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RIDE INDEX PERFORMANCES OF HIGH-SPEED RAILWAY TRAINS

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Abstract— Railway trains are becoming one of the important transportations in nowadays era due to their effective means of travelling from one place to another place. Therefore, many researchers work toward increasing the ride quality and performance of railway vehicles. The aim of this paper is to analyze the ride index of three different trains which are French Thalys Y230A, German ICE 3, and China CRH2 trains through simulation. The model focuses on six degrees of freedom (6DOF) and the Sperling Index Method is used to determine the ride index for each train. Three elements such as mass, spring and damper are connected in series and modelled in

Matlab/Simulink Model	MATLAB/Simulink. The system can be used to simulate the dynamic behavior of the mass under varied conditions, such as a variety of initial conditions, different spring and damping constants, and various external forces. The results obtained from the graph can be concluded as the ride index performances of the trains.

I. Introduction

The comfort of railway vehicles is not only a necessary component of the contemporary railway system, but it is also an important aspect in competing with other modes of transportation [1]. When examining the dynamic behavior of railway vehicles [2], ride quality is one of the parameters considered [3]. The word "ride quality" refers to a wide range of elements, including ride smoothness, noise levels, vibration levels, passenger comfort, safety and stability and track quality [4].

There are many factors that contribute to passenger comfort such as seating, interior ambiance, vibrations, temperature, humidity, climate control, lighting, and noise level. Among all elements influencing passenger comfort, vibrations and oscillations receive special attention due to the importance of their effect on the human body. The vibration will cause tiredness, restlessness, fatigue, and tinnitus.

Thus, these problems attract the researchers and technical communities to grow due to the alarmingly high amplitudes of vibration that were detected [5][6].

Railway vibration is caused by forces created at the point of contact between the train wheel and the rail and these forces are divisible into quasi-static and dynamic components [7]. In the previous research, the ride performance of railway vehicle was improved through active actuator in supressing lateral disturbances from the track [8][9]. However, installation such actuator is costly and cumbersome to the suspension systems. The high-speed train especially has a possibility of creating substantial body vibration which can reduce the ride quality and car body stability. Ride quality is a critical factor that directly impacts passenger comfort, safety, health, and overall satisfaction. Ensuring good ride quality in high-speed train is essential for attracting passengers, especially for long-distance travel, and for promoting the use of rail transportation as an efficient and enjoyable mode of travel.

II. Ride Quality Assessment

There are several methods that can be used to measure the ride comfort. This method focuses on reading the vibration between the railway track and the car body through MATLAB/Simulink program.

A. Smartphone

Nowadays, passengers, equipped with smartphones, may be able to rate their own level of satisfaction with a given voyage. This opens the door to a lot of business opportunities and lets passengers give feedback on the spot about how comfortable their ride was. This gives railway companies information they can use to make rides more comfortable for passengers. The viability of utilizing mobile devices to evaluate road conditions and the ride comfort associated with such conditions is practical [10]. This system can reduce maintenance costs and increase railway safety by identifying track defects and indicating which sections of the track need maintenance.

B. Sperling Method

ISO 2631, EN 12299, and Sperling's method are just some of the tools that have been developed to assess the vibration and ride quality of railway vehicles in various parts of the world [11]. The data comes from the analysis of whole-body vibration presented to determine the ride quality. The ride evaluation can be scaled as in Table 1.

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 stated below.

$$w_z = 4.42(a^{wrms})^{0.03} \quad (1)$$

Table 1: Ride evaluation scale as per Sperling Ride Index [10]

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

C. Vibration Model

According to frequency analysis and statistical analysis, the vibration of railway vehicles can be treated as a random signal with a normal distribution and specific frequency ranges [12]. The modelling procedure is used to evaluate the ride comfort of railway vehicles. To examine he relationships between ride comfort indices of different evaluation methods, multiple ride comfort indices are generated from random signals using the suggested vibration model, and the relationships between two ride comfort indices are determined using curve fitting [13].

III. Mathematical Modelling

The mathematical model of the railway train is derived from six degrees of freedom (6DOF) half-body vertical and pitch model (Figure 1). The equations are reduced from the previously validated 31DOF full body model [14]. The details parameters and variables are tabulated in Table 2 and 3. Train A is French Thalys HST [15], Train B is German ICE 3 [16] and Train C is China CRH2 [17].

i. Vertical body acceleration

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

ii. Body pitch acceleration

$$I_c\ddot{\theta}_c = l_c\{k_{2z}(z_{b1} - z_c - l_c\theta_c) + c_{2z}(\dot{z}_{b1} - \dot{z}_c - l_c\dot{\theta}_c)\} - l_c\{k_{2z}(z_{b2} - z_c + l_c\theta_c) + c_{2z}(\dot{z}_{b2} - \dot{z}_c + l_c\dot{\theta}_c)\} \quad (3)$$

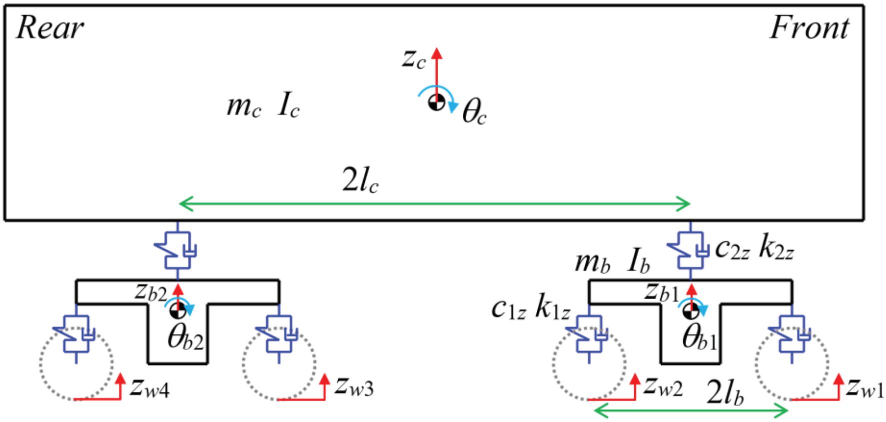

Figure 1: 6-DOF Mass-Spring-Damper system schematic diagram

iii. Vertical front bogie acceleration

$$m_b\ddot{z}_{b1} = -k_{2z}(z_{b1} - z_c - l_c\theta_c) - c_{2z}(\dot{z}_{b1} - \dot{z}_c - l_c\dot{\theta}_c) + k_{1z}(z_{w1} - z_{b1} - l_b\theta_{b1}) + c_{1z}(\dot{z}_{w1} - \dot{z}_{b1} - l_b\dot{\theta}_{b1}) + k_{1z}(z_{w2} - z_{b1} + l_b\theta_{b1}) + c_{1z}(\dot{z}_{w2} - \dot{z}_{b1} + l_b\dot{\theta}_{b1}) \quad (4)$$

iv. Front bogie pitch acceleration

$$I_b\ddot{\theta}_{b1} = l_b\{k_{1z}(z_{w1} - z_{b1} - l_b\theta_{b1}) + c_{1z}(\dot{z}_{w1} - \dot{z}_{b1} - l_b\dot{\theta}_{b1})\} - l_b\{k_{1z}(z_{w2} - z_{b1} + l_b\theta_{b1}) + c_{1z}(\dot{z}_{w2} - \dot{z}_{b1} + l_b\dot{\theta}_{b1})\} \quad (5)$$

v. Vertical rear bogie acceleration

$$m_b\ddot{z}_{b2} = -k_{2z}(z_{b2} - z_c + l_c\theta_c) - c_{2z}(\dot{z}_{b2} - \dot{z}_c + l_c\dot{\theta}_c) + k_{1z}(z_{w3} - z_{b2} - l_b\theta_{b2}) + c_{1z}(\dot{z}_{w3} - \dot{z}_{b2} - l_b\dot{\theta}_{b2}) + k_{1z}(z_{w4} - z_{b2} + l_b\theta_{b2}) + c_{1z}(\dot{z}_{w4} - \dot{z}_{b2} + l_b\dot{\theta}_{b2}) \quad (6)$$

vi. Rear bogie pitch acceleration

$$I_b\ddot{\theta}_{b2} = l_b\{k_{1z}(z_{w3} - z_{b2} - l_b\theta_{b2}) + c_{1z}(\dot{z}_{w3} - \dot{z}_{b2} - l_b\dot{\theta}_{b2})\} - l_b\{k_{1z}(z_{w4} - z_{b2} + l_b\theta_{b2}) + c_{1z}(\dot{z}_{w4} - \dot{z}_{b2} + l_b\dot{\theta}_{b2})\} \quad (7)$$

Table 2: States of the equations

Variable	Description
z_c	Body vertical displacement
$\dot{z}_c$	Body vertical velocity
$\ddot{z}_c$	Body vertical 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
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
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

IV. MATLAB/Simulink Model

In this model, the step, sine wave and random inputs are applied as the disturbances from the track to evaluate the ride index performance for three different trains. The outputs of the model are z_c, θ_c and $\ddot{z}_c$. The body vertical acceleration is implemented in Equation (1) to find the ride index. The input is applied on z_{w1} and z_{w2} to produce body pitch. The inputs are step 0.01 m (Figure 2), sine with amplitude 0.01 m with frequency 2π Hz (Figure 3) and random input between 0.01 m and – 0.01 m (Figure 4).

V. Results and Analysis

Figures 5, 6 and 7 show the output body vertical displacement, pitch angle and vibration or acceleration respectively. From Figure 5, it can be observed that the body vertical displacement Train C is the best since it has the lowest overshoot around 0.006 m and earliest settling to steady state at around 3 s. Train A has overshoot of 0.009 m. However, Train A is better than Train B in term of settling time around 6 s compared to Train B at around 12 s. Body pitch angle follows the performance of body vertical displacement where Train C is the best among others as shown in Figure 6. On the other hand, the body vertical vibration of Train B is the best since it has the lowest overshoot at 0.1 m/s^2 as shown in Figure 7. The ride index is measured from the RMS values of the body vertical vibration. In case of step input, since the body vertical vibration response is damped out, the ride index is neglected. Figures 8 to 10 show the responses from sine input.

In Figure 8, Train A produces the highest body vertical displacement amplitude which is 0.0055 m as compared to Train B 0.0045 m and Train C 0.005 m. However, for body pitch angle, Train A and C have almost the same amplitude which is around 0.6 rad (Figure 9). Train B has the highest amplitude of 0.75 rad. In Figure 10, it can be examined that Train A has the highest amplitude of vibration. Since the ride index is measured from RMS values of the vertical vibration, the ride index of Train A is expected to be the highest. The performance of Train A is getting worse when random inputs are applied on the wheels. This is shown by Figure 11, 12 and 13. Figure 11 shows the body vertical displacements of all trains. The body vertical displacement of Train A is the worse since it has the highest amplitude of 5 x $10^{-}3$ m. The same is happened to the body pitch angle where the amplitude is around 5 x 10^{-4} rad for Train A in Figure 12. In Figure 13, the best performance is by Train B where it amplitude of vertical vibration is around 0.1 m/s^2. Figure 14 shows the MATLAB/Simulink block diagram for the ride index calculation using Equation (1). The RMS and ride index values for each train at different inputs are tabulated in Table 4. According to Table 1, the lower the value of ride index, the better in ride quality and performance.

The ride index for Train C only good for step input where the value is less than 0.01. However, when compared to Train A and B with sine and random inputs, Train C is the awful. Among all trains, riding with Train B is the most pleasant since it has ride index of 1.441 and 1.060 for both since and random inputs.

Table 3: Parameters

Symbol	Quantity	Train A	Train B	Train C
m_c	Railway body mass (kg)	26721	49000	33200
I_c	Railway body inertia (kgm^2)	1150000	2576000	1402000
$2l_c$	Length between bogies (m)	18.7	17.38	17.5
m_b	Bogie mass (kg)	3261	2700	2600
I_b	Bogie inertia (kgm^2)	2870	3330	2600
$2l_b$	Length between wheels (m)	3	2.5	2.5
k_{1z}	Primary suspension stiffness (N/m)	2090000	690000	1176000
c_{1z}	Primary suspension damping (Ns/m)	40000	54000	19600
k_{2z}	Secondary suspension stiffness (N/m)	2450000	603000	193100
c_{2z}	Secondary suspension damping (Ns/m)	40000	29000	98000

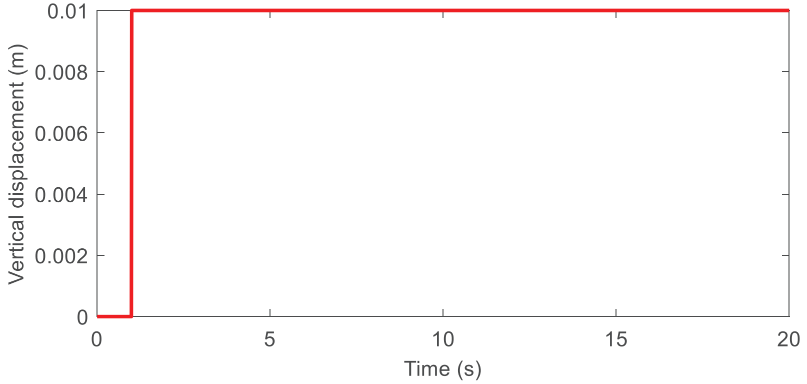

Figure 2: Step input

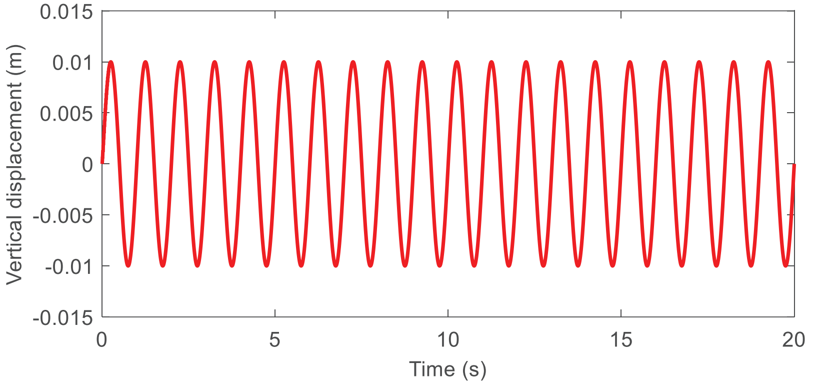

Figure 3: Sine input

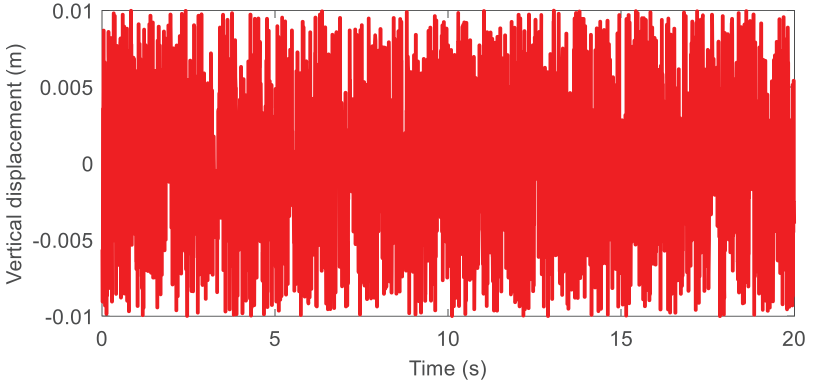

Figure 4: Random input

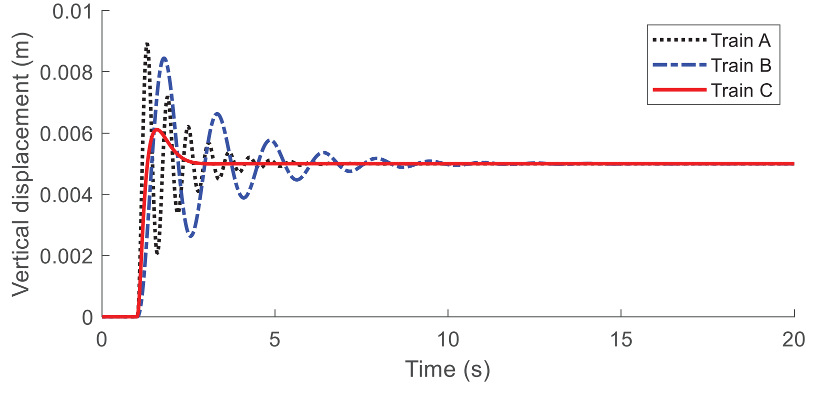

Figure 5: Body vertical displacement from step input

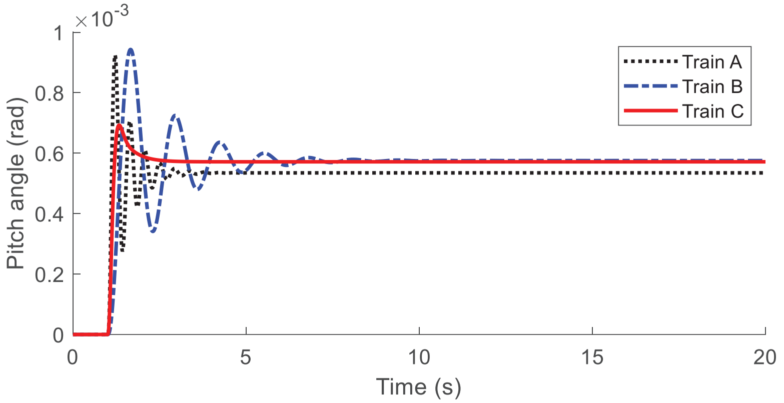

Figure 6: Body pitch angle from step input

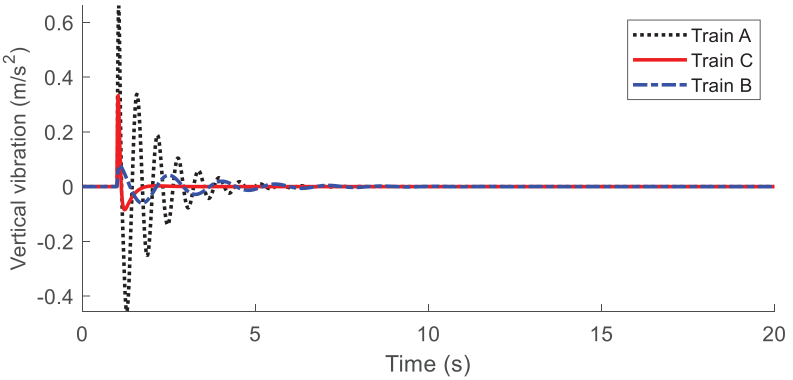

Figure 7: Body vertical vibration from step input

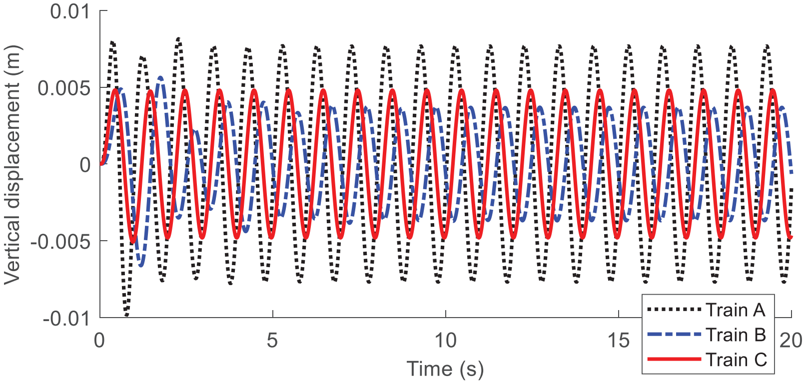

Figure 8: Body vertical displacement from sine input

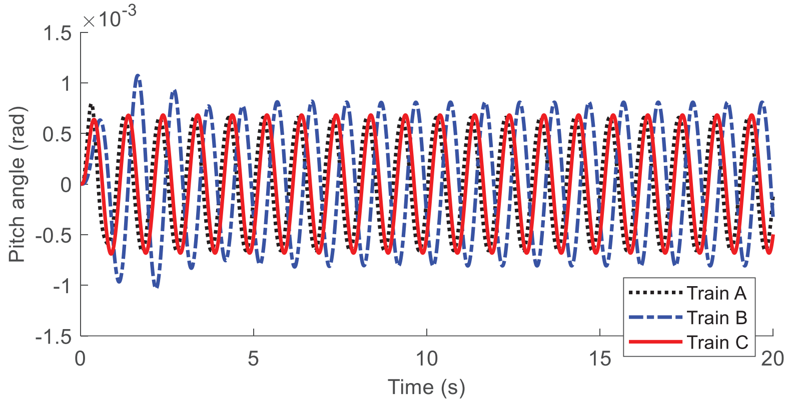

Figure 9: Body pitch angle from sine input

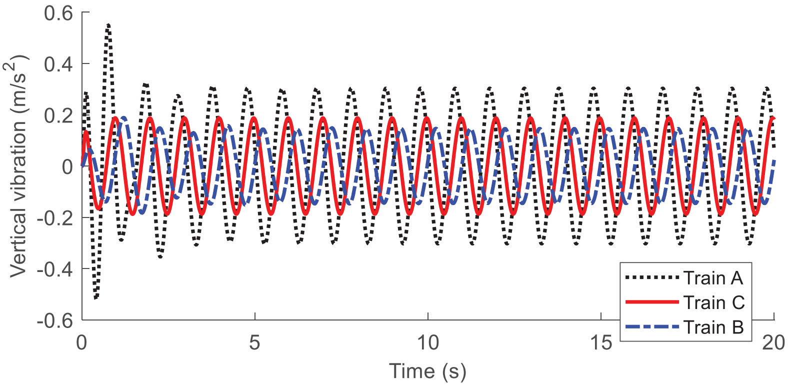

Figure 10: Body vertical vibration from sine input

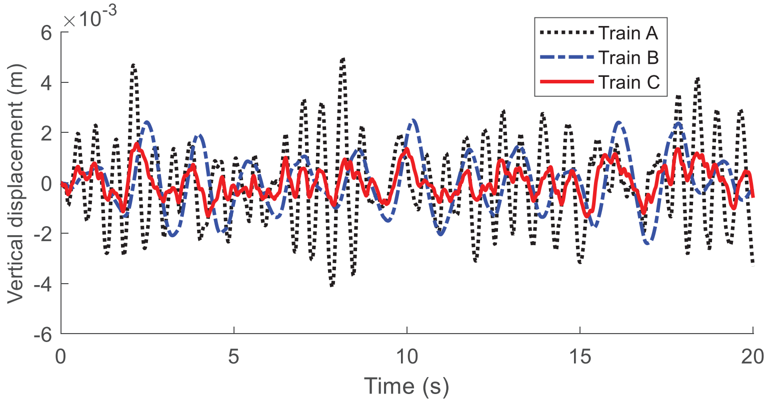

Figure 11: Body vertical displacement from random input

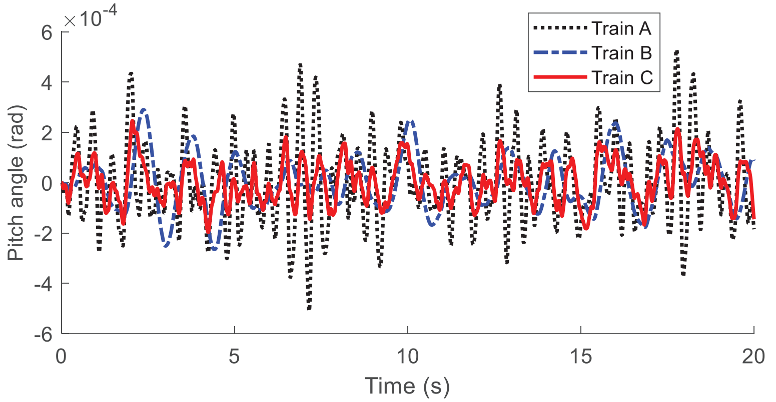

Figure 12: Body pitch angle from random input

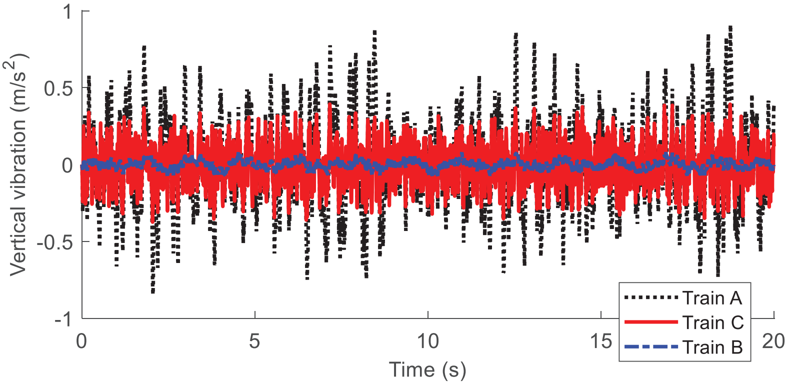

Figure 13: Body vertical vibration from random input

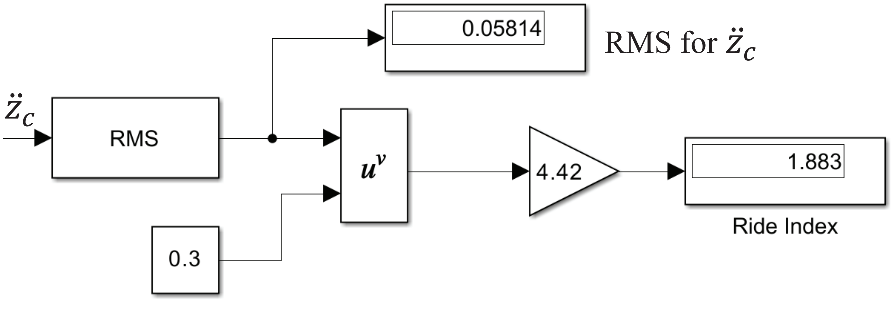

Figure 14: Ride Index blocks in MATLAB/Simulink model

Table 4: Body vertical displacement, z_c and pitch angle, θ_c

Input	Train	A		B		C	
		RMS	Ride Index	RMS	Ride Index	RMS	Ride Index
Step	z_c	<0.01		<0.01		<0.01	
	θ_c	<0.01		<0.01		<0.01	
	$\ddot{z}_c$	<0.01	0.013	<0.01	0.120	<0.01	<0.01
Sine	z_c	<0.01		<0.01		<0.01	
	θ_c	<0.01		<0.01		<0.01	
	$\ddot{z}_c$	0.058	1.883	0.024	1.441	0.1811	2.647
Random	z_c	<0.01		<0.01		<0.01	
	θ_c	<0.01		<0.01		<0.01	
	$\ddot{z}_c$	0.386	3.321	<0.01	1.060	0.1192	2.335

VI. Conclusion

The analysis of a train's ride index is crucial for reducing vibration and improving passenger comfort. The simulation findings reveal that the ride comfort of three types of trains (French Thalys, German ICE 3, and China CRH2) is evaluated under various sorts of inputs (step, sine and random). The Sperling index has been used in calculating the value for ride index performance with several inputs given to the systems. Finally, the results shows that Train B which is the German ICE 3 has the best ride index which is equivalent to ride comfort, quality and performance. The 6DOF half-body train model can further be improved with the implementation of active controls for ride index improvement.

VII. Acknowledgement

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