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Simulation of Drill Pipe Lateral Vibration Due to Riser's Oscillation

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Abstract. This paper attempts to explain the motion behaviour of the marine riser coupled to a drill string when the vortex induced vibration (VIV) is involved. Vibrations have been reported to have a major effect on the drilling performance, affecting the rate of penetration (ROP), causing severe damages to the drilling tools and also reduces the efficiency of the drilling process. There are two major components of drilling tools that are subjected to vibration, namely the marine riser and the drilling string. Analysis of vibration in the marine riser and drill string are two topical areas that have individually received considerable attention by researchers in the past. Though these two subjects are interrelated, borne by the fact that the marine riser encapsulates and protects the drill pipe, there have been few attempts to investigate them together as a unity. Due to the complexities of the models, simplified assumptions were made in order to undertake the investigation by using staggered approach. The results were compared with the experimental and simulation data from the open literature. It was found that the maximum displacement with negative damping occurs at low frequency and rotation speed.

Introduction

Drilling in deepwater presents a significant challenge, Drillstring failures like washout and twist-off can result in expensive non-productive time and loss of equipment. Analysis by the Norwegian oil industry showed that economic losses exceeding 150 million USD in a 10-year period, distributed over 187 cases [1]. Drillstring vibrations are extremely complex because of the random nature of a multitude of factors such as: bit/formation interactions, drill string/wellbore interactions, and hydraulic function. In many cases, it becomes the main cause of premature failure of drill string components and drilling inefficiency. The presence of hydrodynamic pressure, the sea current and the tidal wave further complicate the analysis due to fatigue. In most operation, a riser is needed to protect the drill pipe from external environment and stabilize the drill pipe when the drilling is performed. Despite extensive research in the last four decades, questions remain unanswered because of the complexity in the compounded parameters. [2], [3]

The vortex-induced vibration (VIV) cannot be separated in the design of deepwater riser systems. The VIV can produce a high level of fatigue damage on the riser when exposed to severe current environments in a relatively short period of time [4]. The objective of the current paper, is to explain qualitatively the observed phenomenon of the vortex induced vibration effect at a long span riser to a drill string vibratory mode and the dynamic response.

Methodology

Riser's Vibration Due to Vortex. The vortex excited response is significant only when the structure is excited near the resonant condition. The only significant response is that of a pure mode whose natural frequency is close to the vortex shedding frequency. The excitation frequency f_n is given by :

$$f_n = \frac{v s_t}{D} \tag{1}$$

where s_t is the Strauhal's number, υ is the velocity of fluid, and D is the diameter of cylinder.

In this research, the vortex shedding was analyzed using a 2D fluid dynamic simulation in segmented depth from 1 meter to 500 meters on turbulent current model with a viscosity of 0.0015 N s/m^2. The riser is modelled in the segmented area according to a certain depth to analyse the drag forces that appear on the wall surface. The most widely applied turbulent models were standard K-ε and K-ε RNG model. In the standard K-ε model, the Eddy viscosity determined from a single turbulence length scale and the calculated turbulent diffusion occur only at the specified scale, whereas in reality all scales of motion will contribute to the turbulent diffusion. The operating condition can be seen in Table. 1

Table 1. Riser Operating Condition

Depth (m)	R_e	Operation Pressure (Pa)	Velocity (m/s)
1	1,065,644	1762.44	1.86
60	836,473	704434.00	1.46
100	624,490	1106510.00	1.09
500	486,988	5127230.00	0.85

The result of CFD analysis is the drag coefficient of the riser, the drag coefficient can be transformed into drag force using equation (2) and will be inputted as mechanical force in the mechanical vibration analysis

$$F_D = \tfrac{1}{2} C_D v^2 \rho A \quad (2)$$

where F_D is the drag force, ρ is the fluid density, C_D is the drag coefficient, A is the cross sectional area and v is the velocity.

Drilling String Vibration. Vibrations are extremely complex because of the random nature of a multitude of factors such as bit/formation interactions, drillstring/wellbore interactions, and hydraulics function. The drill pipe inside the riser was modeled in pipe beam 3D with six degrees of freedom, including translations and rotations in the x, y, and z directions. This model is suitable to simulate slender beam structure with large deformation and finite strain. The riser modeled as a shell element with six degrees of freedom with the same material properties as drill pipe.The dimension properties are shown in Table 2.

Table 2 Material properties and parameters used for drill pipe, riser and wellbore

	Drill Pipe	Riser
Wall Thickness, m	0.0083	0.01
Outside diameter, m	0.2032	0.5
Force, kN	100	
Length, m	550	500
Rotational velocity, RPM	80&300	

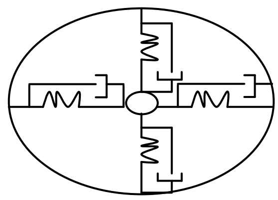

Fig. 1. Schematic of the mechanical model assumption for the coupled drillpipe riser vibrational analysis

Riser-Drill strings Coupling. In order to make the vibratory forces from the riser transmitted to the drillstring, an assumption is made so that the returning mud acts as a medium of force transmission thus providing the means for coupling. The mud was modeled as spring-dashpot elements that connect the drill pipe to the riser, shown in Fig. 1. Using the displacement model, the spring stiffness and damping coefficient can be obtained as follows. Under the influence of an external force ΔF, the drill pipe will be displaced by a distant of ΔL. The stiffness now may be defined as follows:

$$K_{mud} = \frac{\Delta F}{\Delta L} = \frac{\Delta PA}{\Delta V / A} = \frac{\Delta PA^2}{\Delta PV / E} = \frac{A^2 E}{AL} = \frac{AE}{L} \tag{3}$$

where E_{mud} is the bulk modulus of mud, V is the volume of mud, A is the virtual spring cross sectional area, L is the virtual spring length. The effect of mud including added mass and damping can be written, [5–7]:

$$\varsigma = \frac{1}{2}\left(\frac{M_{mud}}{C_m M_{mud} + M_{pipe}}\right) \operatorname{Im} H \tag{4}$$

M_{mud} is the displaced mud mass, H is the coefficient taken from Bessel's Diagram, C_m is the coefficient (3.7), M_{pipe} is the pipe mass.

Results and Discussion

Lateral Vibration. A mechanical modeling using the Finite Element Method is used for drillpipe-riser coupling at 500 meters depth. The drag force outputs from fluid dynamic analysis are used for mechanical force input at the riser surface. Modal analysis is performed to obtain the natural frequency of the system, this analysis using frequency range from 0 to 1000 Hz to obtain the most crucial first three natural frequencies and they can be seen in Table 3. Table 4 shows the drag forces gradually increase as the riser going deeper. The intensity of hydrodynamic force and the differentiation of density as the salinity contribute to the drag force rising. [8], [9]

Table 3. Natural Frequency result of lateral vibration

Source	1st mode(Hz)	2nd mode(Hz)	3rd mode(Hz)
Present Work	1.25	2.93	4.18
Zare *et al.* (2011)	1.23	3.14	4.19
Burgess *et al.* (1987)	1.25	3.15	NA

Table 4. Drag force summarizes

Depth (m)	Force (N)
1	109
60	4310
100	6827
500	7071

Harmonic analysis is performed to obtain the displacement and the harmonic response of the system. The analysis uses frequencies that range from 0 to 250 Hz with 50 frequency sub step. As in the real operation the drill pipe is rotated with several speeds to accommodate the penetration into the well bore. In this model the rotating speeds used are 80 and 300 RPM, to compensate the low and the medium speed at the drill pipe. The result of this analysis is shown in Table 5.

Table 5. Maximum displacement for riser and drill pipe

Rotation	Drill pipe displacement (cm)		Riser displacement (cm)	
(RPM)	0.5 Hz	5 Hz	0.5 Hz	5 Hz
80	4.12	2.05	4.137	2.1
300	0.14	0.19	0.97	1.27

Table 5 show at low RPM the displacement of both riser and drill pipe will rise. It clearly explains the system is under the vortex shedding resonance frequency. With the drill string as low as 80 RPM, the presence of vortex induced vibration will increase both the riser and drillpipe's displacement. The effect of vortex shedding decreases as the frequency and the speed increase. At 80 RPM both the displacement of drill pipe and riser experience their maximum displacement at 0.5 Hz. When the frequency of the system increases to 5 Hz the displacement shows a decreasing trend. Contrary with the 300 RPM speed, while the system frequency increases to 5 Hz the displacement also goes higher. This phenomena explain the resonance of the drillstring with the riser.

This result also recorded the displacement plot of the drill string inside the riser. The drill string profile tends to be shifted away from the riser axis. The energy of the force transferred to the drill string via spring-dot element and shifted the drill strings from its original position. The plots for drill string displacement are shown in Fig. 2 and 3. Fig. 2 shows that the drill strings shifts away from the riser axis. At 0.5 Hz frequency the vibration mode affects the drill string profile to shift at 4.12 cm, while at 5 Hz the displacement reduces up to 2.05 cm. These phenomena show that the energy of vortex shedding to displace the drill string already is lessened due to the riser already withstands the vortex energy as the frequency rises.

The twisting harmonic curve explains the differences between these two frequencies. At 5 Hz, the sinusoidal mode for the elongated drill string prevails at its lower segments, where the twisting of the harmonic curves is visible, their pitches are small and amplitudes are large. On the other hand, at lower frequencies of 0.5 Hz the twisting harmonic is visible along the drill string profile. When approaching to one third section of drill string the pitch and amplitude diminish and then rise in the centre of drill string. The harmonic twisting reaches its maximum point of pitch and amplitude at the depth of 480 meters and diminishes as the drill strings reach 500 meters depth.

Fig. 4 and 5 shows the displacement plot at 5 Hz and 0.5 Hz with 300 RPM speed. As the rotational speed increases, the harmonic amplitude and pitch are recorded very small compared with 80 RPM. The displacement of drill string decrease more and shows that the drill string can withstand the effect of the vortex shedding forces

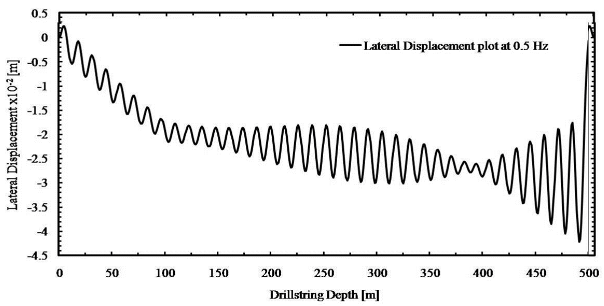

Fig. 2. Lateral Displacement of Drillstring with Vortex Shedding at 0.5 Hz, 80 RPM

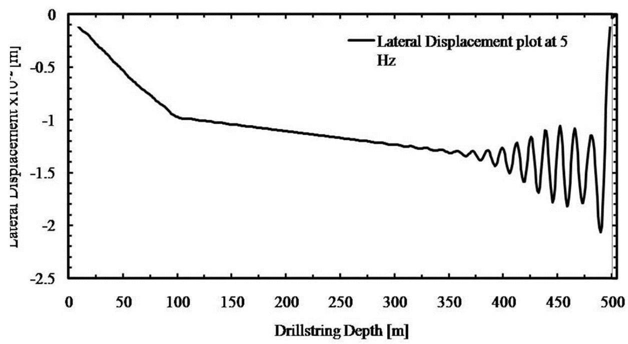

Fig. 3. Lateral Displacement of Drillstring with Vortex Shedding at 5 Hz, 80RPM

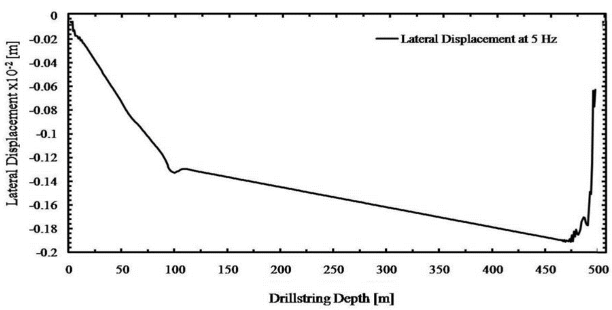

Fig. 4. Lateral Displacement of Drillstring with Vortex Shedding at 5 Hz, 300 RPM

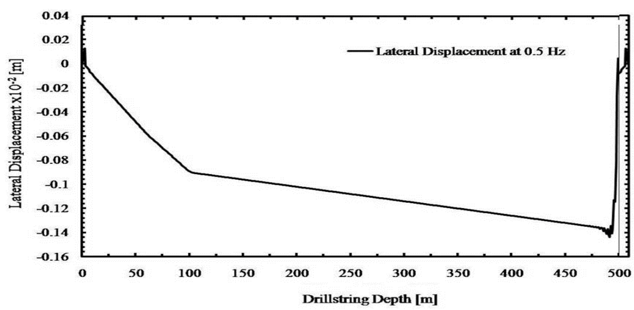

Fig. 5. Lateral Displacement of Drillstring with Vortex Shedding at 0.5 Hz, 300 RPM

Conclusions

A finite element model of a rotating drill string coupled to a riser was developed to mimic the offshore drilling operation. The vortex forces contribute a significant effect to increase the vibration risk by deforming the drill string from its longitudinal axis. The maximum effect of vortex shedding forces occurs at low frequency and low RPM. Rotational speed of drill string also influences the effect of vibration. With a higher rotational speed at 300 RPM, it is concluded that drill string withstand the lateral vibration compared with low RPM at 80 RPM, evidenced bys a small displacement occur in the drill string.

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