Solving specialized well intervention problems

Dynamic FEA models for snubbing buckling and riserless subsea wireline intervention can help design tubing requirements for many different well conditions

Kenneth R. Newman, NOV/CTES, **Charles Overstreet**, Cudd Energy Services and **Pierre Beynet**, BP

A dynamic Finite Element Analysis (FEA) calculation engine has been developed and is being used to solve specialized well intervention problems. This paper summarizes the theory used and documents two applications: 1) the buckling behavior of pipe (or bottomhole assembly) being snubbed through a packer, and 2) wireline being run from a boat to a subsea well to perform an intervention.

When a pipe, or BHA, is being snubbed into a well, large compressive forces are applied to push it through the packer into the well. These forces often cause the pipe to buckle in the surface equipment. Anti-buckling guides are often used to prevent excessive bending. Failures have occurred, especially when snubbing packers into a well. A modeling tool was developed, which calculates the maximum bending and stress in each pipe component or BHA.

Intervening in subsea wells from a boat is much less expensive than using a rig to perform the intervention. There are many questions about how the wireline will behave with the ocean currents, especially when performing an operation, which requires precise force/displacement control, such as operating jars. Currents in the water cause a significant lateral displacement of the wireline. If there is a sudden change in wireline tension at the surface, will the tension be translated through the wireline to the well? Will the wireline motion and its shape in the water absorb the change in tension? Will the operator on the boat be able to determine from his surface tension indicator when jars have fired, or when a plug has released? How should the depth measurement be corrected for the wireline's lateral displacement? If the boat is moving vertically, how much will the wireline tools move up and down in the well?

BUCKLING THEORY

Newman describes the FEA theory used by this calculation engine for a static analysis,[1] while Smalley shows how this theory is used along with a finite-difference scheme in time.[2] The FEA model is run for each time step with the dynamic forces included, forming a dynamic model.

For the snubbing buckling case, the pipe or BHA starts in a straight vertical position, centered in the lubricator/BOP or other stack structure. A pipe can buckle in such a structure in a number of configurations. The helix can be in either direction, the buckling can begin at any rotational position around the structure, etc. These various configurations prevent a static FEA analysis of pipe buckling from converging. However, if a static analysis of the pipe is performed before the compressive load is applied, and then the dynamic analysis is performed while the pipe buckles, the calculation remains stable.

A small destabilizing force must be added to the pipe for the first time step to push it slightly out of line. This destabilizing force is applied at each node along the pipe's length. The direction at each node is varied helically at a user-specified period along the pipe's length. This force determines the pipe configuration, as it buckles, and the final buckled solution. Fortunately, the stresses in the buckled pipe tend to be very similar, no matter which buckled solution is reached.

LARGE DISPLACEMENT THEORY

Fig. 1 shows a single beam element with six Degrees Of Freedom (DOF) at each node. There are three translational DOF along the three axes of the local coordinate system, and three rotational DOF around each axis. Multiple elements are combined[1] to form the desired structure, Fig. 2.

In the case of subsea wireline modeling, the wireline is initially assumed to be perfectly vertical in the global coordinate system. Lateral sea currents cause the wireline to move to a significantly different location. Beam elements work fine with large translational displacements, but rotational displacements about the Y- and Z-axis change the orientation of the element's stiffness. In this application, the element may be rotated by some significant angle, α. A wireline segment has significant strength in the local X direction when it is oriented along the length of the segment,

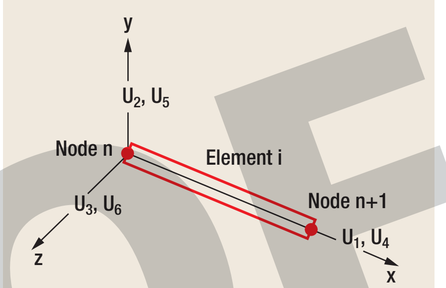

Fig. 1. In a global coordinate system there are six degrees of freedom at each node: displacement (U_1, U_2, U_3) and rotation (U_4, U_5, U_6) in the X, Y and Z axes, respectively.

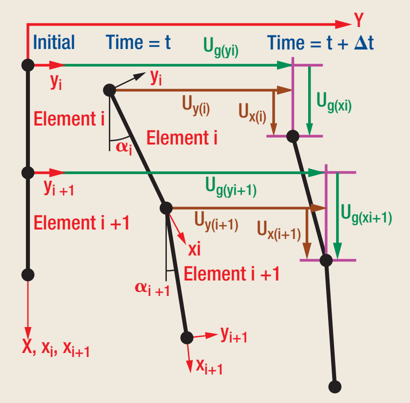

Fig. 2. Multiple elements are combined to form the desired structure, like these two elements with large displacements.

and almost no strength in the local Y and Z directions. If the segment is rotated about the Y and/or Z axis, the stiffness in the X direction is no longer accurate.

To handle this problem, it is necessary to rotate the local coordinate system from its original position to the new position. Since the wireline will be moving laterally during the dynamic simulation, the local coordinate system is moved after each time step. For purposes of discussion consider the situation in Fig. 2 with the wireline at one position, time t, and moving it to another position at time $t+\Delta t$. The global displacements for all DOF are represented by the matrix $\bar{U}$. For this time step they are shown for the X and Y directions. The following points summarize the steps to transform the coordinate system:

- The sum of all of the global displacements for all the time steps must be stored. $\bar{U}g$ is the matrix of the sum of the global displacements for all time steps. $\bar{U}t$ is the matrix of the global displacement from time t to time $t+\Delta t$.

$$\bar{U}_{g(t+\Delta t)} = \bar{U}_{g(t)} + \bar{U}_{t}$$

- The inclination angle, α, and azimuth angle, γ, for the new local coordinate system location is calculated:

$$\alpha_{t+\Delta t} = \tan^{-1}\left[\frac{\sqrt{\Delta y^2 + \Delta z^2}}{L_{x_{t+\Delta t}}}\right]$$

where:

$$L_{x_{t+\Delta t}} = L_o + U_{gx_{i+1}} - U_{gx_i}$$

$$\Delta y = U_{gy_{i+1}} - U_{gy_i}$$

$$\Delta z = U_{gz_{i+1}} - U_{gz_i}$$

$$L_{t+\Delta t} = \sqrt{L^2_{x_{t+\Delta t}} + \Delta y^2 + \Delta z^2}$$

$$\gamma_{t+\Delta t} = \tan^{-1}\left[\frac{\Delta z}{\Delta y}\right]$$

- Once the new values of α and γ have been calculated, a new transformation matrix T must be calculated using the Newman equations.[1]

- The element's length in the new coordinate system will be different than the original element length. This can be compensated for by applying a local force within the element that restores the element to its original length. This applied local force is:

$$R_x = \frac{\left(L_o - L_{t+\Delta t}\right)AE}{L_o}$$

- The dynamic analysis requires the local U displacements in the current local coordinate system for the last three time steps. The new T matrix is used to transform the last three U_g matrices into the current local coordinates. Once this process is completed, the FEA engine can be called for the next time step and the process continues through the dynamic calculation.

Example 1. The operator is faced with a wellbore problem. The objective is to run 1¾-in. coiled tubing to 13,000 ft and drill-out a composite bridge plug. That in and of itself is not necessarily difficult, but other parameters like a hydrogen sulfide (H_2S) environment, the potential of 8,000 psi at surface and conveying the coiled tubing in a large, 7¹⁄₁₆-in.-bore BOP stack, alter the approach substantially.

This operation may have been done before, but how much compressive force the coiled tubing can resist has not been determined by computation. There are documented cases where similar tubulars have buckled in wellbores of similar size. The FEA engine is a tool that can calculate the maximum allowable forces and stresses to provide a unique solution.

One consideration is the H_2S concentration with the partial pressure above allowable values defined in NACE MR0175–2003. Therefore, a 90 ksi-yield material was chosen. The high-pressure environment also means that the snub force and stripper drag must be overcome to enter the wellbore. Triaxial stresses should be calculated at the critical areas above and below the stripper. Standard calculations use a steady-state Von Mises calculation, accounting for standard safety factors. The calculations account for the Bauschinger effect and a safety factor for compression, typical in coiled-tubing applications. For this case, the same approach was used by de-rating the yield stress by 10 ksi, then multiplying by 70% to determine the maximum allowable yield stress.

Since the FEA engine is a transient analysis, it can give more insight to the stresses and forces that take place at different points in the wellbore.

To provide a modeling output, a unique solution was developed. As an example, typical high-pressure strings may have multiple walls, starting from the thinnest at the downhole end and transitioning or tapering to a thicker wall at the core end. This basic philosophy stems from maintaining or maximizing overpull when deeper in the wellbore. This is contrary to what would suit this example.

Since the larger bore is at surface, it is preferred to have a thicker wall when initially entering the wellbore. Obviously, if the string was inverted, the intended depth may never be reached due to tensile failure. So, to optimize both ends of the spectrum a combined "hourglass" concept was introduced. This involved manufacturing a coiled-tubing string with a thick-wall to thinner-wall main portion and then reversing the wall at the string's tail, stepping back to a thick-wall portion at the downhole end. With this approach some overpull was sacrificed, but it allowed a thicker wall when entering the wellbore. The loss in overpull was not that detrimental, compared to the benefit of a thicker wall for the coiled-tubing section that was snubbed into the well.

The FEA engine serves to model the thick-walled coiled tubing upon initial entry and offers a safe working limit in the large-bore BOP stack.

As a portion of a complete pre-job analysis, the FEA engine was used to predict the onset of buckling, when snubbing 1¾-in., 90,000 psi coiled tubing through a 7¹⁄₁₆-in. BOP stack and into 5-in., 23 lb/ft casing under high pressure. The stack modeled consisted of 10 ft of 7¹⁄₁₆-in., 15,000 psi frac valves, a 1-ft crossover spool, a 4¹⁄₁₆-in., 15,000 psi coiled-tubing BOP stack, and enough spool to cover a positive displacement motor assembly (about 15 ft additional). A 15,000 psi, dual, coiled-tubing stripper served as the primary pressure barrier. The FEA engine was focused on buckling at or near the surface.

Since the FEA engine is a general program, it is important that care and intuition is used to properly set the input data, so that the model will accurately depict the scenario. The main constants included: maintaining 8,000-psi external pressure under the stripper and using 1 ¾-in. coiled tubing. This scenario accounts for Bauschinger effects and standard safety factors for collapse and compression. The values are illustrated as a de-rated allowable yield stress.

The 0.203 in.-walled coiled tubing was modeled, while applying zero internal coiled-tubing pressure, and then applying 8,000-psi internal coiled-tubing pressure. Multiple iterations were performed by applying different resistive upward forces that might have been encountered, if the coiled tubing were to strike a ledge in the BOP bore or hit an obstruction in the stack or near surface. The objective was to increase the upward force until the induced stresses reached the de-rated allowable yield stress.

Figure 3 contains the final output graphs at the final time step for the 0.203

in.-wall coiled tubing. This snapshot was not necessarily the worst case, but is used here for illustration. During the iterations, it was necessary to observe the time-step outputs along with the graphs and determine the time when the highest stresses occurred. Then, the input time was modified to stop the calculation at the desired time. The output graphs could then be captured at the time of high stress.

The first two graphs show buckling in the wellbore. The first graph shows the X-Y plane and the second graph shows the X-Z plane. Note that these graphs are distorted, with the Y-axis only showing a span of eight inches, while the X axis shows a depth of nearly 40 ft. The black lines outline the inside of the stack, and the red lines show the buckled pipe.

The third graph shows the maximum radial displacement at all points along the length. The fourth graph shows the wall contact force and the fifth graph shows the Von Mises stress compared to the de-rated allowable yield stress. The FEA engine predicted the maximum attainable force as 11,250 lb.

By using standard modeling practices, as well as de-rating for Bauschinger effects and applying an additional safety factor to account for possible anomalies like rust, pitting, acid etching, mechanical effect, etc., the FEA engine can provide additional information. It is left to the individual to decide how conservative to be using this and other tools, such as force analysis, standard steady-state tri-axial stress analysis and field test data, when determining the best fit for a particular coiled-tubing intervention.

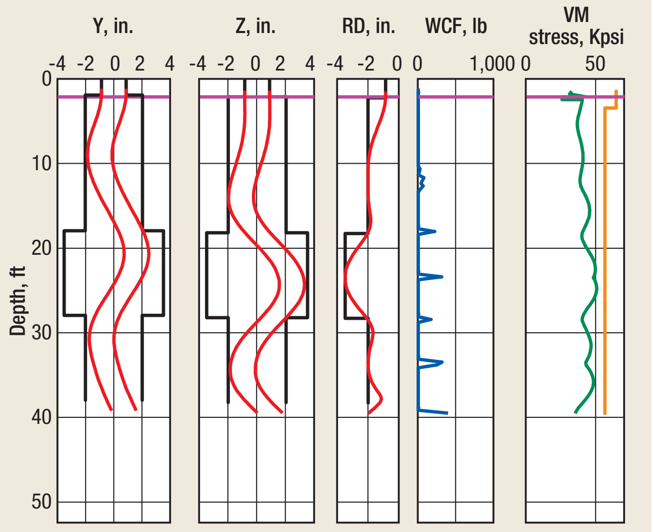

Fig. 3. The final output graphs for 0.203 in.-wall coiled tubing shows tubular buckling in the wellbore, maximum radial displacement (RD), wall contact force (WCF) and Von Mises (VM) stress compared to the de-rated allowable yield stress.

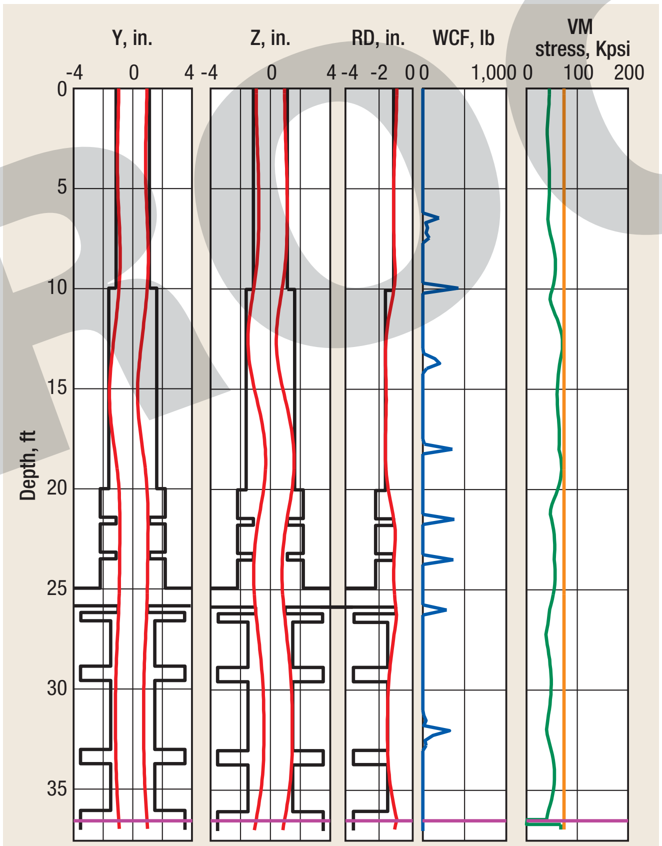

Fig. 4. Example 2 shows that by changing the tubing guide ID and by varying wellhead pressure and drag, a solution was found by using 3-in. ID tubing guides in the BOP stack and a maximum wellbore pressure of 10,500 psi.

Example 2. This scenario further illustrates the FEA engine's versatility. The first requirement was to determine the maximum wellbore pressure or snub force allowable for snubbing 1½-in. jointed tubing into a well with a $7\frac{1}{16}$-in., 15,000 psi, five-ram BOP stack arrangement using a conventional snubbing unit. For this job design, internal tubing guides were to be placed throughout the stack as much as was feasible. A secondary requirement was to size the ID of the tubing guides to maintain as large an ID as possible, while still allowing a sufficient snubbing force.

The purpose for the large BOP stack was that the equipment is overseas and it is easiest to maintain just one standard BOP stack for the applications that arise. In the event that larger tubulars are to be conveyed, the tubing guides can be removed.

The yield strength of the 1.9-in., 2.76-lb/ft tubing was taken as 105,000 psi. A 70% safety factor was used. No reduction for the Bauschinger effect is employed for the jointed tubulars. An additional drag force of 30% of the snub force was induced.

After multiple iterations, changing the tubing guide ID and varying wellhead pressure and the corresponding drag, the solution was found, Fig. 4. For this scenario, using 3-in. ID tubing guides in the BOP stack, the maximum allowable wellbore pressure was 10,500 psi.

SUBSEA WIRELINE INTERVENTION

A well intervention was to be carried out with 0.125-in. slickline from a floating mono-hull vessel at 3,500-ft water depth. One of the first steps in this intervention is to jar the plug out of the tubing, Fig. 5. The FEA model simulated this.

The stroke length of the spang jars is about 30-in. For this example, it was assumed that there is a two-knot current in the Y direction for the upper half and a one-knot current in the same direction for the lower half. The drag forces due to the current were added in the slickline model. When the slickline is held at surface with 1,076 lb of tension, the maximum lateral displacement is 106 ft, Fig. 6. Note that the displacement in the upper half of the slickline is greater than in the lower half due to the higher current. In this situation, there is 800 lb of tension at the spang jars.

It was assumed that the spang jars would release at 1,000 lb of tension. A dynamic simulation was run, in which the slickline was pulled upwards 30-in. at surface and held at that position, Fig. 7. It took 1.5 sec before the bottom force at the jar reached 1,000 lb,

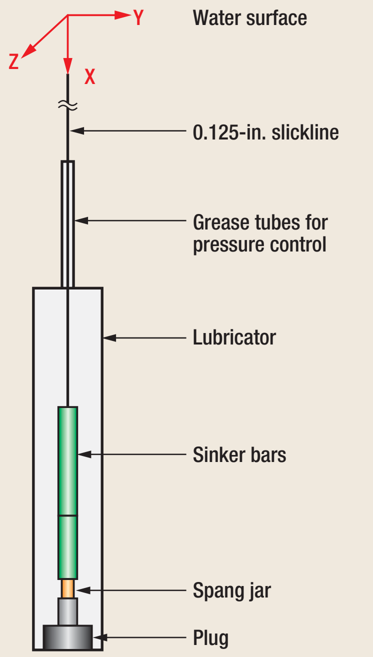

Fig. 5. The FEA model simulated a subsea, slickline-jarring, well intervention with 0.125-in. slickline from a vessel in 3,500-ft water depth.

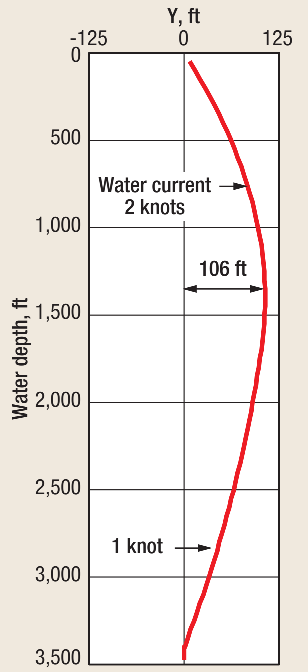

Fig. 6. When the slickline is held at surface with 1,076 lb of tension, the maximum lateral displacement from water currents is 106 ft.

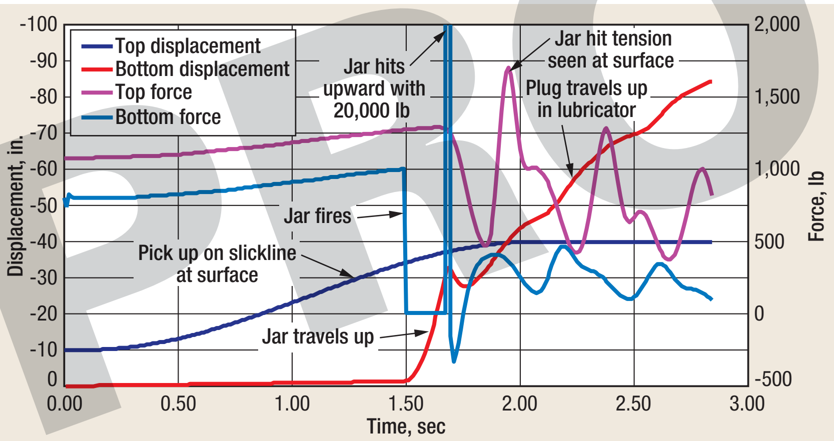

Fig. 7. A dynamic slickline jarring simulation showed that it took 1.5 sec for the bottom force to reach 1,000 lbs to fire the jar.

so the jar could release.

When it released, the bottom force went to zero and the jar started traveling upward. This tension release was seen at surface about 0.2 sec later. The speed of sound in steel is 22,000 ft/sec. Thus, we would expect it to take 0.16 sec for the force change to travel 3,500 ft. The jar travels upward 30-in. and hits with its upward impact. The force imparted by this impact depends upon the stiffness of the jar and plug. In this case, the stiffness was assumed to be 100,000 lb/in. The resulting impact force was about 20,000 lb.

It was assumed that the plug came free due to this impact force. The weight of the plug was added to the jar weight, and both continued to travel upward through the lubricator. During this simulation the maximum lateral slickline travel in the water was about three inches.

The lateral slickline movement did not have a significant impact on the jar's operation. The operator had a very clear indication on surface when the jar released and when it struck. The upward impact was greatly enhanced by having a release mechanism. In fact, it would be very difficult to perform this operation without one. The ocean currents cause tension in the slickline. If there was no release mechanism, the jar would travel to the top of its stroke due to this tension and there would be no jarring action.

CONCLUSIONS

A dynamic FEA model has been developed which can simulate a number of well drilling and intervention problems, including using slickline to perform a jarring operation from a boat to a subsea well. Many job design questions can now be answered with this simulation.

Tubular buckling in a specified cavity was also simulated. Dynamic analysis was used to control the pipe as it buckled to one of many possible solutions. **WO**

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THE AUTHORS

Kenneth R. Newman earned an MS degree in mechanical engineering from the Massachusetts Institute of Technology and invented the Smar-Tract downhole tractor system. He has authored many technical papers, magazine articles and patents, and is a Registered Professional Engineer in Texas. Newman is the Founder and Technical Director of NOV CTES.

Charles C. Overstreet earned a BS degree in mechanical engineering from the University of Texas and has 16 years of oil and gas experience, 15 with Cudd. He has one patent pending and is co-inventor on two patents, as well as having authored many technical papers. Overstreet is a Sr. Corporate Engineer with Cudd Energy Services.

Pierre A. Beynet earned a BS degree from Ecole Centrale de Mechanique de Nantes, and MS and PhD degrees in aeronautical engineering and applied mechanics from the University of Minnesota. He has 37 years of at Amoco and BP in deepwater technology development, design, fabrication and operation. He helped develop deepwater fixed platforms, TLP, spars, tanker and semisubmersible-based production and drilling systems, as well as working on drillstring stability and platform dynamics. Beynet is an Offshore Technology Consultant with BP Amoco.