A Dedicated Control Design Methodology for Improved Tilting Train Performance

Hugo Magalhães[1,2(✉)], Pedro Antunes[1,2], João Pombo[1,2,3], and Jorge Ambrósio[2]

[1] Institute of Railway Research, School of Computing and Engineering, University of Huddersfield, Huddersfield, UK
hugomagalhaes@tecnico.ulisboa.pt
[2] LAETA, IDMEC, Instituto Superior Técnico, Universidade de Lisboa, Lisbon, Portugal
[3] ISEL, IPL, Lisbon, Portugal

Abstract. The development of detailed multibody models of railway vehicles is essential to address industrial problems through computational tools. The assessment of vehicle dynamic performance is one of the studies that can be performed with a multibody software. But when tilting trains are considered, which comprise active suspension elements, control engineering theories are required to estimate the forces developed by the actuators. Despite its importance, in general the details about the tilting control algorithm are unknown. In this work, a dedicated control design methodology is proposed to estimate the control algorithm of a tilting system in order to assure a proper vehicle performance. For this purpose, a detailed multibody model of a tilting train is used to perform a batch of simulations in order to develop an accurate linear model of the tilting system and to study its performance in realistic operation conditions. Thus, the traditional control techniques can be used to assess the tilting system dynamics and to design the control algorithm so that proper tilting performance is ensured. The control algorithm and the tilting performance are tested on a curved and tangent track with track irregularities. The comfort indexes P_{CT} and *RMS* are used here to assess the tilting system.

Keywords: Tilting train · Multibody formulation · Control engineering

1 Introduction

While conventional trains are characterized by passive suspension elements, such as springs and dampers, tilting trains are also characterized by active elements, such as actuators. The modelling of passive suspension elements is based on their mechanical properties, which are known or easy to determine, while the modelling of actuators requires the information of the control algorithm that determine their activation, which is commonly unknown due to confidential reasons. In order to include the actuation of the tilting system in the railway vehicle model, the control algorithm must be estimated, for example, by identifying a control algorithm based on standards that prescribe the

M. Klomp et al. (Eds.): IAVSD 2019, LNME, pp. 72–81, 2020.
https://doi.org/10.1007/978-3-030-38077-9_9

dynamic response of tilting trains [1, 2]. To enable the design of such control algorithm, a control design procedure must be carried out [3].

Tilting trains travel faster than conventional vehicles, decreasing the journey time [4]. During the curve negotiation, the transversal acceleration perceived by the passengers results from the balance between the centrifugal and gravitational accelerations, that is, the non-compensated acceleration (*NCA*) [5]. Since the tilting actuation allows to increase the roll angle of the carbody, the *NCA* is attenuated once the contribution of the gravitational acceleration increases. Nulling *NCA* is possible for a proper tiling angle, however, it has been shown that the full tilting compensation degrades the passengers' comfort due to the increase of the roll angular velocity, being preferable a partial tilting compensation of 60–70% [6]. In turn, the lateral actuation allows not only to reduce the lateral deflection of the secondary suspension, but also to attenuate lateral accelerations induced by the track irregularities. Thus, the tilting actuation tends to react preferably to accelerations developed during curve negotiations; however, in tangent segments, a roll motion of the carbody is verified due to the tilting actuation which deteriorates the comfort level in these segments. The ride quality in this scenario should degrade no more than 7.5% when compared to a conventional train [7].

Conflicting objectives exist in the design of the tilting and lateral actuation [8], namely (*i*) if the tilting actuation is designed to react rapidly, then it will react to track irregularities, even in tangent segments, which degrades the comfort levels of the passengers; (*ii*) the lateral actuation must be designed such that, the centering action must occur for low frequencies, however, the acceleration attenuation action must occur for the range of frequencies related to the track irregularities. To assess the tilting performance two case scenarios must be considered, namely, the negotiation of curved and tangent tracks, being determined comfort indexes in each case. For the curved track the P_{CT} is determined with the lateral acceleration and roll rate of the carbody [2]. For the tangent track with track irregularities, the root mean square (*RMS*) of the lateral acceleration of the carbody is determined [1]. The minimization of such indexes improves the tilting performance.

The governing equations of a multibody model, which are typically non-linear, consist of a set of differential algebraic equations. In the context of control engineering [3], a linear model representing the detailed multibody model of the vehicle is required for the control design [9]. Linear models of tilting trains had shown to be sufficient since their configuration do not vary significantly in its operating conditions [9]. Different approaches can be used to obtain such linear model, namely, by obtaining empirical model based on experimental results [3] or by linearization a multibody model of the tilting train [9]. Once the multibody model is described by a set of differential equations, a Taylor series expansion considering only one term must be used, for a selected configuration of the detailed vehicle model obtained in a given point of the operation [10], leading to the linear model defined by a set of linear differential equations that can be used in a control design procedure [3].

This paper proposes a detailed control design methodology to determine a proper control algorithm of the tilting system. The design of the controllers is performed based on the comprehensive analysis of the tilting system and its operating conditions. The linear model of the tilting system, which is used to support the control design procedure, is obtained from the linearization of the detailed multibody model of the tilting train.

2 Development of Linear Model

2.1 Multibody Model of the Tilting Train

Here, the multibody of the tilting train is used to perform batch of simulations of the vehicle negotiating curved, as the one shown in Fig. 1, and tangent tracks. These numerical results are used not only as reference results to validate the linear model, but also to study the operating conditions for which the tilting system must exhibit proper tilting performance.

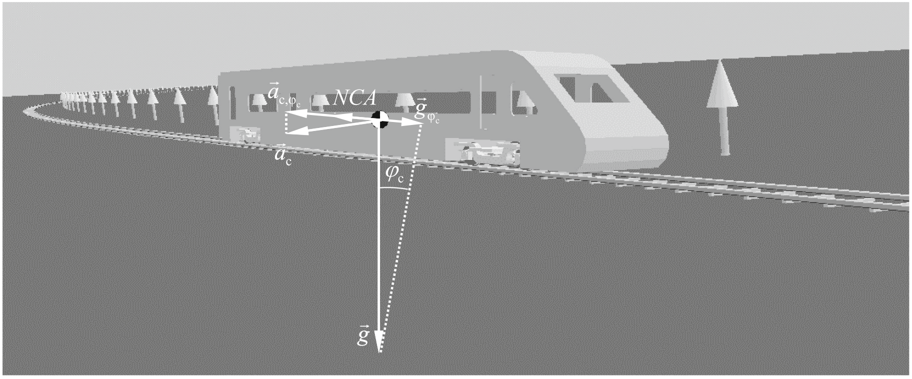

Fig. 1. Non-Compensated Acceleration (*NCA*) obtained from a multibody simulation.

2.2 Linear Model of the Tilting System

A schematic representation of the tilting system acting in the transversal plane of the vehicle longitudinal axis is shown in Fig. 2(a). The tilting system includes the bodies constrained by the tilting and lateral actuators, namely, the carbody and the bolster and bogie frame of the front and rear bogies, as shown in Fig. 2(b). The model of the tilting system is obtained from the linearization of the multibody model [10], in the plane of the cross-section of the vehicle, deemed as transversal plane. The dynamics of the tilting system is represented by the roll motion of the carbody and the lateral motion of the bolster, which are governed by the set of equations:

$$\left[a_{\ddot{\theta}}\ddot{\theta} + a_{\dot{\theta}}\dot{\theta} + a_{\theta}\theta\right] = [H_{\theta}] + \left[b_1^{\theta}d_1^{\theta} + \ldots + b_n^{\theta}d_n^{\theta}\right] \tag{1}$$

and

$$\left[a_{\ddot{y}_b}\ddot{y}_b + a_{\dot{y}_b}\dot{y}_b + a_{y_b}y_b\right] = \left[H_y\right] + \left[b_1^{y}d_1^{y} + \ldots + b_n^{y}d_n^{y}\right] \tag{2}$$

respectively. Here, θ and y_b are the tilting angle of the carbody and the lateral motion of the bolster, respectively, which are controlled variables; the coefficients a and b represent inertia, damping and stiffness properties; H_θ and H_y represent the moment

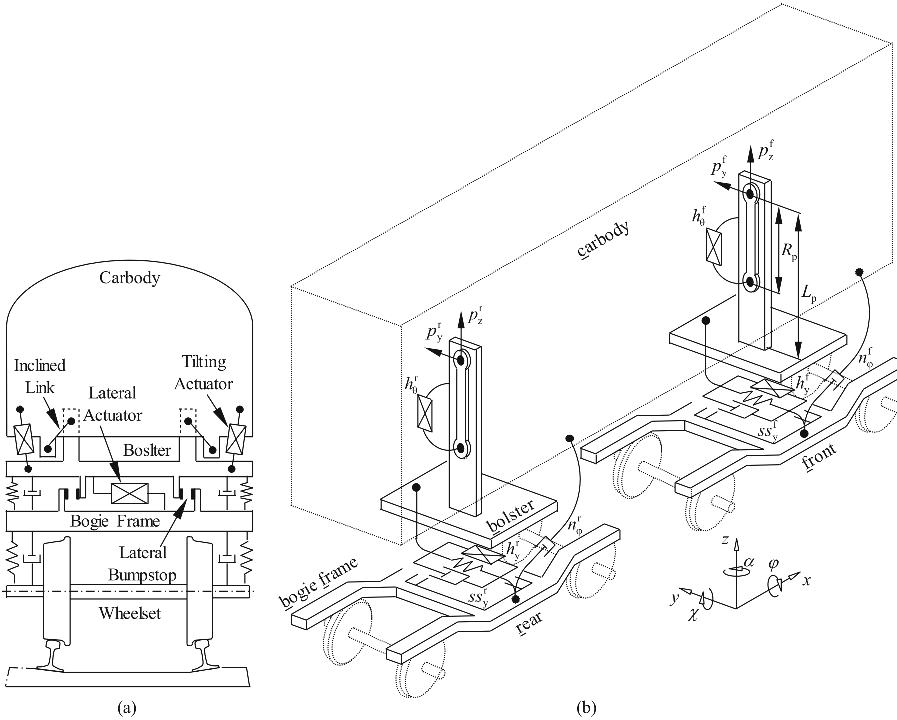

Fig. 2. (a) End view and (b) perspective view of a linear model of a tilting train.

applied in the carbody and the force applied in the bolster by the tilting and lateral actuators, respectively; and $d_1^{\sim}, \ldots, d_n^{\sim}$ are n disturbances of the roll motion of the carbody when the superscript $\sim$ is θ or n disturbances of the lateral motion of the bolster if the superscript $\sim$ is y. The disturbances of the tilting motion are states of the bolsters and bogie frames, such as, lateral, yaw and roll motions, whereas the disturbances of the bolster are states of the bogie frame.

2.3 Model Validation

The simplifications considered in the linearization process and the validation of the linear model are performed based on a batch of simulations where the multibody model of the tilting train operates at different load cases, leading to results of reference. All simulations consist of curve negotiations without actuation. Here, the multibody model of the vehicle is tested in a curved track at three velocities, $0.8v_0$, $1.0v_0$ and $1.2v_0$, being these simulations designated as C1, C2 and C3, respectively. Note that v_0 is the velocity that ensures no cant deficiency, that is, $NCA = 0$ m/s^2 [1, 5].

The simplifications performed in the linearization of the equations that govern the roll dynamics of the carbody and the lateral dynamics of the bolsters affect the model accuracy, for instance, in estimating *NCA*, written as:

$$NCA = \frac{\ddot{y}_b^f + \ddot{y}_b^r}{2} + g\left(\theta + \frac{\varphi_b^f}{2} + \frac{\varphi_b^r}{2}\right) \tag{3}$$

where g is the gravitation acceleration (9.81 m/s^2), $\ddot{y}$ refers to the lateral acceleration, φ refers to the roll angle whereas the subscript b refers to the bolster, while f and r refer to the front and rear bogie. Figure 3, shows that higher values of *NCA* are observed during the curve transition negotiation and it tends to zero in the curve segment, which represents natural nulling tilting. Here, the linear model also shows less accuracy for the simulation 'C3', as observed in Fig. 3, namely, the higher deviation is observed during the curve negotiation, although a reasonable agreement between the linear and the multibody models holds.

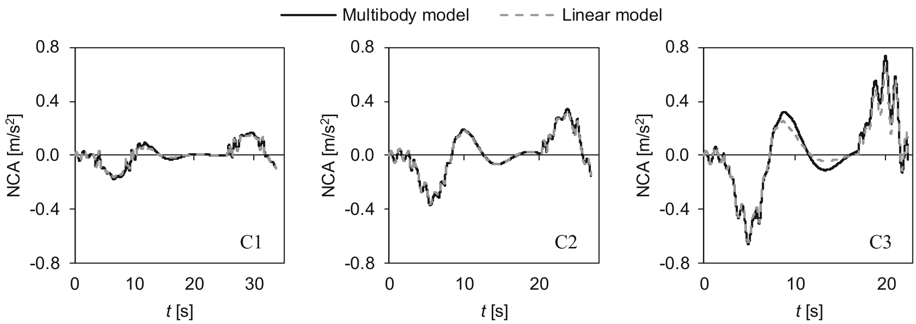

Fig. 3. *NCA* of the multibody model of the tilting train and of the linear model of the tilting system obtained from simulations 'C1', 'C2' and 'C3'.

3 Operating Conditions

3.1 Tangent Negotiations

A frequency analysis is performed to assess the effect of the track irregularities, which are the disturbances of the system, on the states that feed the controllers. For that, multibody simulations of the tilting train negotiating tangent tracks with and without track irregularities at 230 km/h, which is the maximum speed of this vehicle, are performed. From the fast Fourier transform of the lateral acceleration of the bolster 'ddy$_b$', which is one of the states that feed the controllers, it is shown in Fig. 4(a) that the track irregularities excites the tilting system at frequencies around 1.2 Hz, as it is observed in other works [9].

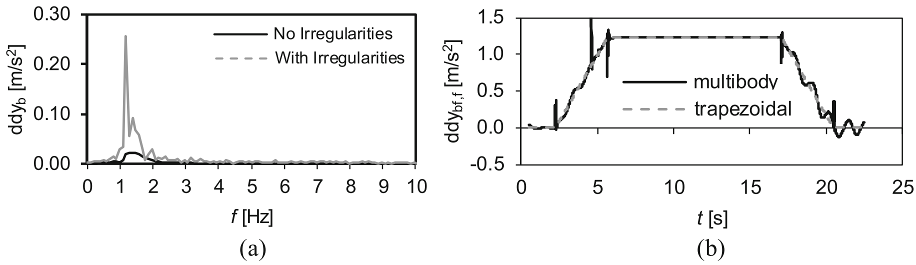

Fig. 4. (a) Fast Fourier transform of lateral acceleration of the bolster obtained from simulations where the vehicle negotiates a tangent track with and without track irregularities at v = 230 km/h. (b) Comparison between the lateral acceleration of the bogie frame of the tilting system obtained from simulation 'C3' and parameterized trapezoidal signals.

3.2 Curved Negotiations

In the batch of simulations where the tilting train is tested in curved negotiations, different track geometries and speeds have been considered. It has been found that the disturbances can be related to the track geometry, vehicle speed and distance between bogies. In particular the disturbances can be approximated to trapezoidal signals as depicted in Fig. 4(b), while the remaining disturbances obtained based on this assumption. For example, the disturbance that represents the lateral displacement of the bogie frame is obtained by double integration of the trapezoidal signal that defines its lateral acceleration. A good agreement between the disturbances obtained from simulation 'C3' and the approximated trapezoidal signal is observed in Fig. 4(b).

4 Design of the Control Algorithm

4.1 Dynamics of the Tilting System

The tilting system comprises three functionalities, namely, the control of the lateral deflection of the secondary suspension, attenuating the lateral acceleration of the bolsters induced by track irregularities and ensuring 70% of full tilt compensation in curve negotiations. To assure each of the three objectives, the three controllers $C_y(s)$, $C_{\ddot{y}}(s)$ and $C_\theta(s)$ are considered. Thus, three analyses of negative feedback closed-loop systems, as illustrated in Fig. 5, are performed to determine what controllers should be selected. In this preliminary analysis, the controllers are set $C_y(s) = C_{\ddot{y}}(s) = C_\theta(s) = 1$ and the disturbances are ignored, that is, $D(s) = 0$. Table 1 lists the controller $H(s)$, the system $G(s)$ and the system output $Y(s)$ for the three cases.

Figure 6 shows the step response of the three systems when excited by a step input $R(s) = 1/s$, where an underdamped motion behavior is observed in all systems. The stabilization time for the lateral displacement and lateral acceleration of the bolster are 1.7 s, while for the tilting motion the stabilization time is 15.1 s. In addition, the steady state error in all cases is approximately 1 which is the value of the reference $R(s)$. Thus, the controllers C_y and C_θ must be designed to reduce greatly the steady state error to

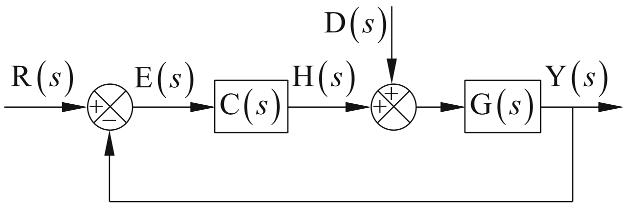

Fig. 5. Block diagram of a generic closed-loop system.

avoid exceeding the limits of the lateral motion of the bolster and tilting motion of the carbody and hence avoiding the bumpstop impacts. In turn, C_θ must also reduce the stabilization time up to 1–2 s, which is typical time for curve transition negotiations. To isolate the tilting actuation from the lateral acceleration induced by the track irregularities, $C_{\ddot{y}}(s)$ must attenuate the lateral accelerations of the bolster at frequencies around 1.2 Hz.

Table 1. Description of the block diagram for the three closed-loop systems.

Closed-loop system	G(s)	H(s)	Y(s)
Lateral displacement of the bolster	$G_y(s) = \frac{1}{a_{\ddot{y}_b}s^2 + a_{\dot{y}_b}s + a_{y_b}}$	$H_y(s)$	$Y_y(s)$
Lateral acceleration of the bolster	$G_{\ddot{y}}(s) = \frac{s^2}{a_{\ddot{y}_b}s^2 + a_{\dot{y}_b}s + a_{y_b}}$	$H_{\ddot{y}}(s)$	$Y_{\ddot{y}}(s)$
Tilting motion of the carbody	$G_\theta(s) = \frac{1}{a_{\ddot{\theta}}s^2 + a_{\dot{\theta}}s + a_\theta}$	$H_\theta(s)$	$Y_\theta(s)$

4.2 Control Design of the Controllers

The lateral controller $C_y(s)$ serves to reduce the lateral deflection of the secondary suspension during curve negotiations. Thus, it has been selected the low pass filter:

$$C_y(s) = \frac{K_y}{\left(1 + \frac{s}{2\pi f_y}\right)^{n_o}} \tag{4}$$

where K_y the gain of the controller, f_y the cut-off frequency and n_o is the order of the filter. Increasing K_y reduces the lateral deflection of the secondary suspension and hence avoid the bumpstop contact in curve negotiations. However, the increase of K_y is limited by the maximum force developed by the lateral actuators. In turn, f_y has been minimized so the controller only reacts to curve negotiations, but its value cannot be too low, otherwise, a large overshoot in the step response leads to a bumpstop impact. The order n_o of the filter allows to attenuate the reaction of the filter for higher frequencies, but it delays its response and, for $n_o > 3$ the system can show unstable behaviour.

To control the lateral acceleration of the bolsters due to track irregularities, a low filter of second order has been selected, being defined as:

$$\mathrm{C}_{\ddot{y}}(s) = \frac{K_{\ddot{y}}}{s^2 + 2\pi f_{\ddot{y}}\xi_{\ddot{y}}s + \left(2\pi f_{\ddot{y}}\right)^2} \tag{5}$$

where $K_{\ddot{y}}$, $f_{\ddot{y}}$ and $\xi_{\ddot{y}}$, respectively, are the gain, the natural frequency and the damping ratio of the controller. Since the track irregularities influence the bolster acceleration mainly at 1.2 Hz, then $f_{\ddot{y}}$ = 1.2 Hz. In turn, $K_{\ddot{y}}$ is increased and $\xi_{\ddot{y}}$ is minimized so the lateral actuation attenuates accelerations 1.2 Hz while not interfering with the other controllers $\mathrm{C}_{\theta}(s)$ and $\mathrm{C}_{y}(s)$.

The controller for the tilting angle consists of a PID and a low pass filter of first order written as:

$$\mathrm{C}_{\theta}(s) = \frac{K_{\mathrm{D}}s^2 + K_{\mathrm{P}}s + K_{\mathrm{I}}}{s}\left(1 + \frac{s}{2\pi f_{\theta}}\right)^{-1} \tag{6}$$

where K_{D}, K_{P} and K_{I} are gains of the PID, while f_{θ} is the cut-off frequency of the low pass filter. The PID serves to control track the tilting angle while the filter serves not only to attenuate the reaction of the tilting actuators to track irregularities, but also to minimize the interaction with the controller $\mathrm{C}_{\ddot{y}}(s)$. In the design of $\mathrm{C}_{\theta}(s)$, special attention is put in the maximum moment applied in curve negotiation scenario and on the settling time of the step response.

4.3 Tilting Performance

The indexes P_{CT} and *RMS* have been determined for the curved and tangent scenario, respectively. Table 2 lists these indexes and it is observed a significant reduction for the P_{CT}, while the *RMS* is slightly lower for the tilting system with actuation, being respected the restriction that the *RMS* should not increase more than 7.5% when actuation is considered [7].

Table 2. P_{CT} of the curve scenario and *RMS* of the tangent scenario.

Comfort index	No actuation	With actuation
P_{CT}	58.3%	35.0%
RMS	12.1%	11.3%

Regarding the quantity *NCA* obtained in the curve scenario shown in Fig. 7(a), it is observed not only that lower absolute values of *NCA* is observed for the model with actuation, but also that the passengers are exposed to a partial compensation since *NCA* tends to 0.7 $\mathrm{m/s}^2$. In the case of the tangent negotiation, the *NCA* shown in Fig. 7(b) is used to determine the comfort index *RMS*. From this plot, no clear differences can be observed, however, note that higher *RMS* is observed when for the system with no actuation, as listed in Table 2, meaning that a better performance is observed for the actuated system.

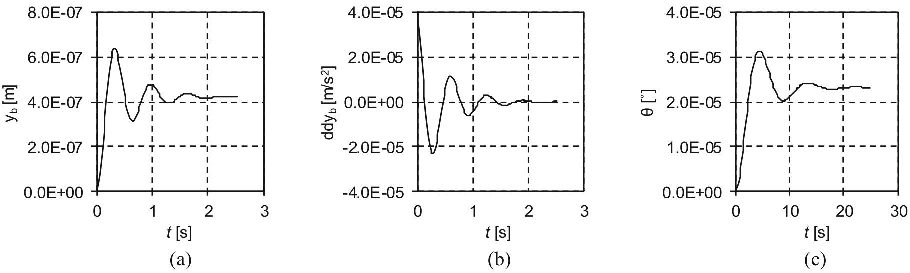

Fig. 6. Step response of the negative closed-loop systems: (a) G_y, (b) $G_{\ddot{y}}$ and (c) G_θ.

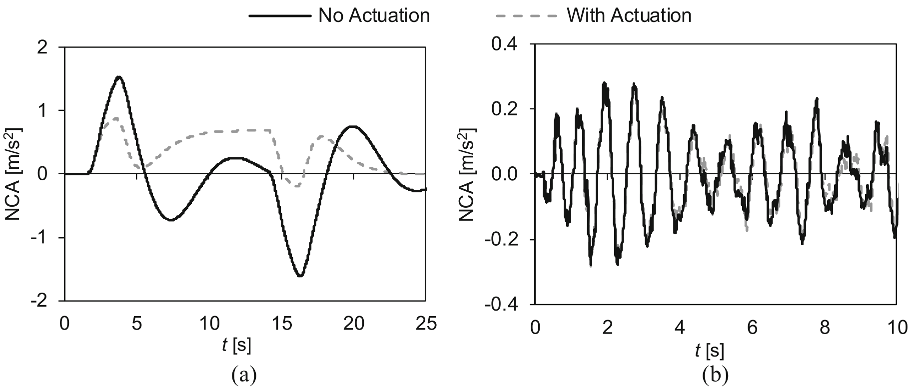

Fig. 7. *NCA* of the tilting system with and without actuation for (a) the curved and (b) tangent case scenarios.

5 Conclusions

The modelling of the actuation forces developed in the tilting system of the tilting train considered in this work required the design of the control algorithm that leads to proper tilting performance. This work led to the development of a robust and accurate linear model of the tilting system, an analysis of the disturbances that represent the range of operating conditions of the tilting train, and the design of the controllers so that the specified design specifications are met. The obtained control algorithm shows not only proper curve negotiations but also shows slightly better performance in tangent negotiation which is the opposite results comparing to some control algorithms [11, 12]. In addition, it has been found that the disturbances of the tilting system can be approximated by trapezoidal signals that are parameterized based on the track geometry, vehicle speed and the distance between bogies.

The next step, as future work, is to integrate the control model in the multibody model through co-simulation. Here, the tilting model receives the states of the multibody model that are used for the feedback, while the multibody receives the forces imposed by the control algorithm. If a similar and proper tilting performance is observed, then, the proposed control algorithm consists of an interesting industrial application, if not, the design of the control algorithm could be performed based on the tilting performance obtained from the multibody simulation rather than the linear model of the tilting system.

References

1. EN 14363: Railway applications - Testing for the acceptance of running characteristics of railway vehicles - Testing of running behaviour and stationary tests (2005)
2. ENV 12299:1999: Railway applications. Ride comfort for passengers. Measurement and evaluation (1999)
3. Ogata, K.: Modern Control Engineering. Prentice Hall PTR, Upper Saddle River (2010)
4. Iwnicki, S., Dahlberg, T.: Handbook of Railway Vehicle Dynamics, pp. 143–179. CRC (2006). https://doi.org/10.1201/9780849333217
5. Magalhaes, H., Madeira, J.F.A., Ambrósio, J., Pombo, J.: Railway vehicle performance optimisation using virtual homologation. Veh. Syst. Dyn. **3114**, 1–31 (2016). https://doi.org/10.1080/00423114.2016.1196821
6. Persson, R.: Tilting trains Enhanced benefits and strategies for less motion sickness (2011)
7. Goodall, R.M., Zolotas, A.C., Evans, J.: Assessment of the performance of tilt system controllers. In: Proceedings of the Railway Technology Conference IMechE, pp. 231–239. Citeseer (2000)
8. Zhou, N., Lv, Q., Yang, Y., Zhang, W.: <TPL-PCRUN> Statement of methods. Veh. Syst. Dyn. **53**, 380–391 (2014). https://doi.org/10.1080/00423114.2014.982136
9. Zhou, R., Zolotas, A., Goodall, R.: Robust system state estimation for active suspension control in high-speed tilting trains. Veh. Syst. Dyn. **52**, 355–369 (2014). https://doi.org/10.1080/00423114.2014.901540
10. Amirouche, F.M.L.: Fundamentals of multibody dynamics, 680 (2006). https://doi.org/10.1007/b137682
11. Zolotas, A.C., Goodall, R.M., Halikias, G.D.: Recent results in tilt control design and assessment of high-speed railway vehicles. Proc. Inst. Mech. Eng. Part F J. Rail Rapid Transit **221**, 291–321 (2007). https://doi.org/10.1243/0954409jrrt43
12. Hassan, F., Zolotas, A.C., Smith, T.: Optimized Ziegler-Nichols based PID control design for tilt suspensions. J. Eng. Sci. Technol. Rev. **10**, 17–24 (2017). https://doi.org/10.25103/jestr.105.02