Dynamic FEA Models for Snubbing Buckling and Riserless Subsea Wireline Intervention
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Abstract
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 in this engine, and documents two of its applications. The first application is modeling the buckling behavior of pipe (or a bottom-hole assembly) being snubbed through a packer. The second application is wireline being run from a boat to a subsea well to perform an intervention. The first step in this intervention is to jar the plug out of the tree.

Introduction
A dynamic FEA engine has been developed for modeling several well drilling and intervention situations. The initial model is for analyzing pipe in a wellbore, discussed in reference 1. The second model is for analyzing well intervention stack stability and stresses, discussed in reference 2. This paper discusses two additional applications which use variations of the same FEA modeling capability. As this powerful calculation technique is refined and applied, many more complicated well drilling and intervention models can be developed.

The theory for this FEA engine has already been discussed in references 1 and 2. The major addition to the theory discussed in this paper involves buckling of a pipe and the large displacements needed to model the behavior of wireline in the ocean between the boat and the subsea well.

When pipe, or a bottom-hole assembly (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 component of the pipe of 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. The current causes a significant lateral displacement of the wireline as it passes through the water. If there is a sudden change in tension of the wireline at surface, will the tension be translated through the wireline to the well, or will the motion of the wireline 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 the jars have fired, when the plug has released, etc? How should the depth measurement be corrected for the lateral displacement of the wireline in the water? If the boat is moving up and down, how much will the wireline tools move up and down in the well? A modeling tool was developed to answer these types of questions.

Theory
Reference 1 describes the FEA theory used by this calculation engine for a static analysis. Reference 2 shows how this theory is used along with a finite-difference (FD) scheme in time. The FEA model is run for each time step with the dynamic forces included, forming a dynamic model.

Buckling Theory
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 possible 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 length of the pipe. The direction at each node is varied helically at a user specified period along the length of the pipe. This destabilizing force determines the configuration of the pipe as
it buckles and thus 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**

Figure 1 shows a single beam element with 6 degrees of freedom (DOF) at each node. There are 3 translational DOF along the 3 axis of the local coordinate system, and 3 rotational DOF around each axis of the local coordinate system. Multiple elements are combined as discussed in reference 1 shown in Figure 2 to form the desired structure. In the case of subsea wireline modeling, the wireline is initially assumed to be perfectly vertical in the global coordinate system, as shown in Figure 2. Lateral sea currents cause the wireline to actually be at 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 stiffness of the element. In this application, the element may be rotated by some significant angle, $\alpha$. A wireline segment has significant strength in the local x direction when it is oriented along the length of the segment, 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 shown in Figure 2 with the wireline at one position at time t and moving to another position at time $t + \Delta t$. The global displacements for all degrees of freedom, are represented by the matrix $\mathbf{U}$. For this time step they are shown for the X and Y directions. The following points summarize the steps used to perform the coordinate system transformation:

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

$$\mathbf{U}_{g(t+\Delta t)} = \mathbf{U}_{g(t)} + \mathbf{U}_t \quad (1.1)$$

- The inclination angle, $\alpha$ and azimuth angle, $\gamma$, for the new local coordinate system location is calculated:

$$\alpha_{t+\Delta t} = \tan^{-1} \left[ \frac{\Delta y^2 + \Delta z^2}{L_{x_{t+\Delta t}}} \right] \quad (1.2)$$

where:

$$L_{x_{t+\Delta t}} = L_o + U_{g_{x_{t+\Delta t}}} - U_{g_{x_{t}}} \quad (1.3)$$

$$\Delta y = U_{g_{y_{t+\Delta t}}} - U_{g_{y_{t}}} \quad (1.4)$$

$$\Delta z = U_{g_{z_{t+\Delta t}}} - U_{g_{z_{t}}} \quad (1.5)$$

$$L_{x_{t+\Delta t}} = \sqrt{L_{x_{t+\Delta t}}^2 + \Delta y^2 + \Delta z^2} \quad (1.6)$$

$$\gamma_{t+\Delta t} = \tan^{-1} \left[ \frac{\Delta z}{\Delta y} \right] \quad (1.7)$$

- Once the new values of $\alpha$ and $\gamma$ have been calculated, a new transformation matrix $\mathbf{T}$ must be calculated using the equations given in reference 1.

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

$$R_x = \frac{(L_o - L_{t+\Delta t})AE}{L_o} \quad (1.8)$$

- The dynamic analysis requires the local $u$ displacements in the current local coordinate system for the last 3 time steps. The new $\mathbf{T}$ matrix is used to transform the last 3 $U_g$ 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.

**Snubbing Buckling Example 1**

The operator is faced with a well bore problem. The objective is to run 1.75 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 (H2S) environment, the potential of 8,000 psi at surface, and conveying the coiled tubing in a large 7 1/16 in. bore blowout preventer (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 well bores 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 H2S 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 well bore. Tri-axial 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 derating 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 well bore.

In accordance with providing 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 down hole end and transitioning or tapering to a thicker wall at the core end. This basic philosophy stems from maintaining or maximizing over pull when deeper in the well bore. This is contrary to what would suit this case. Since the larger bore is at surface, it is preferred to have a thicker wall...
when initially entering the well bore. 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. It involved manufacturing a coiled tubing string with a thick wall-to-thinner wall main portion and then reversing the wall at the tail end of the string to step back to a thick wall at the downhole end. With this approach some over pull was sacrificed, but it allowed a thicker wall when initially entering the well bore. The loss in over pull was not that detrimental compared to the benefit of having a thicker wall for the section of coiled tubing that had to be snubbed into the well.

The FEA engine serves to model the thick walled coiled tubing upon initial entry and offer 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 1.75” 90 Kpsi coiled tubing through a 7 1/16” blowout preventer stack and into 5” 23 lb/ft casing under high pressure. The stack modeled consisted of 10’ of 7 1/16” – 15M frac valves, a 1’ long crossover spool, a 4 1/16” – 15M coiled tubing blowout preventer stack, and enough spool to cover a positive displacement motor assembly (approximately 15 additional feet). A 15M dual coiled tubing stripper served as the primary pressure barrier. The FEA engine was used to focus in on the buckling phenomena 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 up the input data and the fifth graph shows the Von Mises stress compared to length. The fourth graph shows the wall contact force (WCF) maximum radial displacement (RD) at all points along the red lines show the buckled pipe. The third graph shows the nearly 40’. The black lines outline the inside of stack, and the only showing a span of 8”, while the X axis shows a depth of graphs show the tubular buckling in the wellbore. The first two graphs show 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 8”, while the X axis shows a depth of nearly 40’. The black lines outline the inside of stack, and the red lines show the buckled pipe. The third graph shows the maximum radial displacement (RD) at all points along the length. The fourth graph shows the wall contact force (WCF) and the fifth graph shows the Von Mises stress compared to the derated allowable yield stress. The maximum attainable force was predicted by the FEA engine to be 11,250 lb.

By using standard modeling practices, as well as derating 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.

**Snubbing Buckling Example 2**

This scenario further illustrates the versatility of the FEA engine. The first requirement was to determine the maximum wellbore pressure or snub force allowable for snubbing 1 ½” jointed tubing into a well with a 7 1/16” – 15M 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 internal diameter 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 operationally to maintain just one standard BOP stack for the various applications that may arise. In the event, larger tubulars are to be conveyed, the tubing guides can be removed.

The snubbing stack configuration is shown in Figure 7. The yield strength of the 1.900” 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 the wellhead pressure and corresponding drag, the solution shown in Figure 8 was found. For this scenario, using 3 in. internal diameter tubing guides in the BOP stack, the maximum allowable well bore pressure was found to be 10,500 psi.

**Subsea Wireline Intervention**

A well intervention was to be carried out with 0.125” slickline from a floating mono-hull vessel at a water depth of 3,500’. One of the first steps in this intervention is to jar the plug out of the tubing. The FEA model was used to simulate this jarring operation.

Figure 3 shows a schematic of this operation. The stroke length of the spang jars is about 30”. For this example it was assumed that there is a 2 knot current in the Y direction for the upper half of the depth and a 1 knot current in the same direction for the lower half of the depth. The drag forces due to the current were added in the FEA model of the slickline. The resulting lateral displacement of the slickline is shown in Figure 4. When the slickline is held at surface with 1,076 lb of tension, the maximum lateral displacement is 106’. 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 lbs of tension at the spang jars.
It was assumed the spang jars would release at a tension of 1,000 lb. A dynamic simulation was run in which the slickline was pulled upwards 30” at surface and then held at that position. Figure 5 shows the result of this simulation. It took 1.5 seconds before the bottom force at the jar reached 1,000 lbs so the jar could release. When it released, the bottom force went to zero and the jar started traveling upward. This release of tension was seen at the surface about 0.2 seconds later. The speed of sound in steel is 22,000 ft per second. Thus, we would expect it to take 0.16 seconds for the force change to travel 3,500 ft. The jar travels upward 30” 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 travel of the slickline in the water was only about 3”.

For this example, the lateral movement of the slickline in the water did not have a significant impact on the operation of the jar. The operator had a very clear indication on surface when the jar released and when it struck. The upward impact of the jar was greatly enhanced by having a release mechanism. In fact, it would be very difficult to perform this operation without a release mechanism. The ocean currents cause a 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. This paper has demonstrated the use of this model for two very different problems.

Using slickline to perform a jarring operation from a boat to a subsea well has been simulated. Many job design questions can now be answered with this simulation.

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

**Acknowledgement**

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**References**

6 Degrees of Freedom at each node:

- $u_1 = x$ displacement
- $u_2 = y$ displacement
- $u_3 = z$ displacement
- $u_4 = \text{rotation about } x$
- $u_5 = \text{rotation about } y$
- $u_6 = \text{rotation about } z$

Figure 1 - Elements in a Global Coordinate System

Figure 2 – Two Elements with Large Displacements
Figure 3 - Slickline Subsea Jarring Schematic

Figure 4 - Slickline Deflection due to Water Current
Figure 5 - Dynamic Slickline Jarring Simulation Results

Figure 6 - Snubbing Coiled Tubing into a Well
Figure 7 – Snubbing Stack Schematic
Dynamic Finite Element Analysis Engine
Snubbing Calculation Module

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Figure 8 - FEA Output for Snubbing Example 2