Dynamic Response Analysis of Tubular String under Perforating Gun-shock Loads

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Abstract. To obtain the dynamic response of perforating string under gun-shock loads, a dynamical model was established based on the vibration mechanics theory of cantilever beam, and the differential equations of longitudinal vibration were deduced. The dynamic response was obtained according to the measured data of impact loads. The results show that vibration response of displacement, velocity, and acceleration were changed with a cycle of 15ms. The Von Mises stress near the packer is the largest owing to stress concentration. The analytic solution of natural frequency is close to the finite element analysis value, and the maximum relative error is below 5%. The velocity of stress waves propagating along the string approximates 5000m/s, which is close to the theoretical analysis value and is about 3.6 times that of sound velocity in water.

1. Introduction
During the perforating completion of oil and gas wells, when the perforating charges is detonated to complete the perforation, a huge explosion shock wave will be generated, and it will act on the downhole tubular string and perforating liquid, causing severe vibration of the tubular string and downhole tool, resulting in plastic bending of the tubing string and breakage of the mandrel [1,2]. Vibration analysis of the perforation section also concentrates on finite element simulation and ground test. Literature [3] designed a set of perforating tubing string dynamic load ground comprehensive test systems, and another literature [4] designed a set of oil and gas well perforation simulation devices. Literature [5,6] applied the finite element method to the transient dynamics analysis of the perforation tubing string. Literature [7] used a dynamics software to simulate the tubing string dynamics and analyzed the influencing factors on the dynamical tubing string response, such as well depth, pocket length, and tubing string length. In literature [8], a ground simulation test system was established to test the tubing string end pressure and acceleration response of the perforated section. In literature [9], a specific perforating well was used as an example to analyze the influence of well structure on the safety of the tubing string in the perforating section. Schlumberger’s Baumann [10, 11] and others used numerical simulation methods to analyze the dynamic response of downhole strings and tools under impact loads.

Considering the complicated theoretical analysis of the dynamical tubing string response under the impact of explosion, the ground test is expensive and it is difficult to simulate complex environment of the well. The displacement response, velocity, acceleration, and equivalent stress of the tubing string were obtained based on the transient dynamics analysis method, and the reliability of the finite element method was verified by the comparison of natural frequencies.
2. Establishment of the dynamic model of perforation section

The shock wave loads and dynamic response of the perforated section are complex and difficult to describe when using accurate theoretical models. Therefore, this paper used the combination of theoretical analysis and finite element method.

2.1. Physical model

With the tubing string end at the packer and bottom end as the fixed and free ends, respectively, the perforating tubing string can be simplified as the cantilever beam model, as shown in Figure 1(a). The origin is at the packer, with the axis of the tubing string as the x-axis and the y-axis at the vertical downward direction. Let $u(x, t)$ be the longitudinal displacement of the section from the origin $x$ on the tubing string at time $t$. Then, $f(x, t)$ is the longitudinal force on the tubing string of unit length, $L$ is the tubing tubing string length.

The micro-tubing string section with length $dx$ was obtained from the tubing string for force analysis, as shown in Figure 1(b). The one-dimensional longitudinal vibration differential equation of the tube can be obtained as follows:

$$\frac{\partial^2 u(x, t)}{\partial t^2} = \frac{E}{\rho} \frac{\partial^2 u(x, t)}{\partial x^2} + \frac{1}{\rho A} f(x, t)$$  

2.2. Boundary conditions and initial conditions

Prior to perforating, the tubing string is at rest, the initial velocity is zero; the tubing string perpendicular to the wellbore will exhibit a certain elongation under its own weight. However, if weight is the external load, and the tubing string is in the original position as its equilibrium state, then the initial displacement is zero, that is, the initial condition is computed as below:

$$\begin{align*}
  u(x, t) & |_{t=0} = 0 \\
  \frac{\partial u(x, t)}{\partial t} & |_{t=0} = 0
\end{align*}$$

3. Natural frequency and main vibration mode analysis

When the external load on the tubing string is zero, that is, $f(x, t) = 0$ in Eq. 1, the equation for the longitudinal free vibration of the tubing string can be obtained as follows:

$$\frac{\partial^2 u(x, t)}{\partial t^2} = \frac{E}{\rho} \frac{\partial^2 u(x, t)}{\partial x^2}$$

where $v = \sqrt{E/\rho}$ is the velocity at which the vibration wave propagates along the tubing string. The natural frequency and main vibration mode of the tubing string vibration are respectively computed as follows:
\[ f_i = \frac{\omega_i}{2\pi} = \frac{2i-1}{4L} \sqrt{\frac{E}{\rho}} \]  
\[ U_i(x) = \sqrt{\frac{2}{\rho A L}} \sin \left( \frac{2i-1}{2L} \cdot \pi x \right) \]

where \( i \) is a positive integer (i.e., \( i=1, 2, 3, \ldots \)); \( f_i \) is the natural frequency of each order, Hz; \( \omega_i \) stands for the natural angular frequency of each order, rad/s.

4. Displacement response analysis of tubular string under impact load

The explosion shock wave generated after perforating the charges is applied to the bottom end of the tubing string; this shock wave is equivalent to the axial, lateral, and torsional load components applied, with the latter two showing less influence on tubing string vibration. As the initial condition of the tubing string vibration is zero, the initial condition at the regular coordinates is also zero. The displacement response of the tubing string is expanded to an infinite series of regular modes, and the displacement response of the tubing string under impact load is computed as follows:

\[ u(x, t) = \sum_{i=1}^{\infty} \frac{1}{2\pi f_i} U_i(x) \int_0^t q_i(\tau) \sin 2\pi f_i (t - \tau) d\tau \]  

where \( i \) is a positive integer, and \( \tau \) is a time integral variable. The displacement response of the tubing string can be obtained by numerical integration.

5. Finite element analysis (FEA) of the dynamic response of perforating tubing string

The FEA of the tubing string is performed using the transient dynamics analysis module of ANSYS Workbench.

5.1. Establishment of finite element model

The tubing string length, outer diameter, and inner diameter measure 20000, 73, and 62 mm, respectively. The elastic modulus of the tubing material reaches \( 2.06 \times 10^{11} \) Pa, Poisson’s ratio is 0.3, material yield strength is 758 MPa, and density is 7.85 g/cm\(^3\). An eight-node hexahedral solid element is used, and the grid is divided by grid meshing method. The wellbore pressure distribution during the perforating process is collected by the downhole perforation pressure monitor [12]; the load is applied to the bottom end of the tubing string. The peak pressure is 135 MPa, and the loading time is 100 ms.

5.2. Comparison of natural frequency analytical solution and finite element solution

Through the modal analysis method in ANSYS, the first six natural frequencies of the perforation section tubing string vibration can be obtained. Table 1 shows the finite element solution and analytical solution of the natural frequency of the tubing string are close to each other, and the maximum relative error less than 5%, indicating that the results obtained by analyzing the dynamic response of the tubing string with finite element are reasonable and reliable.

| Order | Analytical solution | Finite element solution |
|-------|---------------------|-------------------------|
| 1     | 64.03               | 66.89                   |
| 2     | 192.09              | 200.63                  |
| 3     | 320.15              | 334.39                  |
| 4     | 448.21              | 468.14                  |
| 5     | 576.27              | 601.9                   |
| 6     | 704.37              | 735.65                  |

5.3. Analysis of tube tubing string velocity and acceleration

Figures. 2 and 3 display the curves showing the speed and acceleration of the free and fixed end of tubing string with time. The fixed end of tubing string is limited by the packer, and its speed and acceleration are zero. The speed and acceleration of the free end of tubing string change periodically
with time, and the period is about 15 ms. At \( t = 45 \) ms, the free end of tubing string speed reaches a maximum of 5.35 m/s. When \( t = 10 \) ms, the acceleration reaches the maximum value of 2.95 km/s\(^2\), and the velocity and acceleration amplitudes gradually decrease with time.

![Figure 2. Velocity of constraint end and free end](image1)

![Figure 3. Acceleration of constraint end and free end](image2)

5.4. Equivalent stress analysis of tubing string

According to the finite element transient dynamics analysis, the impact load acts on the tubing string and propagates along the tubing string in the form of elastic stress waves, causing changes in equivalent stress in the tubing string. At \( t = 4 \) ms, the stress wave is transmitted to the end of tubing string of the pipe at the packer (constrained end of tubing string). Figure 7(a) shows the equivalent stress distribution of the constrained end of tubing string. When \( t = 23.6 \) ms, the equivalent stress reaches the maximum value of 305.52MPa and appears on the end of tubing string near the packer. Figure 7(b) displays the equivalent stress distribution diagram at this time. As shown in Figure 4, non-uniform stress distribution occurs on the cross section of the tubing string, resulting in stress concentration. The equivalent stress gradually increases in the radial direction and reaches a maximum at the outer surface of the tubing string.

![Figure 4. Von Mises stress of the constraint end at the time of 4 and 23.6 ms](image3)
Figure 5. shows the maximum equivalent stress versus time curve of the free end of tubing string and the fixed end of the tubing string. The free end of tubing string equivalent stress curve is the same as the impact load curve, and the maximum equivalent stress at the fixed end of tubing string is much larger than the impact load. The maximum equivalent stress of the fixed end of tubing string changes periodically with time, and the period is about 15 ms, and the amplitude gradually decreases. Within 4 ms, the maximum equivalent stress of the fixed end of tubing string is small and causes no considerable change. The equivalent stress is mainly generated by the weight of the tubing string. The stress wave is transmitted at this point at 4 ms, and the equivalent stress suddenly increases. At \( t = 23.6 \) ms, the equivalent stress of the tubing string reaches a maximum of 305.52 MPa and appears at the end of the packer.

![Figure 5. Maximum Von Mises stress of constraint end and free end](image)

6. Conclusion
In this paper, the vibration response of the perforating tubing string under detonation impact load was studied. The following conclusions were obtained:

1. The dynamic model of the perforation section of the tubing string was established, and the partial differential equation of the tubing string vibration was derived.

2. Periodic variations of the velocity, and acceleration of the tubing string with time, and the period is about 15 ms. A stress concentration was observed in the tubing string at the packer, resulting in the largest equivalent stress of 305 MPa, which indicates that the section is dangerous.

3. The stress wave propagation velocity in tubing string is about 5000 m/s, which is close to the theoretical result (5123 m/s) and approximately 3.6 times that of the velocity (1400 m/s) in water.

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