ABSTRACT

Engine is a main vibration source for a tractor. Under its excitation, the engine hood on a tractor may vibrate seriously, because the hood is connected rigidly to the tractor body or frame. As a result, some instruments and lights equipped on the hood may be damaged or can’t work well. In order to control the hood vibration, an engine hood on a wheeled tractor is investigated here. At first, the vibration transmission path was studied. Then the hood vibration was tested and analyzed with the technique of frequency analysis. And the natural vibration characteristics analysis on the hood was also conducted with the technique of experimental modal analysis. Based on the analysis results, some measures for reducing the vibration were applied. So the hood vibration was reduced obviously, and its vibration situation was improved noticeably.

KEY WORDS: tractor, engine hood, vibration control, experimental modal analysis

INTRODUCTION

Tractor’s vibration may affect the durability of its parts and the health of its operator. Here an engine hood on a wheeled medium-sized tractor is studied. The engine hood was supported stiffly at four points. When the tractor works, the engine hood vibrates seriously, which sometimes causes the instruments and lights damaged. In order to decrease the vibration of the engine hood, experimental modal analysis and some vibration tests were conducted. According to the conclusion of the analysis, special elastic supporters were developed. As a result, the vibration of the engine hood was improved obviously.

1. VIBRATION TEST OF THE ENGINE HOOD

According to previous researches and relative documents[1,2], when a tractor works, the engine hood would be excited by the tractor body, but the real vibration sources are ground, the working engine and the transmission system, among them the engine is often the main vibration source[2,7,8]. So when the vibration experiment of the engine hood was conducted, the tractor was parked on a plain ground, and only the engine was working. 18 test points were selected on the engine hood (see figure 1) in order that the vibration situation could be
presented comprehensively. When the experiment was conducted, the tractor’s engine rotated in its highest speed of 2050rpm. The values of vibration acceleration obtained at every test points were listed in Table 1, from which it can be found that the position where the vibration was the most serious is at the point 14. In order to get the overall vibration situation, the average vibration spectrum of the 18 points was calculated (see Figure 2, where G=9.8 m/s^2).

From Figure 2, it could be found that there are three big peaks, where the relevant frequencies are 68.75 Hz, 100 Hz and 343.75 Hz respectively, while the vibration values are 1.487G, 0.605G and 0.489G. Among them the biggest peak is at the frequency of 68.75 Hz, which is the frequency of the second unbalanced inertia force and that of the combustion explosion force. That indicates that the major exciting sources are the combustion explosion force and the second unbalanced inertia force.

### Table 1. Vibration acceleration values of the engine hood (n=2050rpm)

<table>
<thead>
<tr>
<th>position</th>
<th>acc (m/s^2)</th>
<th>position</th>
<th>acc (m/s^2)</th>
<th>position</th>
<th>acc (m/s^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>26.89</td>
<td>7</td>
<td>32.38</td>
<td>13</td>
<td>46.08</td>
</tr>
<tr>
<td>2</td>
<td>18.79</td>
<td>8</td>
<td>16.08</td>
<td>14</td>
<td>51.15</td>
</tr>
<tr>
<td>3</td>
<td>16.25</td>
<td>9</td>
<td>37.89</td>
<td>15</td>
<td>40.86</td>
</tr>
<tr>
<td>4</td>
<td>19.87</td>
<td>10</td>
<td>29.86</td>
<td>16</td>
<td>32.70</td>
</tr>
<tr>
<td>5</td>
<td>35.09</td>
<td>11</td>
<td>48.17</td>
<td>17</td>
<td>44.78</td>
</tr>
<tr>
<td>6</td>
<td>27.76</td>
<td>12</td>
<td>18.70</td>
<td>18</td>
<td>27.72</td>
</tr>
</tbody>
</table>

### 2. EXPERIMENTAL MODAL ANALYSIS OF THE ENGINE HOOD

For a vibration system in engineering, its motion equation will be:

\[ \begin{bmatrix} M \end{bmatrix} \dddot{x} + \begin{bmatrix} C \end{bmatrix} \dot{x} + \begin{bmatrix} K \end{bmatrix} x = \{f(t)\} \]  

(1)

where, \( \{x\}, \{\dot{x}\}, \{\dddot{x}\} \) — the vibration displacement vector, velocity vector and acceleration vector;

\[ [M] \] — mass matrix of the structure;  
\[ [K] \] — rigidity matrix of the structure;  
\[ [C] \] — damping matrix of the structure;  
\[ \{f(t)\} \] — exciting forces vector.

Do Laplace transformation to (1), following equation can be obtained:

\[ [H(s)]^{-1} \{X(s)\} = \{F(s)\} \]

(2)

Where, \([H(s)] = \left( [M]s^2 + [C]s + [K] \right)^{-1}\), the transmission function matrix of the vibration system, which is only determined by such system physical characteristics as mass, rigidity and damping. The \( s \) is the Laplace transformation variable. Substitute \( s = j\omega \) into transmission function matrix, the frequency response function matrix then be obtained,

\[ [H(\omega)] = \left( -\omega^2[M] + j\omega[C] + [K] \right)^{-1} \]

(3)

For a vibration system with \( n \) degrees of freedom, when the excitation is placed at \( P \), and the response is measured at \( L \), then the frequency response function is:

\[ H_{pL}(\omega) = \frac{X_L}{F_p} = \sum_{i=1}^{n} \frac{\Phi_{pi}}{K_{ii} \left( 1 - \omega_i^2 \right)^2 + j \omega_i} \]

where, \( X_i \) — the displacement at response position  
\( F_p \) — exciting force