Design and Development of Tuned Vibration Absorber to Control Engine Body Vibration in Idling and Running Condition

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Abstract:
In the present scenario, the Automobile industries are focusing on economy with high efficiency. As the torque requirement enforces little bit or zero modification in the conventional engine. To achieve the target of high efficiency, light weight vehicle may induce less grip on engine foundation and thus in all petrol/diesel driven car, the engine body vibrates. Our target is to reduce it by using Tuned Vibration Absorber. So, here we are presenting the design and development of such vibration absorber which reduce this vibration and less vibrational energy transmitted to the engine body. By using this Tuned vibration absorber vibration of the engine body is reduced by 65 to 70%.

Keywords: Idling Condition, tuned vibration absorber, engine vibration, vibration control.

1. INTRODUCTION

Due to the unbalanced forces, engine produces the vibratory forces from the engine parts during the operation. The vibration caused by the engine at the supports is combination of torsional vibration and the longitudinal vibration. The torsional vibration is caused at the crankshaft due to engine loads and the fluctuating combustion pressures. The longitudinal vibrations are caused at the mounts by the reciprocating and rotating parts of the engine. They analyzed the mathematical model and take the balancing mass as the design variables. The objective function used in research is the vibratory forces from the engine, transmitted to the engine mounts. The objective function is to be minimized the vibratory outcome of the engine. A machine or system may experience excessive vibration if it is acted upon by a force whose excitation frequency which is nearly coincides with a natural frequency of the machine or system. In such cases, the vibration of the machine or the system can be reduced by using a vibration neutralizer or vibration absorber. By adding an extra mass, spring and damper on the host structure and by calibrating its natural frequency to be equal with the excitation frequency of the host structure, the device can be used in vibration reduction. The device is called tuned vibration absorber (T.V.A). Therefore, by attaching passive T.V.A, it will only suppress the vibration at a single frequency where it has been tuned. Passive tuned vibration absorber maybe effective at low frequency. The idea of the vibration absorber was pioneered by Watts [1] in 1983 and Frahm [2] in 1909. The secondary system with mass-spring was considered as vibration absorber. Tuning of absorber i.e. adjusting either of the m,c,k parameters of the absorber such that natural frequency of the absorber becomes equal to the excitation force frequency. TVA for the electric grass trimmer has attenuated the vibration by 67% [4]. TVA for the control of HAV of the motorcycle handles had given remarkable results [5]. The lightly weighted TVA attached to the handle of the percussive machine allows suppression of the dominating harmonics of machine accelerations [6]. A Dual mass TVA is developed to avoid the resonance condition of a Hand blender and shifting the frequency away from the resonating frequency [7]. So, for controlling the vibration of the engine body a tuned vibration absorber is developed.

2. EXPERIMENTAL SETUP

Figure 1. Experimental setup
The experimental setup is shown in the figure no. 1. The T.V.A is farmed on the engine and the reading of the vibration is taken in the mobile software F.F.T analyzer. Finally the reading of the vibration are compared, the reading of the vibration with the T.V.A and without the T.V.A.

3. THEORETICAL ANALYSIS

3.1 TRANSVERSE VIBRATION OF THE CANTILEVER BEAM

Beams are widely used for structural elements such as floor supports, part of chassis etc. Beams of different dimensions are used for different purpose. A vibrating beam is an elastic distributed mass system which has infinite degree of freedom and hence same number of the natural frequency. But from practical point of view only a few of lower natural frequency are important.
Figure 2. Cantilever beam

As shown in figure a cantilever beam is considered having a mass at its free end. And consider beam as weightless.

The stiffness of beam is given as

\[ k = \frac{3EI}{l} \]  \( (1) \)

The general equation of motion for the undamped free vibration is given as

\[ m\ddot{x} + kx = 0 \]  \( (2) \)

Substituting the value of \( k \) in the above equation,

\[ m\ddot{x} + \frac{3EI}{l}x = 0 \]  \( (3) \)

\[ \omega_n = \frac{3EI}{\sqrt{Pl_m}} \text{ rad/s} \]  \( (4) \)

\[ f = \frac{1}{2\pi} \sqrt{\frac{3EI}{Pl_m}} \text{ Hz} \]  \( (5) \)

3.2 DUNKERLEY’S METHOD

Natural frequencies of structures are evaluated by this method. This method is used to find the natural frequency of transverse vibrations. The load of the system is uniformly distributed. We have used four cantilever beams on a flexible link for that Dunkerley’s equation can be written as;

\[ \frac{1}{\omega_n^2} = \frac{1}{\omega_1^2} + \frac{1}{\omega_2^2} + \frac{1}{\omega_3^2} + \frac{1}{\omega_4^2} \]  \( (6) \)

Where \( \omega_n \), Natural frequency of vibrating engine body = 12.56 rad/sec

\( \omega_1 \), \( \omega_2 \), \( \omega_3 \), \( \omega_4 \), Natural frequency of individual point loads = 25.12 rad/sec (By using Dunkerley’s method)

\[ f = \text{Frequency of engine body} = 2 \text{ Hz} \]

\[ E = 200*10^3 \text{ MPa} \]

\[ I = \frac{b^3h^3}{12} = 341,33 \text{ mm}^4 \]

\[ l = 90 \text{ mm} \]

Using equation no. 5 for finding the mass of individual point load on a cantilever beam

\[ m = 440 \text{ gm} \]

While designing the Tuned vibration absorber the mass taken for individual point on a cantilever beam is 400 grams approximately considering the other factor. The design of the Tuned vibration absorber is shown below.

Figure 3. Tuned vibration absorber

4. RESULTS AND DISCUSSIONS

In figure no. 4 the highest peak is around 105 m/s² acceleration of the vibration of the engine body without T.V.A. The vibration at the idling condition is very high which will decrease the performance of the engine. The wearing of the engine part is more without T.V.A. So the life of the engine part decreases.

Figure 4. Acceleration vs Frequency (Hz) of engine body without T.V.A

In figure no. 5 the highest peak is around 35 m/s² acceleration of the vibration of the engine body with T.V.A. The vibration of the engine body is reduced by about 65-70%. Due to this the performance of the engine body increases and the life of the engine parts also increase. Thus the car efficiency automatically increases with the increase in the performance of the engine.

4.1 Table for comparison of the bar graph with and without T.V.A.

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Acceleration (m/s²)</th>
<th>% Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Without T.V.A</td>
<td>With T.V.A</td>
</tr>
<tr>
<td>0.5</td>
<td>51.4</td>
<td>36.2</td>
</tr>
<tr>
<td>1</td>
<td>98.9</td>
<td>50</td>
</tr>
<tr>
<td>1.5</td>
<td>46.9</td>
<td>28.8</td>
</tr>
<tr>
<td>2</td>
<td>105</td>
<td>28.3</td>
</tr>
<tr>
<td>2.5</td>
<td>62.1</td>
<td>15.6</td>
</tr>
<tr>
<td>3</td>
<td>60.8</td>
<td>18.9</td>
</tr>
</tbody>
</table>

Average reduction = 55.71%

5. CONCLUSION

We conclude that the engine body vibration during idling condition is much more, which is harmful to engine
performance. A FFT analyzer (mobile software) was used to measure the vibration of the engine body, in idling condition maximum acceleration was recorded around 105 m/s\(^2\) at 2Hz frequency. To reduce this T.V.A had been designed and by using that maximum acceleration has reduced to 28.3 m/s\(^2\). Thus the vibration of the engine body is reduced by almost 65 to 70%.

6. REFERENCES


[7] “vibration control of a hand blender with the tuned vibration absorber” bysheth. A