Chapter 6
Vibration Analysis of Railway Wagon Suspension System for Improved Ride Quality using MATLAB Simulink

C. Prithvi, R. Srinidhi, and A. Karthik Hebbar

Abstract In this paper, the dynamic response of the Indian railway wagon suspension system is investigated using MATLAB Simulink under step input condition which simulates the irregularities of the track. In the initial part of the work, a linear dynamic model of the Indian railway wagon suspension system was carried out and later mathematical equations are derived from the model constructed. An equivalent Simulink model in accordance with the equations was constructed in MATLAB Simulink. The system is given a step input to simulate the irregularity of the track and the dynamic response such as displacement, velocity and acceleration of the coach, bogie frame and the wheel were found. From the result, it was observed that for the step input, the major vibrations will occur at the coach and the bogie frame and thus causing discomfort to the passengers. In order to reduce these vibration modifications, it is made to the full suspension model with the use of hydraulic actuator controlled by proportional–integral–derivative (PID) controller. It is then simulated using MATLAB Simulink under the same track condition to study the vibration characteristics. The results of the modified suspension system show greater improvements in comparison with the system without PID controller. The improvization with the modified suspension system is suggested to achieve the travel comfort for the passenger.

Keywords ICF · Simulink · Suspension system · Vibration analysis

6.1 Introduction

Analytical modelling is made to study the dynamic response of the system and to make necessary technical changes to the system for better performance of the

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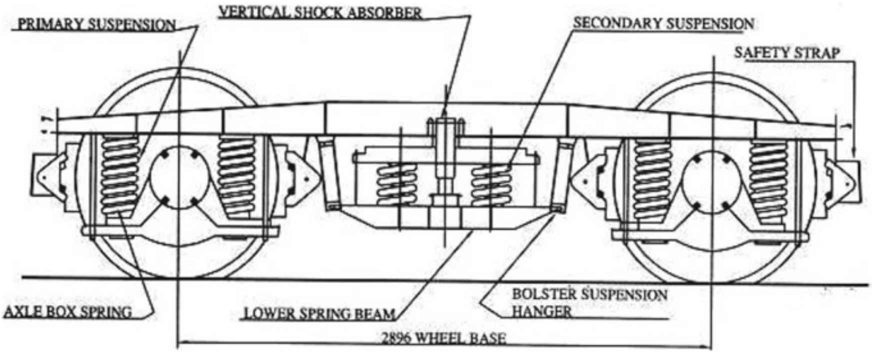


Fig. 6.1 Side view of Indian railway wagon

system. A passenger travelling in Indian railways always experiences some kind of vibration. To minimize these vibrations, the railway wagon has to be redesign or reconstructed with proper technical aspects. In order to bring a technological change to the Indian railway bogie, this project is undertaken. In this project, the vibration characteristics of the Indian railway wagon bogie are studied by analytical modelling the suspension components of the system and the necessary modifications are made to the suspension model to get the better performance of the system. The present work tries to analyse the dynamic performance characteristics of railway suspension model of Indian railway wagon. The Indian railway system bogies are not changed since 1955. The ICF has its own design and structure [1]. To bring a change in its design and structure, extensive modelling and simulation are necessary. The side view diagram and top view diagram of ICF bogie are shown in Figs. 6.1 and 6.2.

The parts of Indian railway wagon system are:

6.1.1 Bogie Frame

The bogie frame railway wagon is manufactured from steel. Bogie frames are sorted into two types, and they are thirteen-ton bogie frame and sixteen-ton bogie frame. All the non-AC coaches are using thirteen-ton bogie frames, and every AC coaches are using sixteen-ton bogie frames. Primary suspension system and secondary suspension system are connected by using brackets with a welding arrangement of bogie frame.

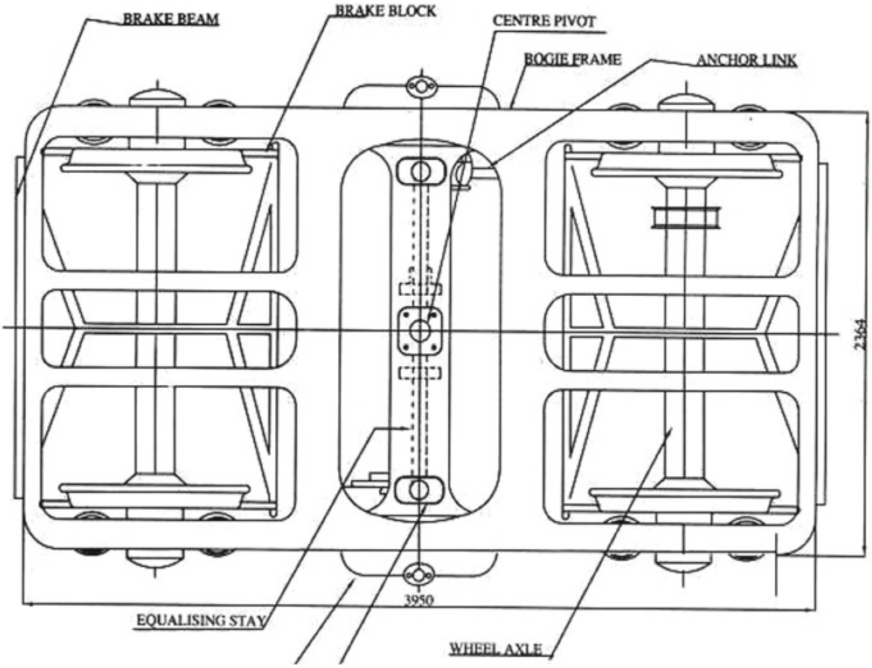


Fig. 6.2 Top view of Indian railway wagon

6.1.2 Bogie Bolster

The central section of the bogie is termed as bogie bolster. It carries the whole weight of a railway coach supported by bogie frame. The centre pivot pin is provided at the mid-section of bogie bolster. Secondary suspension is connected at either bogie face end that connects with bogie bolster. The coach is pivoted to the centre pivot pin that is to the bogie bolster. Centre pivot pin permits the circular motion of the bogie once the rail coach is moving on the arc or curved path. So as to produce the smooth impact for the centre pivot pin, a cylindrical structured member created of metal rubber is placed within the centre pivot.

6.1.3 Secondary Suspension System

Bogie bolster is connected with secondary suspension components using suspension springs and the lower spring beam. The arrangement of bogie bolster cannot have done by the bolting connection or welding connection. Anchor link is used to connect the bogie bolster, and anchor link has hollow rod structure with the housing arrangement. Steel brackets are used to connect the anchor link to the bogie bolster and to the bogie frame. There is hinged arrangement on either side of the anchor link. This allows the movement of bogie bolster during curved line path of the railway [2].

The lower spring beam is used to provide support to the secondary mechanical suspension system. Steel plates are used to manufacture the lower spring beam. The attachment of lower spring beam is on the outer side of the bogie frame using steel made hanger. They are called the Bogie Secondary Suspension Hangers (BSS). The lower spring beam acts as a floating member which is hinged to the bogie frame by the hangers at the top and bottom position. This arrangement allows the vertical or longitudinal movement of the lower spring beam.

The inside part section of the lower spring beam is attached to the bogie bolster by using equalized staying rod. It is manufactured by steel materials like tubes and sheets. It is also fixed on either ends with the lower spring beam and the bogie bolster with by using brackets with the help of welding to the bogie bolster. A pin is used to make the connection between them.

6.1.4 Primary Suspension System

The primary suspension consists of a piston and cylinder arrangement. Wheel axle housing is also provided along with this arrangement. In order to avoid damage, a washer is also used. The sealing ring will act as a piston ring which is placed adjacent to washer. Damper with the oil is used to provide damping effect. The rubber washers are for the arrangement of primary suspension system.

6.1.5 Wheel Set Assembly

The wheel assembly consists of four wheels in two pairs and an axle. The wheels are made by casting and forging process. Roller bearings are used in the Indian railways. The press-fitting arrangement of axle collar with the roller bearing is made. In order to avoid the centre movement of roller, bearing collars are used.

6.2 Full Suspension Model

The model is extended for the full model of Indian railway wagon consisting of four wheels, secondary suspension at both sides and primary suspension at all the wheels [3].

In Fig. 6.3, m_1, m_2 and m_3 represent the masses of coach, bogie frame and the wheel, respectively. The k_1 and c_1 represent the secondary suspension system. The k_2 and c_2 represent the primary suspension system and k_3 represents the stiffness of the wheel.

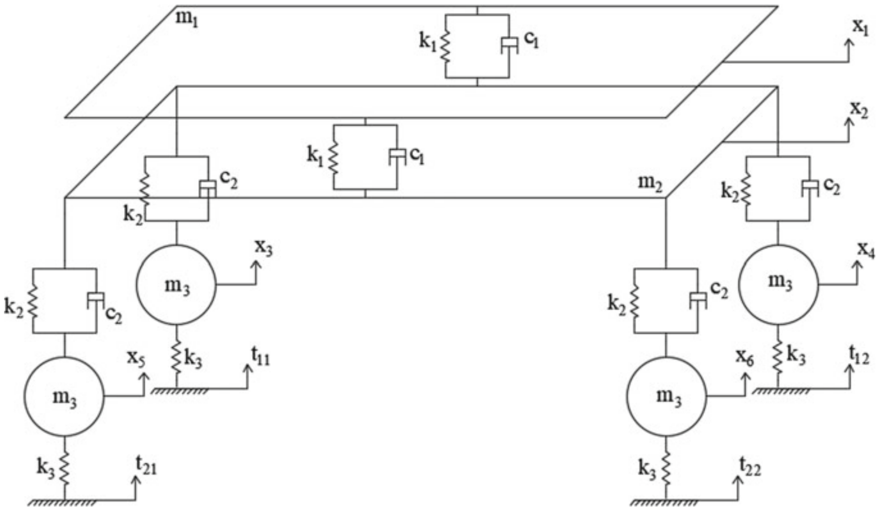


Fig. 6.3 Full linear dynamic suspension model of Indian railway wagon

6.2.1 Equations for the Full Suspension Model

The mathematical equations are developed using Newton's method for the model shown in Fig. 6.3. Newton's second law of motion is applied by considering the free body diagrams of all the masses [4]. The corresponding equations are:

$$m_1\ddot{x}_1 = -2c_1(\dot{x}_1 - \dot{x}_2) - 2k_1(x_1 - x_2) \tag{6.1}$$

$$\begin{aligned} m_2\ddot{x}_2 = {} & -2c_1(\dot{x}_2 - \dot{x}_1) - 2k_1(x_2 - x_1) - c_2(\dot{x}_2 - \dot{x}_3) \\ & - k_2(x_2 - x_3) - c_2(\dot{x}_2 - \dot{x}_4) - k_2(x_2 - x_4) \\ & - c_2(\dot{x}_2 - \dot{x}_5) - k_2(x_2 - x_5) - c_2(\dot{x}_2 - \dot{x}_6) - k_2(x_2 - x_6) \end{aligned} \tag{6.2}$$

$$m_3\ddot{x}_3 = -c_2(\dot{x}_3 - \dot{x}_2) - k_2(x_3 - x_2) - k_3(x_3 - t_{11}) \tag{6.3}$$

$$m_3\ddot{x}_4 = -c_2(\dot{x}_4 - \dot{x}_2) - k_2(x_4 - x_2) - k_3(x_4 - t_{12}) \tag{6.4}$$

$$m_3\ddot{x}_5 = -c_2(\dot{x}_5 - \dot{x}_2) - k_2(x_5 - x_2) - k_3(x_5 - t_{21}) \tag{6.5}$$

$$m_3\ddot{x}_6 = -c_2(\dot{x}_6 - \dot{x}_2) - k_2(x_6 - x_2) - k_3(x_6 - t_{22}) \tag{6.6}$$

6.3 Simulation Using MATLAB Simulink

To get the dynamic response of the system, it needs to be simulated. The simulation of the suspension model of the Indian railway wagon is done by using MATLAB Simulink [5]. The model is simulated for the fixed parameters of the system [6]. The track "t" is simulated for the step input condition [7]. The performance characteristics such as displacement of the coach, bogie frame and the wheel are recorded for the system model.

6.3.1 Simulation of Full Suspension Model

The equivalent Simulink model of the system is created according to Eqs. (6.1)–(6.6) as shown in Fig. 6.4.

The Simulink model consists of six summation blocks representing six equations correspondingly. The displacement, velocity and acceleration can be measured with the scope connected in the circuit. The circuit consists of Integrator and gains which are connected to each other according to the equations.

The model is simulated for the fixed parameters of the system as shown in Table 6.1.

The track input is considered here is step input condition of the track with a time interval of five seconds for each of the wheel. Due to the track disturbance, the maximum displacement is recorded for the coach (m_1) at each of the wheel at regular

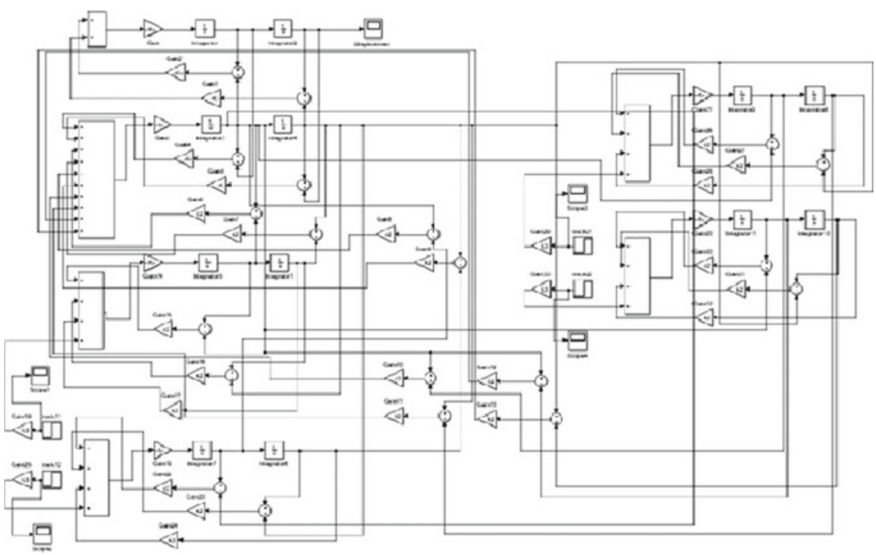


Fig. 6.4 Equivalent Simulink model of full suspension system

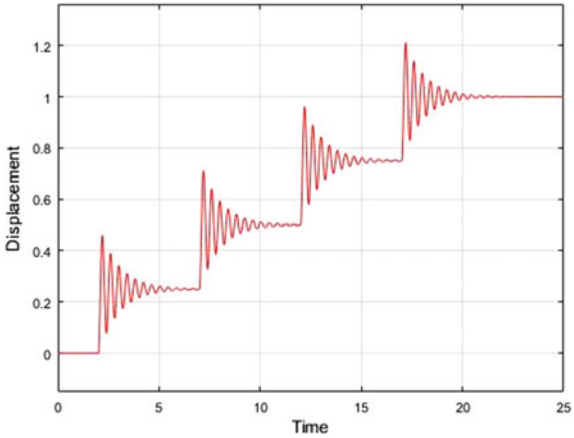


Fig. 6.5 Full model displacements of mass m_1 with respect to time

Table 6.1 Parameters of the system used for full model simulation

Sl. No	Quarter suspension model parameters of Indian railway wagon		
	Parameters	Symbol	Values
1.	Mass of the coach or body	m_1	32,000 (kg)
2.	Mass of the bogie frame	m_2	2615 (kg)
3.	Mass of the wheel	m_3	1500 (kg)
4.	Secondary spring stiffness	k_1	5.8×10^6 (N/m)
5.	Secondary damping coefficient	c_1	60×10^3 (Ns/m)
6.	Primary effective spring stiffness	k_2	7×10^6 (N/m)
7.	Primary effective damping coefficient	c_2	40×10^3 (Ns/m)
8.	Stiffness of the wheel	k_3	35×10^6 (N/m)

time interval of time. The maximum displacement of 0.5 units is measured, and the displacement goes on decreasing with time. The displacement continues again for the next time interval due to the track disturbance at another wheel. Thus, the railway coach undergoes vibration continuously due to the track disturbance (Fig. 6.6).

The displacement results of the bogie frame are comparatively lower than the coach. The maximum displacement of 0.3 units is measured for the bogie frame. The displacement continues again for the next time interval due to the track disturbance at another wheel.

The corresponding velocities of the masses m_1 (coach), m_2 (bogie frame) are recorded and shown in Figs. 6.7 and 6.8, respectively, (Fig. 6.7).

The maximum velocity of 3.8 units is measured for the coach. The velocity goes on decreasing after a step input is passed. The velocity rise is observed again for the next step input at another wheel. Thus, the vibration takes place at each interval

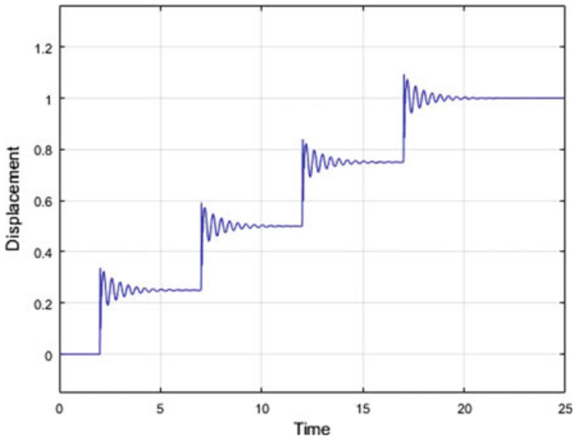


Fig. 6.6 Full model displacements of mass m_2 with respect to time

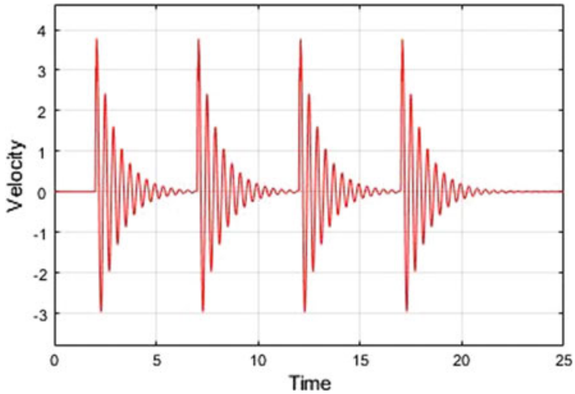


Fig. 6.7 Full model velocity of mass m_1

of step input condition of the track at the different wheels. These effects the ride comfort of the passenger (Fig. 6.8).

The velocity results of the bogie frame are comparatively larger than the coach. The maximum displacement of 22 units is measured for the bogie frame. The velocity continues again for the next time interval due to the track disturbance at another wheel (Fig. 6.9).

The maximum acceleration of 160 units is measured for the coach. The acceleration goes on decreasing after a step input is passed. The velocity rise is observed again for the next step input at another wheel (Fig. 6.10).

The result of full model (coach and bogie frame) shows that the displacement is occurring for every step input condition of the track with the time interval of 5 s. This shows that for the track irregularity presents in a track the system going to vibrate which will affect the passenger ride comfort. The results show that the displacement of the coach (m_1) and bogie (m_2) occurs with a large amplitude for every track disturbance at the four wheels. The maximum velocity is recorded with

Fig. 6.8 Full model velocity of mass m_2

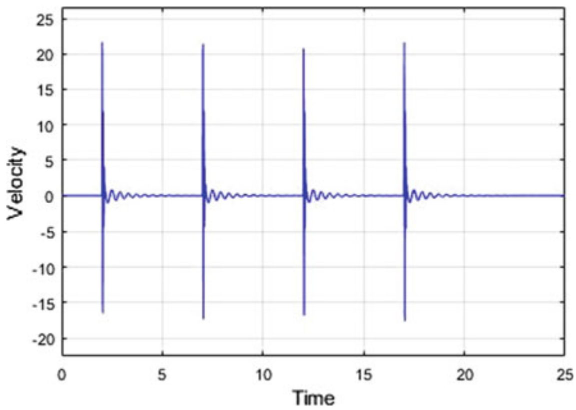


Fig. 6.9 Full model acceleration of mass m_1

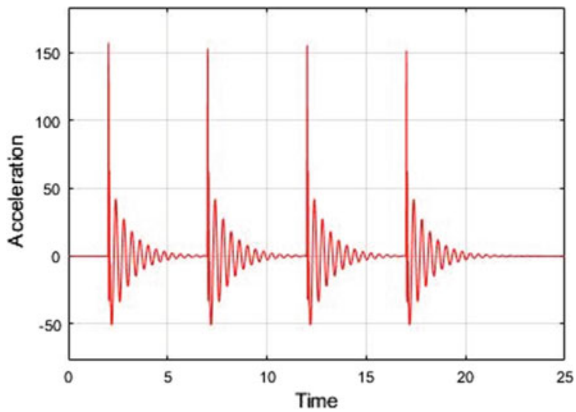
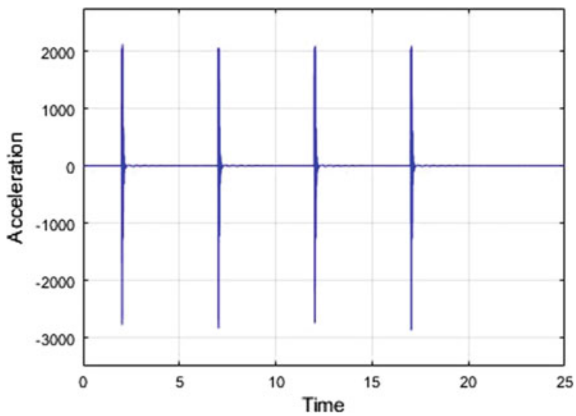


Fig. 6.10 Full model acceleration of mass m_2



a larger settling time for the coach and shorter settling time for the bogie. There is a peak acceleration recorded for the coach and the bogie frame with a larger settling time for the coach and shorter settling time for the bogie. The overall results show that the coach vibrates more when compared to the bogie frame. Hence, it directly affects the person who is travelling in the railway coach.

6.4 Modification

The Vibrations of the bogie has to be minimized for the smooth performance and to achieve the maximum ride comfort for the passenger. Modification to the quarter suspension model of the ICF bogie is made with the application of hydraulic actuator controlled by proportional–integral–derivative (PID) controller in order to minimize the vibration and to achieve the ride comfort for the passenger.

6.4.1 Hydraulic Actuator

A hydraulic actuator consists of a piston and cylinder arrangement that uses hydraulic power for its mechanical movement. The mechanical movement of the piston inside the cylinder can exert a force so that the damping can be achieved. The schematic diagram of hydraulic actuator is shown in Fig. 6.11.

There are two types of hydraulic actuator depending on the movement of piston. They are single acting and double acting cylinder. In single acting, the pressure applied on one side of the piston. In double acting system, the pressure is applied on either side of the piston. The difference in pressure makes the movement from one

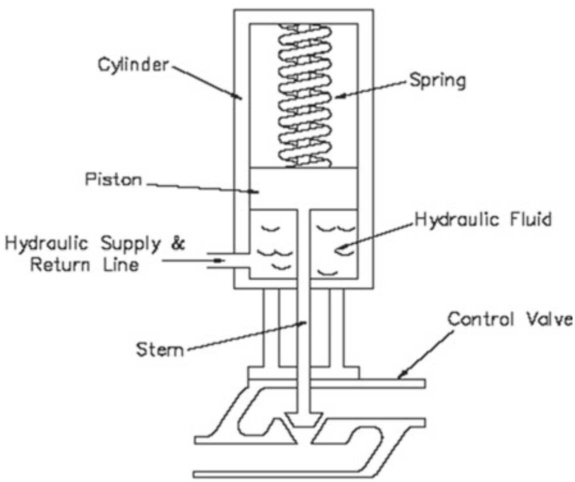


Fig. 6.11 Schematic diagram of hydraulic actuator

side to another. By the application of hydraulic actuator, required damping force can be applied in order to suppress the vibrations [8] of the ICF bogie.

6.4.2 Proportional–Integral–Derivative (PID) Controller

Proportional–integral–derivative (PID) controllers are used in the applications of controlling some of the industrial processes. There are three parameters which act as controllers, and these are combined with each other to produce a desired control signal. A feedback controller sends the output signal to the controller. PID controllers are very flexible and perform the control strategy without having much error.

$$u(t) = K_{\mathrm{p}} e(t) + K_i \int_0^t e(t)\mathrm{d}t + K_{\mathrm{d}} \frac{\mathrm{d}e(t)}{\mathrm{d}t} \tag{6.7}$$

6.4.3 P-Controller

P-controller or proportional produces the output which is proportional to error signal $e(t)$. It compares the output signal with the input signal with the help of feedback controller. The error generated is multiplied proportionally with a proportional constant to get the desired output. The controller output is zero for the error value of zero. There exists a steady-state error in the system but the system is always stable. The proportional constant is denoted by K_{p} and as the K_{p} is increased operation speed is also increased.

6.4.4 I-Controller

Due to steady-state error of the p-controller, I-controller is needed to process the variable and the set point. The steady-state error can be eliminated by using I-controller. It integrates the error produced over a period of time until the error value vanishes to zero. It holds the final control value of the control device at which the error becomes zero. The output of the integral control is decreased when there is a negative error is present. The speed of the control system is decreased, and thus, the stability of the system gets affected. By decreasing the value of $K_{\mathrm{i},}$ the speed of the system can be increased.

6.4.5 D-Controller

The error response cannot be predicted by I-controller. D-controller is used to overcome this problem. The output of the D-controller depends on rate of change of error with respect to time. The system response can be increased by the use of D-controller.

By combining all the three controllers, we can achieve the desired control strategy for the system. The desired signal is called as u(t), and the error signal is called as the $e(t)$. These two signals are fed to the PID controller, by the combination of P-controller, I-controller and D-controller the desired output can be obtained.

6.4.6 Ziegler–Nichols Method of Tuning the PID Controller

In order work efficiently, the tuning of the PID controller is important. The PID controller must be tuned in such a way that dynamic response should be controlled to the desired level. There are different types of tuning methods are used to tune the PID controllers. The one of such method is Ziegler–Nichols method. By the use of this method, we can get the optimum values for P-controller, I-controller and D-controller.

Ziegler–Nichols method is a closed-loop method for tuning the PID controller. It is a method of continuous cyclic damping oscillation method. In this, at first the value of P-controller is kept constant for a particular value by keeping I-controller (K_{i}) and D-controller (K_{d}) values to zero. Proportional gain is increased until the system produces the oscillation at a constant amplitude. Gain at which the system produces constant amplitude of oscillation is called ultimate gain (K_{u}) and the time period at which the constant amplitude of oscillations is achieved is called the ultimate time period (T_{u}).

According to Ziegler–Nichols method for the PID controller after getting the ultimate gain (K_{u}) and the ultimate time period (T_{u}), the optimum values of K_{p}, K_{i} and K_{d} are given by,

$$K_{\mathrm{p}} = 0.6K_{\mathrm{u}} \tag{6.8}$$

$$K_{\mathrm{i}} = 1.2K_{\mathrm{u}}/T_u \tag{6.9}$$

$$K_{\mathrm{d}} = 0.6K_u T_u/8 \tag{6.10}$$

The optimum values of K_{p}, K_{i} and K_{d} are obtained using the above-mentioned formulas.

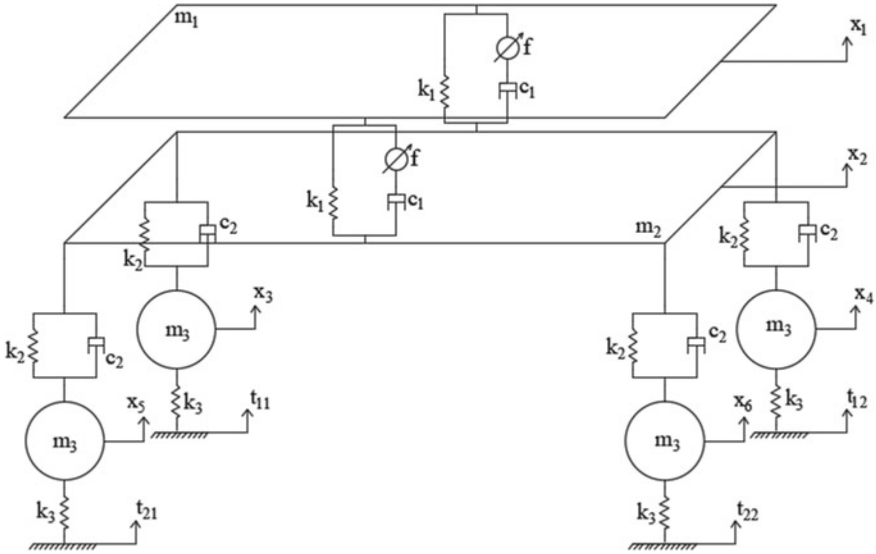


Fig. 6.12 Modified full suspension model of the Indian railway ICF bogie

6.4.7 Modification of Full Suspension Model

The modification is made to the extended model by using the application of hydraulic actuator and PID controller. The hydraulic actuator is connected in series with the damper in the secondary suspension arrangement. Hydraulic actuator is controlled by PID controller which receives the displacement signal from the accelerometer kept over mass m_1 (Fig. 6.12).

6.4.8 Mathematical Equations for the Modified Full Suspension Model

The mathematical equations are developed using Newton's method for the model shown in Fig. 6.13. Newton's second law of motion is applied by considering the free body diagrams of all the masses [4]. The corresponding equations are:

$$m_1\ddot{x}_1 = -2c_1(\dot{x}_1 - f) - 2k_1(x_1 - x_2) \tag{6.11}$$

$$\begin{aligned} m_2\ddot{x}_2 = &-2c_1(\dot{x}_2 - \mathrm{f}) - 2k_1(x_2 - x_1) - c_2(\dot{x}_2 - \dot{x}_3) \\ &- k_2(x_2 - x_3) - c_2(\dot{x}_2 - \dot{x}_4) - k_2(x_2 - x_4) \\ &- c_2(\dot{x}_2 - \dot{x}_5) - k_2(x_2 - x_5) - c_2(\dot{x}_2 - \dot{x}_6) \end{aligned}$$

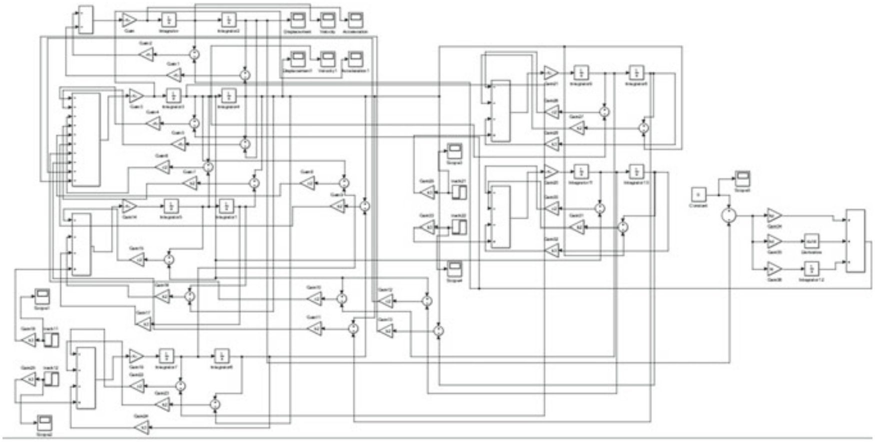


Fig. 6.13 Simulink model of the modified full suspension system

$$- k_2(x_2 - x_6) \tag{6.12}$$

$$m_3\ddot{x}_3 = -c_2(\dot{x}_3 - \dot{x}_2) - k_2(x_3 - x_2) - k_3(x_3 - t_{11}) \tag{6.13}$$

$$m_3\ddot{x}_4 = -c_2(\dot{x}_4 - \dot{x}_2) - k_2(x_4 - x_2) - k_3(x_4 - t_{12}) \tag{6.14}$$

$$m_3\ddot{x}_5 = -c_2(\dot{x}_5 - \dot{x}_2) - k_2(x_5 - x_2) - k_3(x_5 - t_{21}) \tag{6.15}$$

$$m_3\ddot{x}_6 = -c_2(\dot{x}_6 - \dot{x}_2) - k_2(x_6 - x_2) - k_3(x_6 - t_{22}) \tag{6.16}$$

6.4.9 Simulation of Modified Full Suspension Model Using MATLAB Simulink

The Simulink model of the modified full suspension system is made according to the equations. The model is again simulated using MATLAB Simulink for the step input condition of the track t_{11}, t_{12}, t_{21} and t_{22}, taken with a time interval of 5 s at the four wheels, respectively. The Simulink model of the modified suspension system is shown in Fig. 6.14.

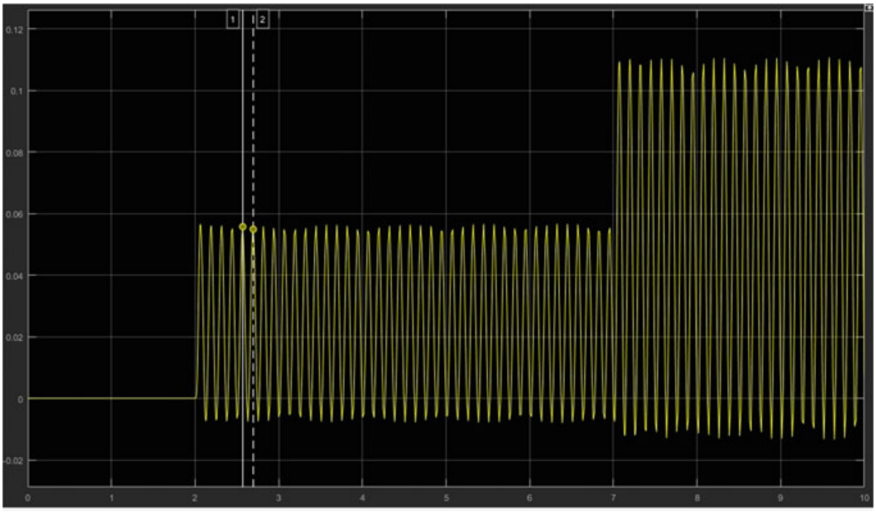


Fig. 6.14 Constant amplitude graph for the value of $K_{\mathrm{p}} = 449$

6.4.10 Ziegler–Nichols Method

According to Ziegler–Nichols method for the PID controller after getting the ultimate gain (K_{u}) and the ultimate time period (T_{u}), the optimum values of K_{p}, K_{i} and K_{d} are given by,

$$K_{\mathrm{p}} = 0.6K_{\mathrm{u}} \tag{6.17}$$

$$K_{\mathrm{i}} = 1.2K_{\mathrm{u}}/T_{\mathrm{u}} \tag{6.18}$$

$$K_{\mathrm{d}} = 0.6K_{\mathrm{u}}T_{\mathrm{u}}/8 \tag{6.19}$$

The optimum values of K_{p}, K_{i} and K_{d} are obtained using the above-mentioned formulas.

The value of K_{p} for which the constant graph is achieved is called the ultimate gain K_{u} which is equal to $K_{\mathrm{p}} = K_{\mathrm{u}} = 449$. The corresponding time period is called the ultimate time period T_{u} and is equal to $T_{\mathrm{u}} = 122.534$ ms. The graph for the value of $K_{\mathrm{p}} = K_{\mathrm{u}} = 449$ is shown in Fig. 6.15.

The ultimate time period T_{u} is taken from the graph and is shown in Fig. 6.16.

Hence, the optimum values of K_{p}, K_{i} and K_{d} are given by,

$K_{\mathrm{p}} = 269.4$;

$K_{\mathrm{i}} = 4397.4694$;

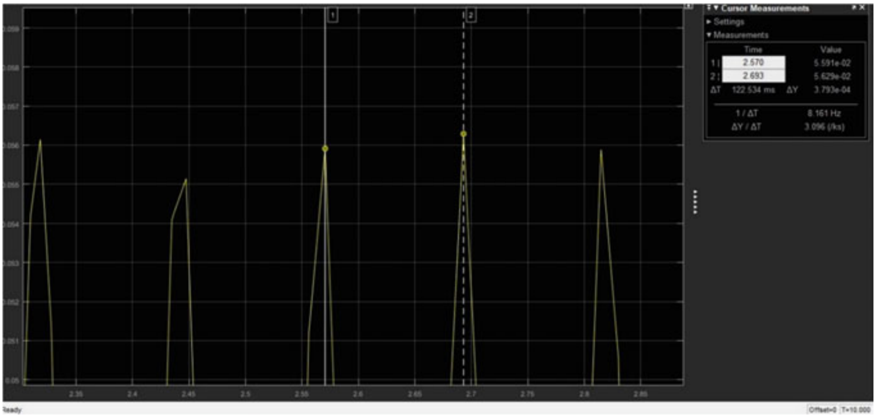


Fig. 6.15 Ultimate time period T_{u} of modified full suspension model

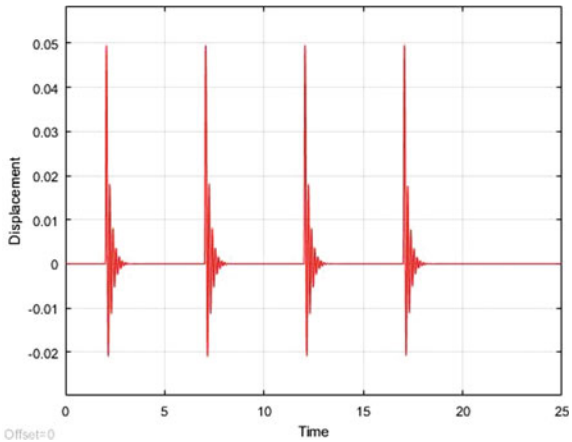


Fig. 6.16 Modified full model displacement of mass m_1

$K_{\mathrm{d}} = 4.1263;$

6.5 Simulation Results of Modified Full Suspension Model

The performance characteristics such as displacement, velocity and acceleration are recorded for the coach (m_1) and the bogie frame (m_2).

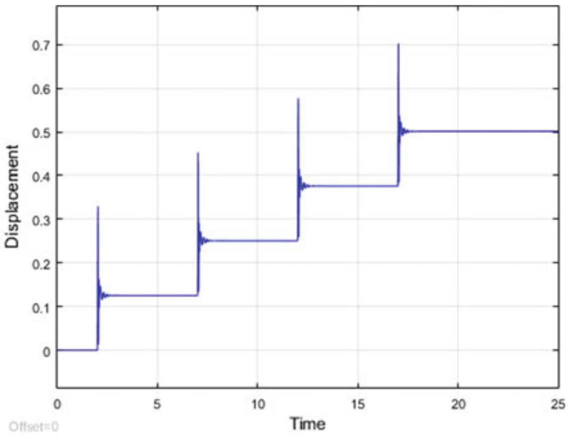


Fig. 6.17 Modified full model displacement of mass m_2

6.5.1 Displacement

The displacement of mass m_1 is shown in Fig. 6.17.

6.5.2 Discussions

The displacement results of the modified quarter suspension model are reduced by the application of hydraulic actuator controlled by PID controller. There is a much lower amplitude of displacement which is recorded for the coach and the bogie frame with a shorter settling time. There is a much lower peak of velocity, and acceleration is recorded for the modified model with a shorter settling time. Hence by using hydraulic actuator controlled by PID controller, we can minimize the vibrations of the railway wagon and so that the travel comfort for the passenger can be achieved.

6.6 Conclusion

Analytical modelling of railway suspension system of Indian railway wagon is done by considering a linear dynamic system model consisting of mass, spring and damper and having three-degree of freedom system. The model is simulated to get the dynamic response using MATLAB Simulink under step input condition of the track. The result shows that the major vibrations occurring at the wheel are transmitted to the bogie and coach of the railway system with a large settling time. These vibrations of the Indian railway wagon affect the passenger ride comfort. The modification to the Indian railway wagon is made by the application of hydraulic actuator controlled by PID controller and again simulated using MATLAB Simulink under step input condition. The result showed improvements over the original suspension model in suppressing the vibrations. Hence, the use of modified suspension system model is suggested in order to minimize the vibrations and to achieve the ride comfort for the passenger.

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