Vibration Controlling Three Wheeler During Idling Condition

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Abstract

Industry accepted method for vibration control of industrial equipment includes Force Reduction, Mass Addition, Tuning, Isolation, and Damping. This paper will briefly introduce tuning method, and describe practical methods for its application. Several scenarios and case studies will be presented, with emphasis on pragmatic solutions to industrial vibration problems. The aim of the present project is to reduce the vibration in three wheeler in the idling condition. The characteristics of the vehicle are; it is generally used for the transportation of goods, this single cylinder engine is just located below the driver's seat. Thus a very high frequency of vibration is transmitted to the handle, seat and back portion of the vehicle. Thus we are making a vibration absorber device that will reduce the vibration of a great amount. The purpose of the project is to provide a good comfort to driver and increase the life of the vehicle. The experimental results demonstrate that the designed vibration absorber has better performance in terms of vibration absorption capacity.

Keywords: Idling Condition, vibration control, vibration absorption capacity

I. INTRODUCTION

Industry accepted method for vibration control of industrial equipment includes Force Reduction, Mass Addition, Tuning, Isolation, and Damping. This paper will briefly introduce tuning method, and describe practical methods for its application. Several scenarios and case studies will be presented, with emphasis on pragmatic solutions to industrial vibration problems. The aim of the present project is to reduce the vibration in three wheeler in the idling condition. The characteristics of the vehicle are; it is generally used for the transportation of goods, this single cylinder engine is just located below the driver's seat. Thus a very high frequency of vibration is transmitted to the handle, seat and back portion of the vehicle. Thus we are making a vibration absorber device that will reduce the vibration of a great amount. The purpose of the project is to provide a good comfort to driver and increase the life of the vehicle. The experimental results demonstrate that the designed vibration absorber has better performance in terms of vibration absorption capacity.

II. DIFFERENT METHODS FOR CONTROLLING VIBRATIONS

1) Force Reduction of excitation inputs due to, for example, unbalances or misalignment, will decrease the corresponding vibration response of the system.
2) Mass Addition will reduce the effect (system response) of a constant excitation force.
3) Isolation rearranges the excitation forces to achieve some reduction.
4) Damping is the conversion of mechanical energy (vibrations) into heat.
5) Tuning (changing) the natural frequency of a system or component will reduce or eliminate Amplification due to resonance.

III. TUNING

Vibration absorbers are devices attached to flexible structures in order to minimize the vibration amplitudes at a specified set of points. Design of vibration absorbers has a long history. First vibration absorber proposed by Frahm in 1909 [Den Hartog, 1956] consists of a second mass-spring device attached to the main device, also modeled as a mass-spring system, which prevents it from vibrating at the frequency of the sinusoidal forcing acting on the main device. If the absorber is tuned so that its natural frequency coincides with the frequency of the external forcing, the steady state vibration amplitude of the main device becomes

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zero. From a control perspective, the absorber acts like a controller that has an internal model of the disturbance, which therefore cancels the effect of the disturbance.

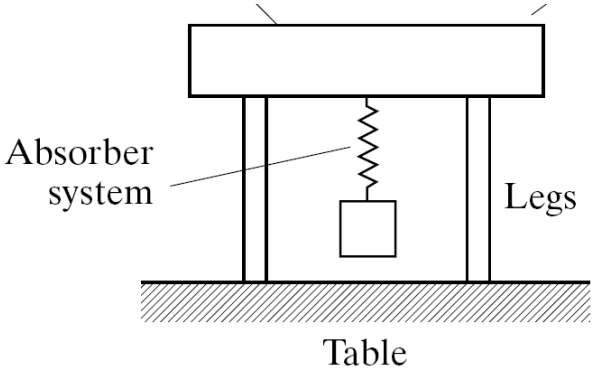


Fig. 1: Tuned vibration absorber

A. Applications:

- Reciprocating machines
- Building excited by an earthquake
- Transmission lines or telephone lines excited by wind blowing
- Vibration absorber in the transmission lines.
- Tuned mass dampers beneath the bridge platform.
- Tuned Mass Damper in the building.

Tuning is a process used to eliminate amplification due to resonance by changing a system or component natural frequency, f n , so that it is no longer coincident with the frequency of a specific force input. Resonance of industrial equipment will amplify vibration response, in theory up to ∞, depending on system damping characteristics.

The Synchronous Amplification Factor (SAF) is a measure of how much 1X vibration is amplified when the system passes through a resonance. Systems with a high effective damping tend to have a low SAF, and systems with low effective damping have a high SAF. The Bode Plot, shown in Figure 3, below, indicates that systems with a high SAF (lightly damped) have a narrow range of resonance with high amplification; systems with low SAF (well damped) have relatively broad range of resonance with low amplification. Note that the range of resonance indicates the amount of tuning (change in the system natural frequency) required to eliminate resonance. Resonant frequencies that are no synchronous exhibit similar behavior.

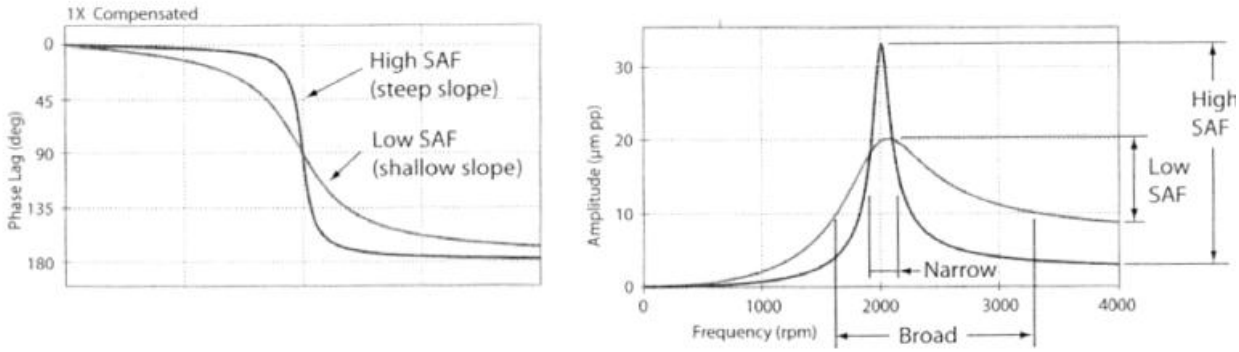


Fig. 2: Bode plot

Before attempting to detune a resonant system, the natural frequency (and damping characteristics) should be determined experimentally, typically by a forced response (bump) test. Once the frequency relationship between force input and system natural frequency, f n, are evaluated, a decision must be made on whether to raise or lower the natural frequency. For a simple (sdof) vibrating system, f n is proportional to the stiffness to mass ratio. Equation 4, below, indicates that adding stiffness will raise the f n, and, conversely, adding mass will lower the f_n.

$$f_n \cong \frac{1}{2\pi}\sqrt{\frac{k}{M}}$$

Where;

f_n = system natural frequency, Hz

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K = stiffness, lb/in [N/m]
M = mass, lb-sec 2 /in [g]

Since increasing mass and/or decreasing stiffness (to lower the f n) may compromise the strength of a machine or its supporting structure, adding stiffness (to raise the f n) is the most common practical application of detuning. When adding stiffeners to a resonant system, the optimal location for end connections is the system antinodes, and installations at nodes will be ineffective. A modal or operating deflection shape analysis is useful for determining the location of antinodes. Care should be taken to ensure that; 1) the added mass of the stiffener does not cancel its stiffening effect (note that pipe has the best stiffness to mass ratio), 2) stiffeners do not introduce new component natural frequencies that are coincident with force input frequencies (ie resonance), and 3) stiffener end connections are as fixed (rigid) as practically possible.

IV. IMPLEMENTATION OF THE PROJECT WORK



Fig. 3: Experimental model –front view

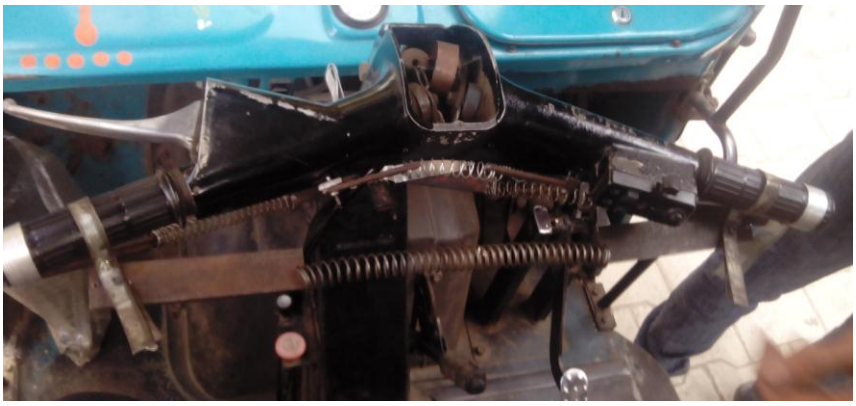


Fig. 4: Experimental model - top view

V. DESIGN PROCEDURE

Natural frequency of handle f_n = 8.75 Hz

Angular frequency of handle ω_n = 55 rad/s

Mass moment of inertia $I = \frac{bh^3}{3} = \frac{(5.5)(30)^3}{3}$

$= 49500\ mm^4$

$\omega_n = \sqrt{\frac{k}{M}}$ Where, $k = \frac{F}{\delta}$

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$$\omega_n = \sqrt{\frac{F}{\delta M}} \quad \textit{Where} \quad \delta = \frac{FL^3}{3EI}$$

So,

$$\omega_n = \sqrt{\frac{3EI}{ML^3}} = \sqrt{3 * 200000 * 49500/0.5 * L^3}$$

L=254 mm

B. Dimensions of spring:

$$\omega_n = \sqrt{\frac{k}{M}}$$

$K_{E=}$ $W_n^2 m$=3025*0.5=1510 N/m where W_n = 55 rad/s

$$\frac{1}{K_E} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3}$$ Where, K_1= K_3=1800 N/mm

Above equation, K_2= 661 N/m

$$K = \frac{Gd}{8*C^3*n}$$ Where, k= spring stiffness

C= spring index
d= diameter of wire
G= modulus of rigidity
n= number of turns

By trial and error method,

Spring diameter D=17 mm
Wire diameter d=1.9 mm
Number of active coils n=38

VI. RESULT ANALYSIS

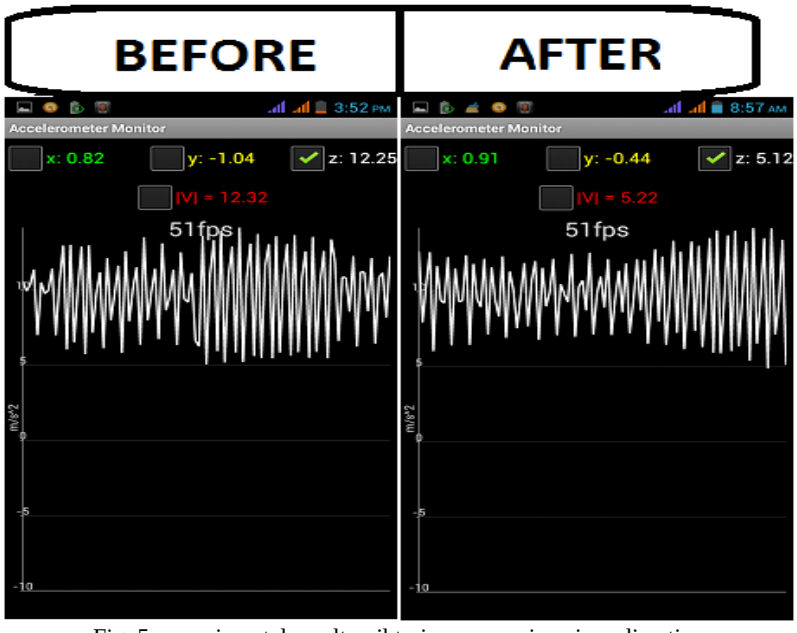


Fig. 5: experimental result , vibtraion comparison in x-direction

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Fig. 6: experimental result , vibtraion comparison in y-direction

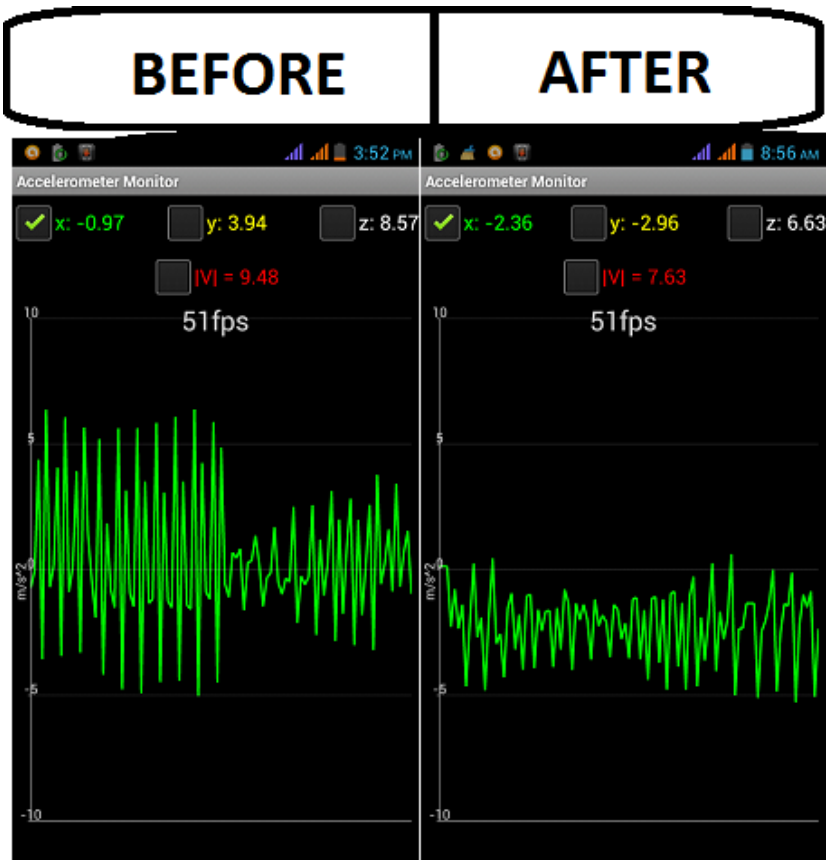


Fig. 7: experimental result , vibtraion comparison in z-direction

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VII. CONCLUSION

As per the experiment, we conclude that there is a significant decrease in vibration of handle in x direction is 20%, y direction is 33% and in z direction is 45%. In idling condition y and z direction is our major focus and we got an appreciable reduction in z direction up to 45% and in y direction up to 33%. Due to the unavailability of FFT analyzer we are not able to measure the exact picks, RMS values but according to stakeholder's opinion, our design is working as per our motive.

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