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Vibration Analysis and Control of Locomotive System

Y. Shireesha
B. Venkata Suresh
Department of Mechanical Engineering
GMRIT, Rajam, India
venkatasuresh.bade@gmail.com
shirisha1248@gmail.com

B. Sateesh
Department of Mechanical Engineering, Vignan Institute of Information Technology, Andhra Pradesh, India

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Vibration is an undesirable phenomenon of ground vehicles like locomotives and vibration control of vehicle suspension system is an active subject of research. The main aim of the present work is to modeling and analysis of locomotive system. The simplified equations for dynamical locomotive are firstly established. Then the dynamical nature of the locomotive without control is investigated, and also active control suspension and passive control suspension are compare and discussed. The obtained simulation shows that suspension of the locomotive with feedback control could decrease the locomotive vibration. According to the above control strategy along with angular acceleration it also reduces the possibility of vibration of the locomotive body, to improves the stability of vehicle operation.

Keywords: locomotive, suspension, feedback and control.

1. Introduction

The payload capacity of a locomotive has not its own, and the main aim is to move the train along the tracks. some trains itself have self-propelled payload-carrying vehicles. Normally these are not considered as locomotives, and referred as railcars, motor coaches or multiple units. vertical accelerations are generated by the variations in road surface, these can be isolated in vehicle body by Vehicle suspension system. It provides passengers inside the vehicle can feel ride is more comfortable. Paolucci et al. [1] explained the ground vibrations produced by a passenger train at Ledsgaard test site, Sweden and simulated numerically through a spectral element discretization of the soil. Triepaischajonsaka et al. [2] introduced a hybrid modeling approach to identify the ground response at different locations. Forrest et al. [3] used a special method to minimize the vibrations transmitted from underground railways and calculated Power-spectral densities and RMS levels of soil vibration due to random roughness-displacement excitation between the rail beam and the masses. Lombaert et al. [4] identified the vibrations induced by railway and a comparison is generally made between dynamic excitation and quasi-static. Jeong -Ryol Shin et al. [5] introduced a size-adjusted vehicle(s)in KTX train arrangement for reducing the vibration. The placed vehicle(s) with adjusted size may release out-of-phase loading and reduces the repercussion phenomenon. Chudzikiewicz [6] explained a general conception and realization of a computer package that is MATLAB to identify railway vehicles dynamic behavior. Sezer et al. [7] done vibration analysis and dynamic modeling using matrix laboratory software. By utilize the generated model, the vibrations produced in car is due to the lateral and two vertical sinusoidal track irregularities and vibrations are reduced by placing fuzzy logic controllers in between the car and bogies. The fuzzy algorithm is used to reduce the vibrations in car. Wu [8] explains the effects of rail vibration absorber on short pitch rail corrugation growth by combining the rolling contact mechanics, wheel–track-absorber dynamics and wear. Croft et al. [9] observed that rail damping is possible mechanism to reduce the noise and reduce the rate of roughness development on the surface of rails. Correia dos Santos et al. [10] conducted experiments which include three fundamental and complementary components that is the characterization of the track, ground and vibrations measurement are generated by railway traffic to know the track and ground vibrations. Stein et al. [11] conducts the simulation study of a vertical seat suspension system by using variable damper.

The present work shows the performance of locomotive engine and car, by analyzed and simulated with passive suspension system. Here we considered the suspension system has a two DOF train model. First establish the simplify dynamic equations for locomotive. The model is analyzed using MAT LAB coding and finally feedback control is implemented to reduce the vibration.

2. Mathematical modeling of locomotive system

The quarter car suspension system governing equations are represented. For the car model the free body diagram drawn. The system can represent by below Free Body Diagrams.

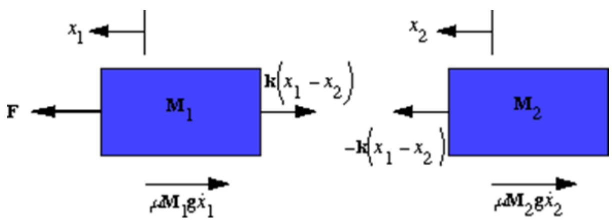

Figure 1 Free body diagram of the locomotive system

where:

M_1 – mass of the engine,
M_2 – mass of the car,
F – force applied,
K – sSpring constant,
μ – coefficient of friction,
g – acceleration due to gravity,
X_1 – displacement of engine,
X_2 – displacement of car.

2.1. *Equation of motion*

Development of propagation models for noise and vibration is an important task in designing solutions due to uncontrolled of noise and vibration. There are two types of approaches are mainly used for modeling and analysis in linear systems, state space formulation and frequency response function (FRF). Initially governing equations of motion for the above system is obtained by drawing free body diagrams and using Newton's Second Law to arrive at the differential equations. Also assumptions consider for different relative positions for the displacements are the static equilibrium point, so it can safely drop the weights out of the equations, leaving the equations in the following form.

From Newton's law, we elaborate that the total of all forces acting on a mass equals the mass times its acceleration. In this work, the forces on M1 are spring, friction and it is applied by the engine. The forces on M2 are the spring and the friction. In the vertical direction, due to the normal force applied by the ground, gravitational force is canceled, so that there will be no acceleration in that direction.

In the horizontal direction the motion equations are followed as:

$$\begin{aligned} M_1\ddot{x}_1 &= F - K\left(x_1 - x_2\right) - \mu M_1 g\dot{x}_1 \\ M_2\ddot{x}_2 &= K\left(x_1 - x_2\right) - \mu M_2 g\dot{x}_2 \end{aligned} \tag{1}$$

2.2. *Modeling of state space equations from second order dynamic system*

The below data is a dashpot system with a force input $f(t)$, simple mass, position $x(t)$ and spring. The following equation explains this system is:

$$m\ddot{x}\left(t\right) + c\dot{x} + K\left(x\right) = f(t) \tag{2}$$

The second order dynamic equation of the suspension system is converted into first order equations in the following form. From Eq. (1) with proper assumptions of the state variables the space equation are explained. This arrangement of framework conditions can now be controlled into state-variable shape. Knowing state-factors are X1 and X2 and the input is F, state-variable conditions will resemble as follows:

$$\begin{aligned} \dot{x}_1 &= v_1 \\ \dot{v}_1 &= \frac{-K}{M_1} X_1 - \mu g V_1 + \frac{K}{M_1} X_2 + \frac{F}{M_1} \\ \dot{x}_2 &= v_2 \\ \dot{v}_2 &= \frac{K}{M_2} - \frac{K}{M_2} X_2 - \mu g V_2 \end{aligned} \tag{3}$$

Equation (1) can be arranged in state space form as:

$$\begin{aligned} \dot{X} &= AX + BU \\ Y &= CX + DU \end{aligned} \tag{4}$$

where: X, Y, A, B, C, D – various matrix orders.
A – space matrix (state),
B – input matrix,
C – output matrix, D – direct transmission matrix,
U – input of the system.
Four matrices A, B, C, and D describe the system behavior, and these will be used to solve the problem.

$$\begin{aligned} \begin{bmatrix} \dot{X}_1 \\ \dot{V}_1 \\ \dot{X}_2 \\ \dot{V}_2 \end{bmatrix} &= \begin{bmatrix} 0 & 1 & 0 & 0 \\ -k/M_1 & -\mu\, g & k/M_1 & 0 \\ 0 & 0 & 0 & 1 \\ k/M_2 & 0 & -k/M_2 & -\mu\, g \end{bmatrix} + \begin{bmatrix} X_1 \\ V_1 \\ X_2 \\ V_2 \end{bmatrix} + \begin{bmatrix} 0 \\ 1/M_1 \\ 0 \\ 0 \end{bmatrix} [F] \\ y &= \begin{bmatrix} 0 & 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} X_1 \\ V_1 \\ X_2 \\ V_2 \end{bmatrix} + [0]\,[F] \end{aligned} \tag{5}$$

The state-space form was controlled from the state-variable and the output equations is described above.

2.3. *Design of control system by the feedback control technique*

The feedback controller design needs system or plant characteristics as input, either in the form of transfer function or state space matrices. However, modern control concepts employ the state-space approach. The main aim of feedback control is to discover a feedback gain which forms stable closed-loop system, so that in the left half of the complex plane all the closed-loop Eigen values are placed. These values control the characteristics of the system response. The pole locations related to closed-loop have a positive impact on time response characteristics like transient oscillations, rise time and settling time. According to the second method of Liapunov stability analysis, the open-loop stability without external input is determined by the Eigen values of the eequation:

$$\{\dot{x}\} = [A]\,\{x\} \tag{6}$$

If all the Eigen values of the matrix $[A]$ have negative, at that point the framework is asymptotically steady. The Eigen estimations of the framework $[A]$ are known as the poles of the first framework. At any point, if any of these poles lie on the right half of the s plane, then the framework is unsteady. The dependability of a direct shut circle framework can be resolved from the area of the closed loop poles in the intricate s plane, in which s are the poles of the framework.
For a second-order control system, the poles can be written as

$$s = -\zeta\omega_n \pm i\omega_d \tag{7}$$

where:

$$\omega_{\mathrm{d}} = \omega_n\sqrt{1-\zeta^2} \qquad i = \sqrt{-1}$$

ω_d – damping frequency,
ζ – damping ratio,
ω_n – undamped natural frequency (rad/s).
Consider full state feedback, the control input can be expressed as:

$$\{u\} =- [G]\,\{x\} \tag{8}$$

were:
$[G]$ – feedback gain matrix,
$\{x\}$ – state vector.
Add Eq. (8) in Eq. (5) gives:

$$\{\dot{x}\}= ([A] - [B]\,[G])\,\{x\} \tag{9}$$

The Eigen estimations of the matrix grid ($[A]-[B][G]$) are the coveted closed –loop poles (p) of the controlled framework, which would supplant the poles of the first framework. It is to be noticed that for a given framework, the matrix $[G]$ is not one of a kind, but depends on upon the desired closed-loop poles (p), which decide the settling time of the reaction. So as to decide the gain matrix $[G]$ for a given framework, the reaction qualities of the control framework is to be studied for a number of cases through computer simulation.
In the present case, gain matrix $[G]$ is computed using place command in MATLAB i.e., G = PLACE (A, B, p). In this system we have large No. of parameters than the quantity of equations to be fulfilled; the additional DOF can be utilized for other purposes such as improving system robustness.

3. Results and discussion

Dynamic behavior of the vehicle can be predicted by different types of results. For this initially response analyses have been done wsithout and with feedback control based on the above response model which is developed in MATLAB and both the results are compared. The main objective of implementing the feedback is to reduce the undesirable vibration in controlled manner. The response analysis of locomotive system without and with feedback control and finally the results are compared. Table 1 gives the input parameters.

Table 1 Input data for the above system is considered as follows

Input data	
M1 = 50 kg	F= 50 N
M2 = 25 kg	u = 0.002 sec/m
g = 9.8 m/s*2	k = 5 N/sec

3.1. *Response analysis of locomotive system without feedback control*

Fig. 2 shows that time response of the system without feedback control. The level of amplitude is high and amplitude is gradually reduces with respect to time. It can be reduced by introducing feedback control. Fig. 3 shows that corresponding frequency response of the system. Table 2 shows the various results of Eigen, damping and frequency values of uncontrolled system.

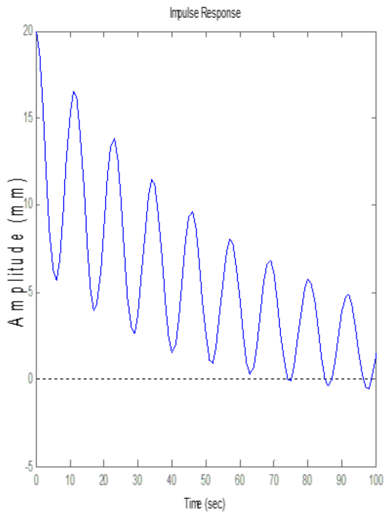

Figure 2 Uncontrolled time response

Table 2 Eigen, damping and frequency values of uncontrolled response

Eigen value	Damping value	Frequency (rad/s)
-1.13e-015	1.00e+000	1.13e-015
-1.96e-002	1.00e+000	1.96e-002
-9.80e-003 + 5.48e-001i	1.79e-002	5.48e-001
-9.80e-003 - 5.48e-001i	1.79e-002	5.48e-001

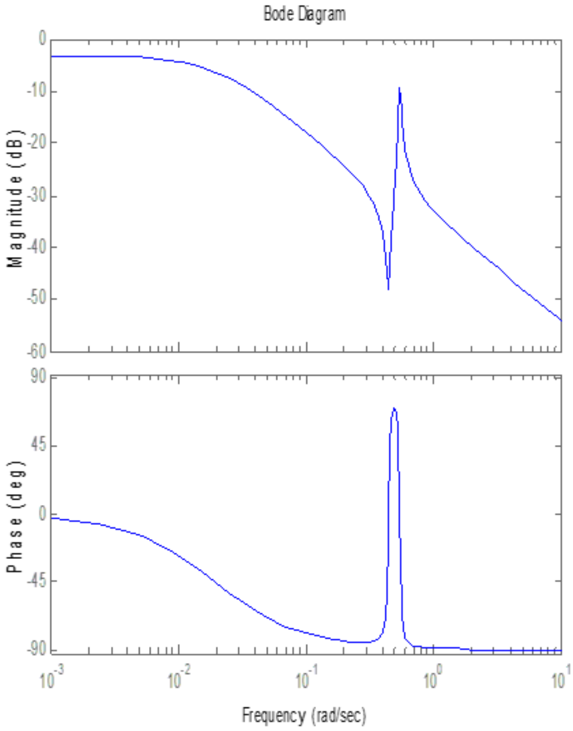

Figure 3 Uncontrolled frequency reaction

3.2. *Response analysis of Locomotive system with feedback control*

So far in the above studies vibration analysis is carried out without control i. e., without giving any feedback to the system. The root locus and bode diagram (frequency response) methods are important for design and analysis of feedback control systems. The analysis is carried out with a variety of graphical tools such as root locus plots, bode plots and feedback control is introduced by calculating the gain from the root locus plot. Finally results are compared without and with control.To implement the feedback control gain is calculated using the root locus plot as shown in Fig. 4. From the bode diagram and root locus figures, it can clearly observe the phase change and magnitude with frequency changes occurring in the level of uncontrolled system with feedback control. The gain value observed from the root locus plot for minimum overshoot is 30.5. This value is used in the control mechanism to reduce the vibration. Similar analysis has been carried out and results are generated for various parameters, but only one set of results of controlled responses is shown in this work.

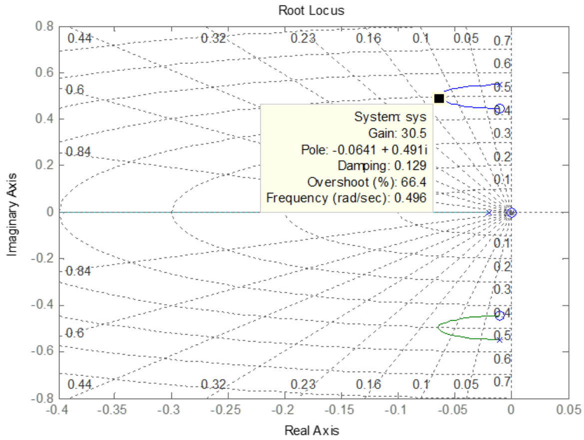

Figure 4 Root focus plot

Table 3 shows the Eigen, damping and frequency values locomotive system after implementation of feedback control. The variation of these values clearly identified with Table 3 which shows the uncontrolled system output. Fig. 5 shows the impulse response of locomotive system with feedback control. The level of amplitude is high with without control (Fig. 2). This amplitude is reduced by implementing feedback control and it almost reaches to zero and stability is obtained within one minute, when compared to uncontrolled response rapid reduction of vibration is noticed. But the stability has been obtained beyond 20 seconds in case of uncontrolled response. So it indicates significance reduction in vibration of the vehicle with the application of feedback control.

Table 3 Eigen, damping and frequency values locomotive system

Eigen value	Damping value	Frequency (rad/s)
-6.34e-017	1.00e+000	6.34e-017
-6.43e-002 + 4.93e-001i	1.29e-001	4.97e-001
-6.43e-002 - 4.93e-001i	1.29e-001	4.97e-001
-5.05e-001	1.00e+000	5.05e-001

Fig. 6 shows the corresponding frequency response. It is observed from the system there is an improvement in frequency of the system when compared to Fig. 3 which represents uncontrolled system frequency response.

The analysis was performed and the results of the controlled and uncontrolled transient responses were compared. Fig. 7 demonstrates the comparison of the results of the controlled and uncontrolled transient reaction plot.

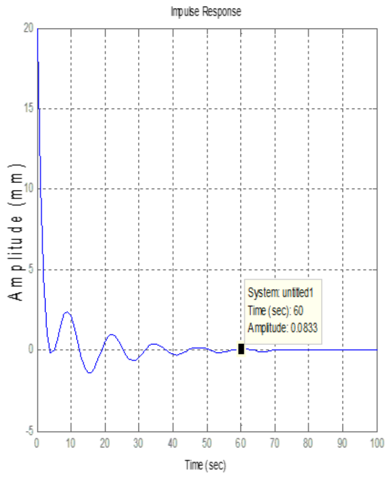

Figure 5 Controlled response

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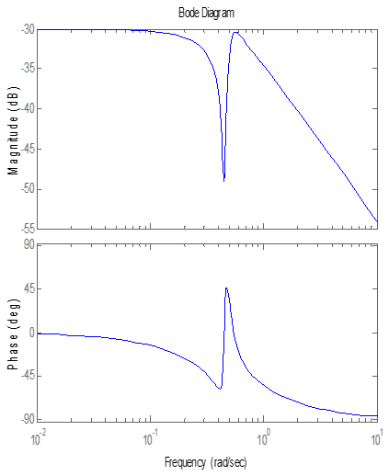

Figure 6 Controlled Frequency response in terms of magnitude and phase angle

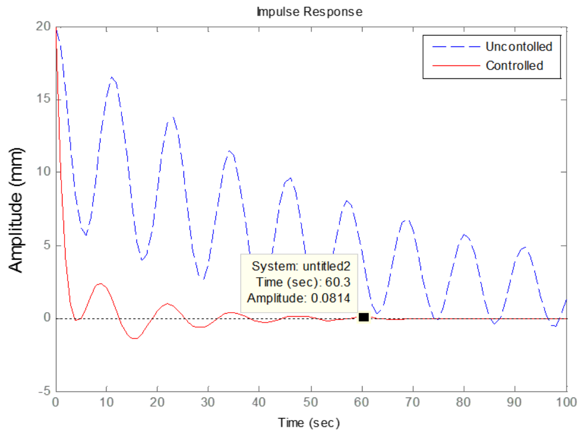

Figure 7 Controlled and uncontrolled responses

4. Conclusions

The passive and feedback control systems have utilized for the analyzing the locomotive system. The above study concluded that two-degree-of-freedom engine car model has developed and analyzed using MATLAB. Initially it is observed that the amplitude of vibration is high by only considering the passive system without any feedback control and it can be reduced by introducing feedback control. From the obtained simulations, it can be expressed that the feedback control can achieve substantial reduction of vibrations of the system than that of passive system.

References

[1] **Paolucci, R., Maffei, A., Scandella, L., Vanini, M., Stupazzini, M.**: Numerical prediction of low-frequency ground vibrations induced by high-speed trains at Ledsgaard, Sweden, *Soil Dynamics and Earthquake Engineering*, 23, 6, 425–433, **2003**.

[2] **Thompson, D.J., Triepaischajonsak, N.**: A hybrid modelling approach for predicting ground vibration from trains,*Journal of Sound and Vibration*, 335, 147–173, **2015**.

[3] **Forrest, J. A., Hunt, H. E. M.**: Ground vibration generated by trains in underground tunnels, *Journal of Sound and Vibration*, 294, 4, 706–736, **2006**.

[4] **Lombaert, G., & Degrande, G.**: Ground-borne vibration due to static and dynamic axle loads of InterCity and high-speed trains, *Journal of Sound and Vibration*, 319, 3, 1036–1066, **2009**.

[5] **Shin, J. R., An, Y. K., Sohn, H., Yun, C. B.**: Vibration reduction of high-speed railway bridges by adding size-adjusted vehicles. *Engineering structures*, 32, 9, 2839-2849, **2010**.

[6] **Chudzikiewicz, A.**: Simulation of rail vehicle dynamics in MATLAB environment, , *Vehicle System Dynamics*, 33, 2, 107–119, **2000**.

[7] **Sezer, S., Atalay, A. E.**: Dynamic modeling and fuzzy logic control of vibrations of a railway vehicle for different track irregularities, , *Simulation Modelling Practice and Theory*, 19, 9, 1873–1894, **2011**.

[8] **Wu, T. X.**: Effects on short pitch rail corrugation growth of a rail vibration absorber/damper, *Wear*, 271, 1, 339–348, **2011**.

[9] **Croft, B. E., Jones, C. J. C., Thompson, D. J.**: Modelling the effect of rail dampers on wheel–rail interaction forces and rail roughness growth rates, *Journal of Sound and Vibration*, 323, 1, 17–32, **2009**.

[10] **dos Santos, N. C., Colaço, A., Costa, P. A., Calçada, R.**: Experimental analysis of track-ground vibrations on a stretch of the Portuguese railway network, *Soil Dynamics and Earthquake Engineering*, 90, 358–380, **2016**.

[11] **Stein, G. J., Múčka, P., Gunston, T. P., Badura, S.**: Modelling and simulation of locomotive driver's seat vertical suspension vibration isolation system, *International Journal of Industrial Ergonomics*, 38, 5, 384–395, **2008**.