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On Improving the Accuracy of Prediction of the Down-hole Drag & Torque in Extended Reach Drilling (ERD)

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Due to the complexity of forces acting on the down-hole tubular strings in extended reach drilling (ERD), the factors which influence the process should be taken into account as much as possible, in order to predict the down-hole drag & torque more accurately. This can help us to identify and prevent the problems related to down-hole drag & torque in ERD. The effects of such factors, as the tubular buckling, the buoyancy, the mechanical resistance and the friction reducer, on down-hole drag & torque, are taken into account in this paper, in order to improve the accuracy of the prediction of the drag and the torque In addition, the optimization problem for controlling the down-hole drag & torque is solved in this paper. The optimum placement of the friction reducers is determined, in order to minimize the down-hole drag & torque, while allowing for a moderate contact between down-hole tubular and wellbore. The consideration of the mentioned effects, and the control methods presented, were used for the optimization analysis of down-hole drag & torque and the drilling operation, in extended-reach drilling. Interpretation of the field data provides us with an insight into the down-hole drag & torque computations.

Keywords: Extended reach drilling, Down-hole drag & torque, Tubular buckling, Mechanical resistance, Friction reducer

1 Introduction

Down-hole drag & torque (D&T) modeling has played an important role in the optimal design and the safe operation control in extended reach drilling. Although several down-hole D&T models have been developed in the past decades, their

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mathematical bases have not changed significantly during the developing process [Mason and Chen (2007)].

The most commonly used method for the prediction of down-hole D&T was originally developed by Johancsik, Friesen and Dawson (1984). This model is referred to as soft-string model, in which the effect of the tubular stiffness is ignored, for convenience in analysis [Mason and Chen (2007)]. The problem was cast in the form of a differential equation, by Sheppard, Wick and Burgess (1987). A three-dimensional soft-string model, based on static equilibrium, was presented by Maidla and Wojtanowicz (1987). Their model takes into account the hydrodynamic viscous drag related to the effect of the pressure drop in the drilling annulus. However, this model overestimates the hydrodynamic viscous drag. He, Sangesland and Halsey (1991) developed a down-hole D&T model, which is similar to the model developed by Johancsik, Friesen and Dawson (1984), whose work takes into account the effect of the axial and circumferential velocities on the down-hole D&T.

The soft-string model is not accurate in tortuous wellbores, because it does not take into account the down-hole tubular stiffness. In order to overcome the shortcomings of the soft-string model, Ho (1988) developed a stiff-string model, based on the large deformation theory of a beam-column. Later, Mitchell and Samuel (2009) developed a comprehensive stiff-string model which accounts for the effect of tubular-wellbore contact position on down-hole D&T. However, the former models were based on a continuous contact hypothesis, and did not need to calculate the contact points between the down-hole tubular and wellbore. Menand,Sellami,Tijani et al (2006) developed a numerical model to calculate the contact points, which can solve the time-consuming problem through the finite element analysis. The new model takes into account the details of the down-hole tubular configuration, and any external forces acting on the down-hole tubular, including hydraulic and temperature-induced force.

There are complex forces acting on the down-hole tubular, and some forces are handled in a very simplified manner or not considered at all. In addition, one needs to solve problems of optimization, in order to implement a down-hole D&T control. Otherwise, the results cannot be directly used in extended-reach drilling, or may be simply replaced by drilling engineers' experience. Therefore, more factors which influence the process should be taken into account, in order to improve the accuracy of prediction and implement the control measures. In this paper, effects of such factors as the buckling of the tubular, the buoyancy of the drilling fluid, the mechanical resistance and the friction reducer, on down-hole D&T are taken into consideration. In addition, the location of the friction reducer was optimized, in order to effectively reduce the down-hole D&T.

2 A model for the Down-hole D&T

The purpose of a down-hole D&T model is to compute the effective axial force, lateral force and torque, by solving the equilibrium equations of forces and moments acting on any differential element of down-hole tubular. The following equations can be obtained, as shown in the authors' previous paper [Sun and Gao (2011)]:

$$\begin{cases} \frac{dF_e}{ds} + EIk\frac{dk}{ds} + w_{bp}t_z - \mu_d w_c(1 - kr_o\cos\theta) = 0 \\ \frac{dM_t}{ds} - \mu_t r_o w_c = 0 \end{cases} \tag{1}$$

$$w_c = \frac{\sqrt{(F_e k + \tau^2 EIk + w_{bp}n_z - \tau k M_t)^2 + \left[w_{bp}b_z - (2\tau EI - M_t)\frac{dk}{ds}\right]^2}}{\sqrt{1 + \mu_t^2 + \tau^2\mu_d^2 r_o^2 + 2\mu_t\mu_d r_o\tau}} \tag{2}$$

$$\begin{aligned} \sin\theta = {} & \frac{w_{bp}b_z - (2\tau EI - M_t)\frac{dk}{ds} + w_c\mu_t r_o k}{w_c\left[1 + (\mu_t + \mu_d r_o\tau)^2\right]} \\ & - \frac{(\mu_t - \mu_d r_o\tau)(F_e k + \tau^2 EIk + w_{bp}n_z - \tau k M_t)}{w_c\left[1 + (\mu_t + \mu_d r_o\tau)^2\right]} \end{aligned} \tag{3}$$

$$\cos\theta = (F_e k + \tau^2 EIk + w_{bp}n_z - \tau k M_t + \mu_t w_c\sin\theta - \tau\mu_d w_c r_o\sin\theta)/w_c \tag{4}$$

$$\begin{cases} t_z = \cos\alpha \\ n_z = \frac{d\alpha}{ds}\frac{\sin\alpha}{k} \\ b_z = -\frac{d\varphi}{ds}\frac{\sin^2\alpha}{k} \end{cases} \tag{5}$$

$$k = \sqrt{\left(\frac{d\alpha}{ds}\right)^2 + \sin^2\alpha\left(\frac{d\varphi}{ds}\right)^2} \tag{6}$$

$$\frac{dk}{ds} = \frac{\frac{d\alpha}{ds}\frac{d^2\varphi}{ds^2} + \frac{d\varphi}{ds}\left(\frac{d\alpha}{ds}\frac{d\varphi}{ds}\cos\alpha + \frac{d^2\varphi}{ds^2}\sin\alpha\right)\sin\alpha}{k} \tag{7}$$

$$\tau = \frac{\sin\alpha\left(\frac{d\alpha}{ds}\frac{d^2\varphi}{ds^2} - \frac{d\varphi}{ds}\frac{d^2\alpha}{ds^2}\right) + \frac{d\varphi}{ds}\left[\left(\frac{d\alpha}{ds}\right)^2 + k^2\right]\cos\alpha}{k^2} \tag{8}$$

whereF_e is the effective force, N; M_t is the torque, N•m; E is the Young's modulus, N/m^2; I is the moment of inertia of down-hole tubular, m^4; k is the curvature, m^{-1}; τ is the wellbore torsion, m^{-1};w_{bp} is the buoyant weight per unit length of down-hole tubular, N/m; w_c is the contact force per unit length between down-hole tubular and wellbore, N/m; μ_d is the axial friction factor which is positive while the tubular is sliding down the wellbore; μ_t is the circumferential friction factor; r_o is the outside radius of down-hole tubular, m; θ is the contact position angle with respect to the normal vector; t_z, n_z, b_z are respectively the vertical components of the tangent vector, the normal vector and the binormal vector; α is the hole angle, degrees; and ϕ is the azimuth angle, degrees.

3 The Effects of Various factors on the down-hole D&T

3.1 Tubular buckling

Down-hole tubular buckling can increase the contact force between the tubular and the wellbore, and thus reduce the efficiency of transfer of the tubular axial force. It is very important to estimate accurately the additional contact force resulting from the down-hole tubular buckling. Based on the previous papers [Dawson and Paslay (1984); Gao (2006)], the critical loads for the sinusoidal buckling and the helical buckling of a down-hole tubular in an inclined wellbore can be respectively expressed as follows:

$$F_{cs} = 2\sqrt{\frac{EIw_{bp}\sin\alpha}{r_c}} \tag{9}$$

$$F_{ch} = 2.75\sqrt{\frac{EIw_{bp}\sin\alpha}{r_c}} \tag{10}$$

where F_{cs} and F_{ch} are respectively the critical loads for sinusoidal buckling and for helical buckling, N; r_c is the effective radial clearance, m; α is hole angle, (degrees). Also, Eq. (9) and Eq. (10) can be generalized for the curved sections in an extended-reach well, by replacing the term $w_{bp}\sin\alpha$ in the above two equations with the normal contact force w_c [He and Kyllingstad (1995)].

The effective radial clearance can be expressed as follows:

$$r_c = \frac{d_c - d_e}{2} \tag{11}$$

$$d_e = \frac{d_{dp}(l - l_{tj})}{l} + \frac{d_{tj}l_{tj}}{l} \tag{12}$$

where d_c is inside diameter (ID) of wellbore, m; d_e is the effective diameter, m; d_{tj} is the outside diameter (OD) of the tool joint, m; l is the length of single pipe, m; l_{tj} is the length of the tool joint, m; d_{dp} is the outside diameter of pipe, m.

The additional contact force caused by the sinusoidal buckling or the helical buckling of a down-hole tubular string is:

$$w_b = \frac{r_cF_e^2}{C_bEI} \tag{13}$$

where w_b is the additional contact force per unit length between the tubular and the wellbore, caused by the tubular buckling, N/m; C_b is the coefficient of buckling, which is equal to 8 for tubular sinusoidal buckling, and 4 for tubular helical buckling, as believed by most relevant researchers at present.The other symbols are the same as the above.

As compared with the scaled laboratory experiments, the full-scale buckling tests of the down-hole tubular have higher credibility. The full-scale buckling tests of 2-7/8", 3-1/2" and 4" drill-pipes while sliding and rotating inside a 7" casing were performed by Mitchell, Moore, Franks et al (2011), which showed that drill-string buckling occurs at loads which are smaller than those predicted by the current models.

3.2 Buoyancy

When the inside and outside of the tubular are submerged in drilling fluids with the same density, the buoyancy factor should be:

$$f_b = 1 - \frac{\rho_m}{\rho_s} \tag{14}$$

where f_b is the buoyancy factor, dimensionless; ρ_s is the density of the tubular material, kg/m^3; ρ_m is the density of the drilling fluid, kg/m^3.

If the drilling fluids inside and outside of the tubular have different densities, the buoyancy factor mentioned in most studies [Aadnoy and Kaarstad (2006); Han (1995); Juvkam-Wold and Baxter (1988)] is:

$$f_{b1} = 1 - \frac{\rho_o A_o - \rho_i A_i}{\rho_s (A_o - A_i)} \tag{15}$$

where ρ_o is the density of the drilling fluid outside the tubular, kg/m^3; ρ_i is the density of the drilling fluid inside the tubular, kg/m^3; A_o is the outside cross-sectional area of the tubular, m^2; A_i is the inside cross-sectional area of the tubular, m^2.

Due to the different sizes of the down-hole tubular body, and the tool joint, the average weight per unit length is widely used in petroleum engineering. The tool joint has not only a heavy impact on the weight per unit length, but also an insignificant impact on the buoyancy. Therefore, the recommended buoyancy factor [He and Kyllingstad (1995)] is:

$$f_{b2} = 1 - \frac{\rho_o g A_o - \rho_i g A_i}{w_p} \tag{16}$$

where w_p is the tubular weight per unit length in air, N/m; g is the gravitational acceleration, m/s^2.

As shown in table 1, when the inside of the tubular is filled with air, the buoyancy factors obtained from Eq. (15) are smaller than those obtained from Eq. (16). There is little difference between f_{b1} and f_{b2} for a 7" casing, but a larger difference for a 5-1/2" drill-pipe (DP), 5-1/2" heavy weight drill-pipe (HWDP) and a 9-5/8" casing. The casing, the drill-pipe, and the heavy weight drill-pipe may be emptied during drilling operations, and the errors in their buoyancy factors will lower the accuracy of prediction of down-hole D&T.

Table 1: Buoyancy factors when the tubular inside filled with air

Tubulars	OD (mm)	Max. OD (mm)	ID (mm)	Weight (kg/m)	ρ_o (g/cm^3)	f_{b1}	f_{b2}	Error (%)
5-1/2" HWDP	139.70	177.80	101.60	82.22	1.15	0.689	0.786	12.30
5-1/2" DP	139.70	177.80	121.36	37.83	1.15	0.403	0.534	24.57
7" casing	177.80	177.80	157.07	43.16	1.15	0.333	0.338	1.68
9-5/8" casing	244.48	244.48	222.38	64.74	1.15	0.151	0.166	8.95

3.3 Mechanical resistance

Due to the complex characteristics of extended-reach drilling, a friction factor represents not only the practical mechanical friction, but also many other down-hole effects, such as the cutting bed, stabilizer, centralizer, wellbore tortuosity and spiraling [Mason and Chen (2007)]. These unwanted contributions can produce an additional mechanical resistance. When the trend of the mechanical resistance is the same as friction, a larger friction factor can be used to calculate the hook load and the rotary table torque more accurately. Otherwise, it would produce a considerable error or even lead to wrong results.

Due to their large size effects, the centralizers and the stabilizers can penetrate into the well wall in the open hole section, which will result in an additional mechanical resistance. The values of mechanical resistance would be different for different centralizers or stabilizers, and be changed while the tubular string slides into or out of the hole. In this paper, the total value of mechanical resistance is considered as the difference between the calculated hook load and the its measured value. For the purposes of calculation, it is simplified that the value of mechanical resistance does not change for every single centralizer or stabilizer in the same movement direction. Calculations are repeated with a changed friction factor and mechanical resistance in an open hole, until the calculated hook load matches with the its measured value. The back calculated value of the friction factor, and the value of mechanical resistance in an open hole, can be used to predict and monitor the down-hole D&T in the same hole section of other adjacent wells.

The significantly higher friction factors are generally associated with the casing running, other than tripping of the drill-string in the same wellbore [Mason and Chen (2007)]. The effect of mechanical resistance is an important factor resulting in this phenomenon, except for the effects of wellbore clearance and tubular stiffness, which are not properly taken into account in the down-hole D&T model. Except for the traditional method based on the friction factors calculated by directional drilling operations, another method, that of combining the slightly larger friction factor with the additional mechanical resistance to the larger size components in an open hole, can be used to predict the drag of the casing running.Usually, the more dangerous case should be analyzed, to ensure the casing running to the target depth.

3.4 Friction reducer

Friction reducers are often used to keep the tubular away from the wellbore, so that the down-hole D&T, and the casing wear, can be reduced in extended reach drilling. At present, a fixed value of the friction factor is widely used to calculate the down-hole D&T of the tubular string with the spaced friction reducers in most relevant computer programs. However, the corresponding depth of a down-hole tool increases with the drilling footage. Therefore, the calculated hook load and the rotary table torque are not in agreement with the measured values in most cases.

In this paper, the friction factor was divided into the open hole friction factor (FF), and the down-hole tool friction factor (FFt). Generally FF and its corresponding depth remain unchanged in a particular interval, and the distance between the bit depth and the corresponding depth of the down-hole friction reducer keeps constant with the increase of drilling footage. Currently, two types of down-hole D&T models are used in extended-reach drilling. One model does not take into account the contact point between the tubular string and the well wall. Another model needs to calculate the maximum deflection of the tubular string, off the wellbore axis between any two adjacent reducers, in order to judge whether the tubular string contacts the well wall or not. Also, the contact force between the friction reducer or tubular and the well wall can be determined simultaneously. In actual engineering practice, the FFt is used for the friction reducer and the FF for the contact point between the tubular and the open hole.

4 Control of down-hole D&T

One of the most effective methods to reduce down-hole D&T is to add friction reducers to the down-hole tubular string. The deflection model developed by Juvkam-

Wold and Wu (1992) is currently used to calculate the spacing of the friction reducers. However, the value of the spacing recommended by adjusting the maximum deflection is not in multiples of length of the single pipe, and is not usually accepted in directional drilling operations.Thus, the placement and number of the friction reducers are often determined by experience. Due to the limitation in the spacing and the number of friction reducers in an actual drilling operation, the placement and number should be optimized in order to realize the ideal down-hole D&T reduction.

Allowing for a proper contact between tubular and the well wall, the optimal placement of friction reducer is recommended by simulating the distribution of down-hole D&T. Generally, the spacing between two friction reducers is the same in entire tubular string. Based on the well structure or drilling operation requirements, the down-hole tubular string can be also divided into two or three sections and the spacing is the same in each section but different from each other. For optimization purposes, the spacing and the number of the friction reducers are first given, and the calculations are repeated by changing the placement of the first reducer until the drag or rotary table torque tends to the minimum value. The rotary drilling mode is preferred in extended reach drilling, and the minimum rotary table torque is selected as the preferred criterion. Therefore, the scheme for an optimal placement of a friction-reducer can be obtained, under the given spacing and the given number of the friction reducers. Then, the above calculations are repeated by changing the number of friction reducers , and the variation of the rotary table torque, with the number of friction reducers and their optimal placement with different numbers, can be obtained. If the spacing needs to be optimized, the above steps are repeated to obtain the optimal placement, the spacing and the number of the friction reducers.

There are mainly four factors which determine the number of friction reducers. These factors are: the friction reduction requirement, the improving efficiency, the available number of friction reducers, and the control requirement of equivalent circulating density (ECD). The more rational placement and the number of friction reducers can be determined by considering the above four factors comprehensively.

5 Case study

The Well A02H03 is an extended-reach well in Liuhua 11-1 oilfield in the South China Sea, China. It was sidetracked from well A2ERW1 which was the first extended-reach well in this oilfield. The well profile of the well A02H03 is shown as Fig. 1. The flowchart of computer program is shown in Fig. 2 for calculating the down-hole D&T in extended-reach drilling.

1. 9-5/8" casing running

Fig. 3 shows the slack-off weight of the 9-5/8" casing (as shown in table 2), with a floating collar. The distance between the floating collar and the casing shoe was 1022.0 m. The density of the drilling fluid was 1.20 g/cm^3. Though the floating and the drilling fluid filling were taken into consideration, the calculated hook loads were far different from the measured values. When the values of mechanical resistance were considered on each centralizer (see table 3), the calculated results were in good agreement with the measured values (see Fig.4). It is noted that the calculated result as shown in table 3 is not a manifestation of the true mechanical resistance of each centralizer, but simply a reasonable explanation for the theoretical analysis and the practical operation. Generally, the friction factor in cased hole is greater than that in open hole (see Fig.4) in this oilfield , mainly because of the better lubrication of the mudstone in open hole.

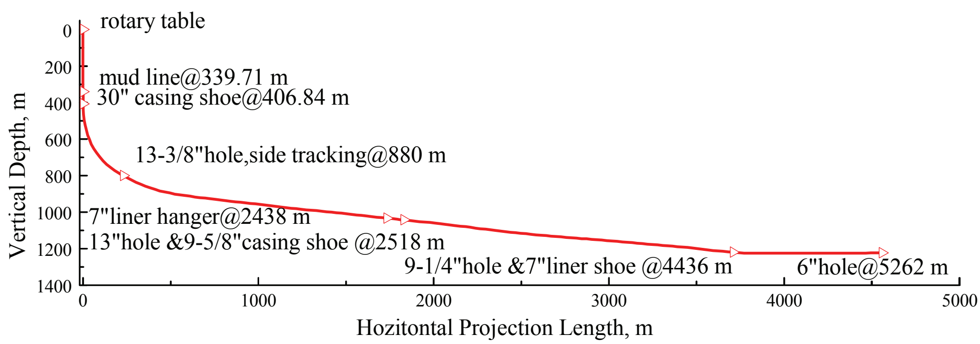

Figure 1: Well profile of well A02H03

Table 2: The data of 9-5/8"casing

Casing	OD (mm)	ID (mm)	Weight (kg/m)	Length (m)
9-5/8" casing	244.475	222.377	64.74	2181.4
9-5/8" casing	244.475	220.497	69.94	338.0

2. 9-1/4" hole drilling

To reduce the rotary table torque and the casing wear, it was decided to pull 5-1/2" drill-string (see table 4) out of the hole, and use one LoTAD friction reducer per stand when the 9-1/4" hole was drilled to reach 3411 m depth. The circumferential friction factor of friction reducer was 0.20. And the

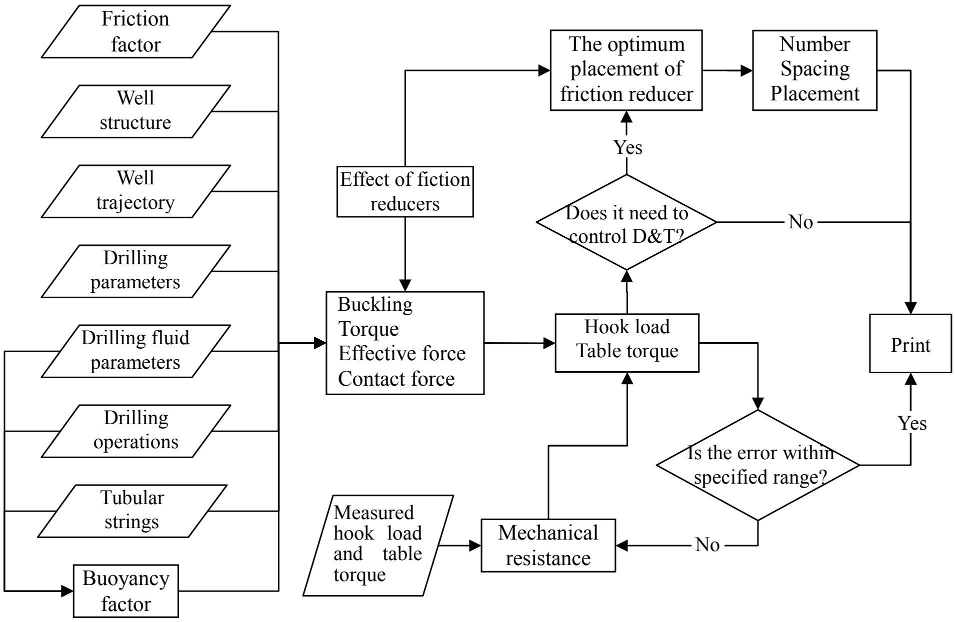

Figure 2: Flowchart of computer program for calculating down-hole D&T

Table 3: Mechanical resistances caused by elastic centralizers

Number	Length to casing shoe (m)	Mechanical resistance (kN)	Number	Length to casing shoe (m)	Mechanical resistance (kN)
1	3.957	4.45	11	300.818	8.90
2	15.974	0.00	12	348.736	8.90
3	58.704	4.45	13	430.645	8.90
4	82.338	0.00	14	513.624	8.90
5	106.182	0.00	15	595.753	8.90
6	123.243	4.45	16	678.452	8.90
7	146.677	0.00	17	760.701	8.90
8	170.671	0.00	18	842.28	8.90
9	194.615	4.45	19	912.752	8.90
10	253.770	8.90			

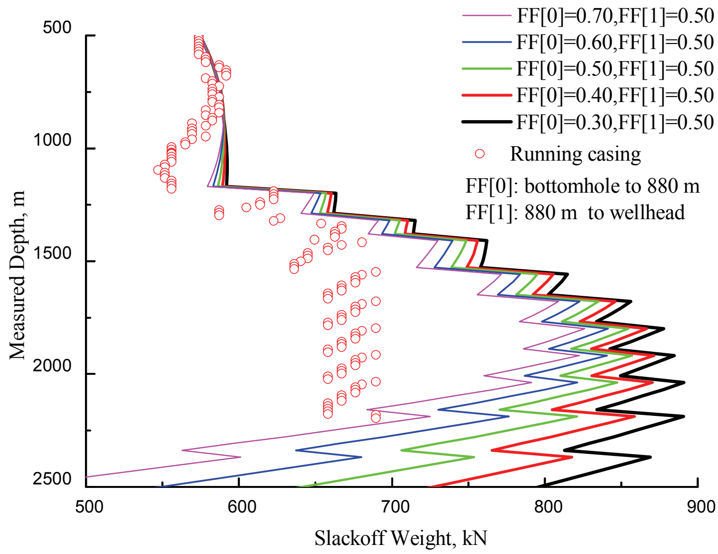

Figure 3: Mechanical resistances not taken into account

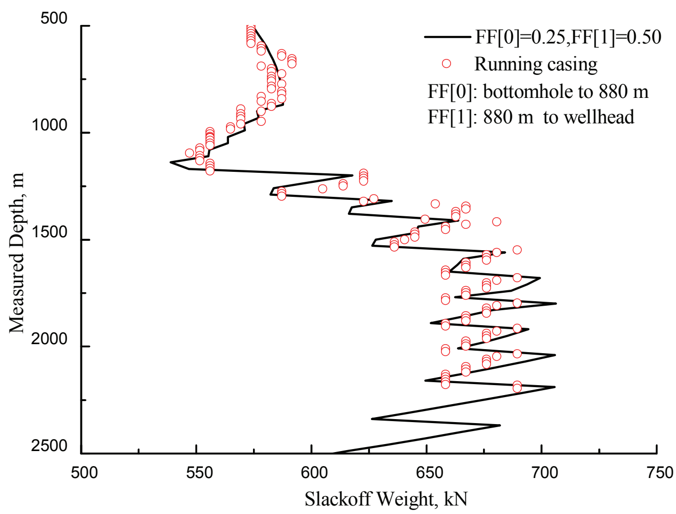

Figure 4: Mechanical resistances taken into account

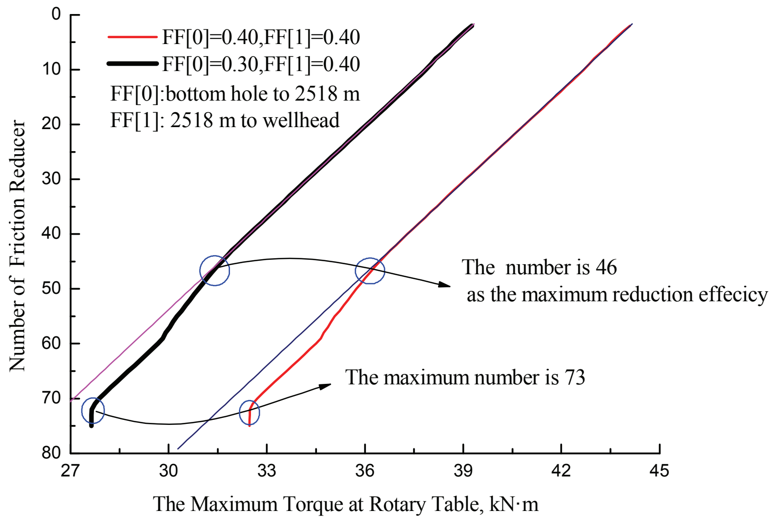

Figure 5: Recommendation for the number of friction reducers in 9-1/4" hole

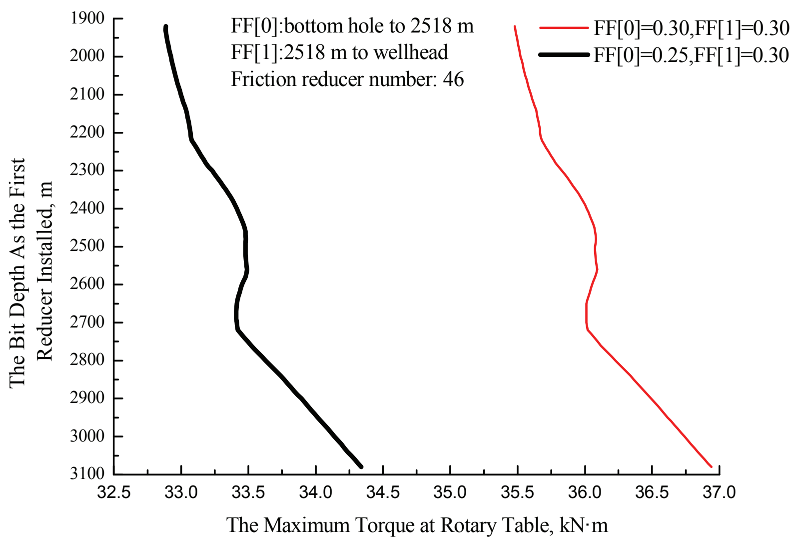

Figure 6: Rational placement of down-hole friction reducers

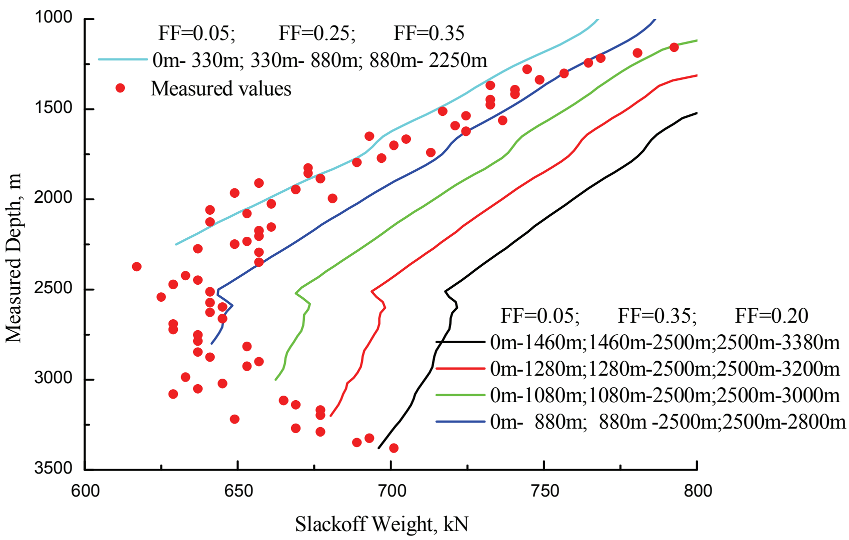

Figure 7: The friction factor with the fixed depth

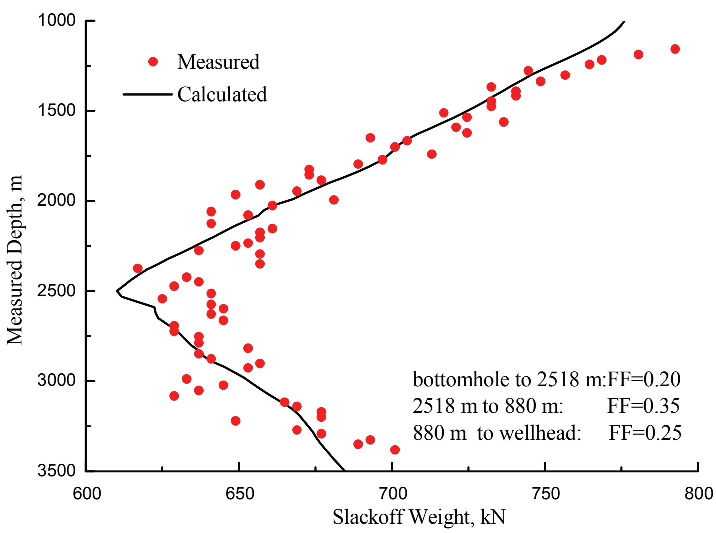

Figure 8: Effect of friction reducers on down-hole drag taken into account

Table 4: 5-1/2" drillstring used in 9-1/4" hole

Drill Pipe (DP)	OD (mm)	Max. OD (mm)	ID (mm)	Weight (kg/m)	Length (m)
5-1/2"HWDP	139.70	184.15	82.55	91.72	70
5-1/2" DP	139.70	177.8	108.61	37.83	4366

density of the drilling fluid was 1.20 g/cm^3. The weight on bit (WOB) was 44.48 kN, and the torque on bit (TOB) was about 6.78 kN·m. Due to a larger window in safe density of drilling fluid, the use of friction reducers and the increase of ECD would not affect the drilling operations. Based on the principle of the minimum torque, the number of friction reducers was determined as 46 for the most efficient reduction, and over 73 when the torque would no longer reduce (see Fig.5). It was decided to use 46 friction reducers, and the first reducer was installed when the drill bit reached a depth of 1920 m (see Fig.6).

As the bottom hole assembly (BHA) consisted of a float valve and a reamer with a small nozzle, the drill-string was filled up per 500 meters, during which the drilling fluid in the annulus flowed in through the small nozzle to result in the fluctuation of the measured slack-off weights . Fig.7 shows the slack-off weights calculated by the conventional method, which made the calculated values hardly consistent with the measured values. As seen in Fig.8, the calculated slack-off weights were in good agreement with the measured values when the method developed in this paper was used and the average axial friction factor of friction reducer was 0.10.

A 5-1/2" drill-string (as shown in table 5), filled with air inside it, was used for the 9-1/4" hole wiper before 7" liner running. It is shown in Fig.9 that the hook loads calculated using Eq. (22) were in good agreement with the measured values, while the empty drill-string was pulled out of the hole and pushed into the hole. The friction factor was 0.30-0.35. However, the friction factor would be wrong if Eq. (21) was used to calculate the buoyancy factor. This indicated that the measured hook loads were higher than the calculated static weights, which seemed to mean that the friction force was in the same direction as the movement of drill-string.Obviously, this is impossible.

3. 6" hole drilling

Fig. 10 shows hook loads while the drill-string (see table 6) slides into the hole before the 6" hole drilling. The density of drilling fluid was 1.03 g/cm^3.

Table 5: The data of 5-1/2" drillstring while making a wiper trip

Drill Pipe (DP)	OD (mm)	Max. OD (mm)	ID (mm)	Weight (kg/m)	Length (m)
5-1/2"HWDP	139.70	184.15	82.55	91.72	56.0
5-1/2"DP	139.70	168.28	121.36	37.83	2619.0
5-1/2" HWDP	139.70	184.15	82.55	91.72	196.4
5-1/2"DP	139.70	168.28	121.36	37.83	349.2
5-1/2" HWDP	139.70	184.15	82.55	91.72	196.4
5-1/2"DP	139.70	168.28	121.36	37.83	145.5
5-1/2" HWDP	139.70	184.15	101.60	82.22	794.8
5-1/2"DP	139.70	168.28	121.36	37.83	79.0

46 LoTAD friction reducers were used by one per stand, and the bit depth was 3130 m when the first reducer was installed. The axial friction factor of LoTAD friction reducer was 0.10. Also, 95 WWT friction reducers were added to the drill-string with one per two drill pipes, and the first reducer should be installed when bit depth was 1045 m. The axial friction factor of the WWT friction reducer was 0.25. The impact of tubular buckling, the filling of drilling fluid, and the friction reducers was considered in calculations. The calculated hook loads were in good agreement with the measured values, when the tubular sinusoidal buckling was taken into account in this calculation. The effective axial force (see Fig. 11) shows that the sinusoidal buckling occurred in hole section from 1045 m to 2130 m when the bit depth was 4000 m. There is a sufficient precision for the critical sinusoidal buckling load and its additional contact force, in many cases. At that moment, the remaining hook load was only 93 kN (the superimposed weight of top drive and traveling block was 533.80 kN). With the increase of the bit depth, the drill-string buckling would be more severe and make the sliding down difficult. Therefore, the drill-string was pulled up to eliminate its buckling, and then was run to the bottom hole by rotating the drill-string with a low rotary speed.

6 Conclusions

1. Due to the fact that the tool joint has an impact on the weight per unit length and buoyancy of the down-hole tubular string, the computational model based on the traditional buoyancy factor method is not preferable to improve the accuracy of prediction of the down-hole D&T, if the densities of the fluid

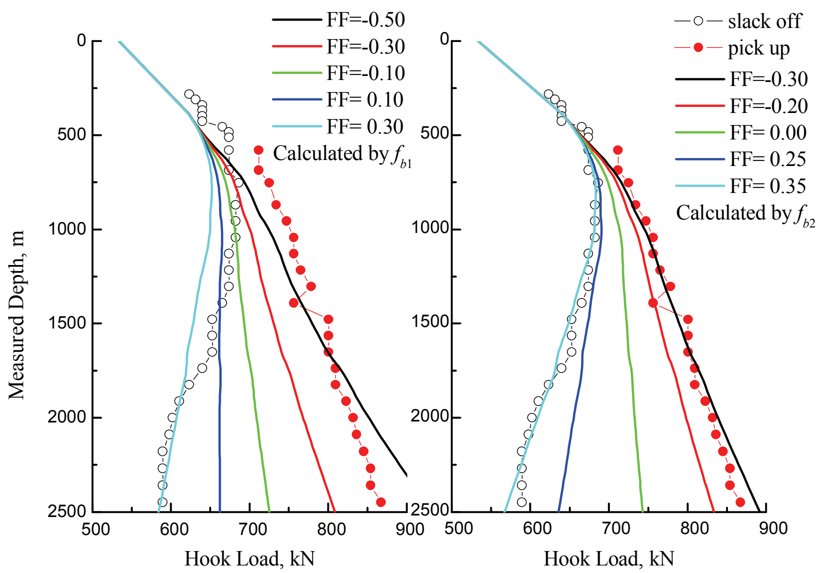

Figure 9: Effect of buoyancy factor on down-hole D&T

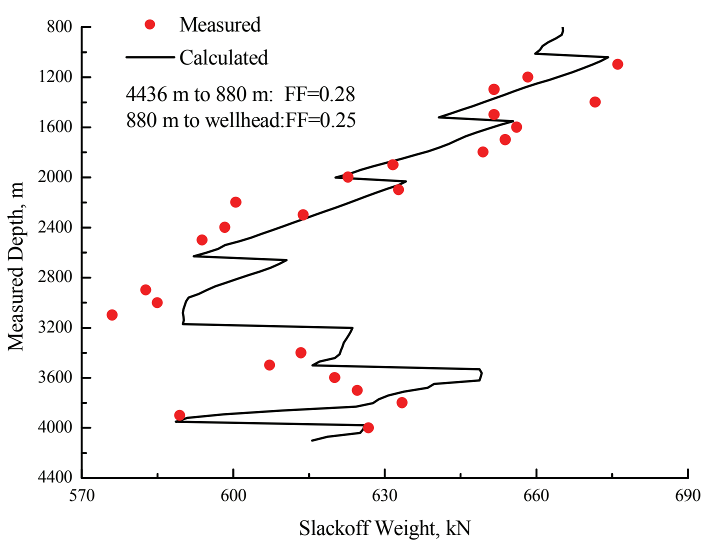

Figure 10: Slackoff weight in 6" hole

Table 6: The data of drillstring for 6" hole

Drill Pipe (DP)	OD (mm)	Max. OD (mm)	ID (mm)	Weight (kg/m)	Length (m)
3-1/2" HWDP	85.17	120.65	57.15	38.18	62
3-1/2" DP	85.17	104.78	70.21	20.95	2894
5-1/2" DP	136.04	180.98	108.61	37.83	4366

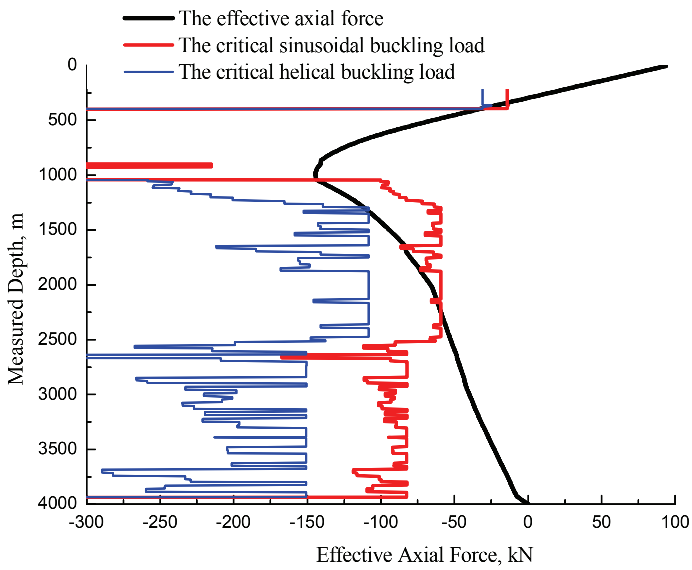

Figure 11: Effective axial force while drillstring sliding down the hole

inside & outside of the down-hole tubular are different. Also, the effect of the tubular buckling on down-hole D&T cannot ignored in the prediction.

2. The effect of mechanical resistance is one of the reasons why there is a higher friction factor in the casing running, other than the tripping of drill-string in the same wellbore. Thus, it is necessary and reasonable to consider the mechanical resistance to the large size components, for improving the accuracy of prediction of the down-hole D&T, in an open hole.

3. The friction factor should be divided into the open hole friction factor (FF) and the down-hole tool friction factor (FFt). A method has been developed to reasonably account for the effect of friction reducers on the down-hole D&T. Thus, the more rational placement and number of friction reducers are

recommended to control effectively the down-hole D&T, while allowing for a moderate contact between down-hole tubular and the well wall.

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