave separator or change the frequency response of drillstring. High-strength shock absorbers must be used because of high dynamic forces acting on it while exciting the drillstring with frequencies (bit revolutions) exhibiting anti-resonance.
3. Combinations of drillpipes from various materials or with different cross-sectional areas could serve as a wave separator too. These combinations can be as follows:
   (a) Drill collars: steel or aluminum alloy drillpipes
   (b) Steel drillpipes: aluminum alloy drillpipes
   (c) Drill collars or conventional steel drillpipes: alternation of aluminum alloy drillpipes and drill collars.

A computer program was developed to do two things. Firstly, to demonstrate the validity of above assumptions by evaluating the ROP based on the mechanical input impedance in dependence of rpm. Secondly, to help design, test, and demonstrate a BHA optimized in regards to resonance behavior.

Fig. 3 shows the mechanical input impedance at the bit face across a BHA run from 2128-2364m averaging a ROP of 3.4 m/h. The actual rpm logs is also plotted, and clearly shows the varying values of rpm intersecting resonance and anti-resonance areas for revolutional speeds between 100-130rpm. The frequent rpm change intersecting resonance and anti-resonances areas makes this example well suited to demonstrate potential ROP gains when operating within preferred rpm regions as calculated by the model, as opposed to operating in undesirable rpm ranges. It is worthwhile to note, that the total percentage spent drilling in either anti-resonance, or resonance rpm is less than 5%. This is believed to indicate an experienced driller who does tend to intuitively avoid drilling in these rpm ranges, to avoid high torque levels, or high vibration levels respectively.

Fig. 4 is a screenshot of the performance analysis of that interval based on the resonance model introduced. Based on the presented model, the following values were calculated (Tab.1). The ROP change in the interval under investigation ranges from almost +14.6% ROP improvement while operating in anti-resonance areas, or desirable rpm areas to about -5.2% when drilling in undesirable rpm ranges.

Also, rpm areas generating excessive dynamic drillstring forces (Fig.5) are being investigated. It is quite obvious that the highest dynamic forces are experienced in the BHA, and that these forces grow with increased rpm, and decrease, due to internal friction, when traveling up the drillstring.

Based on these results, the next step is to further improve performance by utilizing the different wave separator methods introduced above.

Torsional Vibrations

The torsional resonance drilling part investigates a technique to improve penetration rates when drilling very low permeability rocks using drag (PDC) bits. This part looks into the beneficial effects of small axial vibrations in order to decrease the energy dissipated by friction at the bit-rock interface.

This theoretical investigation is based on a new model developed at the University of Minnesota [8], which takes into consideration not only the axial and torsional modes of vibration, but also the coupling between these two modes through bit-rock interaction laws, which account for both frictional contact and cutting processes at the bit-rock interface [9], see Fig. 6.The interaction between bit and rock can be described in four modes:

(a) Cutting, the “normal” case,
(b) Sticking, when the bit rest immobile,
(c) Sliding, with the depth of cut being zero,
(d) Off-bottom.

In the normal” case, either the wearflats could be in contact with the rock or the frictional contact taking place at the wearflats/rock interface vanishes when the bit moves (temporarily) upward, but the bit is still cutting rock. At that moment, the power provided by the drillstring to the bit is transferred towards the cutting process and the system gains in efficiency

The delayed and coupled nature of this interaction is ultimately responsible for the occurrence of self-excited vibrations, which can degenerate into stick-slip oscillations under certain conditions. Figure 7 shows the evolution of the bit RPM (ω), ROP (v), WOB (W) and depth of cut per revolution (δ) during four limit cycles for a typical example. The angular velocity ω is characterized mainly by a mono-frequency signal (corresponding essentially to the natural torsional frequency of the drill pipes) and looks “simple" compared to the other variables, which are display a widespread and higher frequency content as a consequence of the losses of contact occurring at the wearflat-rock interface.

Because of its periodicity, the signal of the depth of cut suggests the formation of a repetitive bottom hole pattern (see Fig. 8), which encourages the self-excited vibrations.

The features of the torsional vibrations that are predicted with this model are well in accordance with field observations. Furthermore, the results reveal that the model predicts apparent rate effects as an inherent outcome of the nature of the bit-rock interface. This approach is in contrast with the classical model where the self-excited torsional oscillations of a rotary drilling system with a drag bit are analyzed by considering only the torsional mode of vibrations of the drill string and by reducing the bit-rock interface to an equivalent frictional contact with a velocity weakening friction coefficient. In other words, the often published weakening torque vs. RPM relationship, assumed in the classical approach to be an intrinsic property of the bit rock interaction, has to be considered not as cause but as a consequence of the self-excited vibrations and more precisely of the losses of contact occurring at the wearflat/rock interface.

The magnitude of the high frequency bit axial vibrations must not reach a threshold beyond which the bit starts to bounce, i.e. when the cutter loose complete contact with the rock, because of the potential severe damage to the bit. it is of interest to determine the conditions when the bit is bouncing, which will be referred to as the K-regime. Figure 9 shows a map depicting the K-regions, as well as the stable regions for different values of the RPM (ω_0) in terms of the nominal weight-on-bit W_o and the bit bluntness λ. The gray region of Fig. 9 corresponds to the K-regime and the white one to the stability region where the contact is lost intermittently between the rock and the wearflats (but with the cutters remaining engaged with the rock). The black region corresponds to the case where the bit is not drilling, but is only in frictional contact. By superimposing the maps obtained for different values of RPM (ω_0), we can determine the evolution of the stability zones with respect to ω_0.

Figure 10 shows how the depth of cut per revolution (δ/λ) varies with the WOB (W_o/λ) for various imposed RPM (ω_o). Increasing the RPM progressively eliminates the axial vibrations, which cause a reduction of the energy available to cutting rock and an increase of the energy dissipated in frictional contact. Increasing the WOB is associated with increasing energy transferred into the axial mode of vibration, and thus increasing amplitude of vibrations, which under certain conditions cause bit bouncing. By increasing the ratio W_o/λ, the energy devoted to the cutting process increases compared to the energy dissipated by friction. Therefore, the amplitudes of the self-excited vibrations enhance the drilling process up to a certain point where the bit starts to bounce, i.e. when the bit loses entirely contact with the rock, not only at the wearflats/rock interface but also at the cutting faces of the cutters.

Fig. 11 shows the promising ROP increase (normalized by the rate of advance ROP_0 in the absence of any vibrations) with the structural parameter ψ (function of the mass of the BHA, and the length of the drillstring) all other parameters remaining constant.

Conclusions

The main contribution of this work is the controlled application of vibration to enhance bit performance. The two distinct approaches are:

- Using resonance vibrations calculations to improve ROP by adjusting the drillstring revolutions, and potentially reflect dynamic load waves back to their source, i.e. the bit, through implementation of an additional drillstring component such as a shock absorber or incorporating different drillstring materials.
- Investigating the positive effects of loosing contact across the chamfers so as to utilize all energy for the cutting process. Parametric maps have been developed to demonstrate quantitative relationships.

Nomenclature

n	positive integer [dimensionless]
m_{BHA}	mass bottomhole assemble [kg]
v_A	longitudinal wave velocity, [m/s]
I	mass moment of inertia [kgm^2]
c_{Tor}	torsional spring constant [kgm^2/s]
ϵ	specific energy [Pa]
φ	Phase shift between F and v

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Figures & Tables

rpm Range	Preferable	Undesirable	Anti-Resonance	Resonance
Percentage of BHA run [%]	32.7	62.4	4.0	0.9
Average ROP [m/h]	3.6	3.2	3.8	3.3
ROP Changes [%]	+8.1	-5.2	+14.6	-0.8

Tab.1, ROP Performance Analysis

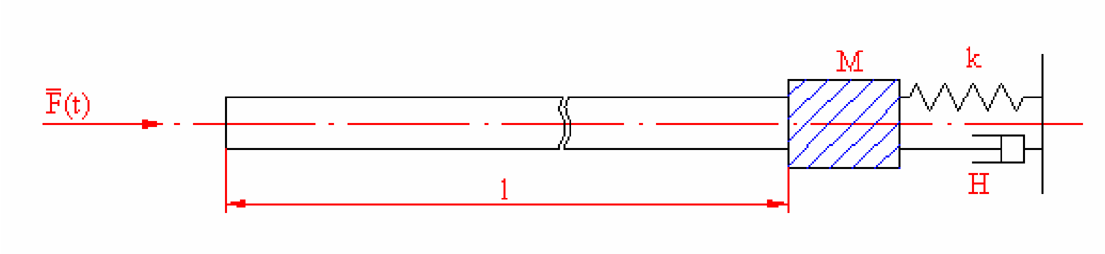

Figure 1, Drillstring model

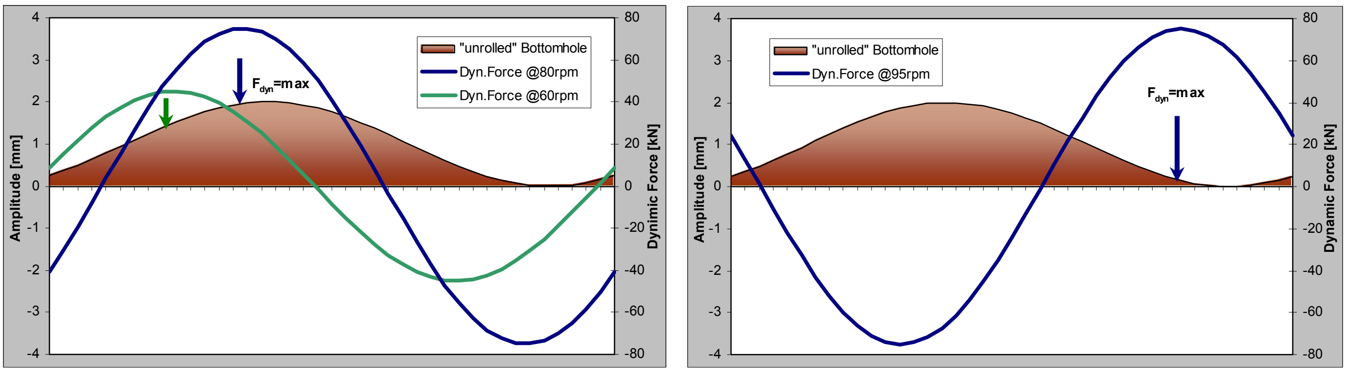

Figure 2a and 2b, Rock Destruction in dependence of Phase Shift

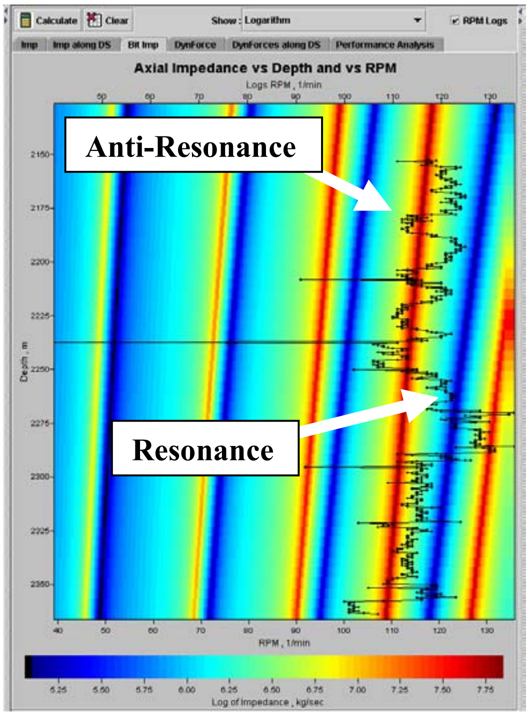

Figure 3, Mechanical Input Impedance at Bit Face

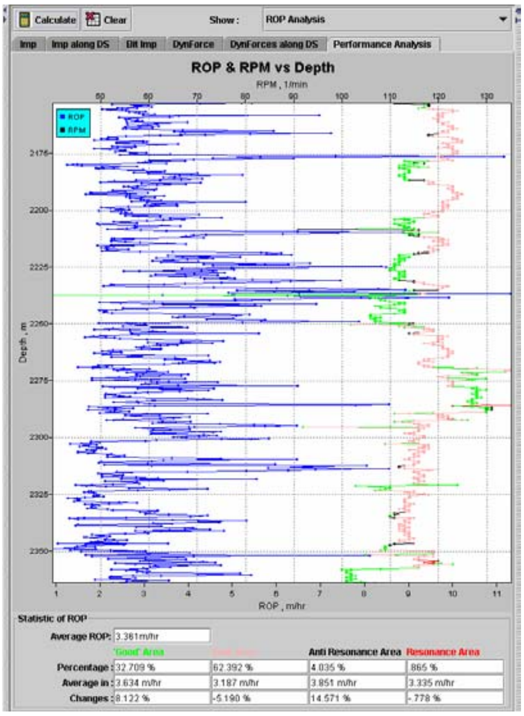

Figure 4, ROP Evaluation

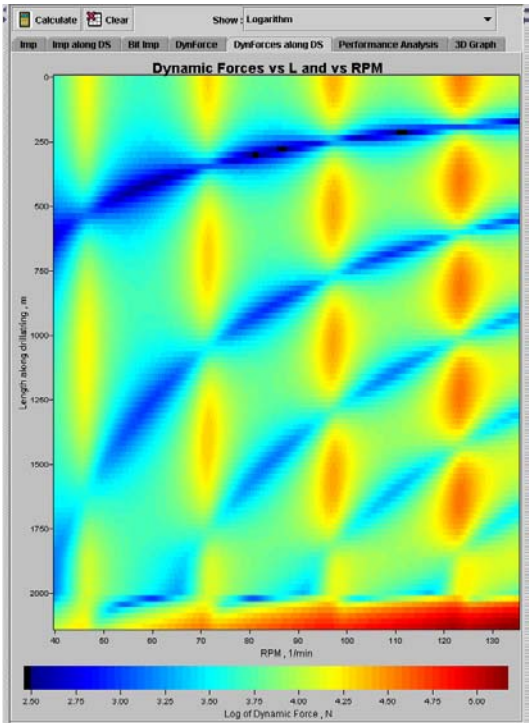

Figure 5, Dynamic Drillstring Forces

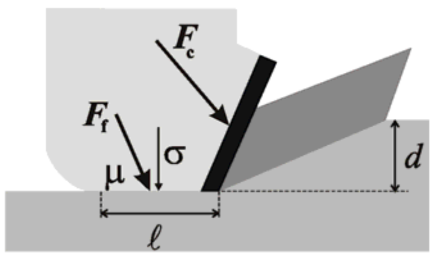

Figure 6, Frictional and Cutting Force Acting on a Cutter [9]

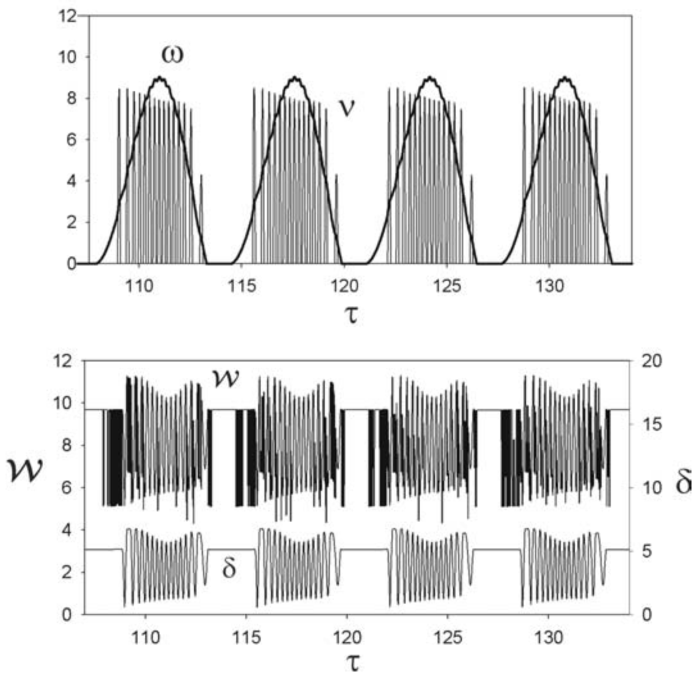

Figure 7, Evolution of the bit RPM (ω), ROP (*v*), WOB (*W*), and depth of cut per revolution (δ)

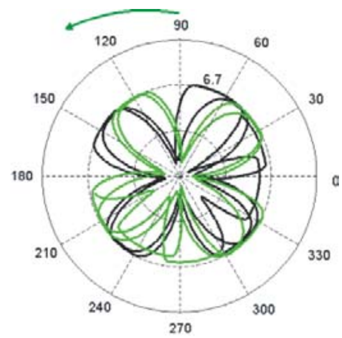

Figure 8, Pattern of Depth of Cut encountering Stick-Slip Torsional Vibrations

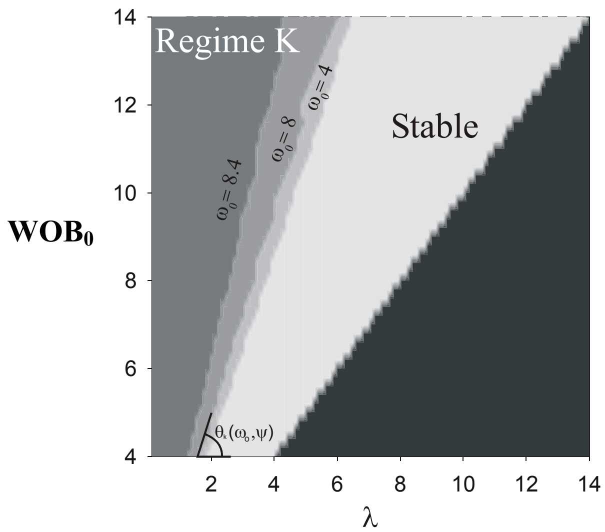

Figure 9, Parametric Stability Map

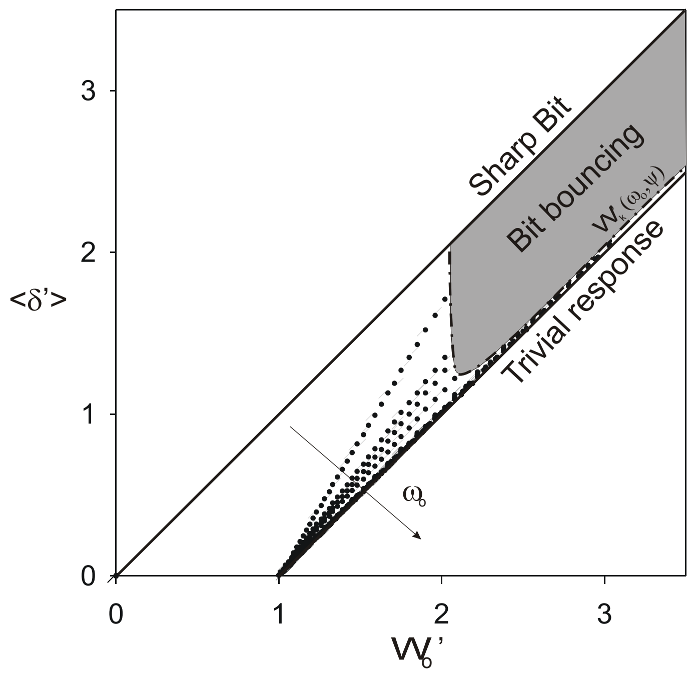

Figure 10: Depth of cut per revolution (δ/λ) versus WOB (W_o/λ) for various rpm

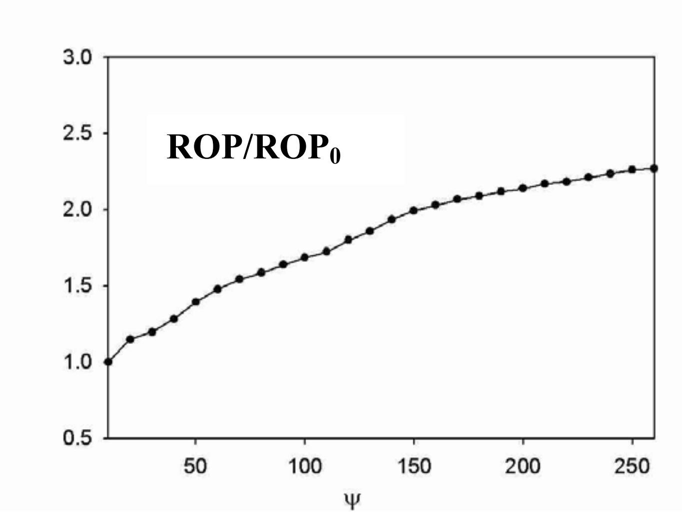

Figure 11, Normalized ROP Increase with Bit Geometry