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IMPROVEMENT OF RAILWAY RIDE INDEX THROUGH PITCH MOTION WITH CONSIDERATION OF LONGITUDINAL DAMPERS

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Abstract— Longitudinal dampers also known as yaw dampers play a crucial role in enhancing the ride comfort of railway vehicles by mitigating longitudinal vibrations, which can significantly impact passenger comfort and safety. This study investigates the enhancement of railway ride comfort by focusing on pitch motion control through the implementation of longitudinal dampers. The research aims to address the significant impact of longitudinal vibrations on passenger comfort, which are often exacerbated by track irregularities, braking, and acceleration

Index, Longitudinal Damper, Anti-Pitch	forces. By integrating longitudinal dampers into the suspension system, the study proposes a method to mitigate these vibrations and improve the overall ride quality. A comprehensive dynamic model of a railway vehicle is developed, incorporating the effects of longitudinal dampers on pitch motion. The model was previously validated through both simulations and real-world experiments, covering a range of operating conditions and speeds. Key performance metrics, including the Sperling's Ride Index and ISO 2631-1 standards, are used to evaluate the effectiveness of the proposed system. With 1.2 weighted primary longitudinal damper, the ride index is improved from 1.21 to 1.098. The results demonstrate that longitudinal dampers significantly reduce pitch-induced vibrations, leading to a marked improvement in ride comfort indices. The study also explores the practical implications of implementing longitudinal dampers, including cost, maintenance, and integration with existing railway infrastructure. The findings suggest that longitudinal dampers are a viable solution for enhancing passenger comfort in modern railway systems, providing a smoother and more stable ride experience. This research contributes to the field of railway engineering by offering a novel approach to ride comfort improvement, emphasizing the importance of addressing longitudinal vibrations through advanced suspension technologies.

I. Introduction

The railway ride index is a critical metric for assessing and improving passenger comfort in railway vehicles. The ride index provides a standardized way to quantify passenger comfort, which is inherently subjective and influenced by various factors such as vibrations, noise, and environmental conditions inside the vehicle [1][2]. By using indices like Sperling's Ride Index and ISO 2631-1 standards, engineers can objectively measure and compare the comfort levels of different railway vehicles and configurations [3]. The ride index helps identify specific sources of discomfort, such as vertical and longitudinal vibrations, which can be addressed through targeted improvements in vehicle design and suspension systems [4]. This allows for more precise interventions to enhance ride quality. By providing detailed insights into the factors affecting ride comfort, the ride index guides the design and engineering of railway vehicles [5]. It helps in optimizing suspension systems, seating arrangements, and other aspects of vehicle design to improve passenger comfort. Improved ride comfort directly translates to higher passenger satisfaction [6, 7]. A smoother and more comfortable ride experience encourages more people to choose rail transport, thereby increasing ridership and revenue for railway operators. The ride index also plays a role in ensuring the safety and stability of railway vehicles. By mitigating excessive vibrations and ensuring a stable ride, it helps prevent potential safety issues that could arise from poor ride quality. Better ride comfort indices often correlate with reduced wear and tear on both the railway vehicles and the tracks [8, 9]. This leads to lower maintenance costs and longer service life for critical components. In past research, the ride performance of railway vehicles was enhanced by employing active actuators to mitigate lateral disturbances originating from the track [10, 11].

Longitudinal dampers also known as yaw dampers are pivotal in enhancing the ride comfort of railway vehicles by effectively mitigating longitudinal vibrations. Longitudinal dampers reduce the amplitude of vibrations transmitted to passengers, thereby improving overall ride comfort. This is crucial for maintaining a pleasant travel experience, especially on long journeys [12]. These vibrations, often caused by track irregularities, braking, and acceleration forces, can significantly impact passenger comfort and safety. By controlling longitudinal vibrations, dampers contribute to the stability of the train, reducing the risk of derailments and other safety hazards. This ensures a safer travel environment for passengers [13]. By absorbing and dissipating the energy from these vibrations, longitudinal dampers help maintain a smoother and more stable ride. This not only improves the overall travel experience for passengers but also contributes to the structural integrity and longevity of the railway vehicles.

The effectiveness of longitudinal dampers can be measured using ride comfort indices such as Sperling's Ride Index and ISO 2631-1 standards [14]. These indices provide a standardized way to evaluate and compare the comfort levels of different railway vehicles [15]. Furthermore, the reduction in vibrations leads to decreased wear and tear on both the train and the tracks, resulting in lower maintenance costs and enhanced operational efficiency [16, 17].

Longitudinal dampers help reduce the wear and tear on both the train and the tracks by mitigating the forces transmitted through the vehicle. This leads to lower maintenance costs and longer service life for critical components.

Thus, the implementation of longitudinal dampers is a critical component in modern railway engineering, ensuring both passenger satisfaction and the reliable performance of the railway system. Enhanced ride comfort and reduced maintenance needs contribute to the overall operational efficiency of railway systems. This can result in cost savings and improved reliability for railway operators

In previous publications, researchers introduced magneto-rheological (MR) dampers to improve the railway ride index. MR dampers provide adaptive damping by adjusting their properties in real time based on the current, amplitude, and frequency of vibrations. This adaptability leads to a significant reduction in vibrations, thereby improving ride comfort [18]. Studies have shown that MR dampers can reduce vertical, lateral, and longitudinal vibrations, leading to a smoother ride experience [19].

The use of MR dampers has been shown to improve ride quality indices such as the Sperling's Ride Index and ISO 2631-1 standards. These indices measure the impact of vibrations on passenger comfort, and improvements in these metrics indicate better ride quality [20]. MR dampers offer superior control over the suspension system, allowing for real-time adjustments to changing track conditions and vehicle dynamics. This results in enhanced stability and safety of the railway vehicle, particularly at high speeds. The ability to adapt to different operating conditions ensures that the vehicle maintains optimal performance and comfort levels.

While MR dampers offer significant benefits in terms of ride comfort and stability, there are valid concerns regarding their cost, particularly in the context of research and development. The development and implementation of MR dampers involve substantial initial costs. These include the expenses for advanced materials, sophisticated control systems, and the integration of these components into existing railway vehicles. For research projects with limited budgets, these costs can be prohibitive.

MR dampers are more complex than traditional suspension systems, requiring specialized knowledge and equipment for maintenance and repairs. This complexity can lead to higher ongoing costs and the need for specialized training for maintenance personnel.

There are alternative technologies and methods for improving ride comfort that may be more cost-effective. For example, optimizing existing suspension systems, improving track maintenance, and using passive damping solutions can provide significant benefits at a lower cost.

In this paper, longitudinal passive dampers are applied to the railway train model to improve its pitch motions and thus improve its ride index.

II. Validated Model

The validated railway dynamic models are essential tools in the development, optimization and improvement of ride index of railway vehicles. Validated dynamic models provide accurate predictions of how railway vehicles will behave under various operating conditions. This includes responses to track irregularities, braking, acceleration, and other dynamic forces [21].

Accurate models help engineers design vehicles that perform reliably and safely. These models are crucial for optimizing ride comfort. By simulating different suspension configurations and damping systems, engineers can identify the best solutions to minimize vibrations and improve passenger comfort [21]. This is particularly important for high-speed trains where comfort is a key competitive factor.

In this paper, the previously validated 31DOF full-vehicle model [22] is used and reduced to 9DOF half-vehicle model for simplification of analysis and focusing on pitch motion performance. The ride index formula is applied on the pitch acceleration since the longitudinal dampers do not affect the vertical motions. The primary longitudinal dampers are attached between bogie and wheel axles (Figure 1). The dampers are attached between the bogie and body for the secondary longitudinal damper (Figure 2).

III. Ride Index

ISO 2631, EN 12299, and Sperling’s method are among the methodologies established to evaluate the vibration and ride quality of railway vehicles globally [23]. The data utilized in these assessments is derived from the analysis of whole-body vibration, which is employed to ascertain ride quality. The ride evaluation can be quantified according to the criteria outlined in Table 1.

Table 1: Ride evaluation scale as per Sperling Ride Index [14]

Ride index, w_z	Vibration sensitivity
1	Just noticeable
2	Clearly noticeable
2.5	More pronounces but not unpleasant
3	Strong, irregular, but still tolerable
3.25	Very irregular
3.5	Extremely irregular, unpleasant, annoying, prolonged exposure intolerable
4	Extremely unpleasant, prolonged exposure harmful

The continuous whole body vibration exposure root mean square average vibration (a^{wrms}) is the root mean square (RMS) value of the frequency weighted acceleration $a^w(t)$ in m/s^2. The value can be calculated by using the Equation (1) [23]. The weighted acceleration for this analysis is using the pitch acceleration in rad/s^2.

$$w_z = 4.42(a^{wrms})^{0.3} \tag{1}$$

IV. Mathematical Modelling

The mathematical model of the railway train is derived for nine degrees of freedom (9DOF) half-body vertical and pitch model (Figure 3) with the additional primary and secondary longitudinal dampers (C_{1x}, K_{1x}, C_{2x}, K_{2x}). The details parameters and variables are tabulated in Tables 2 and 3. Equation (2) to (10) representing the 9DOF model of the train.

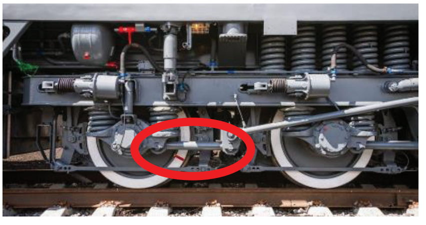

Figure 1: Possible position of primary longitudinal damper [24]

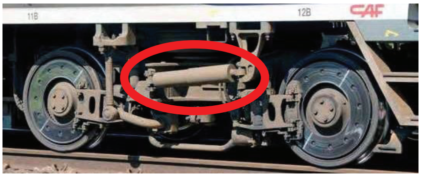

Figure 2: Secondary longitudinal damper [25]

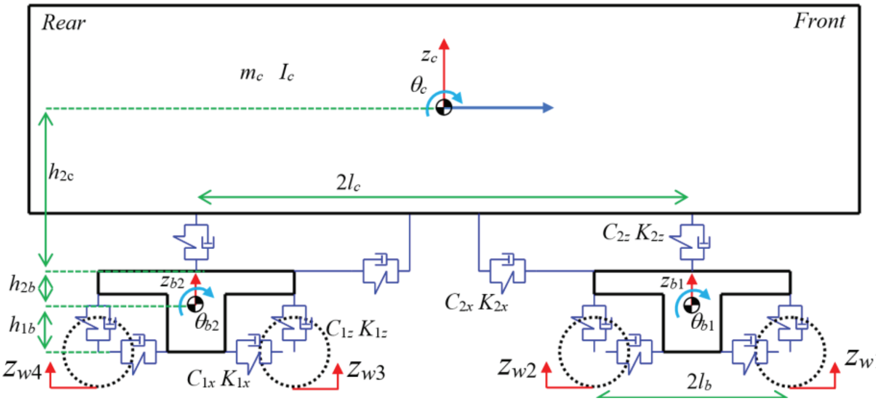

Figure 3: 9DOF mass-spring-damper system schematic diagram

i. Vertical body acceleration

$$m_c\ddot{z}_c = k_{2z}(z_{b1} - z_c - l_c\theta_c) + c_{2z}(\dot{z}_{b1} - \dot{z}_c - l_c\dot{\theta}_c) + k_{2z}(z_{b2} - z_c + l_c\theta_c) + c_{2z}(\dot{z}_{b2} - \dot{z}_c + l_c\dot{\theta}_c) \tag{2}$$

ii. Longitudinal body acceleration

$$m_c\ddot{x}_c = K_{2x}(x_{b1} + h_{2b}\theta_{b1} - x_c + h_{2c}\theta_c) + C_{2x}(\dot{x}_{b2} + h_{2b}\dot{\theta}_{b2} - \dot{x}_c + h_{2c}\dot{\theta}_c) + K_{2x}(x_{b2} + h_{2b}\theta_{b2} - x_c + h_{2c}\theta_c) + C_{2x}(\dot{x}_{b2} + h_{2b}\dot{\theta}_{b2} - \dot{x}_c + h_{2c}\dot{\theta}_c) \tag{3}$$

iii. Body pitch acceleration

$$I_{cy}\ddot{\theta}_c = l_c\{K_{2z}(z_{b1} - z_c - l_c\theta_c) + C_{2z}(\dot{z}_{b1} - \dot{z}_c - l_c\dot{\theta}_c)\} - l_c\{K_{2z}(z_{b2} - z_c + l_c\theta_c) + C_{2z}(\dot{z}_{b2} - \dot{z}_c + l_c\dot{\theta}_c)\} - h_{2c}\{K_{2x}(x_{b1} + h_{2b}\theta_{b1} - x_c + h_{2c}\theta_c) + C_{2x}\left(\dot{x}_{b2} + h_{2b}\dot{\theta}_{b2} - \dot{x}_c + h_{2c}\dot{\theta}_c\right) + K_{2x}(x_{b2} + h_{2b}\theta_{b2} - x_c + h_{2c}\theta_c) + C_{2x}\left(\dot{x}_{b2} + h_{2b}\dot{\theta}_{b2} - \dot{x}_c + h_{2c}\dot{\theta}_c\right)\} \tag{4}$$

iv. Vertical front bogie acceleration

$$m_b\ddot{z}_{b1} = -k_{2z}(z_{b1} - z_c - l_c\theta_c) - c_{2z}(\dot{z}_{b1} - \dot{z}_c - l_c\dot{\theta}_c) + k_{1z}(z_{w1} - z_{b1} - l_b\theta_{b1}) + c_{1z}(\dot{z}_{w1} - \dot{z}_{b1} - l_b\dot{\theta}_{b1}) + k_{1z}(z_{w2} - z_{b1} + l_b\theta_{b1}) + c_{1z}(\dot{z}_{w2} - \dot{z}_{b1} + l_b\dot{\theta}_{b1}) \tag{5}$$

v. Longitudinal front bogie acceleration

$$m_b\ddot{x}_{b1} = -\{K_{2x}(x_{b1} + h_{2b}\theta_{b1} - x_c + h_{2c}\theta_c) + C_{2x}(\dot{x}_{b1} + h_{2b}\dot{\theta}_{b1} - \dot{x}_c + h_{2c}\dot{\theta}_c)\} + K_{1x}(x_{w1} - x_{b1} + h_{1b}\theta_{b1}) + C_{1x}(\dot{x}_{w1} - \dot{x}_{b1} + h_{1b}\dot{\theta}_{b1}) + K_{1x}(x_{w2} - x_{b1} + h_{1b}\theta_{b1}) + C_{1x}(\dot{x}_{w2} - \dot{x}_{b1} + h_{1b}\dot{\theta}_{b1}) \tag{6}$$

vi. Front bogie pitch acceleration

$$n\ I_{by}\ddot{\theta}_{b1} = l_b\{K_{1z}(z_{w1} - z_{b1} - l_b\theta_{b1}) + C_{1z}(\dot{z}_{w1} - \dot{z}_{b1} - l_b\dot{\theta}_{b1})\} - l_b\{K_{1z}(z_{w2} - z_{b1} + l_b\theta_{b1}) + C_{1z}(\dot{z}_{w2} - \dot{z}_{b1} + l_b\dot{\theta}_{b1})\} + h_{2b}F_{2xf} - h_{1b}\{K_{1x}(x_{w1} - x_{b1} + h_{1b}\theta_{b1}) + C_{1x}\left(\dot{x}_{w1} - \dot{x}_{b1} + h_{1b}\dot{\theta}_{b1}\right) + K_{1x}(x_{w2} - x_{b1} + h_{1b}\theta_{b1}) + C_{1x}\left(\dot{x}_{w2} - \dot{x}_{b1} + h_{1b}\dot{\theta}_{b1}\right)\} \quad (7)$$

vii. Vertical rear bogie acceleration

$$m_b\ddot{z}_{b2} = -k_{2z}(z_{b2} - z_c + l_c\theta_c) - c_{2z}(\dot{z}_{b2} - \dot{z}_c + l_c\dot{\theta}_c) + k_{1z}(z_{w3} - z_{b2} - l_b\theta_{b2}) + c_{1z}(\dot{z}_{w3} - \dot{z}_{b2} - l_b\dot{\theta}_{b2}) + k_{1z}(z_{w4} - z_{b2} + l_b\theta_{b2}) + c_{1z}(\dot{z}_{w4} - \dot{z}_{b2} + l_b\dot{\theta}_{b2}) \quad (8)$$

viii. Longitudinal rear bogie acceleration

$$m_b\ddot{x}_{b2} = -\{K_{2x}(x_{b2} + h_{2b}\theta_{b2} - x_c + h_{2c}\theta_c) + C_{2x}(\dot{x}_{b2} + h_{2b}\dot{\theta}_{b2} - \dot{x}_c + h_{2c}\dot{\theta}_c)\} + K_{1x}(x_{w3} - x_{b2} + h_{1b}\theta_{b2}) + C_{1x}(\dot{x}_{w3} - \dot{x}_{b2} + h_{1b}\dot{\theta}_{b2}) + K_{1x}(x_{w4} - x_{b2} + h_{1b}\theta_{b2}) + C_{1x}(\dot{x}_{w4} - \dot{x}_{b2} + h_{1b}\dot{\theta}_{b2}) \quad (9)$$

ix. Rear bogie pitch acceleration

$$I_{by}\ddot{\theta}_{b2} = l_b\{K_{1z}(z_{w3} - z_{b2} - l_b\theta_{b2}) + C_{1z}(\dot{z}_{w3} - \dot{z}_{b2} - l_b\dot{\theta}_{b2})\} - l_b\{K_{1z}(z_{w4} - z_{b2} + l_b\theta_{b2}) + C_{1z}(\dot{z}_{w4} - \dot{z}_{b2} + l_b\dot{\theta}_{b2})\} + h_{2b}\{K_{2x}(x_{b2} + h_{2b}\theta_{b2} - x_c + h_{2c}\theta_c) + C_{2x}(\dot{x}_{b2} + h_{2b}\dot{\theta}_{b2} - \dot{x}_c + h_{2c}\dot{\theta}_c)\} - h_{1b}\{K_{1x}(x_{w3} - x_{b2} + h_{1b}\theta_{b2}) + C_{1x}\left(\dot{x}_{w3} - \dot{x}_{b2} + h_{1b}\dot{\theta}_{b2}\right) + K_{1x}(x_{w4} - x_{b2} + h_{1b}\theta_{b2}) + C_{1x}\left(\dot{x}_{w4} - \dot{x}_{b2} + h_{1b}\dot{\theta}_{b2}\right)\} \quad (10)$$

Table 2: Parameters [27]

Symbol	Quantity	Value
m_c	Railway body mass (kg)	40000
I_c	Railway body inertia (kgm^2)	2080000
$2l_c$	Length between bogies (m)	17.5
m_b	Bogie mass (kg)	3040
I_b	Bogie inertia (kgm^2)	3930
$2l_b$	Length between wheels (m)	2.5
K_{1z}	Primary vertical suspension stiffness (N/m)	530000
C_{1z}	Primary vertical suspension damping (Ns/m)	90200
K_{1x}	Primary longitudinal suspension stiffness (N/m)	340000
C_{1x}	Primary longitudinal suspension damping (Ns/m)	10000
K_{2z}	Secondary vertical suspension stiffness (N/m)	1180000

C_{2z}	Secondary vertical suspension damping (Ns/m)	39200
K_{2x}	Secondary longitudinal suspension stiffness (N/m)	10000000
C_{2x}	Secondary longitudinal suspension damping (Ns/m)	100000
h_{2c}	Vertical distance from car to secondary longitudinal suspension (m)	0.8
h_{2b}	Vertical distance from bogie to secondary longitudinal suspension (m)	0.5
h_{1b}	Vertical distance from bogie to primary longitudinal suspension (m)	0.22

Table 3: States of the equations

Variable	Description
z_c	Body vertical displacement
$\dot{z}_c$	Body vertical velocity
$\ddot{z}_c$	Body vertical acceleration
x_c	Body longitudinal displacement
$\dot{x}_c$	Body longitudinal velocity
$\ddot{x}_c$	Body longitudinal acceleration
θ_c	Body pitch angle
$\dot{\theta}_c$	Body pitch angular velocity
$\ddot{\theta}_c$	Body pitch angular acceleration
z_{b1}	Front bogie vertical displacement
$\dot{z}_{b1}$	Front bogie vertical velocity
$\ddot{z}_{b1}$	Front bogie vertical acceleration
x_{b1}	Front bogie longitudinal displacement
$\dot{x}_{b1}$	Front bogie longitudinal velocity
$\ddot{x}_{b1}$	Front bogie longitudinal acceleration
θ_{b1}	Front bogie pitch angle
$\dot{\theta}_{b1}$	Front bogie pitch angular velocity
$\ddot{\theta}_{b1}$	Front bogie pitch angular acceleration
z_{b2}	Rear bogie vertical displacement
$\dot{z}_{b2}$	Rear bogie vertical velocity
$\ddot{z}_{b2}$	Rear bogie vertical acceleration
x_{b2}	Rear bogie longitudinal displacement
$\dot{x}_{b2}$	Rear bogie longitudinal velocity
$\ddot{x}_{b2}$	Rear bogie longitudinal acceleration
θ_{b2}	Rear bogie pitch angle
$\dot{\theta}_{b2}$	Rear bogie pitch angular velocity
$\ddot{\theta}_{b2}$	Rear bogie pitch angular acceleration
z_{w1}	Wheel 1 vertical disturbance
z_{w2}	Wheel 2 vertical disturbance

z_{w3}	Wheel 3 vertical disturbance
z_{w4}	Wheel 4 vertical disturbance

V. MATLAB/Simulink Model

In this model, the sine wave is applied as disturbance from the track to evaluate the ride index performance. The outputs of the model are the vertical body acceleration, longitudinal body acceleration, pitch body acceleration, vertical front and rear bogies accelerations, longitudinal front and rear bogies accelerations and front and rear pitch accelerations. The body pitch acceleration is implemented in Equation (1) to find the ride index. The input is applied on z_{w1} to produce body and front bogie pitch motions. The input is sine wave with amplitude 0.01 m with frequency 2π Hz as shown in Figure 4.

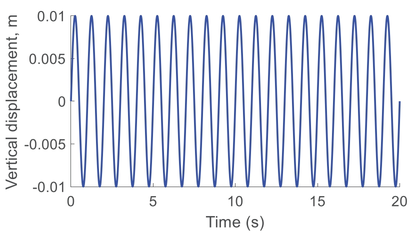

Figure 4: Sine input

VI. Results and Analysis

Figures 5, 6 and 7 show the output body vertical and longitudinal and pitch angle accelerations (vibrations) respectively.

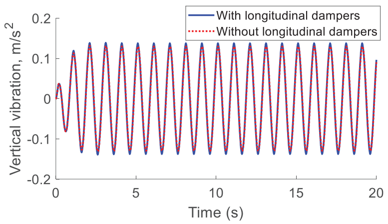

Figure 5: Body vertical vibration

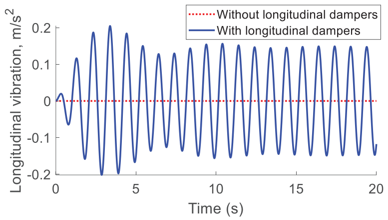

Figure 6: Body longitudinal vibration

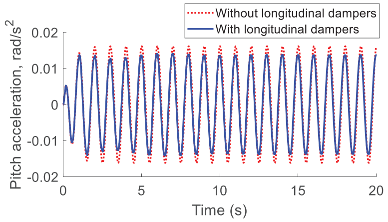

Figure 7: Body pitch acceleration

From Figure 5, it can be observed that the body vertical vibration with longitudinal dampers is slightly higher (amplitude 0.137 m/s^2) than without longitudinal dampers (amplitude 0.129 m/s^2). This due to the minor effect of longitudinal dampers. In longitudinal vibration output as in Figure 6, since there are no longitudinal dampers, there is no longitudinal body vibration. This is accurately representing the previous finding when the longitudinal motions are neglected [26]. Yet, the apparent longitudinal vibration of the system with longitudinal dampers is considered important for the analysis. In Figure 7, with the application of longitudinal dampers, it can be observed that, the pitch angle acceleration is reduced moderately from 0.016 rad/s^2 to 0.012 rad/s^2. The improvement is about 25% as stated in Table 4. The output is later used to determine the ride index.

Table 4: Performance at original parameters

Performance	Without Longitudinal Dampers	With Longitudinal Dampers	Improvement (%)
$\ddot{\theta}_c$ (rad/s^2)	0.016	0.012	25
Ride index	1.272	1.224	3.8

Figures 8, 9 and 10 establish the output performance of the front bogie's vertical, longitudinal and pitch accelerations (vibrations) respectively.

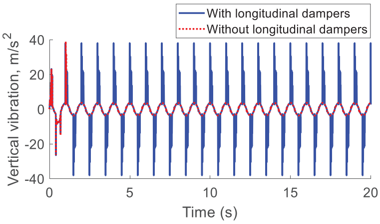

Figure 8: Front bogie vertical vibration

In Figure 8, the vertical vibration with longitudinal dampers is obviously significant compared to the system without longitudinal dampers starting at time = 1.49 s. The different in amplitude is around 90.5% where the amplitudes are 38.05 m/s^2 for the system with longitudinal dampers and 3.62 m/s^2 for the system without longitudinal dampers. The same effect is monitored with the pitch vibrations (Figure 10). The different in amplitude is 88.1%. The longitudinal dampers drastically increase the pitch vibration from 0.456 rad/s^2 to 3.819 rad/s^2.

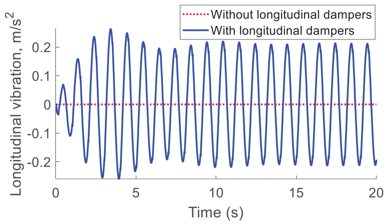

Figure 9: Front bogie longitudinal vibration

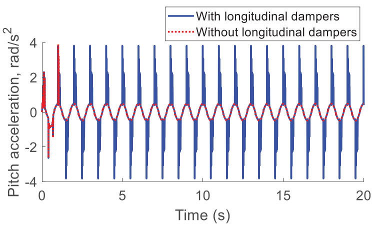

Figure 10: Front bogie pitch acceleration

Obvious effect of longitudinal dampers can be seen in Figure 9 where the longitudinal vibration is recorded at 0.2 m/s^2 instead of zero longitudinal vibration with the system without longitudinal dampers. The vibrations are significantly affected by the implementation of longitudinal dampers. The same remarks are observed for the vibrations of rear bogie as shown in Figures 11, 12 and 13. The amplitude of vertical vibration is increased from 0.045 m/s^2 to 0.052 m/s^2 as shown in Figure 11 which is about 15.6%.

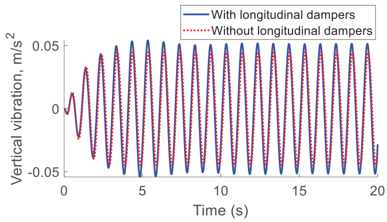

Figure 11: Rear bogie vertical vibration

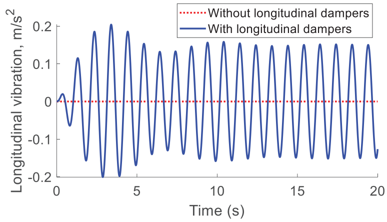

Figure 12: Rear bogie longitudinal vibration

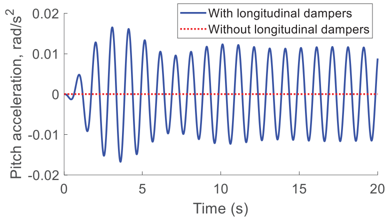

Figure 13: Body pitch angle from random input

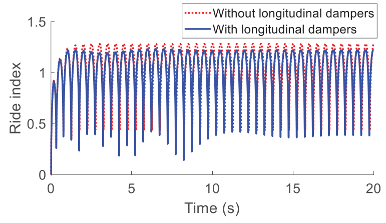

Figure 14: Ride index

In Figures 12 and 13, there is no longitudinal vibration for the system without longitudinal dampers. This is due to the absent of secondary longitudinal dampers in rear bogie. Therefore, there are no longitudinal motions are transferred from the body to the rear bogie. Since the effects are substantial to the vibrations, the longitudinal dampers are important in the derivation of the equations and cannot be neglected. Figure 14 shows the ride index performance of the railway vehicle using body pitch vibration values. The lower the value of the ride index, the better the ride performance. It can be observed that, the average maximum ride index is about 1.272 for the model without longitudinal dampers, while the average maximum ride index with the longitudinal dampers is 1.224. The ride performance is improved about 3.8% as stated in Table 4.

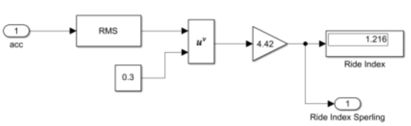

Figure 15: Pitch acceleration ride index blocks in MATLAB/Simulink model

Figure 15 shows the ride index block model derived from Equation (1). Table 5 summarize the ride index values at different weighted of primary and secondary longitudinal dampers. For example, the weighted 1.5 to secondary longitudinal dampers means the simulation is continued and performed with values of 1.5 times the original parameters. Where C_{2x} = 1.5 x 100 kNs/m = 150 kNs/m and K_{2x} = 1.5 x 10 MN/m = 15 MNs/m. Without longitudinal dampers, the ride index is 1.281. With longitudinal dampers at original parameters [27], the ride index is 1.216. The improvement of ride is about 5.1%. For the primary longitudinal suspensions, with the increment of weighted from 0.5 to 1.2, the ride index is reduced linearly. Nevertheless, with weighted 1.5, the ride index is increased. This proved that the optimize primary longitudinal dampers parameters is about 1.2 weighted. For the secondary longitudinal suspensions, the increment of weighted from 0.5 to 1.5, the ride index is improved from 1.229 to 1.211. This confirms that, the higher the values of secondary longitudinal dampers parameters, the better the ride performance.

Table 5: Ride index using $\ddot{\theta}_c$

Weighted to secondary longitudinal dampers	Ride Index
0 (no longitudinal dampers)	1.21
1.5	1.211
1.2	1.214
1.0 (original values)	1.216
0.8	1.220
0.5	1.229
Weighted to primary longitudinal dampers	**Ride Index**
1.5	1.106
1.2	1.098
1.0 (original values)	1.216
0.8	1.252
0.5	1.271

VII. Conclusion

In previous research, the 6DOF model neglected the longitudinal motions [26]. The findings are accurate with the results in this paper with 9DOF with weighted is 0 where there are no longitudinal dampers. The primary and secondary longitudinal dampers affect significantly on the performances of the railway dynamics, especially the longitudinal and pitch vibrations of body, front bogie and rear bogie. This effect cannot simply be neglected since the actual railway vehicle is having these dampers. The body pitch vibration has improved 25 %. With these dampers, the ride index determined from the pitch vibrations is improved 3.8 %. With the increment of weighted parameters, the ride index is linearly enhanced. The 9DOF model can be further used to develop primary and secondary active suspension systems.

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