A SIMULATION STUDY ON THE INFLUENCE OF SUSPENSION SYSTEM PARAMETERS ON TRACTOR VIBRATION CHARACTERISTICS

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ABSTRACT: In order to study the influence of the position of farm implement on the vibration characteristics of the tractor, this paper takes the Dongfanghong-LX754 tractor and Huaiweng 1LS-740 subsoiler as the research objects to carry out the studies, it constructs a simplified geometric model for the tractor and the farm implement using the CATIA software and adopts MATLAB to compile the random road surface file; then, based on the ADAMS software, it modifies the tire parameter text and imports the random road surface file to construct a virtual prototype for the subsoiler-tractor-road surface system; after that, through simulation, this paper studies the influence of suspension system parameters on the vibration characteristics of the tractor, such as the vertical vibration acceleration of the tractor driver, the angular acceleration of the pitch vibration at the position where the seat is installed, and the vertical vibration acceleration of the mass center of the tractor. The research results show that when the angle between the inner lift arm and the horizontal direction increased from 20° to 70°, the peak vertical vibration acceleration of the driver decreased from 30.09 m/s² to 17.61 m/s², the peak vertical vibration frequency increased from 1.36 Hz to 2.12 Hz, the angular acceleration of the pitch vibration at the seat position decreased from 11.12 rad/s to 7.56 rad/s, and the peak vertical vibration acceleration of the mass center of the tractor decreased from 42.42 m/s² to 29.33 m/s². This study provides a certain reference for the design of the vibration damping system of the tractor.

KEYWORDS: Tractor; Vibration; Acceleration; Suspension System; Position of Farm Implement

1 INTRODUCTION

Under normal conditions, the tractors for agricultural use in China are running on roads with poor conditions such as rural gravel roads or dirt roads. In the case of tractor with suspension system, due to the existence of equipped farm implements, it will cause great vibrations to the tractor when driving on roads with poor conditions, resulting in a significant reduction in the driving steadiness of the tractor and the ride comfort of the driver (Wang, 2015; Zheng et al., 2019). Researchers at home and abroad have found that the vibration of the tractor will cause certain damage to the service life of the tractor parts and the driver's body and psychology (Dou et al., 2019; Sakai, 1999; Kumar et al., 2001). Zhu et al. (2014) established a vibration model for tractor with suspended farm implements, and studied the influence of the mass of the farm implements on the vibration characteristics of the suspension system of the tractor; their research results showed that, when the speed of the tractor remains unchanged, the increase in the mass of the farm implement will result in a decrease in the natural frequency of the vertical and pitch vibrations of the tractor with suspension system. Gu Gu et al. (2014) adopted Pro/E and ADAMS to establish a tractor and road surface model, and simulated and analyzed the mutual influence of vibrations in the three directions of vertical, roll and pitch. Muzzamil et al. (2004) studied the driver's vibration under different field conditions, and their research results showed that the suspended farm implements had an impact on the vibration, while the influence of the field type on the vibration was not significant. Zhu and Zhu (2013) analyzed the influence of tire inflation pressure and forward speed on the vibration of unloaded tractor through experiments, and the results showed that the tractor vibration was greatly affected by the tire inflation pressure and forward speed of the tractor, and the field soil moisture also had a large impact on the vibration of the front axle of the tractor.

This paper takes the tractor with suspended farm implements as the research object, and constructs a virtual prototype for the subsoiler-tractor-road surface system based on ADAMS; then simulation analysis is conducted on the tractor suspended with
farm implements at different positions and the unloaded tractor to explore the influence of different farm implement positions on the vibration of the tractor, so as to provide certain reference for the design of the vibration damping system of the tractor.

2 MATERIALS AND METHODS

2.1 Modeling of the entire tractor mechanical system

There are many causes for the vibration of the tractor. During transportation, the vibration of the tractor is related to many factors such as the driving speed of the tractor, the mass of the iron counterweight, the farm implements, the diameter of the tire, and the elastic properties, etc. If all these factors are taken into consideration, the modeling and simulation will be very complicated, therefore, this study only concerns about the vibration caused by the position of the farm implement and the bumpy road surface. Under the influence of the external environment, the tractor and the farm implement will generate multi-degree-of-freedom vibrations, including the tractor’s surge/sway/heave, and pitch/roll/yaw motions, the vertical motions of the four wheels and the steering of the front wheels, and the heave and pitch motions of the farm implement. Since the frequency below 20 Hz has a greater impact on the human body, this study only considers the vibrations with a frequency below 20 Hz, the vibrations with a frequency above 20 Hz, such as the vibration caused by the operation of the engine and the transmission system of the tractor had not been taken into consideration (Zhou et al., 2017). In the constructed tractor mechanical system, the tractor engine, transmission system and other structures were simplified into mass blocks that were fixed to the chassis of the tractor. Under the condition that the mass center, mass, and the installation positions of each component of the farm implement and the tractor are accurate, since the other components are relatively rigid, it’s not necessary to emphasize their specific shapes and structures, therefore, they were simplified to a rod structure (Miao et al., 2012). The model of the cab and the chassis was constructed according to the requirements of mass, front wheel track, rear wheel track, and the wheelbase between front and rear wheels; and the installation positions of each component were determined, so as to ensure the spatial location relations between the components. Based on the fact that the average body weight of adult males aged 18 years and above in China is 66.2 kg, in this study, the human body was simplified as a square iron block with a side length of 20.4 cm.

The simplified three-dimensional modeling of the tractor and the subsoiler was completed in CATIA (Figure 1, Figure 2), the model format was converted to stp, with the help of Pro/E, the three-dimensional model of the tractor with suspended farm implement was converted to a x_t format file. Using the File Import command in ADAMS, the constructed models were imported into ADAMS, and the simplified three-dimensional model of the tractor with suspended farm implement was obtained, then the parameters of the mass and properties of each component were selected and the materials of each component were set.

![Fig. 1 Model of the entire tractor](image1)

![Fig. 2 Geometric model of the tractor with suspended farm implement](image2)

2.2 Construction of tire and random road surface model

In this study, the Fiala tire model was adopted, according to formulas (1)-(4), the parameters of the tire were compiled into the tire property file (Crolla et al., 1990; Lines and Murphy, 1991; Zhang, et al. 2012).

\[ K = 172 - 69.69r + 5.6Y + 0.527K_w r P \] (1)

In formula (1), \( K \) is the radial stiffness of the tire, KN/m; \( r \) is the rim radius, m; \( K_w \) is the tire width, m; \( Y \) is the service life, a; \( P \) is the tire pressure, Pa.

\[ C = C_0 + 0.7P \] (2)
In formula (2): $C$ is the radial damping of the tire, KN·s/m; $C_0$ is the radial damping of the tire when tire pressure is 0, KN·s/m.

$$K_1 = \frac{C_1 \pi r_1}{\sigma_1 100} \quad (3)$$

$$K_2 = \frac{C_2}{\sigma_2} \quad (4)$$

In formulas (3)-(4): $K_1$ and $K_2$ are respectively the lateral stiffness and the longitudinal stiffness, N/(º); $C_1$ and $C_2$ are respectively the cornering stiffness and the slip stiffness, N/rad; $\sigma_1$ and $\sigma_2$ are the free length of the tires, m; $r_1$ is the dynamic radius of the tire, m.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Front wheel</th>
<th>Rear wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius /mm</td>
<td>550</td>
<td>700</td>
</tr>
<tr>
<td>Coefficient of static friction</td>
<td>0.95</td>
<td>0.95</td>
</tr>
<tr>
<td>Coefficient of kinetic friction</td>
<td>0.75</td>
<td>0.75</td>
</tr>
<tr>
<td>Cornering stiffness / (N/rad)</td>
<td>60 000</td>
<td>120 000</td>
</tr>
<tr>
<td>Longitudinal slip stiffness / (N/mm)</td>
<td>495.25</td>
<td>437.16</td>
</tr>
<tr>
<td>Camber stiffness / (N/rad)</td>
<td>6 000</td>
<td>12 000</td>
</tr>
<tr>
<td>Free radius /mm</td>
<td>553</td>
<td>705</td>
</tr>
<tr>
<td>Coefficient of rolling resistance</td>
<td>0.15</td>
<td>0.15</td>
</tr>
<tr>
<td>Radial damping ratio</td>
<td>0.04</td>
<td>0.04</td>
</tr>
<tr>
<td>Radial stiffness / (N/mm)</td>
<td>348.48</td>
<td>560.82</td>
</tr>
</tbody>
</table>

Different forms of trigonometric series were used to simulate the random road surface. The spatial frequency was $n_{mid,i} (=1, 2, \ldots, m)$, the road surface model adopted a sine wave function with a standard deviation of $\sqrt{G_q(n_{mid,i})\Delta n_i}$, and the sine wave function can be expressed as (Xu, 2007; Zhang and Zhong, 2004; GB/T7031-2005, 2005):

$$q_i(a) = \sqrt{2G_q(n_{mid,i})\Delta n_i \sin(2\pi n_{mid,i}a + \omega_i)} \quad (5)$$

$$q(a) = \sum_{i=1}^{m} \sqrt{2G_q(n_{mid,i})\Delta n_i \sin(2\pi n_{mid,i}a + \omega_i)} \quad (6)$$

In formulas (5) and (6), $a$ is the longitudinal position of the road surface; $\omega_i$ is a random number in $[0, 2\pi]$.

For the random road surface, the longitudinal position was set to $a$, and the horizontal position was set to $b$:

$$q(a, b) = \sum_{i=1}^{m} \sqrt{2G_q(n_{mid,i})\Delta n_i \sin(2\pi n_{mid,i}a + \omega_i(a, b))} \quad (7)$$

where, $\omega_i(a, b)$ is a random number in interval $[0, 2\pi]$ at any point $(a, b)$ on the road surface.

The H-class road surface established in MATLAB is shown in Figure 3.

2.3 Construction of the virtual prototype model for the tractor-subsoiler-road surface system

The three-dimensional model of the assembled subsoiler-tractor was imported into ADAMS, the material properties were set, the random road surface with different classes was compiled in MATLAB based on the sine wave superposition method, the format of the created txt files was altered to RDF format, and the files were saved. According to the movement relations between the components of the tractor with suspended farm implement, the engine, gearbox, transmission gear and cab were regarded as rigid bodies and simplified to mass blocks installed on the chassis. Revolute joints were set between the farm implement, the three-point suspension system and the tractor, and fixed joints were set between the simplified driver model and the seat, and then the constraints of the virtual prototype model were completed. The “Create wheel and tire” function of the Special force module in ADAMS was used to add the compiled tire and road surface files, and revolute joints were set between the tire and the drive axle. The constructed virtual prototype model of the subsoiler-tractor-road surface system is shown in Figure 4.

2.4 Test design

The vibration test was carried out in the Mechanical and Electrical Engineering School of Anhui Agricultural University to verify the accuracy of the constructed simulation model. The
tractor model was Dongfanghong-LX754, and the suspended farm implement was a subsoiler. Because the focus of this study was the position of the farm implement, a hydraulic device was set to change the position of the farm implement. The test runway was a H-class random road surface with a length of 50m, which was basically the same with the road surface in the simulation. The driving speed of the tractor was 10 km/h, and the data was collected three times; the sampling frequency was 200 Hz, and the average value was taken in the end. According to the provisions of GB/T 10910-2004 Agricultural Wheeled Tractors and Field Machinery - Measurement of Whole-Body Vibration of the Operator, on the H-class standard random road surface, the vibration acceleration of the tractor with a 45° angle between the inner lift arm and the horizontal direction was collected. The installation positions of the acceleration sensors were: one at the position where the seat was installed, one directly below the center of the front axle, one directly below the center of the chassis, and one directly below the center of the rear axle. The vibration test equipment is shown in Figure 5.

3 RESULTS AND ANALYSIS

3.1 Comparison of test results and simulation results

See Table 3 for the results of the vibration simulation test of the tractor with suspension farm implement, for the peak acceleration of vertical vibration at the position directly below the chassis center, the peak acceleration of vertical vibration at the position where the seat was installed, the peak acceleration of vertical vibration at the position of the front axle, and the peak acceleration of vertical vibration at the position of the rear axle, the relative error of the test results and simulation results was 8.52%, 9.56%, 7.04%, and 10.83%, respectively, which had good consistency and can basically meet the content of this study.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Test results</th>
<th>Simulation results</th>
<th>Relative error%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peak vertical vibration acceleration at the bottom center of the chassis</td>
<td>38.35</td>
<td>35.08</td>
<td>8.52</td>
</tr>
<tr>
<td>Peak vertical vibration acceleration at the position where the seat is installed</td>
<td>37.32</td>
<td>33.75</td>
<td>9.56</td>
</tr>
<tr>
<td>Peak vertical vibration acceleration of the front axis</td>
<td>62.10</td>
<td>57.73</td>
<td>7.04</td>
</tr>
<tr>
<td>Peak vertical vibration acceleration of the rear axis</td>
<td>49.71</td>
<td>44.32</td>
<td>10.83</td>
</tr>
</tbody>
</table>

3.2 The influence of the position of farm implement on the vibration acceleration of the tractor

The simulation results are shown in Figure 6. A, B, and C are respectively the vertical vibration acceleration of the mass center of the driver, the angular acceleration of the pitch vibration at the seat position, and the vertical vibration acceleration of the mass center of the tractor when the tractor was moving at a speed of 10 km/h, and the angle between the inner lift arm and the horizontal direction was 20°, 45°, and 70°. When the angle between the inner lift arm and the horizontal direction was 20°, the peak vertical vibration acceleration of the mass center of the driver was 30.09 m/s², the peak angular acceleration of pitch vibration at the seat position was 11.12 rad/s², and the peak vertical vibration acceleration of the mass center of the tractor was 42.42 m/s². When the angle between the inner lift arm and the horizontal direction was 45°, the peak vertical vibration acceleration of the mass center of the driver was 25.73 m/s², the peak angular acceleration of the pitch vibration at the seat position was 8.85 rad/s², and the peak vertical vibration acceleration of the mass center of the tractor was 35.08 m/s². When the angle between the inner lift arm and the horizontal direction was 70°, the peak vertical vibration acceleration of the mass center of the driver was 21.21 m/s², the peak angular acceleration of the pitch vibration at the seat position was 7.56 rad/s², and the peak vertical vibration acceleration of the mass center of the tractor was 29.33 m/s². In conclusion, with the increase of the angle between the inner lift arm and the horizontal direction, the
vertical vibration acceleration of the driver, the angular acceleration of the pitch vibration at the seat position, and the vertical vibration acceleration of the mass center of the tractor had all decreased to some extent.
3.3 Acceleration power spectrum

Under the condition of different positions of the suspended farm implement, the time-domain signal of the vertical vibration acceleration of the mass center of the driver was obtained through simulation; the power spectral density of the vertical acceleration of the mass center of the driver was calculated by Fourier transform in the ADAMS post-processing module, the results are shown in Figure 7. With the increase of the angle between the inner lift arm and the horizontal direction, the peak frequency of vertical vibration of the driver increased from 1.36 Hz to 2.12 Hz.

3.4 The influence of the suspended farm implement on the tractor

Figure 8 shows the vibration acceleration of the driver and the tractor when the unloaded tractor was moving at a speed of 10 km/h and the angle between the inner lift arm and the horizontal direction was 70º. It can be seen from the figure that, the peak vertical vibration acceleration of the driver was 24.59 m/s², the peak frequency of the vertical vibration of the driver was 0.78 Hz, and the peak angular acceleration of the pitch vibration at the seat position was 5.73 rad/s², the peak vertical vibration acceleration of the mass center of the tractor was 37.84 m/s². Compared with the
unloaded tractor, for the tractor with suspended farm implement, the vertical vibration accelerations of the driver and the tractor were increased to a certain extent, while the peak frequency of the vertical vibration of the driver and the angular acceleration at the seat position were decreased to some extent.

Fig. 8 Measured values of the vibration of the driver under the condition of unloaded tractor

4 DISCUSSION

In this study, a hydraulic device was used to control the installation position of the farm implement, the data of the vibration of the driver and the tractor were measured, and the vibration data of unloaded tractor and tractor with suspended farm implement during the driving were compared. The paper studied the influence of the position of farm implement on the vibration characteristics of the tractor, and concluded that, under the condition of the tractor with suspended subsoiler, with the increase of the position of the farm implement, the natural frequency of the vertical vibration of the driver increased as well, the peak frequency of the vertical vibration of the driver increased from 1.36Hz to 2.12Hz, both the vertical vibration acceleration of the mass center of the driver and the angular acceleration of the pitch vibration of the seat position decreased, where the peak acceleration of the vertical vibration of the mass center of the driver decreased from 30.9 m/s² to 17.61 m/s², and the peak angular acceleration of the pitch vibration of the seat position decreased from 11.12rad/s² to 7.56rad/s². Under the condition of unloaded tractor, both the vertical vibration accelerations of the driver and the mass center of the tractor had increased to some extent, and the vertical vibration frequency of the driver and the angular acceleration of the seat position decreased to some extent. For the driver of the tractor with suspended farm implement, with the decrease of the angle between the inner lift arm and the horizontal direction, the vibration acceleration of the driver, the vertical vibration frequency of the driver, the angular acceleration of the seat position, and the vibration acceleration of the mass center of the tractor all increased to some extent, resulting in greater damages to the health of the driver, and the
driver’s comfort had been decreased. This study provides a certain reference for the design of the vibration damping system of the tractor.

5 CONCLUSION

Due to the limitation of various aspects, when building the virtual prototype model, this study had simplified the tractor engine, the driver and the transmission system to some extent, and the friction force at the hinges of the tractor with suspended farm implement had not been taken into consideration, which had inevitably brought some errors in the simulation results. Therefore, in the follow-up studies, it is necessary to improve the models of the tractor, the farm implement and the driver’s body, so as to improve the accuracy of simulation analysis.

6 ACKNOWLEDGEMENT

This work was supported by the Chinese National Key Research and Development Program (No. 2017YFD0700104).

REFERENCES