Coupling Dynamic Analysis of Fracturing Truck Frame during Operation

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Abstract. In order to study the effect of vibration of the engine and fracturing pump on the truck frame during the fracturing operation, lumped parameter method and state-space method were used to build multi-body dynamics coupling model between the balanced suspension, tires, engine, fracturing pump and the frame. Based on this model, the MATLAB/Simulink software was applied to simulate the characteristic of rotational speed, stiffness, and damping of the 2500 fracturing truck during the fracturing operation. The results show that the maximum intensity of vertical vibration is hundreds of times that of pitch motion. The vibration frequency range of 0–10 Hz affects vibration most evidently; the variation of stiffness of the rear suspension and the damping of the front and central suspension has a great influence on amplitude, while the variation of damping of rear suspension and the stiffness of the front and central suspension have a great influence on frequency of peak.

1. Introduction

With the continuous exploitation of coal bed methane, shale gas resources and the tapping of potential oil production in the original oil and gas field in China, large-scale CNC fracturing trucks are widely used in various oil fields, and the working capacity of the fracturing equipment needs to be continuously improved. The frame is the bearing foundation of large-scale CNC fracturing trucks, which is a heavy and complex space structure and plays a key role in the whole truck driving and fracturing operations [1–3]. On the basis of 1/4 truck model, Guo calculated the distribution law of dynamic load according to the statistical data of road roughness in typical road condition, studied the reasonable selection of truck damping and limit stroke, and improved the maximum speed of the truck [4]. Chen established the train-bridge-automobile coupling model, and analyzed the influences of the increase in the number of trucks on the vertical acceleration, derailment coefficient and wheel loading shedding rate of trains and trucks [4]. Therefore, it is very important to study the vibration characteristics of the frame during the operation of the fracturing truck.

In the paper, according to the actual situation of fracturing operation, the truck engine and fracturing pump as power subsystem, and the interaction between the tire subsystem and the frame subsystem, the suspension subsystem and the frame subsystem is taken into account to established multi-body coupling dynamics simulation model of fracturing truck operation process and deduce the
system motion equation. The coupled model is simulated and analyzed by MATLAB/Simulink software, which provides a reference for the vibration simulation analysis of the fracturing truck operation process.

2. Coupling system modeling and equilibrium equation

2.1. Fracturing truck composition and operation process

The fracturing truck mainly consists of a chassis and upper part. Top-loading equipment mainly includes platform engine, gearbox system (transmission gear, torque converter), fracturing pump, the water tank, and other high-pressure pipelines and matching valves.

In the fracturing truck operation, the chassis engine’s power drives the drive motor of the platform engine, making the platform engine work, and the chassis engine is closed at this time. The power generated by the engine is transmitted to the power end of the fracturing pump through the hydraulic transmission box and driveshift, which drives the fracturing pump for fracturing operation.

2.2. Physical model

In order to analyze the vibration problems during the fracturing truck operation from different parameters, a 1/2 truck model in the vertical longitudinal section was established to simulate and analyze the vibration response problems caused by the engine and the fracturing pump during the fracturing truck operation [5-7]. The following assumptions are made for the model:

- The body, frame, and cab are assumed to be rigid bodies due to their great stiffness relative to the suspension.
- The frame is symmetrical to the longitudinal axis, only considering the vertical motion and the pitching motion of the frame, the fracturing pump and the engine during the operation, ignoring the influence of the lateral swing;
- For the fracturing pump, the platform engine and other components are used the mass to present;
- The role of leaf springs and tires on the frame is a simplified model, ignoring all non-linear factors, considered a linear spring and damping system.

Based on the above assumptions, the coupled system model shown in Figure 1 is established. The parameters are described in Table 1.

![Figure 1. Coupled system model](image-url)
| Name                                | Symbol | Unit   | Numerical value |
|-------------------------------------|--------|--------|-----------------|
| Engine mass                         | $m_e$  | kg     | 7500            |
| Fracturing pump mass                | $m_p$  | kg     | 9400            |
| Frame mass                          | $m_b$  | kg     | 16000           |
| Balance suspension mass             | $m_s$  | kg     | 400             |
| First axle tire mass                | $m_{t1}$ | kg   | 400             |
| Second axle tire mass               | $m_{t2}$ | kg   | 500             |
| Third axle tire mass                | $m_{t3}$ | kg   | 1000            |
| Fourth bridge tire mass             | $m_{t4}$ | kg   | 1000            |
| Truck engine inertia                | $I_e$  | kg-m$^2$ | 3451           |
| Fracturing pump inertia             | $I_p$  | kg-m$^2$ | 2107           |
| Frame inertia                       | $I_b$  | kg-m$^2$ | 300997        |
| Balance suspension inertia          | $I_s$  | kg-m$^2$ | 351            |
| First bridge center to truck gravity center distance | $l_1$ | m   | 4.709            |
| Second bridge center to truck gravity center distance | $l_2$ | m   | 3.097            |
| Suspension center to truck gravity center distance | $l_3$ | m   | 3.196            |
| Engine center to truck gravity center distance | $l_4$ | m   | 1.909            |
| Fracturing pump gravity center to truck gravity center distance | $l_5$ | m   | 4.364            |
| Engine gravity center to front suspension point distance | $l_6$ | m   | 1.179            |
| Engine gravity center to rear suspension point distance | $l_7$ | m   | 1.179            |
| Fracturing pump gravity center to front suspension point distance | $l_8$ | m   | 0.820            |
| Fracturing pump gravity center to rear suspension point distance | $l_9$ | m   | 0.820            |
| Balanced suspension length          | $l_{10}$ | m     | 1.450           |
| Front suspension stiffness          | $k_{sf}$ | kN-m$^{-1}$ | 457         |
| Medium suspension stiffness         | $k_{sm}$ | kN-m$^{-1}$ | 457         |
| Rear suspension stiffness           | $k_{sr}$ | kN-m$^{-1}$ | 583         |
| First axle tire stiffness           | $k_{t1}$ | kN-m$^{-1}$ | 1500        |
| Second axle tire stiffness          | $k_{t2}$ | kN-m$^{-1}$ | 1500        |
| Third axle tire stiffness           | $k_{t3}$ | kN-m$^{-1}$ | 3000        |
| Fourth axle tire stiffness          | $k_{t4}$ | kN-m$^{-1}$ | 3000        |
| Engine front suspension stiffness   | $k_{e1}$ | kN-m$^{-1}$ | 750         |
| Engine rear suspension stiffness    | $k_{e2}$ | kN-m$^{-1}$ | 750         |
| Fracturing pump front suspension stiffness | $k_{p1}$ | kN-m$^{-1}$ | 900         |
| Fracturing pump rear suspension stiffness | $k_{p2}$ | kN-m$^{-1}$ | 900         |
| Engine front suspension damping     | $c_{f1}$ | kN-s-m$^{-1}$ | 75         |
| Engine rear suspension damping      | $c_{f2}$ | kN-s-m$^{-1}$ | 75         |
| Fracturing pump front suspension damping | $c_{p1}$ | kN-s-m$^{-1}$ | 90         |
| Fracturing pump rear suspension damping | $c_{p2}$ | kN-s-m$^{-1}$ | 90         |
Front suspension damping \( c_{sf} \) kN-m \( s^{-1} \) 91.4
Middle suspension damping \( c_{sm} \) kN-m \( s^{-1} \) 91.4
Rear suspension damping \( c_{sr} \) kN-m \( s^{-1} \) 163.8
First axle tire damping \( c_{t1} \) kN-m \( s^{-1} \) 5
Second axle tire damping \( c_{t2} \) kN-m \( s^{-1} \) 5
Third axle tire damping \( c_{t3} \) kN-m \( s^{-1} \) 10
Fourth axle tire damping \( c_{t4} \) kN-m \( s^{-1} \) 10

2.3. Truck engine balance equation
According to the displacement coordination relationship, the relative displacement \( z_{e1} \) and \( z_{e2} \) of the front and rear suspension of the engine and the subsystem of the truck frame are expressed as:

\[
z_{e1} = z_b - \theta_b (l_4 + l_6) \tag{1}
\]

\[
z_{e2} = z_b - \theta_b (l_4 + l_6) \tag{2}
\]

Where \( z_b \) is the vertical motion displacement of the truck body and \( \theta_b \) is the elevation angle of the truck body.

The stress \( F_{e1} \) and \( F_{e2} \) of the front and rear suspension of the engine and the frame joint are:

\[
F_{e1} = k_{e1} (z_e - \theta_e l_6 - z_{e1}) + c_{c1} (\dot{z}_e - \dot{\theta}_e l_6 - \dot{z}_{e1}) \tag{3}
\]

\[
F_{e2} = k_{e2} (z_e + \theta_e l_7 - z_{e2}) + c_{c2} (\dot{z}_e + \dot{\theta}_e l_7 - \dot{z}_{e2}) \tag{4}
\]

Where \( z_b \) is the vertical motion displacement of engine and \( \theta_e \) is the elevation angle of the engine.

According to the Lagrange motion equation, the vertical motion balance equation and the pitching motion balance equation of the engine are respectively:

\[
m_e \ddot{z}_e + k_{e1} + k_{e2} - P_e = 0 \tag{5}
\]

\[
l_6 \ddot{\theta}_e - l_6 F_{e1} + l_7 F_{e2} - M_e = 0 \tag{6}
\]

Where \( P_e \) is the engine unbalanced force and \( M_e \) is the engine unbalanced torque.

2.4. Fracturing pump balance equation
According to the coordination of displacement, the relative displacement of the front and rear suspension of the fracturing pump and the frame \( z_{p1} \) and \( z_{p2} \) are respectively:

\[
z_{p1} = z_p + \theta_p (l_5 - l_2) \tag{7}
\]

\[
z_{p2} = z_p + \theta_p (l_5 + l_2) \tag{8}
\]

The force at the joint between the frame and the fracturing pump is:

\[
F_{p1} = k_{p1} (z_p - \theta_p l_5 - z_{p1}) + c_{p1} (\dot{z}_p - \dot{\theta}_p l_5 - \dot{z}_{p1}) \tag{9}
\]

\[
F_{p2} = k_{p2} (z_p + \theta_p l_5 - z_{p2}) + c_{p2} (\dot{z}_p + \dot{\theta}_p l_5 - \dot{z}_{p2}) \tag{10}
\]

Where \( z_p \) is the vertical motion displacement of fracturing pump and \( \theta_p \) is the elevation angle of the fracturing pump.
According to the Lagrange motion equation, the vertical motion balance equation and the pitching motion balance equation of the fracturing pump are respectively:

\[ m_p \ddot{x}_p + F_{p1} + F_{p2} - P_p = 0 \]  

\[ I_p \ddot{\theta}_p - l_g F_{p1} - 9F_{p2} - M_p = 0 \]

Where \( P_p \) is the fracturing pump unbalanced force and \( M_p \) is the fracturing pump unbalanced torque.

### 2.5. Tire balance equation

From the displacement coordination relationship, the relative displacement \( z_{ff} \) and \( z_{mm} \) at the connection between the front and rear suspension and the frame joint are respectively:

\[ z_{ff} = z_b - \theta_{b} l_1 \]  

\[ z_{mm} = z_b - \theta_{b} l_2 \]

From the force balanced relationship, the forces \( F_{ff} \) and \( F_{mm} \) of the front and rear suspension and the frame joint are:

\[ F_{ff} = k_{sf}(x_{t1} - z_{ff}) + c_{sf}(\dot{x}_{t1} - \dot{z}_{ff}) \]  

\[ F_{mm} = k_{sm}(x_{t2} - z_{mm}) + c_{sm}(\dot{x}_{t2} - \dot{z}_{mm}) \]

Where \( x_{t1} \) and \( x_{t2} \) are the vertical displacements of the first and second bridge tires. The vertical motion balance equations of tire 1 and tire 2 are:

\[ m_{t1} \ddot{x}_{t1} + F_{ff} + k_{t1}x_{t1} + c_{t1}\dot{x}_{t1} = 0 \]  

\[ m_{t2} \ddot{x}_{t2} + F_{mm} + k_{t2}x_{t2} + c_{t2}\dot{x}_{t2} = 0 \]

### 2.6. Balance suspension balance equation

According to the displacement coordination relationship, the relative displacement \( z_{rr} \) of the rear axle suspension and the frame joint is:

\[ z_{rr} = z_b + \theta_{b} l_3 \]

The force at the joint between the frame and the balanced suspension is:

\[ F_{rr} = k_{sr}(x_{s} - z_{rr}) + c_{sr}(\dot{x}_{s} - \dot{z}_{rr}) \]

The force of tire 3 is:

\[ F_{t3} = -k_{t3}\left(\ddot{x}_{s} - \frac{1}{2}\dot{\theta}_{s} l_{10}\right) - c_{t3}\left(\dot{x}_{s} - \frac{1}{2}\dot{\theta}_{s} l_{10}\right) \]

Where \( x_{s} \) is the balanced suspension vertical displacement and \( \theta_{s} \) is the elevation angle of balanced suspension.

The force of tire 4 is:

\[ F_{t4} = -k_{t4}\left(\ddot{x}_{s} + \frac{1}{2}\dot{\theta}_{s} l_{10}\right) - c_{t4}(\dot{x}_{s} + \frac{1}{2}\dot{\theta}_{s} l_{10}) \]

The vertical motion equation of the balanced suspension and the balanced equation of the pitching motion are respectively:
2.7. Frame balance equation

The vertical motion balance equation of the frame is:

\[
(m_x + m_y + m_z) \ddot{x} - F_{x1} - F_{x2} - F_{x3} - F_{x4} + F_{rr} = 0
\]

(23)

\[
\left( \frac{1}{4} m_x l_1^2 + \frac{1}{4} m_y l_1^2 + I_y \right) \ddot{\theta}_y + \frac{1}{2} l_1 F_{y1} - \frac{1}{2} l_1 F_{y2} = 0
\]

(24)

The pitching motion balance equation of the frame is:

\[
m_y \ddot{\theta}_y + F_{ry1} (l_4 + l_6) + F_{ry2} (l_4 - l_7) - F_{ry1} (l_5 + l_9) + F_{ry2} (l_5 + l_9) + F_{ry1} l_1 + F_{ry2} l_2 - F_{rr} l_2 = 0
\]

(25)

\[
I_\theta \ddot{\theta}_y + F_{y1} (l_4 + l_6) + F_{y2} (l_4 - l_7) - F_{y1} (l_5 + l_9) + F_{y2} (l_5 + l_9) + F_{y1} l_1 + F_{y2} l_2 - F_{rr} l_2 = 0
\]

(26)

3. Calculation and analysis

3.1. Calculation model

For the 2500 fracturing truck, the fracturing operation process was simulated by MATLAB/Simulink software. The main dynamic simulation parameters used are shown in Table 1.

In the calculation and analysis, the sine wave in the continuous signal source is selected to build the input signal, and the output module uses the scope to observe and display the simulation result. In the middle, the state space equation module and differential link are used for conversion. The Simulink simulation module built is shown in Figure 2. The equation of state is set according to the parameter in Table 1. Start time and end time are set respectively, and an ode45 solver is used for the solution.

![Figure 2. Simulation model in Simulink](image-url)
3.2. Result analysis

3.2.1. Vibration form analysis
In order to study the main vibration mode of the truck body, the engine speed of the truck is set to 1900 r/min, the gear position is 9 gears, the transmission ratio is 0.8, the truck engine and the fracturing pump are in the highest speed operation state which the vibration is the most severe. Through the previous Simulink simulation model, the time-velocity-acceleration curve of the vertical motion of the truck body (Figure. 3), the time-angular velocity-angular acceleration curve of the truck body pitching motion (Figure. 4) and the power spectral density curve of the truck body vibration (Figure. 5) are obtained.

It can be seen from Fig.3 and Fig.4 that the curve exhibits a certain periodicity, which is due to the fact that the input excitation is simplified to a sine and cosine curve. It can be seen from Fig. 6 that in the vertical and pitching motions of the truck body, the low frequency, especially in the 0~10 Hz region, has the greatest influence on the vibration of the truck body, and the power spectral density near 1 Hz reaches the peak value of 0.1 m²·s⁻³ and 0.0012 m²·s⁻³ respectively, while the high-frequency region has a smaller effect, with minimum values of 1.41×10⁻⁸ m²·s⁻³ and 1.29×10⁻¹⁰ m²·s⁻³.

It can be seen from Fig.3~5 that the truck body is most sensitive to vertical vibration, and its vibration intensity and pitching vibration are different by two orders of magnitude, which indicates that the fracturing truck vibration is mainly vertical motion during the fracturing operation, and the pitch direction vibration is weaker.
3.2.2. Stiffness effect

In order to study the influence of the suspension stiffness of the truck body on the truck body vibration, the acceleration variation in the range of 0-10 Hz (large frequency influence area) is observed by changing the suspension stiffness. Due to the fact that the front and middle suspension area are close to each other, and the stiffness is exactly the same, the front and middle suspension are considered together, and the rear suspension is considered separately. The stiffness values were compared in groups 250 kN/m, 400 kN/m, 550 kN/m, and 700 kN/m. In order to more intuitively observe the relationship between frequency and acceleration, the time domain acceleration curve obtained by Simulink is transformed by the Spectrum RMS of DATS software to obtain the change curve of acceleration with the vertical vibration frequency of the truck body[8], as shown in Fig.6-7.

It can be seen from Figure.6 that the influence of the changes of front and middle suspension parameters on the vertical acceleration of the truck body is mainly reflected in the change of the corresponding frequency of the peak value. The increase of stiffness parameters makes the corresponding frequency of the first and second peak shift to the right, while the influence on the amplitude is relatively small.
As can be seen from Figure 7, the influence of rear suspension stiffness on the vertical vibration of the truck body is also reflected in the amplitude change and peak change. With the increase of the rear suspension stiffness, the amplitude of vertical vibration will decrease: the first peak value decreases from 0.85 m·s\(^{-3}\) to 0.52 m·s\(^{-3}\); the second peak amplitude also has a similar rule, but the change is small. The frequency corresponding to the first and second peaks moves with the change of rear suspension stiffness, but there is no obvious regularity.

3.2.3. Damping effect
Figure 8 and Figure 9 show the change of vertical motion acceleration with suspension damping. The damping of front and middle suspension has a great influence on the amplitude. With the increase of damping, the first peak value decreases gradually. Then, the change of rear of suspension damping has a small impact on the amplitude of the vertical movement of the truck body and a large impact on the frequency corresponding to the peak value. It can be considered that within 120kN·s·m\(^{-1}\)~200kN·s·m\(^{-1}\), as the damping increases, its corresponding peak right shift.
4. Conclusion

- In this paper, the dynamic model of the fracturing truck operation process is established by the lumped parameter method and the dynamics simulation analysis is trucked out. The velocity and acceleration curves of the vertical motion and the pitching motion of the fracturing truck are obtained. By comparison, it is found that the vertical motion was the main vibration form of fracturing truck.
- By analyzing the power density distribution curve of the vibration frequency of the truck body, it is found that the low frequency (0~10 Hz) has the greatest influence on the vibration of the truck body.
- The influence of suspension stiffness and damping on vibration spectrum is reflected in the following aspects: the front and middle suspension spring stiffness mainly affect the frequency corresponding to the peak value, while the rear suspension spring stiffness has a great influence on the amplitude. The increase of front and middle suspension damping is beneficial to decrease the small value, while the increase of rear suspension damping has little effect on the amplitude value but can make the frequency corresponding to the peak shift to the right. Therefore, from the perspective of vibration reduction, the rear suspension can adopt greater stiffness, while the front and middle suspension can adopt greater damping.

5. References

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