Design and development of alternative method for vibration issues in Locomotive Wagon wheels

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Abstract. Locomotive transportation is one of the important sectors in the field of transportation. Hence, passengers’ experience plays a vital role in the improvement of the public transportation. The present research article addresses the vibrational issues that occur due to lack of damping in the public transportation. The study focuses on reducing vibrations in locomotive transportation systems by the use of Anti-vibrational pad (AVP). Various results obtained from the experimentations involving modal analysis, random vibration analysis, harmonic response analysis, and fatigue analysis are validated using Ansys workbench. The study also compares the obtained experimental data with the existing approach. The study resulted in a positive outcome, with the possibility of being able to build a practical and more stable damping set-up in public transportation, which in-turn can improve the passengers’ experience.

1. Introduction

Locomotive transportation is one of the important sectors in the field of transportation. In India, around 23 million passengers travel to different places every day, which gives us a chance of improvising the present system and make it more comfortable than before. Hence, the research focusses on improving the passenger experience for the locomotive. Transportation noise has adverse effects on sleep structure, affects the heart rate (HR) during sleep, and maybe linked to cardiovascular disease. The suspension system is the fundamental unit of an automotive which mainly contributes to the passengers’ experience[1]. Like cars, boats, and planes, it can also induce motion sickness to ride a train. Commuter trains have a tendency to rock and stop. Double-decker trains can be perfect for scenery viewing but they also appear to sway. Higher-end bullet trains can travel a little smoother but then no train is known to be totally free from the causes of movement sickness.

The typical locomotive suspension system consists of shock absorber spring between the bogie and frame. The common types of suspension system used are coil springs, leaf springs, and rubber airbags in modern bullet trains[2]. Axle box suspensions absorb shocks between both the bogie frame and the axle bearings. The suspension of the axle box typically consists of a spring between the bogie frame and the axle bearings to allow up-and-down movement, and sliders to stop lateral motion[3]. The modern design uses solid rubber springs[4].
AVP is used when heavy types of machinery like turbines, CNCs, etc are been bolted to the ground so that the vibration is damped before going into the ground. This made us think, AVP as a potential material to eradicate our problem as stated above. The material used for AVP is Polyethylene, which has good flexibility and damping properties when moulded as required.

1.1 Train Chassis
A Forged bed made up of steel and high stiffness spring, bar construction, with simple horn guides installed, allowing vertical movements between the axle boxes. A cast-steel equalizer beam or bar on the axle boxes rested on them. The bar is fitted with two steel coil springs, and the bogie frame balanced on the coils. The result was to allow the bar to act as a compensating lever between the two axles and to use both springs to relieve shocks from either axle. This all makes up a Train chassis as seen in figure 1.

![Figure 1. Chassis of Bogie](image)

2. Problem statement
Use of Anti-vibrational pad (AVP) to reduce vibrational issues in locomotive public transportation and hence improve passengers’ travel experience

3. Methodology
A step by step procedure was carried out from design to simulation using tools like SOLIDWORKS 2016 and ANSYS 18.2 version.

3.1 Design of chassis
The initial step involved in the material selection are by collecting all the necessary material properties which are stated in table 1.

| Properties          | Anti-Vibration Pad |
|---------------------|--------------------|
| Density             | 920 kg/m³          |
| Youngs modulus      | 82000 Pa           |
| Poisson ratio       | 0.49               |
3.1.1 3D modelling

Based on the dimensions provided by the Indian Railways, the model was built in SOLIDWORKS 2016 using features like extrude, extrude cut and revolve. Also, the model was cleaned up by removing the parts and features which were not needed for the analysis to minimize the computational time. The final model for the analysis is given in the figure 2. This model consists a set of 4 small springs and set of 2 heavy duty springs rated for locomotive application.

![SOLIDWORKS model of Train chassis and Springs](image)

**Figure 2.** SOLIDWORKS model of Train chassis and Springs

3.2 Determining load and boundary conditions

- Compression/radial
- Tangential force
- Bonded contact
- Cycle time = 1 sec
- Force applied by the bogie on chassis = 120000N
- Number of modes = 50
- Number of intervals for Harmonic response = 50

4. Simulation study

The succeeding approach was followed to solve the problem and the same was checked for both the setups during the analysis using the Pure penalty approach, the model obtained is shown in figure 3. Figure 3(a) is the geometry obtained for the Initial setup, figure 3(b) is the geometry obtained for the proposed setup which has cylindrical shaped Anti-vibration Pad (AVP) as shown in the figure. The analysis software here used is ANSYS 18.2 version.

![Geometry obtained in ANSYS 18.2 workbench](image)

**Figure 3.** Geometry obtained in ANSYS 18.2 workbench (a) Without AVP, (b)With AVP
4.1 Type of analysis
As we are not considering the non-linearity behavior, we are going forward with Modal[5], Random Vibration and Harmonic response as our major form of analysis as shown in figure 4. Also Fatigue analysis is been carried for the proposed model.

![Figure 4. Analysis setup in ANSYS](image)

4.2 Contact generation
The springs are in no separation connection with the rest of the chassis(bonded). Also, the axle of the wheel is having a no separation connection(bonded) as seen in figure 5.

![Figure 5. Connections generated in ANSYS workbench](image)

4.3 Mesh generation
A Tetragonal mesh was obtained for the model as shown below in figure 6. Based on the element size, the following data were obtained for the 3 different cases of meshing as seen in table 2.

![Figure 6. Mesh generation in ANSYS](image)
It is evident from the above statistics that a higher number of nodes and elements is obtained at finer element sizes (70 mm). The results obtained are more reliable when the number of nodes and elements is significantly higher. On the other hand, when the number of nodes is more, it takes additional computational power.

To improve and check the convergence of the results, two methods were used:

a) H-type: This technique involves altering the global size of the element set during the meshing process either by raising or lowering the size of the element without changing the type of mesh being used in simulations. Which results may not always converge\[6,7].

b) P-type: This approach focuses on the form of mesh used in the analysis, keeping the size of the element constant, which means the order of the elements is changed. Higher-order means more reliable results, but it needs more computational time, noticeably\[8,9].

Given the increase in mesh size deformation has been observed and there has been no significant change in deformation observed in 5 iterations over a range of 240 mm to 70 mm element size, leading to a convergence of the results.

The following types have been used to infer about convergence and precision, hence the convergence method of type H is used throughout the analysis.

4.4 Pre-processing for simulation

a) Modal analysis

As illustrated in the figure 7 modal analysis is an important method of analysis of the vibration characteristics. We can determine the characteristics of chassis vibration and get a modal parameter.
That way, it is an important basis for under the dynamic load structure design[10]. We created in the SOLIDWORKS, a three-dimensional model for chassis assembly. This provides a theoretical basis for locomotive chassis structure optimization. The figure 7 shows the boundary conditions for modal analysis.

![Figure 7. Fixed support of Modal Analysis for Initial setup (Without AVP)](image)

b) Random vibration analysis
This analysis can be used to calculate the structure ’s response to random loads such as turbulences or vibrations. Results are presented as statistical values because the exact load history is not known. Anyway, to perform random vibration analysis in ANSYS, the following step must be followed by providing the excitation curve as PSD (Power Spectral Density). The faces fixed for the analysis is shown in figure 8.

![Figure 8. Fixed support of Random Vibration for Initial setup (Without AVP)](image)
c) Harmonic and Fatigue analysis
Modal analysis was performed in the ANSYS and the natural vibration frequency and effect was obtained in normal frequency response[11]. The calculated forces like normal force were applied by the bogie to the chassis with a force of 120000N and the analysis is carried out in the no-slip condition. Same boundary conditions are used for Fatigue analysis as shown in figure 9.

![Figure 9](image)

**Figure 9.** Loading and Boundary conditions for Harmonic and Fatigue analysis

5. Results and discussion
The analysis was carried out in ANSYS 18.2 workbench, and the following results have been obtained, according to the simulation study mentioned.

5.1 Modal analysis
For the existing model and the proposed model with finer element size, the individual analysis was performed to obtain the most efficient results in the stipulated amount of time. The number of modes found was 50. Figure 10 and figure 11 shows the deformation for mode 8 and mode 50. And the modal frequencies are mentioned in table 3 and table 4 respectively.

Case 1: Without Anti-Vibration Pads (AVP)

![Figure 10](image)

**Figure 10.** (a) Deformation in Mode 8 (Without AVP), (b) Deformation in Mode 50 (Without AVP)
Case 2: With Anti-Vibration Pads (AVP)

Figure 11. (a) Deformation in Mode 8 (With AVP), (b) Deformation in Mode 50 (With AVP)

Table 3. Modal frequency for Initial setup

| Sr. No | Mode | Frequency (Hz) |
|--------|------|----------------|
| 1      | 1    | 6.5819         |
| 2      | 4    | 8.9928         |
| 3      | 8    | 48.738         |
| 4      | 16   | 86.986         |
| 5      | 32   | 148.05         |
| 6      | 50   | 209.55         |

Table 4. Modal frequency for Proposed setup

| Sr. No | Mode | Frequency (Hz) |
|--------|------|----------------|
| 1      | 1    | 7.4479         |
| 2      | 4    | 13.209         |
| 3      | 8    | 49.749         |
| 4      | 16   | 63.239         |
| 5      | 32   | 103.06         |
| 6      | 50   | 150.40         |

5.2 Random Vibration analysis
The PSD can be generated from design specifications or taken from a time-dependent random signal. The PSD unit is squared amplitude per unit frequency. Here we simulate our own frequency to find the mode shapes required for random vibration analysis (because it uses the technique of modal superposition). This will be in the form of field variables RMS values (RMS stress, RMS displacement etc.)

Case 1: Without Anti-Vibration Pads (AVP)  
Figure 12 and figure 13, illustrates directional deformation and stress obtained for Initial Setup with 6 DOF’s

Figure 12. (a)Directional deformation in X-axis, (b)Directional defamation in Y-axis
Case 2: With Anti-Vibration Pads (AVP)

Figure 14 and 15, illustrates directional deformation and stress obtained for Proposed Setup with 6 DOF’s, table 5 and 6 states the physical properties of both the setups obtained during the analysis.

Table 5. Physical quantities of random vibration for Initial Setup (Without AVP)

| mode | Frequency | Displacement (mm) | Deformation in X direction (mm^2/Hz) | Deformation in X direction (mm) | Deformation in Y direction (mm) | Deformation in Z direction (mm) |
|------|-----------|-------------------|-------------------------------------|---------------------------------|-------------------------------|-------------------------------|
| 1    | 4.9008    | 0.79263           | 0.128195869                         | 2.7273                          | 6.2471                        | 2.7579                        |
| 2    | 6.3705    | 0.8956            | 0.125908384                         |                                 |                               |                               |
| 3    | 6.7307    | 0.57649           | 0.049376843                         |                                 |                               |                               |
| 4    | 7.1608    | 0.77195           | 0.083217909                         |                                 |                               |                               |
| 5    | 9.0911    | 1.0928            | 0.131360544                         |                                 |                               |                               |
| 6    | 10.101    | 0.9803            | 0.095137916                         |                                 |                               |                               |
Table 6. Physical quantities of random vibration for Proposed Setup (With AVP)

| mode | Frequency | Displacement (mm) | mm²/Hz | Deformation in X direction (mm) | Deformation in Y direction (mm) | Deformation in Z direction (mm) |
|------|-----------|-------------------|--------|-------------------------------|-------------------------------|-------------------------------|
| 1    | 7.4478    | 0.87544           | 0.102902225 | 0.71821                       | 0.3381                         | 1.528                         |
| 2    | 9.2513    | 1.0614            | 0.121774233 |                              | 0.3381                         | 1.528                         |
| 3    | 10.321    | 0.855             | 0.070828893 |                              | 0.3381                         | 1.528                         |
| 4    | 13.209    | 1.3799            | 0.144153532 |                              | 0.3381                         | 1.528                         |
| 5    | 13.483    | 1.3813            | 0.14510768  |                              | 0.3381                         | 1.528                         |
| 6    | 28.206    | 1.1485            | 0.046764952 |                              | 0.3381                         | 1.528                         |

Table 7. Comparison of results obtained for both the setups in Random Vibration[12]

| Sr No | Type of Setup                  | Deformation in Y direction (mm) | Deformation in X direction (mm) | Deformation in Z direction (mm) |
|-------|--------------------------------|---------------------------------|---------------------------------|--------------------------------|
| 1     | Initial Setup (Without AVP)    | 6.2471                          | 2.7273                          | 2.7579                         |
| 2     | Proposed Setup (With AVP)      | 0.3381                          | 0.71821                         | 1.528                          |

The changes seen in the directional deformation are noticeable and is decreased by 94.59%, 73.66% and 44.59% in X, Y and Z axis respectively as mentioned in the above table 7.

5.3 Harmonic Response

We did the analysis of harmonic simulation, based on modal analysis to obtain the stress, deformation and frequency response curves with respect to the phase angle between them.

Case 1: Without Anti-Vibration Pads (AVP)

Figure 16 and figure 17 illustrates equivalent stress, total deformation and frequency response obtained for Initial Setup with 6 DOF’s

![Figure 16](image1.png)  
![Figure 17](image2.png)

**Figure 16.** (a) Equivalent stress, (b) Total deformation
Figure 17. Frequency response obtained in Harmonic Response (Without AVP)

Case 2: With Anti-Vibration Pads (AVP)

Figure 18 and figure 19 illustrate equivalent stress, total deformation, and frequency response obtained for Proposed Setup with 6 DOF’s.

Figure 18. (a) Equivalent stress, (b) Total deformation

Figure 19. Frequency response obtained in Harmonic Response (With AVP)
Table 8. Comparison of the results obtained from Harmonic Response analysis

| Type        | Load (N) | Number of intervals | Stress (MPa) | Deformation (mm) | Max Amplitude (mm) | Frequency for max Amplitude (Hz) |
|-------------|----------|---------------------|--------------|------------------|--------------------|----------------------------------|
| Without AVP | 120000   | 50                  | 128.51       | 1.3228           | 194.53             | 3                                |
| With AVP    | 120000   | 50                  | 21.119       | 0.9636           | 170.76             | 4                                |

Percent decrease in stress observed in the system is 83.56%, in deformation is 27.15% and in the amplitude is 12.22%. These values are obtained for 50 intervals of harmonic response. Detailed values are displayed in table 8.

5.4 Fatigue analysis

Analyzing fatigue has helped us, designers, to estimate the fatigue life of the entire system. This analysis offers data on fatigue life, crack distribution, damage, and strength that you can use to make informed decisions to ensure product quality and maximize fatigue life preventing premature product failure in the field[13,14]. The life of the chassis is obtained shown in figure 20.

![Figure 20. Life obtained in fatigue analysis](image)

6. Result and discussion

a) Modal analysis

After running this analysis for 50 modes we observe that the frequency change after mode number 6 noticeably in both the cases and is reduced in our proposed setup which is desired.

b) Random vibration analysis

Directional deformation were obtained for all the dimensions using the PSD method of analysis. Also, a decrease in the deformation is seen in the proposed setup which is a favourable result for us.

c) Harmonic Response analysis

The maximum amp. attend by the setup which does not have AVP is 194.53mm @3Hz, whereas the setup with AVP has max amp. 170.76mm @4Hz. This tells us that the boogie without AVP has its resonating frequency as 3Hz, that means the train when its natural frequency is 3Hz will start resonating similarly for the AVP setup. From this, we can conclude that the AVP setup dampens the deformation noticeably and from the graph, we can say that it is STABLE setup.

d) Fatigue analysis

As seen from the analysis, the fatigue life for the proposed setup is $1 \times 10^6$ which is almost near to the infinite life, which implies that it is safe for us to use this setup for a practical purpose.
7. Conclusion
The analysis was carried out on the chassis of the train with and without the AVP set-up to check for modal deformation, resonant frequency, stresses, and harmonic responses that occur. The modal analysis showed an increase in frequency after mode number 6 for both the set-ups. Directional deformation by Random vibration analysis showed a reduction in deformation in the proposed set-up. Results of the harmonic response analysis method showed significant damping of deformation for the AVP set-up with a noticeable difference in the maximum amplification attend by the system to be 170.76 mm at 4 Hz as compared to the existing set-up being 194.53 mm at 3 Hz. The proposed set-up can be used in a real-time scenario as the fatigue life theoretically resulted in $1 \times 10^6$, nearing to an infinite lifetime. From the task, along with the results, it is possible to add AVP’s to the present setup at a very marginal cost.

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