IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684, p-ISSN: 2320-334X
PP 60-74
www.iosrjournals.org

A study of Vibration Analysis for Drill String Using Finite Element Analysis

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ABSTRACT: *With energy consumption and the development of exploitation technology of oil and gas underground, the drilling engineering has developed toward the direction of deep well .The problem of fatigue of drill string has a great effect to drilling engineering. Strong vibration is one of the main reasons that induce drill string fatigue. This paper contains the study about vibration analysis for drill string considering with and without damping effect using Finite Element Analysis (FEA).The aim of this paper is to apply ANSYS software to determine the natural vibration modes and forced harmonic frequency response for drill string. This analysis is to find the natural frequency and harmonic frequency response of drill string in order to prevent resonance for drill string. From the result, this analysis can show the range of the frequency that is suitable for drill string which can prevent maximum amplitude and stress.*

I. INTRODUCTION

Drill strings are used in drilling for oil and gas. A drill string is essentially a long pipe with a drill bit at its lower end. Drill strings experience instability leading to excessive vibrations causing bit failure. Since a drill string is essentially a long series of pipes connected together with a bit at the end, it behaves as a string; hence, the name "drill string." Bits come in many varieties, but can be loosely classified as roller-cone and Polycrystalline Diamond Compact (PDC) bits. The flexibility of drill strings is the root cause of excessive vibrations [1]. These vibrations take place in three-dimensional space; namely axial, torsional (including bit whirl), and lateral [2]. Lateral vibrations are controlled through track bits and the placement of supports close to the bit. Torsional vibrations including bit whirl are the most destructive because they lead to bit movement in a backward direction relative to the rock [3] – [6]. This results in breakage of the cutters. Replacing the bit requires the removal of what might be a mile-long drill string, a costly process. It has been shown that torsional vibrations can be controlled by controlling axial vibrations [7]. This is because controlling axial vibrations keeps the bit firmly against the rock and does not allow the torsional energy to be released. Control of axial vibrations can accomplish by employing a shock absorber.

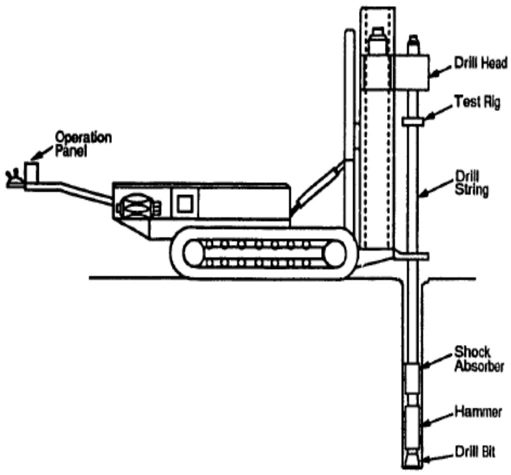

Figure 1. Drilling machine

Reference [12] show that the study of natural frequency, consider a beam fixed at one end and having a mass attached to the other, this would be a single degree of freedom (SDoF) oscillator. Once set into motion it will oscillate at its natural frequency. For a single degree of freedom oscillator, a system in which the motion can be described by a single coordinate, the natural frequency depends on two system properties; mass and stiffness. The circular natural frequency, ωn, can be found using the following equation:

$$\omega_n^2 = k/m$$

Where:
k = stiffness of the beam
m = mass of weight
ωn = circular natural frequency (radians per second)
From the circular frequency, the natural frequency, fn, can be found by simply dividing ωn by 2π. Without first finding the circular natural frequency, the natural frequency can be found directly using:

$$f_n = (1/2\pi)(k/m)^{1/2}$$

Where:
fn = natural frequency in hertz (1/seconds)
k = stiffness of the beam (Newton/Meters or N/m)
m = mass of weight (kg)
For the forced harmonic frequency, the behavior of the spring mass damper model need to add a harmonic force in the form below. A force of this type could, for example, be generated by a rotating imbalance.

$$F = F_0 \cos(2\pi f t).$$

Then, the sum the forces on the mass are calculate using following ordinary differential equation:

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos(2\pi f t).$$

The steady state solution of this problem can be written as:

$$x(t) = X \cos(2\pi f t - \phi).$$

The result states that the mass will oscillate at the same frequency, f, of the applied force, but with a phase shift φ.
The amplitude of the vibration "X" is defined by the following formula.

$$X = \frac{F_0}{k} \frac{1}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}}.$$

Where "r" is defined as the ratio of the harmonic force frequency over the undamped natural frequency of the mass–
Spring–damper model

$$r = \frac{f}{f_n}.$$

The phase shift, φ, is defined by following formula. The base

$$\phi = \arctan\left(\frac{2\zeta r}{1-r^2}\right).$$

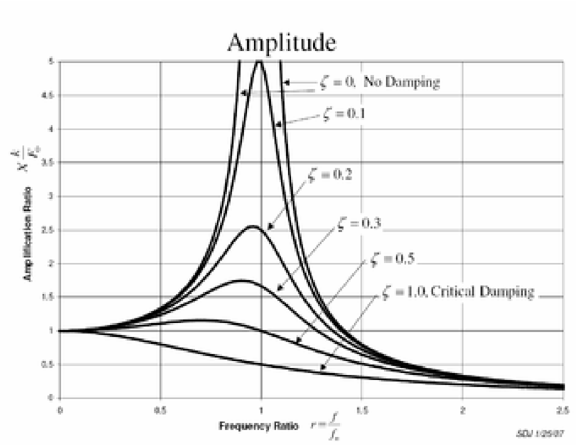

Figure 2. The frequency response of the system

The plot of these functions, called "the frequency response of the system", presents one of the most important features in forced vibration. In a lightly damped system when the forcing frequency nears the natural frequency (r=1) the amplitude of the vibration can get extremely high. This phenomenon is called resonance (subsequently the natural frequency of a system is often referred to as the resonant frequency). In rotor bearing systems any rotational speed that excites a resonant frequency is referred to as a critical speed.

If resonance occurs in a mechanical system it can be very harmful – leading to eventual failure of the system. Consequently, one of the major reasons for vibration analysis is to predict when this type of resonance may occur and then to determine what steps to take to prevent it from occurring. As the amplitude plot shows, adding damping can significantly reduce the magnitude of the vibration. Also, the magnitude can be reduced if the natural frequency can be shifted away from the forcing frequency by changing the stiffness or mass of the system. If the system cannot be changed, perhaps the forcing frequency can be shifted (for example, changing the speed of the machine generating the force).

The following are some other points in regards to the forced vibration shown in the frequency response plots.
At a given frequency ratio, the amplitude of the vibration, X, is directly proportional to the amplitude of the force $F0$ (e.g. if double the force, the vibration doubles)

• With little or no damping, the vibration is in phase with the forcing frequency when the frequency ratio $r < 1$ and 180 degrees out of phase when the frequency ratio $r > 1$

• When $r < 1$ the amplitude is just the deflection of the spring under the static force $F0$. This deflection is called the static deflection δst. Hence, when $r < 1$ the effects of the damper and the mass are minimal.

• When $r > 1$ the amplitude of the vibration is actually less than the static deflection δst. In this region the force generated by the mass ($F = ma$) is dominating because the acceleration seen by the mass increases with the frequency. Since the deflection seen in the spring, X, is reduced in this region, the force transmitted by the spring ($F = kx$) to the base is reduced. Therefore the mass–spring–damper system is isolating the harmonic force from the mounting base–referred to as vibration isolation. Interestingly, more damping actually reduces the effects of vibration isolation when $r > 1$ because the damping force ($F = cv$) is also transmitted to the base.
This analysis is to find the natural frequency and harmonic frequency response of drill string in order to prevent resonance for drill rod. From the result, this analysis can show the range of the frequency that is suitable for drill string which can prevent maximum amplitude.

II. DESIGN OF DRILL STRING ASSEMBLY

A. In drilling machine bull gear is used to rotate the drill rod main shaft. Its Specification is: Radius =278 mm, Thickness = 70 mm.

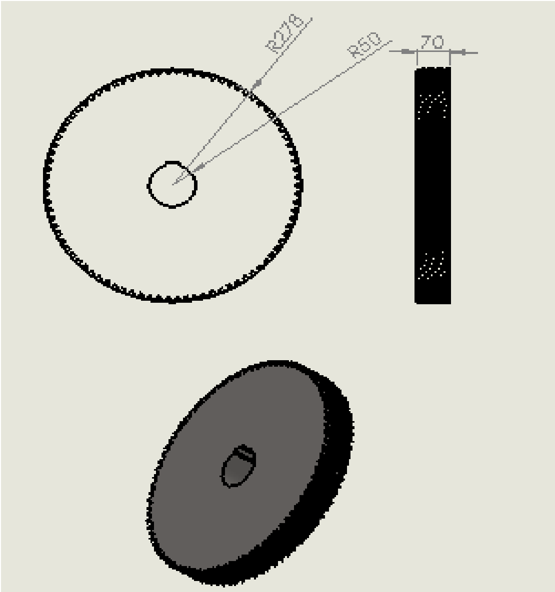

Figure 3: bull gear

B. The bull shaft is connected with bull gear. It transmits the power to drill string. Its Specification is: Length -508 mm, dia-140 mm

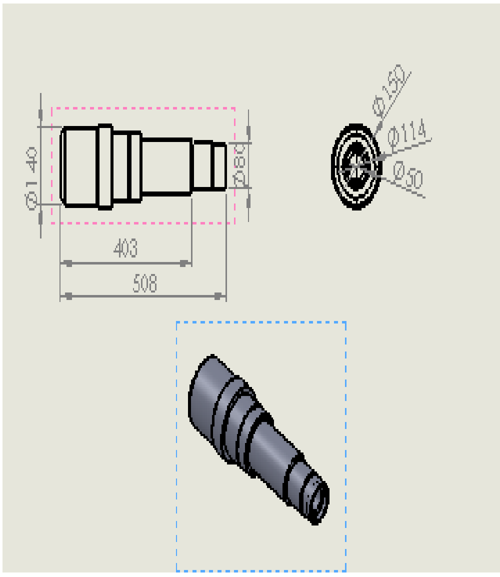

Figure 4: Bull shaft

C. Taper roller bearing is used in drill head gear box. Because it can carry high axial and radial loads. Its Specification is: OD- 187 mm, ID - 90 mm, Thickness - 46 mm

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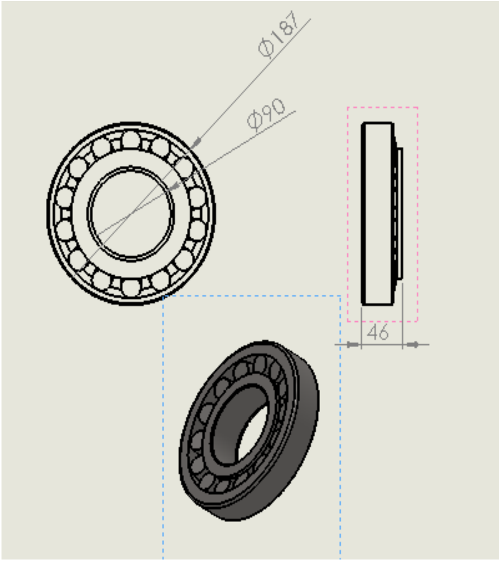

Figure 5: taper roller bearing

D. The drill rod upper end is connected with bull shaft and bottom end is connected with drill bit. Its specification is:
Drill rod: length = 5902mm, OD = 134
Drill bit: length = 359 mm, OD = 266

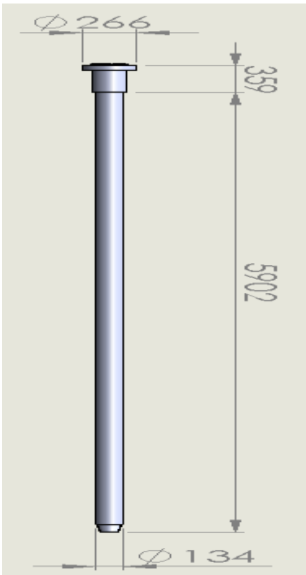

Figure 6: drill rod with drill bit

E. In drilling penetration is achieved by the repeated application of a large impulsive force to a rotating rock drill bit in contact with the rock.

Figure 7: drill rod assembly

III. MESH STRATEGY

The details of mesh strategy are defined in Table 1 and Figure 8.An appropriate mesh is selected to make sure this meshing can solve in 1 hour duration. This mesh is applied to whole object as one body meshing.

Table 1: Details of meshing strategy

Object Name	*Mesh*
State	Solved
Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Use Advanced Size Function	Off
Relevance Center	Coarse
Element Size	Default
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge Length	2.6741e-004 mm
Statistics	
Nodes	60453
Elements	28433
Mesh Metric	None

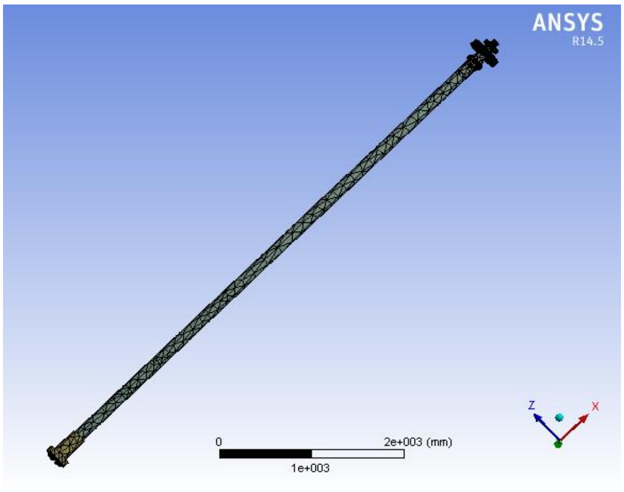

Figure 8: Actual mesh of drill string

IV. BOUNDARY CONDITION AND APPLIED LOAD

This section described the details of applied load and boundary condition of natural vibrations and harmonic analysis.

A. Natural Vibration Analysis

A modal analysis is performed with number of modes is 10.The details of the support is in Table 2 and Figure 9.

Table 2: Details of boundary condition

Object Name	Fixed Support	Frictionless Support
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	2 Faces	1 Face
Definition		
Type	Fixed Support	Frictionless Support
Suppressed	No	

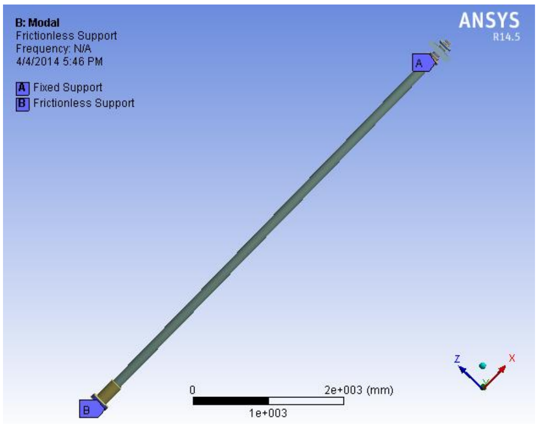

Figure 9: Actual fixed support on top and frictionless support at bottom

B. Harmonic Frequency Response Analysis

In the harmonic frequency response analysis, the fixed support is exactly same condition in Figure 8.In this analysis, 2MPa pressures is applied to the upper half of the drill bit on one side of the drill string and to the lower half of the other side for a frequency range from zero to 3 times the frequency of the tenth vibration mode. This 2MPa pressure is applied normal to the surface according to the Table 3 and Figure 9.

Table 3: Details of applied pressure and fixed support

Object Name	*Force 2*	*Pressure*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	1 Face	
Definition		
Type	Force	Pressure
Define By	Vector	Normal To
Magnitude	-21218 N	2.0684 MPa (step applied)
Phase Angle	0. °	
Direction	Defined	
Suppressed	No	

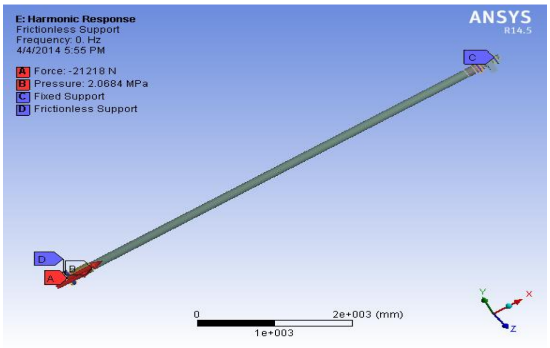

Figure 10: The actual applied load in drill string.

V. RESULT

These results for natural vibration analysis and harmonic frequency response analysis is done using ANSYS 14.5

A. Result of Natural Vibration Analysis

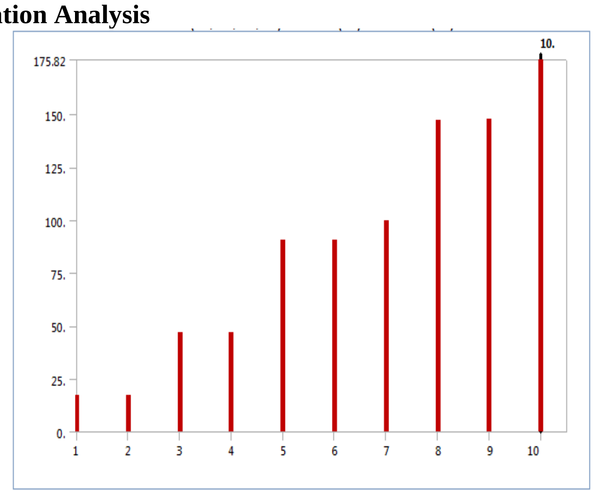

Figure 11:Result of frequency corresponding to 10 modes for normal Vibration analysis.

From these result, 10 lowest vibration frequencies are:

Table 4: 10 lowest frequencies for natural vibration analysis

Mode	Frequency [Hz]
1.	17.161
2.	17.19
3.	46.727
4.	46.8
5.	90.445
6.	90.56
7.	99.986
8.	147.41
9.	147.6
10.	175.82

B. Result of Harmonic Frequency Response Analysis without Damping

In this harmonic frequency response analysis, frequency range need to be set up from zero to 3 times the frequency of the tenth vibration mode. In Table 4, tenth vibration mode is 175.82 Hz.

3×the frequency of the 10 *th* vibration mode
= 3× 175.82
= 527 Hz
From this result, 0-527 Hz frequency range is applied.

Table 5: Applied frequency in Harmonic Frequency Response Analysis

Object Name	*Frequency Response*	*Phase Response*
State	Solved	
Scope		
Scoping Method	Geometry Selection	
Geometry	1 Face	
Spatial Resolution	Use Average	
Definition		
Type	Directional Deformation	Normal Stress
Orientation	X Axis	
Suppressed	No	
Options		
Frequency Range	Use Parent	
Minimum Frequency	0. Hz	
Maximum Frequency	527. Hz	
Display	Bode	
Frequency		183. Hz
Duration		720. °
Results		
Maximum Amplitude	1.8343 mm	
Frequency	179.18 Hz	
Phase Angle	180. °	0. °
Real	-1.8343 mm	97.708 MPa
Imaginary	0. mm	0. MPa
Amplitude		97.708 MPa
Reported Frequency		179.18 Hz

All the result is from one vertex as in the Table 5.This point 179.18 HZ is selected because this point is the maximum displacement and maximum stress occurred in the Figure 12 & 13.The Table and figure value shows that maximum amplitude at 1.8343 mm and maximum stress at 26706 Mpa.

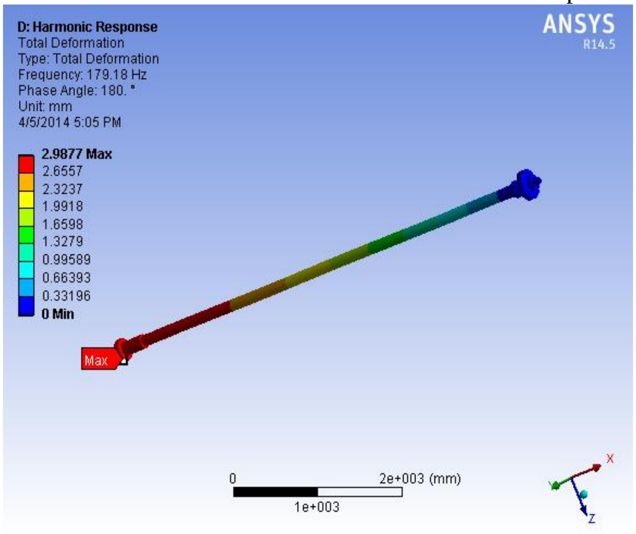

Figure 12: Deformation Analysis without damping

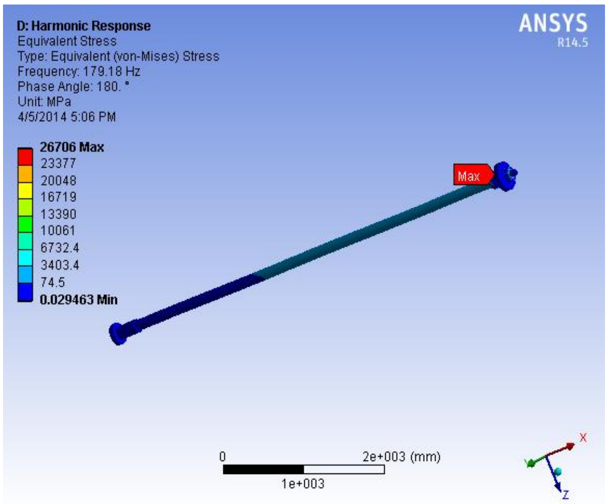

Figure 13: Stress Analysis without damping

Table 6: Harmonic Frequency Response Analysis

Tabular Data

	Frequency [Hz]	Amplitude [mm]	Phase Angle [°]
1	10.54	6.7233e-002	0.
2	21.08	6.8013e-002	0.
3	31.62	6.9333e-002	0.
4	42.16	7.1271e-002	0.
5	52.7	7.3919e-002	0.
6	63.24	7.7427e-002	0.
7	73.78	8.201e-002	0.
8	84.32	8.8e-002	0.
9	94.86	9.5833e-002	0.
10	105.4	0.10644	0.
11	115.94	0.12113	0.
12	126.48	0.1425	0.
13	137.02	0.17597	0.
14	147.56	0.23319	0.
15	158.1	0.36491	0.
16	168.64	0.8786	0.
17	179.18	1.8343	180.
18	189.72	0.43385	180.
19	200.26	0.24178	180.
20	210.8	0.16576	180.

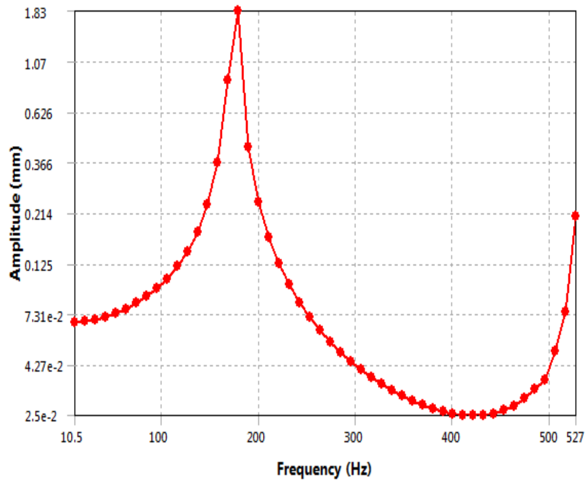

Figure 14: Details of X-axis result for directional deformation

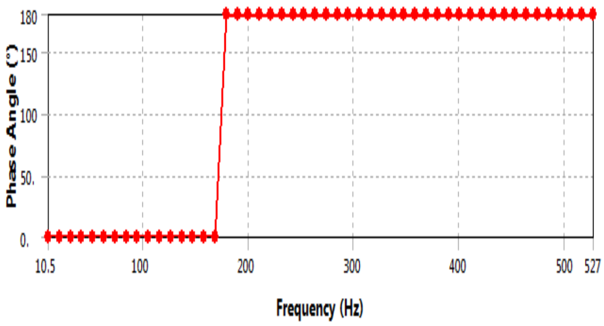

Figure 15: Details of X-axis result for phase angle

C. Result of Harmonic Frequency Response Analysis with Damping

From logarithmic decrement of displacement value, 0.2 damping ratio is selected for vibration control. The same frequency range is set for 0 – 527 HZ.

Table 6: Applied damping ratio in Harmonic Frequency Response Analysis

Object Name	*Analysis Settings*
State	Fully Defined
Options	
Range Minimum	0. Hz
Range Maximum	527. Hz
Solution Intervals	50
Solution Method	Mode Superposition
Cluster Results	No
Modal Frequency Range	Program Controlled
Store Results At All Frequencies	Yes
Output Controls	
Stress	Yes
Strain	Yes
Nodal Forces	No
Calculate Reactions	Yes
General Miscellaneous	No
Damping Controls	
Constant Damping Ratio	0.2

Table 7: Harmonic Frequency Response Analysis with damping

Object Name	*Frequency Response*
State	Solved
Scope	
Scoping Method	Geometry Selection
Geometry	1 Face
Spatial Resolution	Use Average
Definition	
Type	Directional Deformation
Orientation	X Axis
Suppressed	No
Options	
Frequency Range	Use Parent
Minimum Frequency	0. Hz
Maximum Frequency	527. Hz
Display	Bode
Results	
Maximum Amplitude	0.17886 mm
Frequency	168.64 Hz
Phase Angle	-79.434 °
Real	3.2796e-002 mm
Imaginary	-0.17583 mm

All the result is from one vertex as in the Table 5.This point 168.64 HZ is selected because this point is the maximum displacement and maximum stress occurred in the Figure 16 & 17.The Table and figure value shows that maximum amplitude at 0.178 mm and maximum stress at 2417 Mpa.

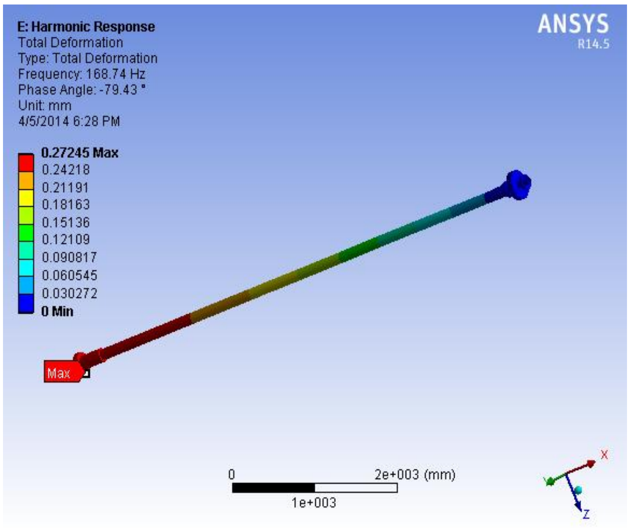

Figure 16: Deformation Analysis with damping

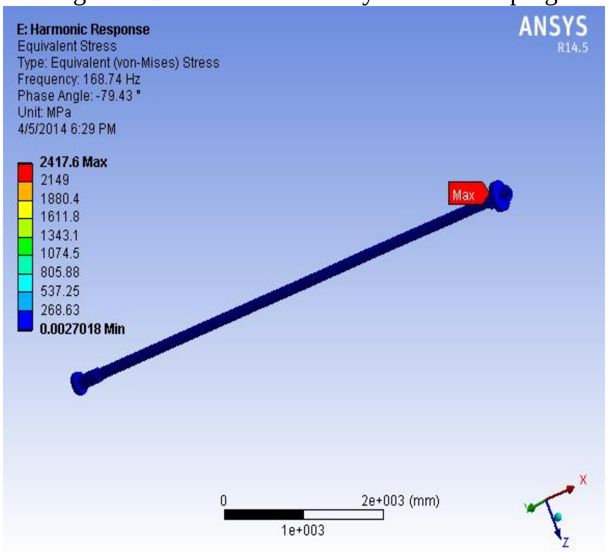

Figure 17: Stress Analysis with damping

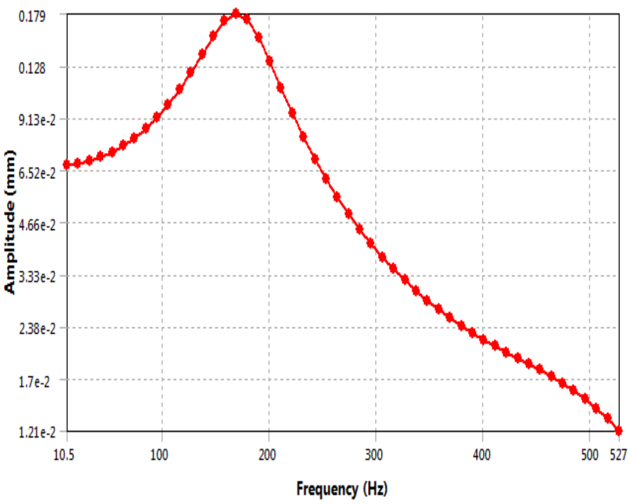

Figure 18: Details of X-axis result for directional deformation

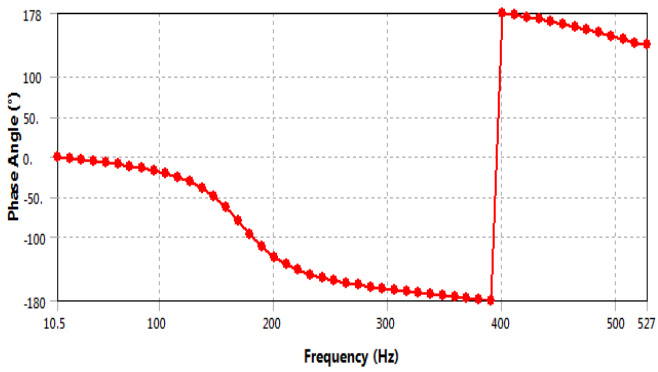

Figure 19: Details of X-axis result for phase angle

VI. CONCLUSION

From Figure 12 until Figure 19, the conclusion is:

(a) In this analysis, pressure and force is applied to drill string as in Figure 9 as a normal to that surface. This is meaning that force is mainly applied to X-axis. Due to this reason, only result for X-axis is more considerable in this harmonic analysis.

(b) For the X-axis, the first maximum amplitude for normal stress and directional deformation are happen at 179.18 Hz. At this frequency, the resonance is occurred.

(c) In this analysis, first resonance is happen when the ratio of harmonic forced frequency over natural frequency is,

r = resonance in harmonic forced frequency/ modal natural frequency

= 179.18/175.82

= 1.019 ≈ 1

(d) In order to prevent the resonance, frequency ratio need to be setup to be less than 1.When r<<1 the amplitude is just the deflection of the spring under the static force F0. This deflection is called the static deflection δst. Hence, damping effect is included in this analysis.

(e) In this study, damping ratio can set to 0.2 from the Harmonic frequency analysis in order to prevent resonance. so,

= 168.74/175.82

= 0.9 < 1

At this condition amplitude change from 1.83 to 0.178 mm, and stress also reduced at 26706 to 2417 Mpa.

(f) As a result of vibration data analysis it is conduced that vibration controlled through a damping properties- rather than drilling variables.

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