A STUDY ON EFFECT OF DRAG & TORQUE ON BUCKLING OF DRILLSTRING IN HORIZONTAL WELLS

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ABSTRACT

Tubular Buckling is defined as lost in the original rectilinear status due to axial compression load. Tubular buckled in a horizontal section of oil or gas wells may cause problems such as: Casing wear/failure, Eliminates the transmission of axial load to the Bit, Drillpipe fatigue, Bit direction change, Severe drag and torque, Tubing seal failure, Connection failure.

According to what mentioned above, it would be necessary to make an investigation on the effect of drag and/or torque on buckling of tubular inside horizontal wellbore (the so-called torque-drag-buckling relationship), the study depends mainly on conducting experimental tests, and utilize the results to predict the tubular behavior under different circumstances of torque and drag, also these results were compared and verified with a finite element model and some of theoretical published approaches.

Results obtained from this study indicates that torsion load had a little effect on buckling of tubular in horizontal section, this effect increase with decrease in pipe stiffness. Helical buckling tends to increase highly drag force, and thus eliminates transmission of axial load to the bit. Drag force tend to increase with torsion load as a result drag increase helical buckling. For prediction of helical buckling Lubinski and Woods, Gao Deli and Chen et. al provided a good agreement with one-end hinge state, and Dellinger equation is the best for two-end hinge case.

Key word: *Drillstring mechanics, Tubular Buckling Prediction, Drag and Torque Prediction.*

1. INTRODUCTION

Buckling is defined as a lost in original rectilinear of a tubular due to axial compression load, even a large bridge can collapse due to buckling, when applied to compressive loads. The buckling of tubular inside wellbore has been the subject of man researches and articles in the past. These theories provide critical loads that cause sinusoidal and helical buckling in perfect and smooth well geometry. These theories are sometimes conflicting and have the common weakness: hole friction and natural tortuosity are ignored in the modeling. Knowing the buckled tubular configuration is important to prevent failures and determine the possibility of further enforce to a tool along a horizontal well. Critical load (P_{cr}) is the axial load at which a tubular started to bend. Sinusoidal Buckling is a minimum axial load at which a tubular displays a sinusoidal buckling shape, it can be expressed by the following equation:

$$\theta(z) = \sum_{n=1}^{k} \theta_{on} \sin \frac{n_{1/2}\pi z}{l} \tag{1}$$

Helical Buckling took place as axial compressive load increase beyond critical buckling. The developed helix configuration is expressed in the following mathematical form:

$$\theta = 2n\pi z / L \tag{2}$$

2. THE SIGNIFICANT OF THE STUDY

Buckling of a tubular (casing, drillstring, tubing, and coiled tubing) is a critical problem presented in oil/gas field drilling operations for many years, the importance of studying this phenomenon is because it cases many serious problems (eliminate axial load transmission, drillstring failure, casing wear and failure …etc.). Torque and drag are important factors in horizontal and extended reach drilling and can be the limiting factor in how far the well can be drilled, and they associated with each other and may be more effective in extended reach and horizontal well.

3. LITERATURE REVIEW

Lubinski and Woods[26] investigated on helical buckling of drillstring in gas/oil wells and conducted experimental tests for buckling of a vertical drillstring and developed the following equation:

$$F_{hel} = 2.8(EI)^{0.504}(W)^{0.496}(\frac{\sin\alpha}{r})^{0.511} \quad (3)$$

Lubinski continue studying the effects of changes in temperature and pressure. The study led to the derivation of the helix pitch-force relationship:

$$p^2 = \frac{8\pi^2 EI}{F} \quad (4)$$

The above equation is valid only for weightless pipe.

Paslay and Bogy[27] studied stability of a circular rod lying on the low side of an inclined circular hole. They developed the following expression for critical compressive load:

$$F_{crit} = 2\left(\frac{EI\rho Ag}{r}\sin\phi\right)^{1/2} \quad (5)$$

Dellinger investigated Lubinski and Woods results and derived the following formula:

$$F_{hel} = 2.93(EI)^{0.479}(q)^{0.522}(\frac{\sin\alpha}{r})^{0.436} \quad (6)$$

R. F. Mitchell[20] developed and solved the equilibrium equation for helical buckling of a tube. He concluded that packer has a strong influence on the pitch of the helix, and provided the following sets of equations:

$$EI\frac{d^3}{dz^3}u_1 + F\frac{d}{dz^2}u_1 - F_1 = 0$$

$$and,$$

$$EI\frac{d^3}{dz^3}u_2 + F\frac{d}{dz^2}u_2 - F_2 = 0 \quad (7)$$

Cheatham et al[29] studied helical buckling phenomena using Lubinski derivation of the helix-pitch-force relationship, he derived the following bending strain energy for the helix:

$$E_b = 8\pi^4 EILr^2/(L/n)^2 \quad (8)$$

And the external work by force F is:

$$W_e = 2FL\pi^2 r^2/(L/n)^2 \quad (9)$$

Wu and Juvkam-Work[30] investigated the frictional drag of helical buckling pipes in extended reach and horizontal wells. The differential equation for static axial force balance is given by:

$$F_{cr} = \frac{4EI}{rR}\left[1 + \sqrt{1 + \frac{rR^2 w\sin\bar{\theta}}{4EI}}\right] \quad (10)$$

Y. W. Kwon[17] provided the following solution for helical buckling yield shapes of pipes:

$$\frac{2\pi}{P} = \frac{d\theta}{dz} = a_0 + a_1 z + a_2 z^2 + a_3 z^3$$

$$where: a_0 = \left(\frac{F}{2EI}\right)^{\frac{1}{2}}; \quad a_1 = \frac{a_0}{3L}\left[\frac{1}{2} + \frac{2}{3}a_0^2 L^2 - (6\tfrac{1}{4} + 2\tfrac{2}{3}a_0^2 L^2 + \tfrac{4}{9}a_0^4 L^4)^{\frac{1}{2}}\right]$$

$$a_2 = -\left(\frac{a_1}{2L} + \frac{3a_1^3}{2a_0}\right); a_3 = \left(\frac{a_0}{3L} + \frac{2}{3}a_1\right)a_0^2; \quad L = \frac{F}{W} \quad (11)$$

Robert F. Mitchell[18] solved the equilibrium equations by an approximate method. This solution gives a method for evaluating the range of application of Lubinski results. Mitchell contact force equation can be expressed as:

$$W_n = \frac{rF^2}{4EI}; \quad (For \quad C_0 = 0) \quad (12)$$

Yu-che Chen, Yu-Hsu Lin, and J. B. Cheatham Chen et al.[16] presented the following theoretical equation for predicting the helical buckling of a pipe in a horizontal hole:

$$F^{\#} = 2\sqrt{\frac{2EIW_e}{r}} \tag{13}$$

Wu Jiang [24] provided the following equation to predict the torque and drag force inside a horizontal section:

$$T = \frac{\mu L D_{TJ} w_B \sin\alpha}{2000} \quad ; \quad D = \mu L w_B \sin\alpha \tag{14}$$

In addition to this, Wu developed the following equations to determine the critical buckling load for large and small curvature of a wellbore respectively:

$$F_{cr} = \frac{4EI}{rR}\left[1+\sqrt{1-\frac{rR^2 w\sin\bar{\theta}}{4EI}}\right]$$

$$F_{cr} = \frac{4EI}{rR}\left[1+\sqrt{\frac{rR^2 w\sin\bar{\theta}}{4EI}-1}\right]$$

Jiang Wu and H. C. Juvkam-wold [30] presented the following equation to estimate the load required to cause a tubular to buckle helically in horizontal wellbore:

$$F_{hel} = 2\left(2\sqrt{2}-1\right)\sqrt{\frac{EIW_e}{r}} \tag{15}$$

And under torsion load effect becomes:

$$\overline{F}_h = \sqrt{\frac{(8EI-2TP/\pi)W}{R}} - \frac{\pi T}{P} \tag{16}$$

Stefan Miska [11] gives a theoretical approach to predict buckling behavior in inclined and horizontal wellbore also considering the effect of torque using conservation of energy and the principle of virtual work. Miska concluded that torque tends to reduce critical buckling force and this reduction tend to reduce critical buckling force, and that the reduction depends on torsion value, wellbore inclination, and large clearances. The force F can be determined as:

$$F = \frac{4\pi^2 EI}{P^2} - \frac{2\pi T}{P} + \frac{w\sin\alpha}{2\pi^2 r} p^2$$

$$F = \frac{8\pi^2 EI}{P^2} - \frac{3\pi T}{P} \tag{17}$$

Miska also provided the following formula to determine helical buckling of weightless tubular-buckling:

$$F_{PB} = 4\sqrt{\frac{2EIW\sin\alpha}{r}} \tag{18}$$

X. He[31] developed the following theoretical module for determining the interaction between torque and helical buckling:

$$f_r \approx \frac{rF^2}{4EI}\left(1+\frac{T}{\sqrt{EIF/2}}\right) \tag{19}$$

Gao Deli et. al [8] presented a real configuration of helically buckled tubular by describing an analytical solution based on equilibrium method. Gao developed the following critical helical buckling load equation:

$$F_{crh} \geq 2.802\sqrt{\frac{EIq}{r}} \tag{20}$$

Gao formula was found to be in accordant with results obtained by Chen's approach.

4. LABORATORY EXPERIMENT

A laboratory Experiment was conducted to simulate and analyze interactions between buckling, drag, and torque using a steel pipe (represent drillstring) inside a glass tube as horizontal wellbore. The test's results can be recorded on an industrial computer (Fig (1)). The experimental tests were conducted and results were obtained from experiments using a linear displacement transducer (LVDT) that placed at the moving end to measure axial movement, which is a measure of buckling level. The instrument system consists of two load cells. The sensors are connected to a personal computer to record the data for analysis.

The pipe tested is 6.5 m long and two different sizes (OD=32mm, and 27mm; ID=25mm, and 21mm respectively), the inside diameter of the glass tube is 149mm (Fig (1)).

The tests were conducted under three different states:

1. Fixed-End on both sides

2. One-End Hinge

3. Two-End Hinge

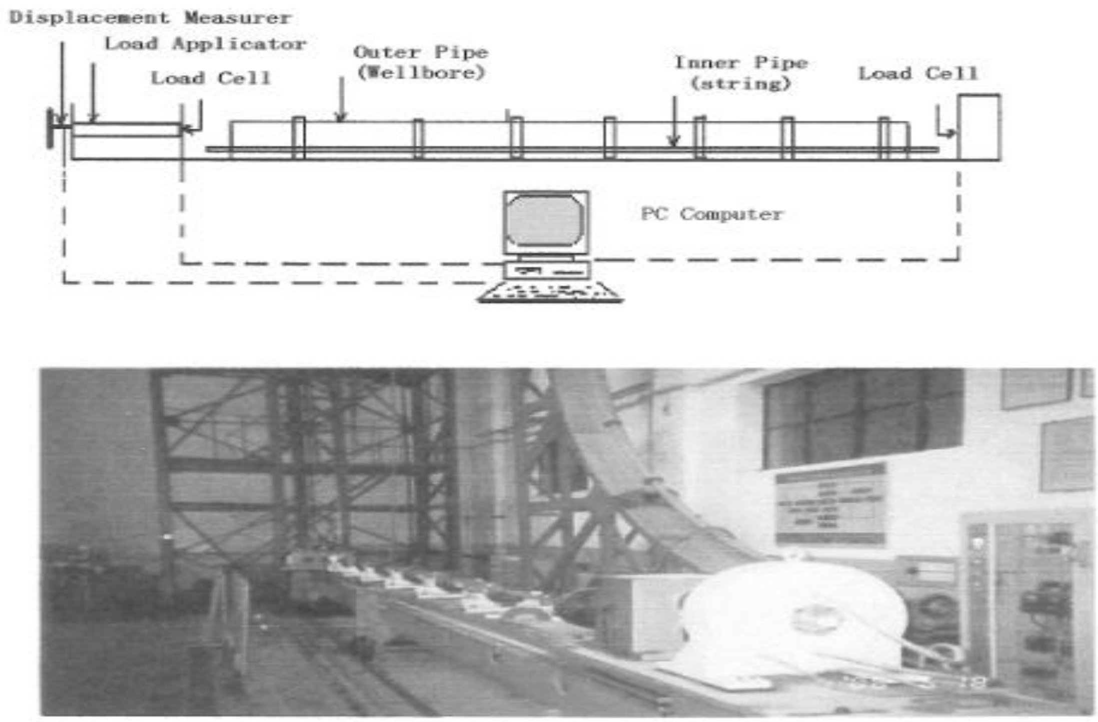
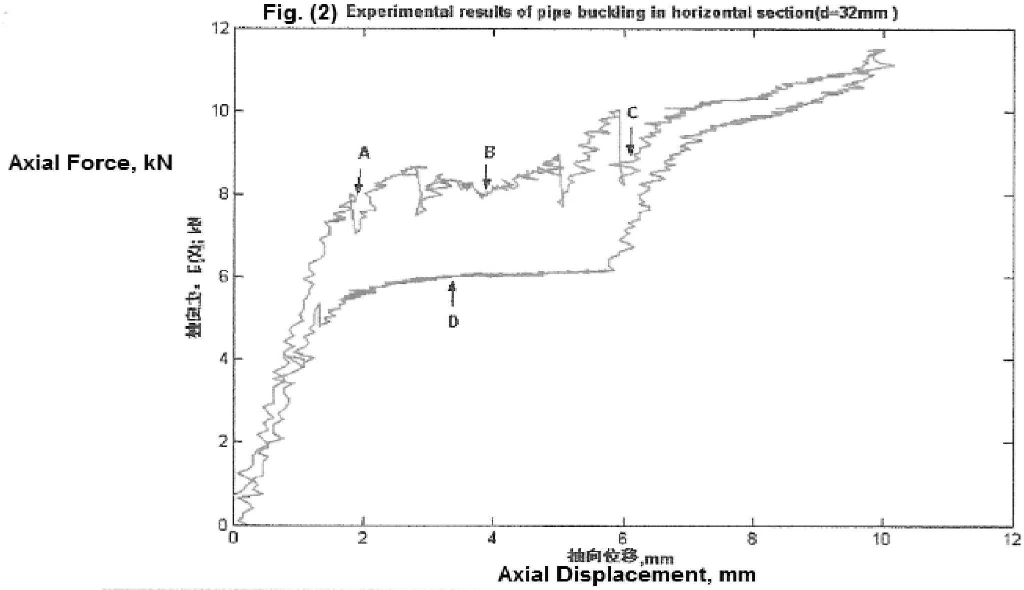


Figure (1) a schematic diagram and a picture of experiment instrument.

The test results can be plotted as shown in figure (2), which represents the relationship between the displacements and axial forces obtained from tests. Point A indicates that pipe buckled into a sinusoidal shape in the down-end of the experiment specimen, point B shows that helix has been initiated; point C denotes that a fully developed helix was observed, and in point D helix disappeared. Unloading path was proven to represent the no friction condition, also all analytical solutions assumed this condition, so this path will be taken as the actual experimental results to make our comparison analysis.

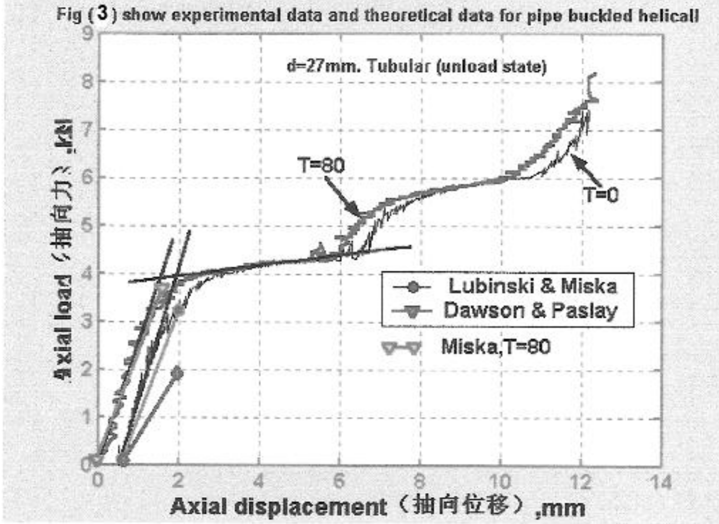
Fig. (2) Experimental results of pipe buckling in horizontal section(d=32mm)



Comparing Between The experimental Results and the Theoretical Methods: utilizing the theoretical methods when compared with experiment results. The critical buckling in a horizontal section using Lubinski, Paslay et al., and Stefan Miska results were compared with the experiment results as shown in Fig (3). For fixed tubular case the estimated value of Paslay equation is lower than experiment results where for one-end hinge and two-end hinge comes closed. Stefan Miska's equation that used to predict critical buckling under torsion load (Torque (T)=0.0) only gives a relatively good results with fixed condition. It was observed that little effect found on critical buckling prediction when torsion load (T=80N.m) for a 32mm. tubular, and no effect for 27mm. tubular.

Calculations of Critical helical Buckling with no Torque: many theoretical methods had been published to predict critical helical buckling. Wu Jiang correlation provides poor prediction of helical buckling load under fixed ends. Wu's concludes that Miska and Chen approach predict

Fig (3) show experimental data and theoretical data for pipe buckled helicall



only the average helical buckling value. Methods like Chen, Miska, and Lubinski and Wood give quite acceptable results. Chen predicts a value lower than the measured result with more than 30% error percent. This proves Wu discussion on applicability of this equation that it is only represents average axial load during helical buckling process. For torsion load applied to 32mm tubular show no significant effect. Wu Jiang approach denoted a reduction in buckling load due to 100 N.m.

The critical helical buckling for no torsion load case:

5. TORQUE-DRAG-BUCKLING RELATIONSHIP

Torque and drag are important factors in extended reach drilling and horizontal drilling and completion. Since buckling is a phenomenon that increases both torque and drag, a comprehensive study of its effects on torque and drag were needed, also a discussion on torque-buckling load-drag relationship is important in order to understand its actual effects on buckling. The simultaneous estimation of torque-drag-buckling is believed to be complex as shown in the diagram drawn below:

It is suggested that investigation on the torque-drag-buckling relationship could be conducted through studying some parameters that have direct or indirect relation with torque and/or drag, utilizing experiments data, and verifications of some published correlations, the parameters mentioned above are:

- Contact force: it has a close relation with parameters such as lock-up of tubular, variations in axial load at upper-end and down-end …etc.
- Buckling length,
- Pitch length, and
- Dimensionless down-end with respect to dimensionless length.

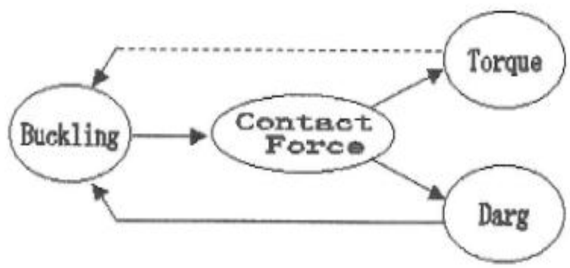


Fig. (4) Schematic diagram of possible relationships between torque, drag, buckling and contact force.

6. BUCKLING LENGTH DETERMINATION [33]:

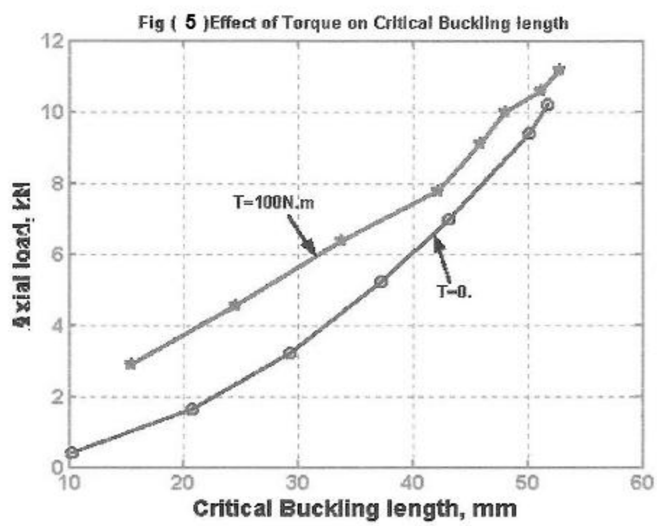
The critical buckling length (L_{cr}) is derived from the following differential equation:

$$EI\frac{d^4y}{dx^4} + F\frac{d^2y}{dx^2} = \varepsilon \tag{23}$$

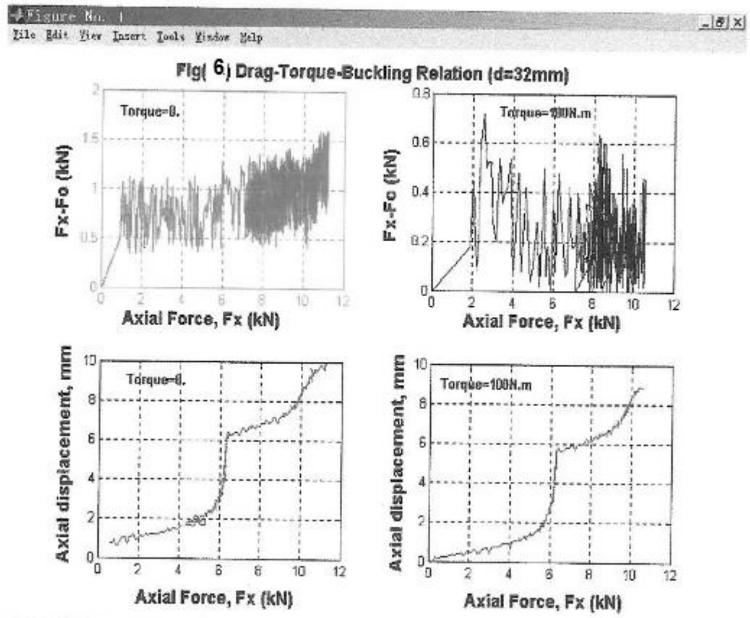
The solution of above equation is:

$$L_{cr} = 2\pi\sqrt{\frac{EI}{F}\frac{1}{12}} \text{ (For fixed ends)} \tag{24}$$

Utilizing the critical buckling length equation proved that significant of torque on buckling length for 32mm. tubular diameter with 100 Nm. applied torque (Fig. (5)). the figure shows that torsion load has a significant effect on buckling length for low axial load and tend to have little effect as the axial load increase, but 27mm. diameter tubular has no significant effect from 80Nm. torsion load. This results show that the influence of torque is effective on large radial clearance and string flexibility.



Effect of Drag On Helically Buckled tubular: By plotting drag experimental data against axial load, as shown in fig.(6). It is obviously clear from the relationship between contact force, buckling, and torque that it get a major effect on each other.



7. DRAG

The lock-up condition: a differential equation for static axial force balance is given by(6):

$$\frac{dFx}{dx} = W_e cos\alpha - \mu N \quad (25)$$

$$N = \frac{rF^2}{4EI} + W_e\ sin\alpha \quad (26)$$

Where: X = axial coordinate

$\alpha =$ Inclination angle of wellbore

$W_e cos\alpha =$ Pipe unit weight component in the axial direction

N= radial contact force due to helical buckling

Fig.(7) presents proportional relationship between drag force(kN) and axial load shows the effect of helical buckling on drag force, and also prove that torque effectiveness on large radial clearances, and string flexibility. In this figure, it is clear that helical buckling tends to decrease the high drag force value in the right side of the curve. This prove that under different end support conditions, the helical buckling help in decreasing the drag force.

Fig. (8) shows the relation between drag and axial force under no torque and 100 N.m torsion. It is quite clear the relation has entirely changed.

8. EFFECT OF CONTACT FORCE ON LOCKING-UP TUBULARS[30]

After helical buckling the drag becomes very large. The following is an axial load distribution for helically buckled pipe in horizontal wellbore under friction:

$$F(x) = 2\sqrt{\frac{EIW_e}{r}}\ \ tan\left\{\mu W_e x\sqrt{\frac{r}{4EIW_e}} + arctan\left(F_o\sqrt{\frac{r}{4EIW_e}}\right)\right\} \quad (27)$$

Where:

F_o = the axial load at the down-end of the buckled portion of pipe (x =0.);

F(x) = axial load at the x point;

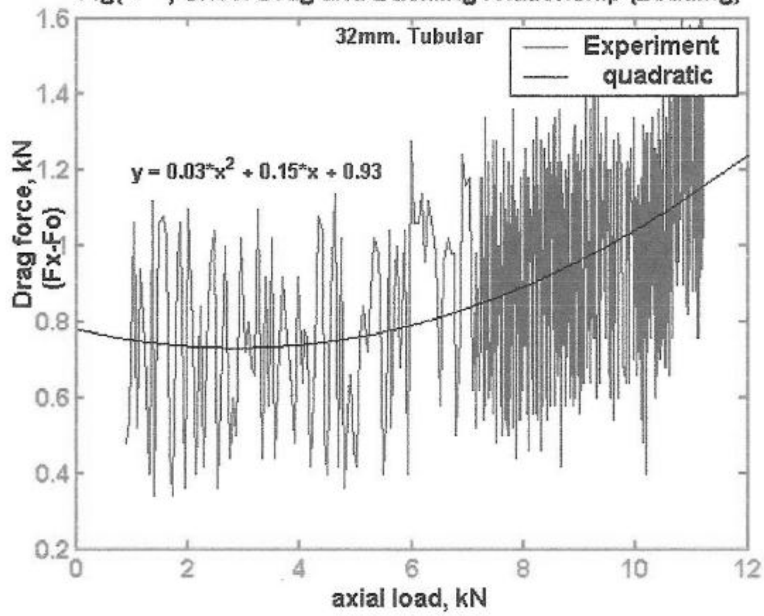
F_{cr} = axial compressive load to initiate sinusoidal buckling of pipe.

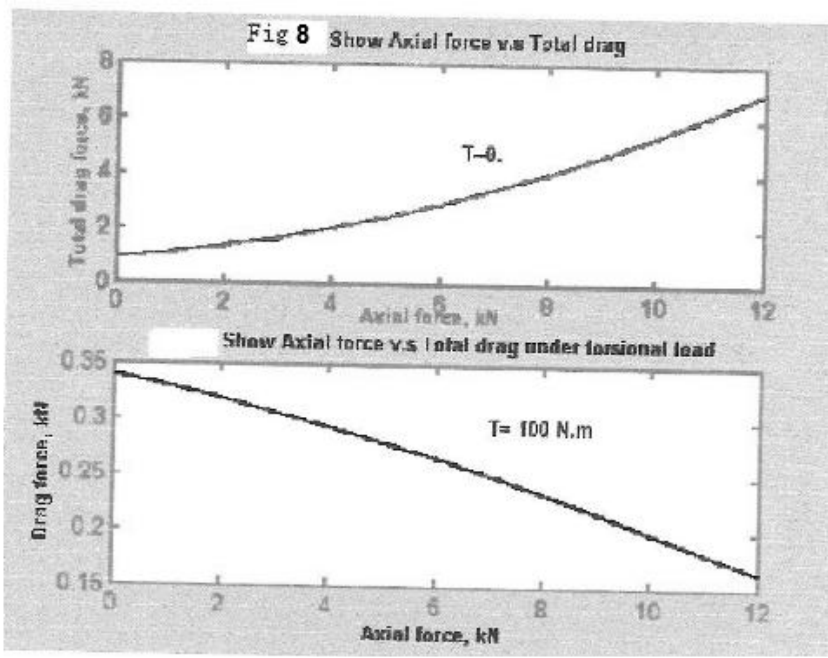
For Simplification the equation (27) can be written as:

$$F(x)^* = \tan(x^* + \arctan(F_o^*))$$

Where: $F(x)^* = \frac{F(x)}{F_{cr}}$; $F_o^* = \frac{Fo}{F_{cr}}$; $x^* = \frac{\mu W_e x}{F_{cr}}$

Fig(7) Show Drag and Buckling relationship (Loading)





So one can conclude that the relationship between torque, drag and buckling is very complex, because every parameter has an affect on the others.

Figure (9) shows a comparison between axial load distributions calculated under the drag of helically buckled pipe and that calculated under a drag of unbuckled pipe. From this figure, one can conclude that axial force for the buckled pipe becomes much larger that unbuckled. The dimensionless axial load $F(x)^*$ has a tendency to approach infinity at a certain dimensionless pipe length. This indicates the lock-up of the pipes in horizontal wellbore due to helical bukling. The equation define lock-up is as follows:

$$x^* = \frac{\pi}{2} - \arctan(F_o) \qquad (28)$$

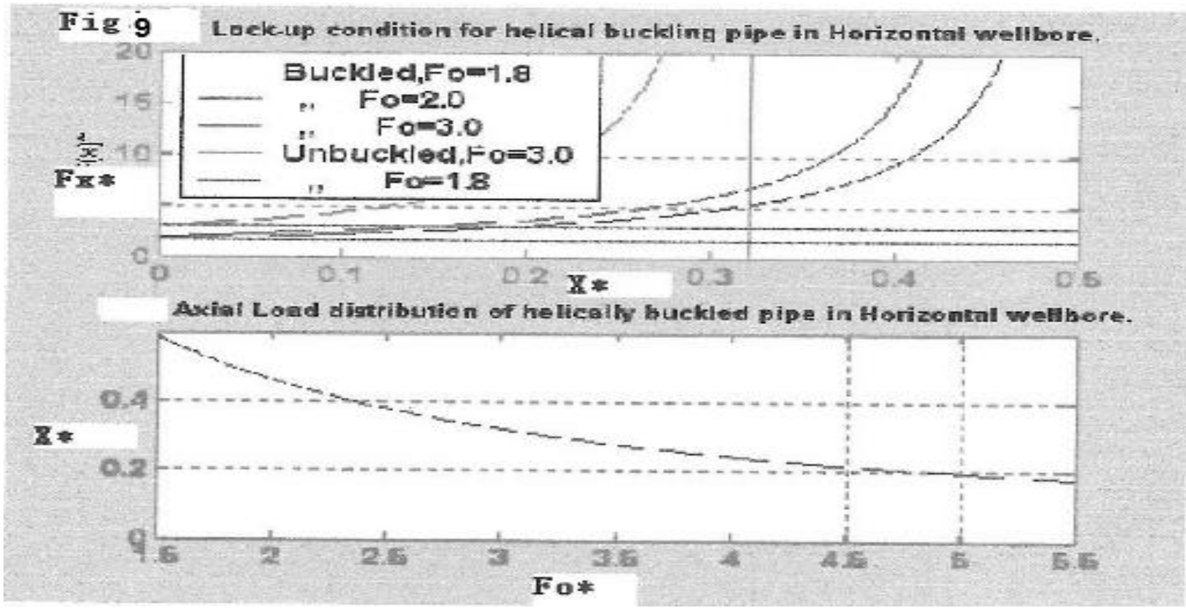
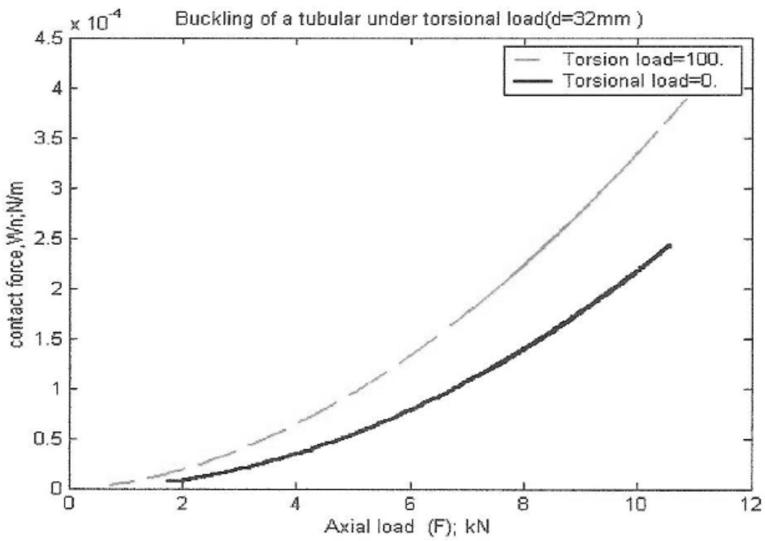


Fig. (10) shows the "lock-up" criteria, here comes the conclusion that down-end axial load can be increased when the helically buckled pipe length x^* is still less than the value defined in eq.(28). The theoretical solution gives a quite close agreement with the experimental results.

Fig (10)



9. CONCLUSION

According to our experiment analysis and the comparativeness of experiment results with that predicted by some theoretical method on the issue of buckling and parameters that affect buckling process and behavoir, the following conclusions obtained:

The quite better method to estimate critical buckling load are Lubiniski and Stefan Miska methods for fixed-end conditions.

Torsion load has a little effect on buckling of tubular and this effect increase with the decrease in pipe stiffness

For prediction of helical buckling load, Lubinski an Woods, Gao Deli and Chen et.al methods provide a good results for two-end hinge boundary.

Helical buckling increases highly the drag force, and so it eliminates the transmission of axial load to the other end.

Under torional load drag force is decreased, as a result drag decreases helical buckling, so it causes an increment to the axial transmitted load.

The torsion load effect on buckling behavior of a tubular through the study of critical buckling length equation shows that torsion load affects critical buckling length.

The effect of torsional load on buckling can be detected through the determination of contact force and that the torque tends to reduce buckling load and increase contact force.

Fig (11) show experimental data and theoretical data for pipe buckled helically

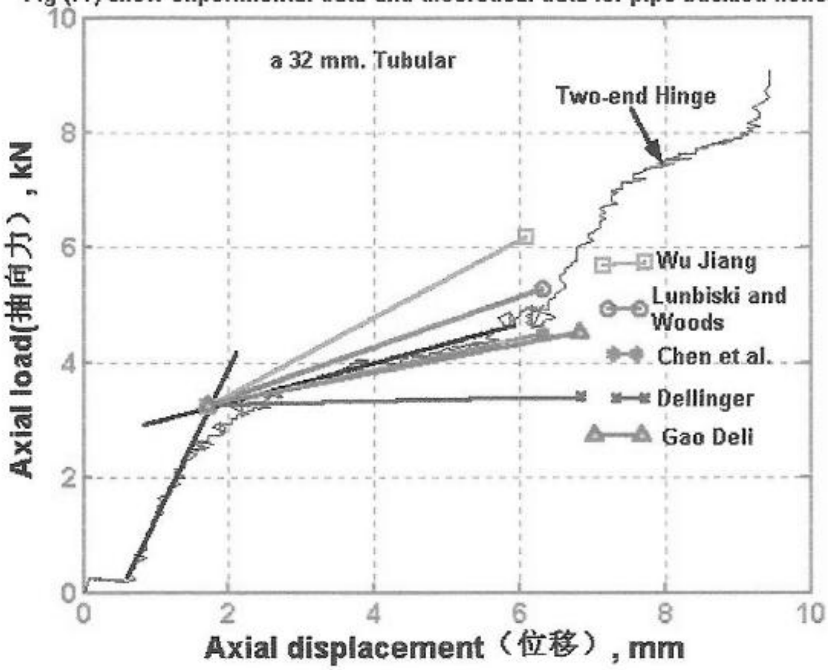
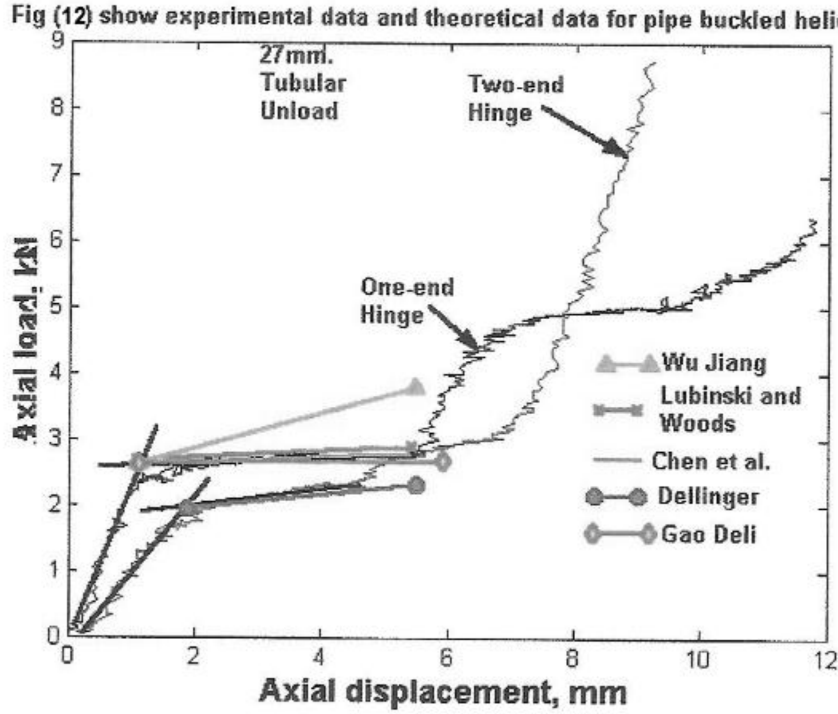


Fig (12) show experimental data and theoretical data for pipe buckled helically



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