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Stefan Miska*, Mengjiao Yu*, Yi Zhang*, Włodzimierz Miska**

MODEL OF DRAG AND TORQUE FOR CASING RUNNING IN EXTENDED REACH AND HORIZONTAL WELLS

1. INTRODUCTION

In conventional vertical wells the operation of casing running normally takes a small portion of the time and cost associated with drilling and completion of a well. In other words, the operation of casing running is considered to be rather simple as compared to other operations. The casing is suspended at the surface and moves into the wellbore due to its weight. At times to avoid excessive hook load the casing is run partially empty to increase the buoyancy effects. However, in extended reach highly inclined wells a great deal of planning and modeling is frequently needed to make sure the casing will reach the desired setting depth. Consequences of not getting the casing to the planned depth are time consuming and very costly particularly in offshore wells.

Proper management of processes related to casing running starts with the optimization of the wellbore trajectory. Here as the criterion of optimization we use the maximization of the slack off weight needed to overcome the axial drag forces generated due to the contact of casing and the wall of wellbore. In some instances the slack off weight may not be sufficient for moving casing and a pushing force at the top of the hole must be applied to drive casing to the desired depth. In any case the casing must be run to desired depth and subsequently cemented to meet the well objectives.

Because of friction the contact forces generate the drag forces that always oppose the direction of motion. Therefore accurate calculations of the contact forces are of critical importance for proper well plan design and the subsequent predictions of the length of high angle/horizontal well that can be cased off if needed.

A considerable reduction in axial drag forces is achieved by casing rotation. When casing is rotated the drag force due to pipe contact with the wellbore is overcome by the torque. Typically in drag and torque type of analysis we assume that if the pipe is rotated the

* The University of Tulsa
** Wydział Wiertnictwa, Nafty i Gazu AGH, Kraków

axial drag is nil. In this case however, with an increase in length of the highly inclined part of the wellbore, we observe a considerable increase in the rotary torque which in turn may be detrimental to the pipe and couplings mechanical integrity and in some cases even exceed the maximum torque that can be delivered by the rotating mechanism.

Another method that is used by industry to reduce the axial drag in a horizontal portion of the well is to run casing only partially filled with drilling fluid or just empty to reduce its weight. By proper management of buoyancy one can float casing and avoid contact forces. Reduction in contact forces makes however the casing vulnerable to helical buckling. It is well know that when the pipe is helically buckled high contact and drag forces develop. Yet another approach is to reduce the wellbore fiction coefficient by adding various lubricants to the drilling fluid. In any case, the designer needs a model that accurately describes all major factors that control the contact and drag forces.

At present there are two types of drag and torque models that are mentioned in the literature. For practical design applications a number of computer programs are available from different vendors. Perhaps the most widely used is the so called Soft-String Model proposed by [6] for evaluating forces in drillstrings during tripping in and out operations in 3D wells. The model assumes that the drillstring is a weighted cable (belt) without stiffness. Subsequently Ho [7] included the pipe stiffness and proposed equations valid for large deformations. However, Ho's model provides only four equilibrium equations, i.e., three for force balance and one for moment balance, which are applicable to the case of pipe running in or pulling out of the hole. A model that accounts for pipe stiffness is called a Stiff String Model.

Gonet [1] used 3D formulation and a system of simplified equations for the purpose of calculating the distance between casing centralizes.

Maidla and Wojtanowicz [13] used the Soft String Model for predicting casing running for four directional wells with the well path restricted to the vertical plane. They compared calculated versus actually measured hook loads and found that the agreement was good except for the case a well that while drilling unintentionally developed 3D path.

Mason at al [4, 5] presented a number of papers devoted to various issues related to operational difficulties and predictions of casing loads in extended reach well.

More recently, Mitchell [3] develop a system of equations valid for the segments of wellbore designed using the minimum curvature (zero torsion) approach. Mitchell included the pipe stiffness in his formulation and derived equations for calculating both the axial force and torque. Mitchell pointed out that the moment equilibrium equation must include the distributed moment due to the drag force.

In this paper we basically use a similar approach but include the effect of wellbore torsion as explained in the section below.

2. MATHEMATICAL MODEL

The mathematical model for a drag and torque is a system of equations that can be used for calculation of forces and moments in the pipe as well the contact forces that are generated due to the interaction between a pipe and wellbore. Here we present a development of a three dimensional model that is a subject of several simplifying assumptions but still very useful for practical design applications. The major simplifying assumptions are as follows:

– Pipe is in a continuous contact with the wellbore (effects of tool joints, couplings and wellbore irregularities (tortuosity) are ignored).
– Inertia effects due to pipe sliding and or rotation are ignored.
– Drilling fluid flow effects are not considered.
– Friction force is modeled using the Coulomb friction concept.

To develop the model we use Frenet-Serret system of coordinates and as we show in Appendix A and the desired equilibrium equations of forces and moments are as follows:

Equilibrium of forces:

t – component

$$\frac{dF_t}{ds} - \kappa F_n + w_e \vec{t} \bullet \vec{k} + \vec{w}_d \bullet \vec{t} = 0 \tag{1a}$$

n – component

$$\frac{dF_n}{ds} + \kappa F_t - F_b \tau + w_e \vec{n} \bullet \vec{k} + \vec{w}_d \bullet \vec{n} + \vec{w}_c \bullet \vec{n} = 0 \tag{1b}$$

b – component

$$\frac{dF_b}{ds} + F_n \tau + w_e \vec{b} \bullet \vec{k} + \vec{w}_d \bullet \vec{b} + \vec{w}_c \bullet \vec{b} = 0 \tag{1c}$$

where:

$\vec{t}, \vec{n}, \vec{b}$ – unit vectors in tangent, normal and binormal directions,
$\vec{k}$ – unit vector in z direction,
F_t, F_n and F_b – are the tangential (axial), normal and binormal components of the pipe force vector $\vec{F}$,
$\vec{w}_c$ – unit contact force vector,
$\vec{w}_d$ – unit drag force,
w_e – pipe unit weight in fluid,
κ – pipe curvature,
τ – pipe torsion.

Equilibrium of moments:

t – component

$$\frac{dM_t}{ds} + \vec{m} \bullet \vec{t} = 0 \tag{2a}$$

n – component

$$\kappa M_t - EI\kappa\tau - F_b + \vec{m} \bullet \vec{n} = 0 \tag{2b}$$

b – component

$$EI\frac{d\kappa}{ds}+F_n+\vec{m}\bullet\vec{b}=0 \tag{2c}$$

where:

- M_t – moment in tangential direction (torque),
- m – distributed moment along the string,
- EI – pipe bending stiffess.

It is important to notice that in addition to pipe curvature κ the equilibrium equations also contain the pipe torsion τ which should not be confused with the twist. For practical calculations we assume the pipe curvature and torsion are the same as wellbore curvature (dogleg severity) and torsion which are calculated from the following equations:

Well bore curvature:

$$\kappa=\sqrt{\left(\frac{dI}{ds}\right)^2+\sin^2 I\left(\frac{dA}{ds}\right)^2} \tag{3}$$

Wellbore torsion:

$$\tau=\frac{1}{\kappa^2}\left\{\cos I\left[2\left(\frac{dI}{ds}\right)^2\left(\frac{dA}{ds}\right)+\sin^2 I\left(\frac{dA}{ds}\right)^3\right]+\sin I\left[\left(\frac{dI}{ds}\right)\frac{d^2A}{ds}-\left(\frac{dA}{ds}\right)\frac{d^2I}{ds}\right]\right\} \tag{4}$$

Where: I(s) and A(s) are the hole inclination angle and azimuth along the well trajectory. Derivation of Eqn(3) and Eqn(4) is provided in Appendix B.

The first derivatives with respect to the measured depth s are the conventional build and turn rates. It is easy to notice that if the hole azimuth is constant the wellbore torsion is nil.

3. LIMITATIONS DUE TO BUCKLING

If the pipe is subjected to high compression it may buckle and our 3D drag model discussed so far is not applicable any more.

It can be shown that if the friction is ignored the compressive force required to initiate lateral (sinusoidal) buckling can be calculated from the following equation:

$$F=2\sqrt{\frac{EIw_c}{r}} \tag{5}$$

where:

- F – compressive force in the string,
- EI – pipe bending stiffness,
- w_c – unit contact force,
- r – pipe radial clearance.

As a first approximation the unit contact force can be calculated using the soft string model:

$$w_c = \pm \frac{F}{R} + w_e \sin I \tag{6}$$

The positive sign pertains to buidup sections of wellbore while the negative is for drop off sections. For the case of drop off segments the contact force can be small and a relatively small compressive force may buckle the pipe. On the other hand, in a build up section the wellbore curvature increases the contact force and it takes high compression to buckle the pipe. In other words, a positive wellbore curvature (dogleg severity) has stabilizing effect on drill pipe. Eqn (5) is also valid straight inclined wells.

4. CASE STUDIES

Case 1: 2-D ERW from BP Well Profile

Figure 1 shows the trajectory of an Extended-Reach-Well drilled and cased off by British Petroleum as in SPE 52842.

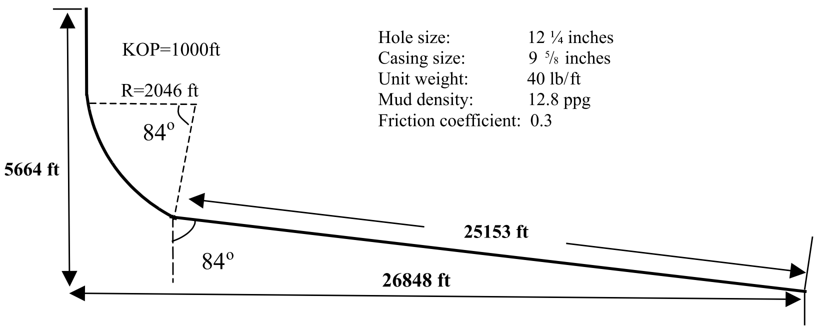

Fig. 1. Well profile for Case 1

Casing flotation is a widely used technique in the industry for casing running. The main benefit of casing floatation is the reduced effective weight can lower the friction encountered in casing running operations. Table 1 shows the properties of the casing and drilling fluid used in this study.

Table 1

Parameters of casing floatation

Casing OD	Casing ID	Unit weight	Effective weight	Density of fluid outside casing	Density of fluid inside casing
(in)	(in)	(lb/ft)	(lb/ft)	(ppg)	(ppg)
9 5/8	8.835	40	32.4	12.8	12.8
9 5/8	8.835	40	25.7	12.8	10.7
9 5/8	8.835	40	13.9	12.8	7

Figure 2 shows the axial force along the casing. Figure 3 shows that casing flotation can significantly reduce the unit contact force (hence the drag force). With a compressive force of 40 klbs at the surface (Fig. 2), casing with effective weight 32.4 lbs/ft (472.8 N/m) or 25.7 lbs/ft (375 N/m) can not reach the designed depth of 29153 ft (8886 m) due to the tension required at the bottom of the casing. However, if the effective weight is reduced to 13.9 lb/ft (202.8 N/m), the unit contact force can be significantly reduced (Fig. 2) and therefore the casing can be run to the designed depth successfully. One can also see that because of the wellbore curvature (dogleg severity) the unit contact force is increased in the build part of the wellbore (Fig. 3).

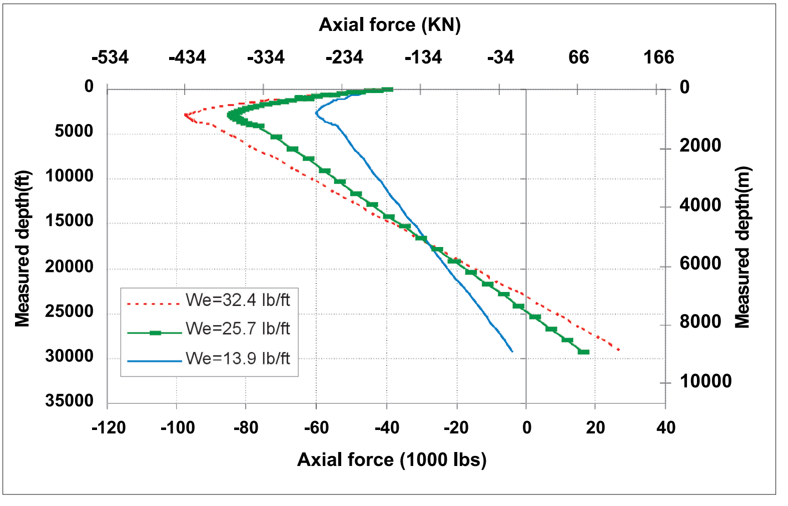

Fig. 2. Effect of casing floatation on axial force transfer

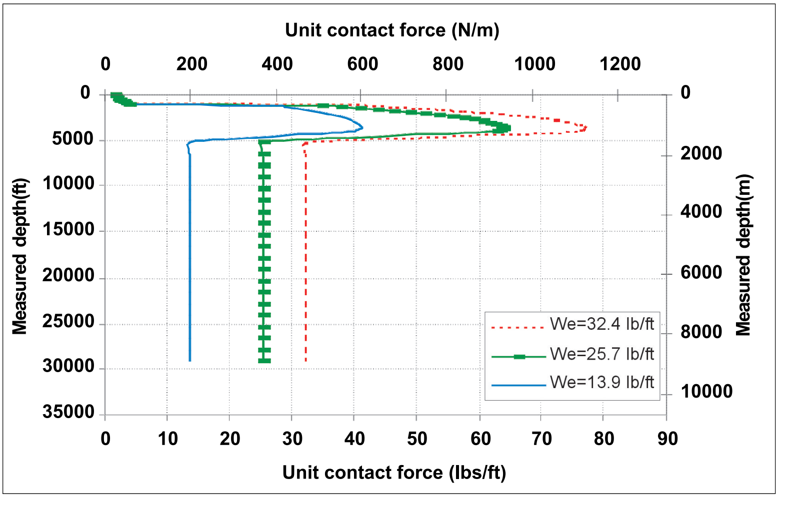

Fig. 3. Effect of casing floatation on contact force

Although casing flotation is an effective way in casing running operation, reducing the effective weight of the casing is associated with an increasing risk of buckling of the casing. Care must be taken not to compromise the benefit of this technique due to casing buckling caused by the reduced unit weight.

5. CASE 2: 3-D WELLBORE TRAJECTORY

Table 2 shows the survey data (measured depth [ft], inclination angle [degrees] and azimuth [degrees]) of a 3-D wellbore as given in SPE 84246. The wellbore is 13.5 inches (0.34 m) in diameter. A 9-5/8 inch (0.244 m) OD and 8.835-inch (0.224 m) ID casing with a unit weight 40 lbs/ft (583.7 N/m) is run into the wellbore

Table 2
Survey data

Measured depth (ft)	Inclination angle (degrees)	Azimuth (degrees)
0	3.75	45
702.5	5.5	45
1964.5	29.75	77.05
4250	29.75	77.05
5086.3	29.75	77.05
8504.1	80.89	300.71
8828	90	297.31
9151.9	99.11	293.92
9901.7	120	285

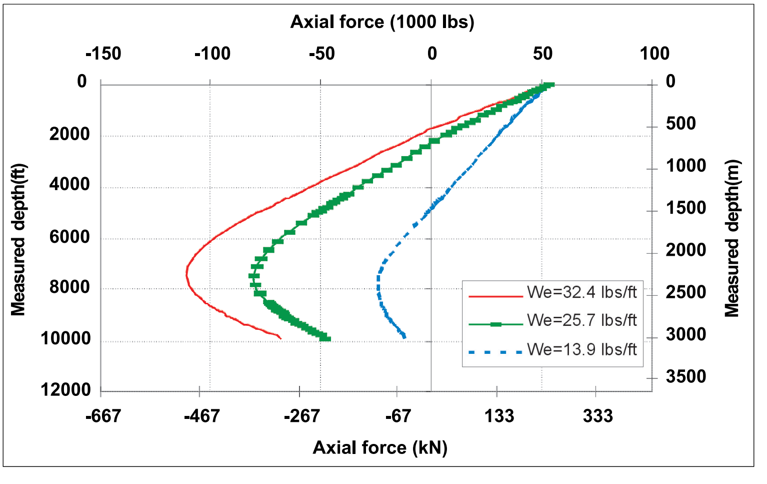

Fig. 4. Effective axial forces vs. measured depth for different unit weight

Figure 4 shows the effect of the unit pipe weight on axial force transfer. Effective unit weights of 32.4 lbs/ft (472.8 N/m), 25.7 lbs/ft (375 N/m), and 13.9 lbs/ft (202.8 N/m) are investigated based on the parameters of the well trajectory. A 53 klbs (235.7 kN) hook load was recorded at surface is 53 klbs (235.7 kN). Although casing floatation can reduce the effective weight and resulting in a smaller unit contact force as shown in Figure 5, the corresponding running weight is also reduced as shown in Figure 4.

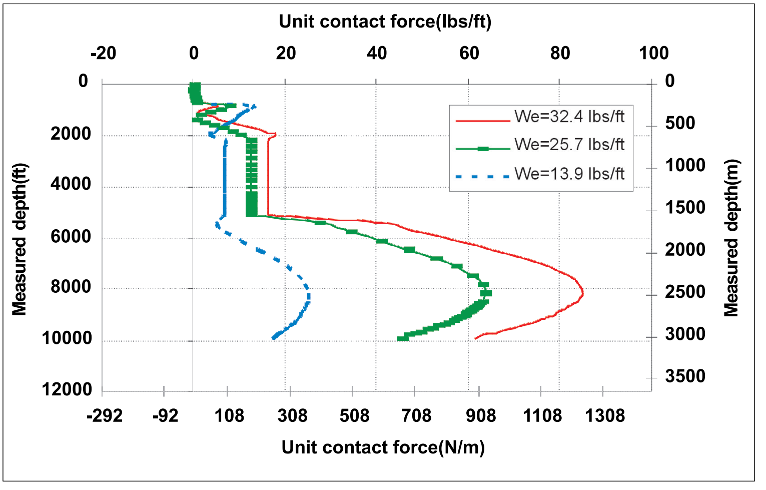

Fig. 5. Unit contact forces vs. measured depth for different unit weight

6. APPENDIX A

To derive the force and moment equilibrium equations we will consider a differential pipe element as shown in Figure (A-1). For the purpose of this analysis we chose a right hand *x*, *y*, *z* system of coordinates with the z axis pointing down and the conventional unit vectors $\vec{i}$, $\vec{j}$, $\vec{k}$. It will be also very useful to introduce the right hand Frenet – Serret system of coordinates with the unit vectors $\vec{t}, \vec{n}, \vec{b}$ as also depicted in Figure A-1. The position vector $\vec{r}(s)$ is drawn from the origin of the rectangular coordinates to the pipe element's mass center.

If the pipe is moving slowly so the inertia effects can be neglected and the vector sum of all forces on the pipe segment gives:

$$\vec{F} + \Delta\vec{F} + \vec{w}\Delta s - \vec{F} = \vec{0} \qquad \text{(A-1)}$$

where $\vec{F}$ is the force in the pipe (internal pipe force), $\vec{w}$ is the total (resultant of all external forces) force per unit length applied to the pipe, and Δs is the length of the pipe segment. In the limit as $\Delta s \to 0$, the change in force $\vec{F}$ due to applied load vector $\vec{w}$ is given by the following equation:

$$\frac{d\vec{F}}{ds} + \vec{w} = \vec{0} \qquad \text{(A-2)}$$

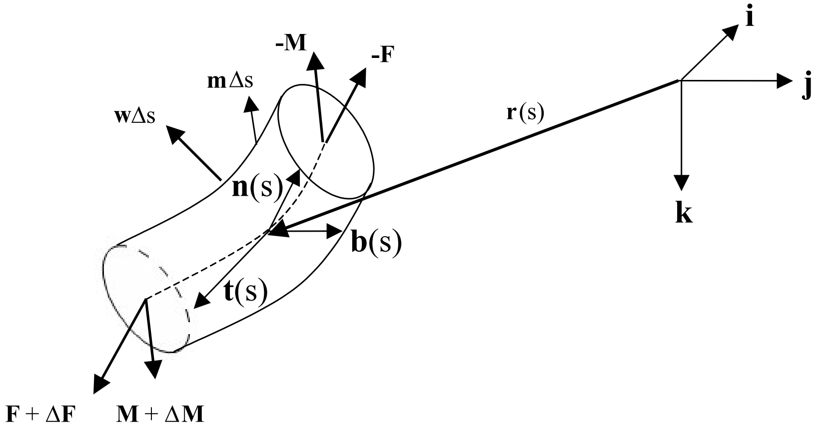

Fig. A-1. Schematic drawing of a system coordinates x, y, z

In general, the pipe force $\vec{F}$ will consist of the tangential F_t (axial) and two shear forces in normal F_n and binormal F_b directions so we write:

$$\vec{F} = F_t\vec{t} + F_n\vec{n} + F_b\vec{b} \tag{A-3}$$

Typically in a drag and torque type of analysis we consider only three external forces due to pipe effective weight (weight of pipe in fluid) $\vec{w}_e$, normal to the pipe contact force $\vec{w}_c$ and force due to friction (drag force) $\vec{w}_d$ and we write:

$$\vec{w} = \vec{w}_e + \vec{w}_c + \vec{w}_d \tag{A-4}$$

In our system of coordinates the gravity is pointing down and so is the z axis, therefore the effective unit pipe weight vector is:

$$\vec{w}_e = w_e\vec{k} \tag{A-5}$$

As already stated, the unit contact force $\vec{w}_c$ is normal to the pipe hence it is in the $\vec{n}-\vec{b}$ plane. If the pipe is sliding the drag force is in tangent $\vec{t}$ direction. If pipe is rotated we assume the drag force is also tangent to the pipe but in the $\vec{n}-\vec{b}$ plane.

In a similar manner as for the force balance we can show that the vector moment equilibrium equation is as follows:

$$\frac{d\vec{M}}{ds} + \vec{t}\times\vec{F} + \vec{m} = \vec{0} \tag{A-6}$$

where:

$\vec{M}$ – internal moment (pipe moment),
$\vec{m}$ – applied moment per unit length (distributed moment) due to the drag force,
$\vec{t}$ – unit tangent vector.

For a circular pipe the moment vector $\vec{M}$ in the pipe consist of the bending moment (pointing in the binormal direction) with a magnitude equal to the product of pipe bending stiffness (EI) and pipe curvature κ and torque (pointing in the tangential direction) hence we write:

$$\vec{M} = EI\kappa\vec{b} + M_t\vec{t} \tag{A-7}$$

where:

EI – pipe bending stiffness,

κ – pipe curvature,

M_t – magnitude of moment (torque) required for pipe rotation.

The distributed moment $\vec{m}$ associated with the drag force $\vec{w}_d$ is a cross product of pipe radius and drag force:

$$\vec{m} = \vec{r}_p \times \vec{w}_d \tag{A-8}$$

where $\vec{r}_p$ is pipe radius vector.

Substituting Eqn (A-3) and Eqn (A-4) to Eqn (A-2) gives:

$$\frac{d\left(F_t\vec{t}\right)}{ds} + \frac{d\left(F_n\vec{n}\right)}{ds} + \frac{d\left(F_b\vec{b}\right)}{ds} + \vec{w}_e + \vec{w}_c + \vec{w}_d = \vec{0} \tag{A-9}$$

Eqn (9) is a general form of the force equilibrium in three dimensions using vector notations. To obtain the corresponding scalar components we firstly calculate the derivatives with respect to the measured depth s and then take the dot products of tangent, normal and binomial vectors. After some rearrangements we obtain Eqn (1a), Eqn (1b) and Eqn (1c) in the body of the paper.

For a circular pipe with the bending stiffness EI the vector moment $\vec{M}$ in the pipe is given by Eqn (A-7).

Differentiating Eqn (A-7) with respect to s yields:

$$\frac{dM(s)}{ds} = EI\left(\frac{d\kappa}{ds}\vec{b} - \kappa\tau\vec{n}\right) + \frac{dM_t}{ds}\vec{t} + M_t\kappa\vec{n} \tag{A-10}$$

The vector product of the unit tangent vector and the force in pipe is:

$$\vec{t} \times \vec{F} = \vec{t} \times \left(F_a\vec{t} + F_n\vec{n} + F_b\vec{b}\right) = F_n\vec{b} + F_b\vec{n} \tag{A-11}$$

Substituting Eqn (A-10) and Eqn (A-11) to Eqn (A-6) gives:

$$\left(EI\frac{d\kappa}{ds} + F_n\right)\vec{b} + \left(M_t\kappa - EI\kappa\tau - F_b\right)\vec{n} + \frac{dM_t}{ds}\vec{t} + \vec{m} = \vec{0} \tag{A-12}$$

Eqn (A-12) is a general form of the moment balance written in a differential form.

It is important to notice that Eqn (A-12) contains not only pipe curvature and torsion but also the change of curvature $\frac{d\kappa}{ds}$ along the pipe

To obtain the corresponding three scalar equations we multiply (dot product) Eqn (A-12) by the unit tangent, normal and binormal vectors. In such a manner we obtain Eqn (2a), Eqn (2b) and Eqn (2c) in the body of the paper.

7. APPENDIX B

The position vector 2as depicted in Fig (A-1) is:

$$\vec{r} = x(s)\vec{i} + y(s)\vec{j} + z(s)\vec{k} \tag{B-1}$$

Where: s- measured depth and $\vec{i}, \vec{j}, \vec{k}$ – unit vectors.

Tangent vector is:

$$\vec{t} = (\sin I \cos A)\vec{i} + (\sin I \sin A)\vec{j} + (\cos I)\vec{k} \tag{B-2}$$

where: I and A hole inclination and azimuth angles respectively

The curvature vector is:

$$\vec{K} = \frac{d\vec{t}}{ds} \tag{B-3}$$

Differentiating Eq (B-2) we obtain:

$$\vec{K} = \left[\cos I \cos A\left(\frac{dI}{ds}\right) - \sin I \sin A\left(\frac{dA}{ds}\right)\right]\vec{i} + \\ + \left[\cos I \sin A\left(\frac{dI}{ds}\right) + \sin I \cos A\left(\frac{dA}{ds}\right)\right]\vec{j} - \sin I\left(\frac{dI}{ds}\right)\vec{k}$$

Hence the wellbore curvature (*dogleg severity* – DLS) is

$$\left|\vec{K}\right| = K = DLS = \sqrt{\left(\frac{dI}{ds}\right)^2 + \sin^2 I\left(\frac{dA}{ds}\right)^2} \tag{B-4}$$

In general, both hole inclination angle and azimuth change with depth and so is the dogleg severity and the contact force between the pipe and wellbore. For practical purposes we use various piece-wise approximations.

From differential geometry we know that the torsion, τ, of a 3D curve is defined as follows:

$$\frac{d\vec{b}}{ds} = -\tau\vec{n} \qquad \text{(B-5)}$$

where:

$\vec{n}$ – normal vector,

$\vec{b}$ – bi-normal vector,

τ – torsion.

It can be shown that the normal and bi-normal vectors can be expressed as below:

Normal:

$$\vec{n} = \frac{1}{K}\left[\cos I \cos A\left(\frac{dI}{ds}\right) - \sin I \sin A\left(\frac{dA}{ds}\right)\right]\vec{i} + \\ + \frac{1}{K}\left[\cos I \sin A\left(\frac{dI}{ds}\right) + \sin I \cos A\left(\frac{dA}{ds}\right)\right]\vec{j} - \frac{1}{K}\sin I\left(\frac{dI}{ds}\right)\vec{k} \qquad \text{(B-6)}$$

Bi-normal:

$$\vec{b} = \left[-\frac{1}{K}\sin A\left(\frac{dI}{ds}\right) - \frac{1}{K}\cos I \sin I \cos A\left(\frac{dA}{ds}\right)\right]\vec{i} + \\ + \left[+\frac{1}{K}\cos A\left(\frac{dI}{ds}\right) - \frac{1}{K}\cos I \sin I \sin A\left(\frac{dA}{ds}\right)\right]\vec{j} + \frac{1}{K}\sin^2 I\left(\frac{dA}{ds}\right)\vec{k} \qquad \text{(B-7)}$$

Substituting Eqn (B-6) and Eqn (B-7) to Eqn (B-5) and solving for τ yields Eqn (4) in the body of the paper.

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