Petroleum Technology Development Journal

Simulation of Tubular Buckling and Its Effect on hole Tortuosity

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Published in:
Petroleum Technology Development Journal (ISSN 1595-9104)
An International Journal

January 2012 - Vol. 1

Abstract
Buckling of tubulars inside wellbore has been the subject of many research work. However, these theories have always followed the same assumptions that the wellbore has a perfect and unrealistic geometry (vertical, horizontal, deviated or curved) and the friction is ignored, conditions relatively far from actual field conditions. How tubulars buckle in actual conditions remains a big question, in a naturally tortuous wellbore with friction and rotation. Can we apply conservatively theories developed for perfect well conditions (no tortousity, no friction) to actual well conditions? This work gives the needed answers to these questions in comparing results obtained from existing models with results obtained from an advanced model dedicated to drillstring mechanics. This work presents the new developments integrated in a recent advanced model for drillstring mechanics and enabling to take into account the buckling phenomenon in any actual well trajectory. It also 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 set up that enables confirmation of theoretical features. The model helps us to know the buckling load above which the tubular will buckle.

Introduction

The term simulation can be defined as a process of creating a model of an existing or proposed system in order to identify and understand those factors which control the system and/or predict the future behaviour of the system.

Buckling of drillstrings has been a serious problem in oil/gas field operations for many years. Some of the associated problems are casing failure, drillstring failure and drillstring lock up, casing wear, increasing in drillstring drag and torque. Therefore, the buckling of radial-constrained drillstring in a wellbore has become a topic of interest in the area of petroleum engineering in recent years. Due to the complexity of the problem, conflicting results are still reported. Gao et al[1], included torsion when evaluating the buckling load, of a weightless tubular for the case of a constrained Euler sinusoidal buckling. The impart of torsion is taken into account via the dimensionless parameters $\sqrt{2}T/\sqrt{FEI}$, in which T is the torsional load, F the axial compressive load, E Young's modulus of the isotropic material, and I the moment of inertial of the string's cross-section.

This parameter showed that when neglecting friction, the influence of torsion on axial compression force was relatively small, suggesting that the model by neglecting the torsion effect was a good approximation. However, experimental results were nearly double the buckling load predicted by Wu and Juvkam-Wold's[2] model. On the other hand, numerical results for helical buckling only agreed well existing experimental results for helical buckling and only agreed well with existing experimental results for compressive loads alone. Based on the

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[1] Gao, D.L. Liu F.W. and Xu B.Y. An analysis of helical buckling of long tubulars in horizontal wells. SPEJ, 1998, 50931 (4): 517-523
[2] Wu, J. and Juvkam-Wold H.C. Study of helical buckling of pipes in horizontal wells. SPE, 1993, 25503(7): 867-876; see also Wu, J. and Juvkam-Wold H.C. Discussion of tubing and casing buckling in horizontal wells. JPT, 1990, 42(3): 1062-10.

principle of energy balance, Wu concluded that the larger the torsional load, the lower the buckling load[3]; torsion loads had a very little effect on the buckling length (Pitch); and the tube would not buckle under torsional loads alone. Que et al[4]. calculated the normal force per unit length at the constraining wall for a helically buckling cylinder under compression and torsion and indicated that the role of torsion was also very little. Based on analysis of the deformation and forces, Gao et al[5] derived a buckling equation for a weightless elastic beam in a plane curved wellbore.

Torsion is shown to have a small effect on the buckling load but it controls the direction of the helical buckling. However, some researches suggest that torsional loads have significant effect on buckling loads which makes the research of utmost importance

Literature Review

Several researches have attempted to present the theory and practice of simulation of tubular buckling and its effect on hole tortuosity. A review of the literature reveals a variety of solutions and addressing various problems area.

Rezmer-Cooper[6], stated that representations of the drillstring and bottom hole assembly (BHA) that do not include the individual component stiffness have been used with some success in analyzing and preventing problems in extended reach drilling. As curve rates have increased in directional wells, the effects of the previous neglected stiffness have led to errors in the side force calculations that underpin all torque and drag analysis. Full finite-element analysis can solve this problem. Additionally, tortuosity can be added to provide a rippled effect to a well plan that simulates the micro doglegs that occur in the drilled well, enabling more realistic bounds to be placed on the drag and torque losses and the forces experienced by the drillstring.

According to Santos[7], wellbore instability problems have been attributed mainly to rock-fluid interaction, especially when drilling shaley formations with water-based fluids. However, recent studies showed that other events during drilling may contribute much more decisively in causing the problems than the rock-fluid interaction and swelling. The drillstring vibration is one of these events. Identification of the real cause of the problems is essential to reduce the exploration costs in challenged environments, such as the ones faced today by the oil industry. Also, suggestions in term of the drilling fluids were also made, since this can contribute decisively to strong reduce the incidence of the problems. However, the adopted solution should be in a completely different direction from what has been done so far. Robnett, et al[8], focus on measurements of drillstring torque, have detailed the nature of analysis of stick-slip phenomenon and have shown its detrimental effects on drillstring components. Using surface rotary motor current, drilling

[3] Wu, J. "Torsional load effect on drill-string buckling". SPEJ, 1997, 37477(6): 703-710

[4] Qiu, W. Miska, S. and Volk, L. Drill pipe/coiled tubing buckling analysis in a hole of constant curvature. SPEJ, 1998, 39795(3): 385-395

[5] Op cit

[6] Rezmer-Cooper, **Field Data Supports the use of Stiffness and Tortousity in Solving Complex Well Design Problems**. 2001

[7] Santos, H. and Placido J.C.R. "Consequences and Relevance of Drillstring Vibration on Wellbore Stability" SPE/IADC, 1997, 52820

[8] Robnett, E. "Analysis of the Stick-Slip Phenomenon Using Downhole Drillstring Rotation Data" SPE/IADC, 2007, 52821

contractors and surface logging companies have attempted to monitor drillstring torque so as to recognise and correct for this torsional instability. According to them, recent MWD tool developments using downhole drillstring rotational speed have established a superior method to identify stick-slip and establish its severity. Measuring actual rotational behaviour of the lower BHA gives a better insight to the torsional movement of the drill bit during the drilling process, without the problems involved when inferring this behaviour from surface torque readings, obtained at the opposite end of the drillstring.

Jun Agata, et al[9], summarizes the results of a study to assess the mechanical integrity of expandable tubular in monobore wells, and understanding the tubular material behaviour as it is being expanded. An experiment was setup to simulate the expansion process, and the development of a finite element model. Test results show that the variation of wall thickness around the circumference of the pipe is amplified when the axial load is increased, which results from the restraint condition in the wellbore. For both the experimental and the numerical simulations, a strong correlation was observed between pre- and post-expansion pipe wall thickness variations, and it was also concluded that the magnitude of the wall thickness variations of the expanded pipes are dependent on the work hardening of the original non-expended pipe. In addition, the collapse resistance of the expanded pipe was found to have degraded when the restraint condition was applied.

Stephane Menand et al[10], considered axial force transfer as an issue in deviated well when friction and buckling phenomenon take place. The general perception is that once drill pipe exceeds convectional buckling criteria, axial force cannot be transferred down hole anymore. Their work shows that even though buckling criteria are exceeded, axial force transfer could be still good if drill pipe is in rotation. Results from a drillstring mechanics model show how axial force is transferred down hole in many simulated field conditions: sliding, rotating, with or without doglegs.

A comprehensive buckling model was developed by Gao D.L. et al.[11], where a group of fourth order non-linear ordinary differential equations, is derived with application of the principle of minimum potential energy. Then the equations were normalized to make the solutions independent of the wellbore size, type of pipe and mud. The critical sinusoidal buckling load of tubing was analyzed under different boundary conditions typical for drilling and well completion applications. The results show that the effects of boundary conditions can be neglected when the dimensionless length of tubing is greater than is 6. The dimensionless length of pipe depends on the clearance between tubing and the wellbore, the bending stiffness of the tubing, unit weight, and is proportional to length of the pipe. The authors also investigated the effects of friction on both sinusoidal and helical buckling for a long pipe with dimensionless length greater than 6. The results show that friction will significantly increase the critical loads for both sinusoidal and helical buckling and reduced the amplitude of sinusoidal post buckling configuration.

[9] Jun, A. Mitsuru, S. Hitoshi, A. and Hidefumi, T. Evaluating the Expandability and Collapse resistance of Expandable Tubulars. SPE/IADC, 2009, 1199552

[10] Stephane, M. Sellami, H. Bouguecha, A. and Isambourg, P. Axial Force Transfer of Buckled Drill Pipe in Deviated Wells. SPE/IADC, 2003, 119861

[11] Gao, D.L. Liu F.W. and Xu B.Y. An analysis of helical buckling of long tubulars in horizontal wells. SPEJ, 1998, 50931 (4): 517-523

Methodology

The basic mathematical model used in most torque and drag software has not changed significantly since its original inception. By contrast, software user interfaces have improved dramatically as computer hardware, processor power, and software functionality have evolved. On the basis of more than 20 years of industry access to torque-and-drag string-simulation software, the time is right to reflect on the state of current models and identify future requirements.

Soft-String Model

The soft-string model ignores any tubular-stiffness effects; this means that the pipe is treated as a heavy cable, chain, or rope lying along the wellbore. It also means that axial tension and torque forces are supported by the string and contact forces are supported by the wellbore. This model assumes that the loads on the string result solely from the combined effects of gravity and frictional drag, a result of contact between the string and the wellbore.

Virtually all torque-and-drag string models assume that the drillstring is made up of short elements joined by connections that transmit tension, compression, and torsion but not bending moment. The calculations start at the bottom of the string and proceed upward to the surface. Each short element contributes increments of torque, axial drag and weight. Forces and torque values are summed to produce the cumulative loads on the drillstring. Initial conditions usually are specified at the bit, and for drilling operations, the input would be weight-on-bit (WOB) and bit-torque values.

The most significant assumption is that the string is in connection contact with the wellbore over its entire length. This also means that radial clearance effects are ignored and that the bending moment is not considered in the model.

The basic governing equations are given below in natural curvilinear coordinates for the soft-string model. The effects of the internal and external fluids, with pressures P_i and P_o, are taken into consideration by using the effective tension, T_e.

$$T_e = T + P_o A_o - P_i A_i$$

Where:

T = Actual axial tension

A_i = Drill string section area defined by inner diameter D_i

A_o = Drill string section area defined by outer diameter D_o

Replacing the dry weight density ρ_s, by the submerge density, ρ_v

$$\rho_v = \rho_s - \rho_f$$

Where : ρ_f is the fluid density

With the substitutions made, equilibrium of the soft string model is described as follows (while tripping out):

$$d(T E_t)/dS + N - fNE_t + \rho E_g = 0$$

Where:

E_t = Tangential direction, positive up hole

N = Distribution normal contact force

f = Friction coefficient

ρ = Submerge weight density
E_g = Vector of submerge drill string weight per unit length

Stiff-String Model

The stiff-string model account for string stiffness (flexural rigidity) and radial clearance by allowing initially unknown sections of the string not to be in contact with the wellbore. Higher sidewall forces occur as stiff tubular are forced around curved sections, and reduced sidewall forces occur as the pipe straightens. Variation of contact area between a string component and the wellbore also will occur.

Concentrated bending moments at stabilizers and casing centralizers, as well as at drill pipe connections, also must be included in a comprehensive stiff-string model. The stiff-string method is intended to produce a more realistic analysis of the configuration, stresses and loads acting on the string and borehole.

$$dE_n/dS = -k_bE_t + k_nE_b$$

Tubular Buckling Approach

According to the conceptual approach used to model the areas of interest in the well by taking the horizontal section as an example. When drilling or tripping into the well, the tubular in a horizontal section is in compression, due to frictional force and/or bit weight. The axial compression load increases up hole from the bottom of the tubular as a result of the frictional force. When the axial compressive load exceeds the sinusoidal buckling load limit for the tubular, it will buckle into sinusoidal shape.

When the axial compressive load increases to its helical buckling load limit, the tubular will be buckled helically. The helical buckling of the tubular will in turn generate a large contact force between the tubular and the wellbore wall due to the confinement by the wellbore wall. This buckling produces a large frictional force. When drilling or running into the well, the frictional force is now much larger than that for an unbuckled, straight tubular. Therefore, the axial load distribution will no longer be linear, but a quickly rising non linear curve.

For tubular in other types of wellbores, such on a vertical or inclined wellbore, large frictional force will be generated if the tubular is helically buckled, and the axial load distribution of the tubular will be very different from that of a straight, unbuckled tubular.

Since the axial load distribution of buckled is not the same as that of the unbuckled tubular, we must develop a new model to deal with tubular buckling and the large frictional drag due to the buckling. This necessary to correctly predict bit weight for an extended-reach and/or horizontal well when there is buckling of a tubular in some part of the wellbore. It is also very important to for us to predict when and how the tubular can be locked up in the wellbore due to the extreme frictional drag resulting from tubular buckling. When the tubular is locked up, it cannot be pushed farther into the hole. No increase in bit weight from further slack off of weight at the surface.

Sinusoidal buckling in a horizontal welbore

When the axial compression load along the coiled tubing reaches the following sinusoidal buckling load F_{cr}, the initial (sinusoidal or critical) buckling of the coiled tube will occur in the horizontal wellbore.

$$F_{cr} = 2\sqrt{EIW_e/r}$$

Where:

F_{cr} = Sinusoidal buckling load, lbf

E = Young's modulus, psi

I = Moment of inertia of pipe, in^4

W_e = Effect weight of tubular, lb/in.

r = Radial clearance between pipe and wellbore, in.

A more generalized sinusoidal buckling load equation for highly inclined wellbores (including the horizontal wellbore) is:

$$F_{cr} = 2\sqrt{EIW_e \sin\theta/r}$$

For tubular in a curved wellbore, the sinusoidal buckling loads is:

$$F_{cr} = 4EI/rR[1 + \sqrt{1 + rR^2W_e \sin\theta/4EI}$$

Helical buckling in a horizontal wellbore

When the axial compressive load reaches the following helical buckling load F_{hel} in the horizontal wellbore, the helical buckling of coiled tubing then occurs:

$$F_{hel} = 2(2\sqrt{2} - 1)\sqrt{EIW_e/r}$$

A more general helical buckling load equation for highly inclined wellbores (including the horizontal wellbore) is:

$$F_{hel} = 2(2\sqrt{2} - 1)\sqrt{EIW_e \sin\theta/r}$$

Buckling in a vertical wellbore

In a vertical wellbore, the buckling of coiled tubing will occur if the coiled tubing becomes axially compressed and the axial compressive load exceeds the buckling load in the vertical section. This could happen when we "slack-off" weight at the surface to apply bit weight for drilling and pushing the coiled tubing through the build section and into the horizontal section.

Wu and Juvkam-Wold (1992) derived the following buckling load equation for the initial buckling of tubular in vertical wellbores:

$$F_{cr,b} = 2.55(EIWe^2)^{1/3}$$

Helical buckling in vertical wellbores

A helical buckling load for weighty tubular in vertical wellbores was also derived recently through an energy analysis to predict the occurrence of the helical buckling.

$$F_{hel,b} = 5.55(EIWe^2)^{1/3}$$

This helical buckling load predicts the first occurrence of helical buckling of the weighty tubular in the vertical wellbore. The first occurrence of helical buckling in the vertical wellbore will be a one-pitch helical buckle at the bottom portion of the tubular, immediately above KOP. The upper portion of the tubular in the vertical wellbore will be tension and remain straight.

When more tubular weight is slacked-off at surface, and the helical buckling becomes more than one helical pitch, the above helical buckling load equation may be used for the top helical pitch of the helically buckled tubular.

The top helical buckling load $F_{hel,t}$ is calculated by simply subtracting the tubular weight of the initial one-pitch of helically buckled pipe from the helical buckling load Fhel,b which is defined at bottom of the one-pitch helically buckled tubular.

$$F_{hel,t} = 5.55(EIWe^2)^{1/3} - W_e L_{hel}$$
$$F_{hel,t} = 0.14\ (EIWe^2)^{1/3}$$

The length of the initial one-pitch of helical buckling or the first order helical buckling is:

$$L_{hel} = (16\pi^2 EI/W_e)^{1/3}$$

For the tubular in curved wellbore, the helical buckling load limit is (Wu, 1992):

$$F_{hel} = (12EI/rR)\ [1 + \sqrt{1 + rR^2 W_e \sin\theta / 8EI}]$$

Results and Discussion

Axial Force Experiment

A 15-m, 0.60-kg/m steel pipe (outside diameter = 13.5 mm) was inserted into a transparent plastic tube (inside diameter = 42.2 mm) representing the wellbore. The pipe and tube sizes were selected by using a scaling criterion that considers the ratio of the size of the hole outside diameter of the pipe. The tube was sufficiently flexible to produce doglegs along the path to study the tortuosity effect on buckling. Both fixed- and free-end support can be provided to the pipe. A motor is fixed at one end to make the pipe rotate, and it enables torque and rotation speed to be recorded vs. time. The experimental facility allows axial load to be applied at one end of the pipe and load to be measured at both ends of the pipe. The compressive axial load is applied by a hand-controlled hydraulic jack, enabling the experiment to be paused at any position for visual inspection and additional measurements. The displacement of the pipe is measured by a linear variable displacement transducer at the top end of the pipe, and the loads and displacements are recorded by a computer data acquisition system. A typical experiment consists of loading the pipe to given compressive load and then unloading the pipe back to its original state, with or without rotation. The μ between the pipe and the plastic tube has been measured to be from 0.25 to 0.38.

Results

For the situation, compressive top and bottom loads are measured to determine axial-force transfer. If the top load is equal to the bottom load, then axial-force transfer is 100% efficient. The following four cases were considered for a horizontal well.

- Case 1 – without dogleg in sliding mode
- Case 2 – without dogleg in rotating mode
- Case 3 – with dogleg in sliding mode
- Case 4 – with dogleg in rotating mode

Each experiment is compared to the numerical model.

Sliding vs. Rotating

Fig. 4.1 shows axial-force transfer (top load vs. bottom load) for a non-rotating and rotating pipe when the pipe is loaded (increasing compressive load) or unloaded (decreased compressive load) in a horizontal well (Cases 1 and 2, respectively).

In loaded case without rotation (similar to a slack-off or sliding situation), the bottom load is less than the top load, as a result of sliding friction. Once the pipe without rotation is unloaded, the relative motion between the pipe and the tube becomes opposite, and this is why the bottom load is greater than the top load (the pipe is pulled out of the hole). As observed experimentally by numerous authors, this hysteresis effect is the result of friction because the relative motion between the pipe and the tube is opposite when the pipe is loaded or unloaded. The current model is able to reproduce this hysteresis effect.

For the rotating case, the hysteresis effect has disappeared almost completely in the experiments; that is, the loading curve is quite similar to the unloading curve. This effect depends on the ratio of the rotation speed of the pipe to the loading/unloading rate. The current model is able to reproduce correctly observation made in the laboratory. For the axial-force transfer in sliding mode for the horizontal well with or without doglegs (Cases 1 and 3, respectively), the weight transfer is affected by the dogleg because approximately 70% of the top load is transferred to the bottom load (case 3) when the top load reaches 5000N. Experimental and numerical results show that axial-force transfer is very good, even though conventional helical-buckling criteria are exceeded. At a compressive load more than two times the conventional critical helical load, the axial-force transfer still is very good.

Summary

A set of equilibrium equations have been developed for the buckling analysis of a drillstring in a wellbore which is subjected to axial compression, torsion, and the gravity simultaneously. Detailed derivations are presented. Numerical results of a simple example indicate that torque has a significant impact on the helical buckling load. The proposed model will provide a theoretical basis for the buckling analysis of a drillstring constrained in a wellbore.

Conclusions and Recommendations

The laboratory tests and numerical simulations demonstrated that the axial-force transfer is quite good even though drill pipe is helically buckled and that helical buckling decreases axial-load transfer in sliding mode, but not in rotating mode. These observations confirm field case studies where it was possible to drill with compressive loads significantly higher than standard critical-buckling loads.

The torque-drag analysis is thus a significant step toward providing a true predictive directional drilling program that can be used both in the office as a planning aid and in the field as a monitoring and advisory tool. By coupling an overall predictive drilling program with a trouble analysis program which accounts for the effects of the deviation on torque and drag, basic elements of a directional drilling simulator are provided that will effectively enable one to drill a well on a computer.

Recommendations

There are several recommendations that can be made based on the analysis of simulation of tubular buckling and its effects on hole tortuosity. These include:

- Simulation is recommended to predict when and how the tubular can be locked up in the wellbore due to the extreme frictional drag resulting from tubular buckling.
- To achieve a uniform probability of failure of individual components of structural systems and result in material savings as well as in a more intensive evaluation of actual structural reliability
- For production optimization.
- Provision of a drilling system using long guage bits will significantly reduce hole spiraling, one form of micro-tortuosity, which is intended by use of the drill bit designed to improve many facets of the drilling operation.