Buckling of Tubulars in Simulated Field Conditions

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Summary

First, this paper presents the new developments integrated in a recently advanced model for drillstring mechanics and takes into account the buckling phenomenon in any actual well trajectory. Second, this paper shows the influence of tortuosity and friction on the buckling phenomenon for some practical and critical cases met in the drilling industry. These tortuosity and friction effects are demonstrated with an experimental setup that confirms theoretical features. Finally, we compare results obtained from existant models with results obtained from our new model to evaluate the tortuosity and friction effects on the critical buckling load found in the literature.

Introduction

The ever-increasing complexity of well trajectories and drillstring designs has renewed and amplified the importance of understanding the buckling behavior of well tubulars inside wellbores. A realistic model is essential to make such complex field operations a success. For example, a fine prediction of axial-force transfer in a long horizontal or extended-reach-drilling well without compromising the drillstring-mechanical integrity is important.

While many equations have been derived for perfect vertical, inclined, horizontal, and curved wellbore without friction or rotation effect, no theory has been developed or applied to actual well conditions—that is, for a drillpipe rotating in a naturally tortuous wellbore. After a brief literature review, an advanced model dedicated to drillstring mechanics is presented and used to predict the onset of buckling in actual well conditions (with rotation and friction effect). A comparison of the model then is proposed with an experimental setup that reproduces the tortuosity of a wellbore.

State of the Art

Introduction. Buckling occurs when the compressive load in a tubular exceeds a critical value beyond which the tubular is no longer stable and deforms into a sinusoidal or helical shape. The sinusoidal buckling (first mode of buckling) corresponds to a tube that snaps into a sinusoidal shape. This first mode of buckling is sometimes called lateral buckling, snaking, or 2D buckling. The helical buckling (second mode of buckling) corresponds to a tube that snaps into a helical shape (spiral shape).

The first work dedicated to the buckling behavior of pipes in oil-well operation was initiated by Lubinski (1950; Lubinski et al. 1962). Since then, many theoretical works and/or experimental studies have been developed to better understand the buckling phenomenon. The aim of the following section is to select and present briefly the different improvement steps made during the last 50 years, in terms of theoretical and experimental works, going from the basic buckling models to more complicated modeling. A comprehensive literature review can be found in Cunha (2003), referring the most important contributions on the subject of buckling of tubulars inside wellbores.

Theoretical Works. The first theories were developed for perfect vertical wellbore without friction by Lubinski (1950; Lubinski et al. 1962). Then, the buckling behavior of drillpipes in inclined wellbore was first proposed by Dawson and Paslay (1984), based on earlier work by Paslay and Bogy (1964). The authors came to the following known critical buckling load for sinusoidal mode:

$$F_{\sin} = 2\sqrt{\frac{EI\ \ \omega \sin(Inc)}{r}}, \quad (1)$$

where EI is the stiffness of the pipe, ω is the buoyed linear weight of the pipe, Inc is the inclination of the wellbore, and r is the radial clearance between the pipe and the wellbore. The authors showed that for high angles of inclination, the drillpipe becomes more resistant to buckling because of the support and constraint provided by the wellbore. The critical force given by Eq. 1 is considered by the authors as the onset of buckling in an inclined hole and is widely used in the drilling industry.

Having derived equations for straight wells (vertical, inclined, and horizontal), some authors extended some existing equations for curved borehole (Schuh 1991) or developed new theories for tubular strings in curved wells (Kyllingstad 1995).

These equations, derived for perfect vertical, inclined, horizontal, and curved wellbore, are subject to simplifying assumptions: The pipe is continuous, without rotation, and the friction between the buckled drillpipes and the constraining wellbore is often ignored. This is why some authors tried to take into account the friction effect (Mitchell 1986, 1996), which has a strong influence on tubular loads but without providing an equation to account for that effect. Others (Duman et al. 2003; Mitchell and Miska 2006) investigated the effect of tool joint on the buckling behavior of drillpipes in horizontal wellbores and concluded that the presence of tool joints may increase the critical helical-buckling load up to 20%.

Some equations also were developed (Wu 1997) to take into account the effect of torsional load or torque during drilling. It has been found that the critical buckling load is reduced when torsional load (torque) is present (Wu 1997; He et al. 1995). Recently, Mitchell (2007) studied the buckling behavior of rotating drillpipes inside a straight horizontal wellbore and found that the critical buckling load of a rotating pipe is 78% of the one obtained for a nonrotating pipe.

Although there seems to exist a general consensus for the onset of buckling (sinusoidal mode), there is some controversy regarding the solution for critical force of helical buckling of tubulars inside wellbores (Cunha 2003; Aasen and Aadnøy 2002). Indeed, the equation for critical helical buckling in a straight deviated wellbore is given by:

$$F_{\mathrm{hel}} = \lambda\sqrt{\frac{EI\ \ \omega \sin(Inc)}{r}}, \quad (2)$$

where λ varies from 2.83 to 5.65 depending on the author (Cunha 2003; Aasen and Aadnøy 2002).

Buckling is known to increase the contact forces between the drillpipes and the wellbore and as a consequence increases both torque and drag. An analytical expression of the contact force for a helical pipe has been developed by Mitchell (1986):

$$W_{\mathrm{contact}} = \frac{rF^2}{4EI}, \quad (3)$$

where F is the axial compressive load in the buckled pipe. Eq. 3 has been verified in laboratory but is valid only for a given

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