Journal of Engineering and Technological Sciences

Research Paper

Curving Performance Analysis of a Freight Train Transporting 50-Meter-long Rail Using Multibody Dynamics Simulation

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Abstract

Long rails are normally used in highspeed railways to minimize the number of rail joints and the dynamic impact force that follows. However, transporting long rails using a freight train requires multiple wagons for each rail section, presenting potential safety and loading gauge issues, especially when going through curves. Thus, a safety assessment needs to be done prior to actual transport. Computational simulation can be used for preliminary assessment. Finite element analysis can be used to incorporate the flexibility of the rails into the analysis but requires significant manpower and computer power to perform. In this study, an alternative method to model rail flexibility using a multibody approach is presented. The rails are sectioned into multiple rigid bodies along their length and interconnected using rotational joints. The stiffness coefficient of the joints is defined as a function of the actual rail's physical properties. This modelling technique results in a simplified multibody model that retains the original rail elastic properties. Simulations of the constructed rail model hauled using a freight train were done and the results were compared to on-track test measurements of the same configuration. The comparison generally showed good agreement, showing this modelling technique's ability and accuracy to simulate the case.

Keywords: *finite element; flexible body; freight; multibody simulation; railway.*

Introduction

Vehicle-track dynamic interaction is amplified as train speed increases. Highspeed train wheels are subjected to a large dynamic impact force every time they pass a rail joint or discontinuity [1-3]. In addition, the presence of joints also impacts rail irregularities over that track section. For this reason, long rails or continuous welded rails are normally used in highspeed railway construction in place of short ones [4]. This trend continues to have a profound presence in newly built highspeed railways. The Jakarta-Bandung highspeed railway currently under construction in Indonesia conforms to the norm, using 50-meter-long rails.

The rails for Jakarta-Bandung highspeed railway construction are shipped to Cilacap harbor by sea and then hauled to Bandung using a freight train. The railway route from Cilacap to Bandung includes a mountainous section with numerous small-radius curves and multiple turnouts. The smallest curve radius on the route is 150 meters.

Transporting long rails on twisty mountainous routes is logistically challenging. Due to their length, four flat wagons are needed to transport each one. The rails are laid on flat wagons and are constrained by stoppers at either side to limit load movement. Constraining a single load to four wagons presents a potential safety and loading gauge issue, especially when going through curves. During curving, the rails are bent as the train is navigating the curve. This bending results in outward forces acting on the wagon at both ends of the rails and inward forces acting on the wagon at the midpoint of the rails. These forces acting on the wagons could potentially lead to derailment of the train, while also presenting track shift issues due to the large lateral force acting on the wheel-rail contact points.

Safety against derailment must be assessed prior to actual transport runs. This is normally done based on the value of derailment coefficient Y/Q at the most critical point of the operating route—usually the smallest radius curve [5,6]. An on-track test run will yield the most accurate result. However, on-track testing poses a degree of

ISSN: 2337-5779

DOI: 10.5614/j.eng.technol.sci.2023.55.2.8

safety issue and requires substantial time and cost to undertake. Computational simulation can be done prior to on-track testing as a preliminary study to assess the feasibility of such a transport.

Multibody simulation, wherein the objects are modelled as interconnected bodies, can be used to conduct the assessment. The finite element method is favorable for analyzing a bent rail due to its similarity to a beam being deflected by forces being applied to it. On the other hand, integrating an elastic finite element model of rails into a multibody model of the freight train for the simulation is both labor intensive and computationally expensive, as it constitutes a two-step process of finite element modal analysis and multibody dynamics analysis with an added modal parameter [7]. Modelling the rails directly as multibody system circumvents the need for a prior finite element analysis. However, to maintain the merit of the simulation-based assessment, the multibody model of the rails should be constructed in such a way that it represents all of the rail's fundamental characteristics, including its elasticity.

This study aimed to demonstrate the use of a multibody simulation for a simplified preliminary safety assessment of a freight train carrying 50-meter-long rails through small radius curves.

In the present work, the simplified modelling technique of long rails was demonstrated by modelling it as a multibody system. In the model, the rails are sectioned along their length into multiple rigid bodies that are interconnected by elastic joints. The joint stiffness used in the model is a function of the rail's modulus of elasticity, moment of inertia, and length, derived from the cantilever beam deflection equation. This way, the multibody model of the rail preserves the elastic properties of the rail beam without having to resort to finite element modelling. The rail model was integrated into a freight train model of four flat wagons to form the final model. The final model was then simulated to run on a small radius curve at different operating speeds to assess the safety of such haulage. Safety against derailment for the cases are assessed on wheel-rail forces, with additional recorded variables of flatbed accelerations and the force the hauled rails inflict on the stopper frames. The simulation results are provided, along with a comparison between them and on-track test measurements.

The rest of this paper is arranged as follow: Section 2 contains a detailed explanation of the modelling of 50-meter-long rails and a freight train as multibody models, as well as the simulation setup of the freight train-hauled rail system going through a small radius curve. The safety assessment of the haulage based on the simulation results and its comparison with the on-track test measurements are discussed in Section 3. The concluding remarks in Section 4 end the paper.

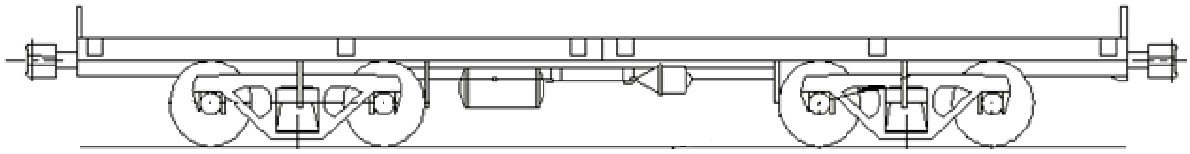
Multibody Simulation of Long Rail Haulage

Simulation Object

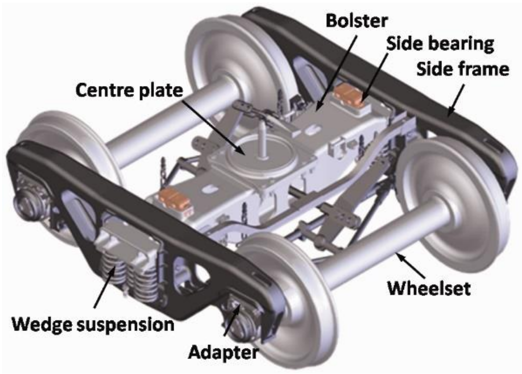
The rails for the Jakarta-Bandung highspeed railway will be transported from Cilacap to Bandung using a freight train. The freight wagons have three-piece bogies with a single-level suspension system. The suspension system is made up of a cluster of coil springs and wedge dampers between the bogie frames and the bolster. Figure 1 shows an illustration of the flat wagon and three-piece bogie. The wagon features a flatbed with a length of 13.8 meters. To accommodate 50-meter-long rails, four flat wagons are used to transport them. The rails are laid on the wagon flatbed in a ten abreast configuration. The rails are constrained in the lateral direction by two stopper frames on each wagon.

For the simulation, freight train and rails were modelled as a multibody system. It consisted of four subsystems of the flat wagon and a subsystem of the rails. Each of the four wagon subsystems was connected to each other representing wagon coupling. The rail subsystem was then attached to every wagon to represent the load constraint by the stopper frames. The hierarchy of the multibody model is shown in Figure 2. The detailed modelling setup for the train and the rails is explained in the following sections.

DOI: 10.5614/j.eng.technol.sci.2023.55.8



(a)



(b)

Figure 1 Flat wagon illustration: (a) wagon [8], (b) three-piece bogie [9].

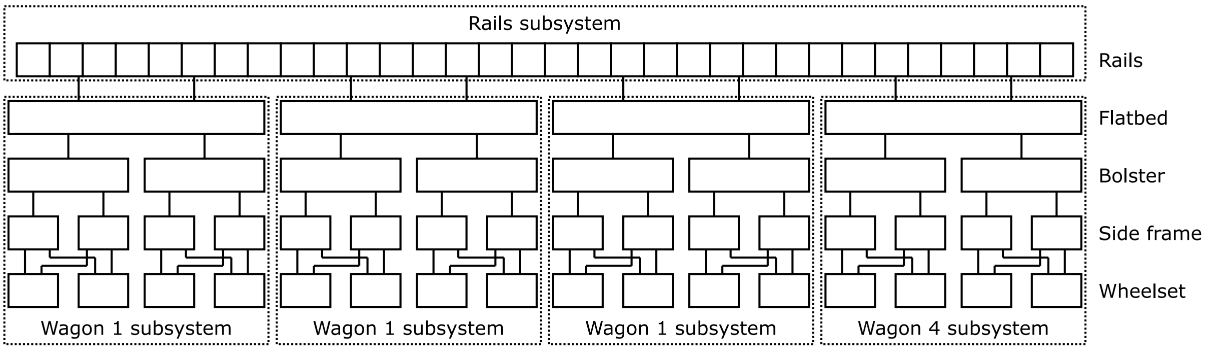


Figure 2 Model hierarchy of freight train and rail multibody system.

Freight Train Model

The freight wagons were modelled as a multibody system comprising eleven interconnected bodies. The bodies represent four wheelsets with axle boxes, four bogie side frames, two bolsters, and a flatbed with stopper frames. As shown in Figure 1, wheelsets form the base of the model. Then, the bogie frames were connected directly to the wheelsets, the bolster to the frames, and the flatbed to the bolster.

The bogie side frames in the model are connected directly to the axle boxes, as this type of bogie does not have primary suspension. Vertical support of the bogie side frames was modelled as a contact force element between the frame and the wheelset axle box they sit on, while the clearance in the longitudinal and lateral direction was modelled as a nonlinear force element with stiffness parameters representing a bump stop. There are two side frames in each bogie, positioned on either side of the wheelset with a bolster bridging the two.

A cluster of coil springs and a pair of friction wedges make up the central suspension of the bogie, connecting the frames with the bolster, as shown in Figure 3. Coil springs provide an elastic connection between the bolster and the bogie side frames in the vertical direction. In the model, the coil spring is represented by a linear force element. Meanwhile, the wedge pair provides friction damping in vertical lateral directions and was modelled as a friction force element. Working together, the coil spring cluster and friction wedges resist asymmetrical loads and holds the bogie frame square in-plane.

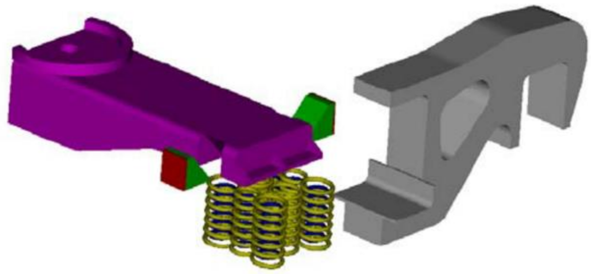


Figure 3 Coil springs and friction wedge of the three-piece bogie [10].

A flatbed was then introduced to complete the wagon model. It was connected to the two bogies' bolsters using a center plate. The center plate is represented in the model by a force element having stiffness and damping in six directions, three translational and three rotational, and a contact force element in the vertical direction. In addition to the center plate, side bearers were placed on the bolster to limit the flatbed's movement in relation to the bogie. They were modelled as contact force elements with predefined gaps to simulate the vertical clearance. On the flatbed, stopper frames for the rail were modelled as four simple posts, due to the detailed design of the frame not being available yet during modelling and simulation. They were positioned ten feet away from the center of the flatbed in the longitudinal direction and forty millimeters to the side of the outermost rail in the lateral direction. A schematic of the stopper frame is shown in Figure 4.

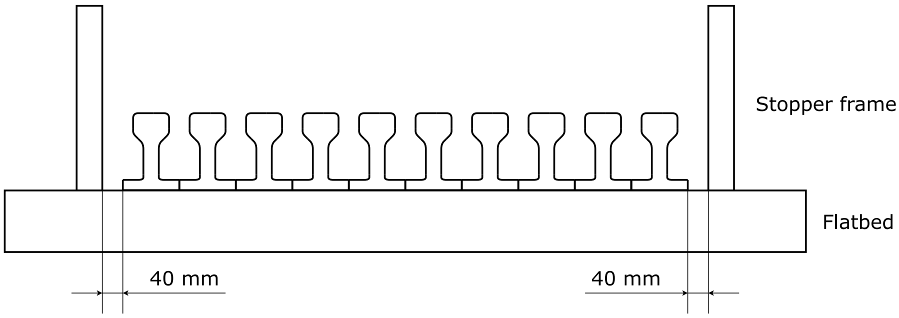


Figure 4 Schematic of stopper frame for hauling rails.

Long Rail Model

Ten 50-meter-long rails were modelled as a multibody system with elastic joints. In the model, the group of rails was sectioned along its length into 32 interconnected rigid bodies. Each body represented a group of ten 1.5625-meter-long rails in a ten abreast configuration. It was connected to the bodies next to it using a rotational joint on the vertical axis of a predefined stiffness coefficient. A schematic of the long rail model is shown in Figure 5.

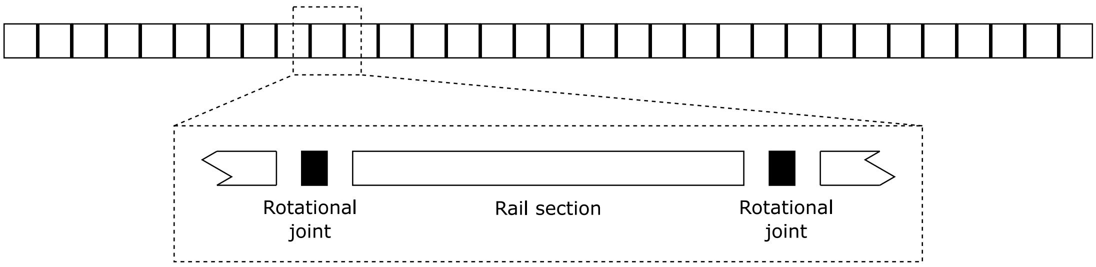


Figure 5 Long rail model schematic.

DOI: 10.5614/j.eng.technol.sci.2023.55.8

The stiffness coefficient for the rotational joints was derived from the deflection equation of a cantilever beam. The deflection of a cantilever beam is a function of its modulus of elasticity, moment of inertia, length, and loading condition. Angular deflection of a cantilever beam with a length L subjected to a moment of M is governed by Eq. (1) [11]:

$$\theta = \frac{ML}{EI} \tag{1}$$

Angular bending stiffness of the beam, which is defined by the applied moment divided by the resulting angular deflection, can then be obtained. By rearranging Eq. (1), the angular stiffness coefficient for the rotational joints of the beam model is described in Eq. (2):

$$\frac{M}{\theta} = \frac{EI}{L} \tag{2}$$

By defining the model's joints stiffness coefficient as a function of the actual rail's physical properties, the connection between each section in the model represents the bending properties of the rails being modelled. Combined with adequate sectioning of the rail along its length, the rail model acts as an elastic body of 50-meter-long rails. Instead of the two-step process of finite element modal analysis and modal incorporation into the multibody model, this modelling technique simplifies the modelling process of long rails by modelling it directly as a multibody system without losing its elastic properties.

The rail model was then integrated into the freight train model as a subsystem. The rail multibody model was placed on top of the wagon models. To attach the rails to the wagons, force elements with linear stiffness and damping coefficient in vertical, roll, and yaw directions were used. One force element was used per body of a rail section. Contact force elements with limited clearance were then added between the posts on the flatbed and the rails to model rail lateral movement constraint. The rails are held in the longitudinal direction at one of their ends, constraining their behavior to that of a cantilever beam. In this analysis, friction between the feet of the rails and between the rails and the flatbed was neglected.

Equation of Motion

For the multibody system, which involves constraints such as the rail vehicle and track system, the equation of motion is written in a set of differential and algebraic equations as presented in Eq. (3) [12]:

$$\left.\begin{aligned} M\ddot{q} + C_q^T\lambda = Q_e + Q_v \\ C(q,t) = 0 \end{aligned}\right\} \tag{3}$$

M is the mass matrix of the system, ***C*** is the vector of the system constraint, ***q*** is the vector of generalized coordinates, $\boldsymbol{\lambda}$ is the vector of the Lagrange multiplier, $\boldsymbol{Q_e}$ is the vector of externally applied force, and $\boldsymbol{Q_v}$ is the vector of inertia forces, which is quadratic to the velocity that arises from differentiating the kinetic energy with respect to time and the system's generalized coordinates. The external forces applied to the bodies in the system include the suspension forces, the wheel-rail contact forces, and the traction and braking forces. The vector of system constraints describes the mechanical joints and track trajectory or curved track.

Modelling the long rails as a multiple rigid body system with elastic joints will increase the size of the mass matrix as well as the generalized coordinates vector of the rigid bodies involved in the system. However, it eliminates the modal mass matrix and its associated modal stiffness matrix as well as the elastic coordinates of the deformable bodies, which are more complex in nature and have to be calculated using external finite element analysis software prior to the multibody dynamic simulation. Thus, overall, it can reduce the time and resources needed to perform the analysis.

Simulation Cases

Multibody simulations of the model were set up to assess the freight train's safety against derailment throughout its way from Cilacap to Bandung. The case chosen for simulation was a curve of 150-meter radius, representing the smallest radius curve found on its planned route. This is considered a critical point due to its nature of inflicting large lateral wheel force, increasing the risk of derailment. The simulations were done using a commercial multibody simulation software package.

The simulations were done in two phases, baseline simulation and real scenario simulations. The baseline simulations consisted of a simulation run of an empty freight train model and a simulation run of a loaded freight train model. The track was set to have fully even microgeometry features on both baseline simulations to focus on the effect of the curve. After the baseline simulations had been completed, simulation scenarios with the inclusion of track irregularities were done.

The macrogeometry of the curves used in the simulation took cues from real curves found on the planned route. Transition curve length, cant, and gauge widening were mirrored from real curves, whereas the constant radius curve length was set to accommodate the whole freight train in the simulation. The transition curve length for the 150 m-radius curve was 42.5 meters long, the cant was 85 mm high, and the gauge was widened by 20 mm from its nominal value of 1,067 mm. The constant radius curve length was set at 50 meters in the simulation.

The microgeometry of the track used in the haulage scenario simulations was set to represent rail irregularities in the actual track. Rail irregularities amplify dynamic forces between vehicle wheel and rail, which increases the risk of derailment [13,14]. The software's template microgeometry file UIC Bad was used in the simulation. Its standard deviation and peak value are listed in Table 1. A track twist was added at the entry transition curve section. The twist was configured based on GMRT 2141 standard. It had a semispan of six meters with a grade of 1:150 [15].

Table 1 Microgeometry data of UIC bad template file.

Orientation	Standard deviation (mm)	Peak (mm)
Lateral	4.95	15.00
Vertical	6.57	23.75

The safety against derailment was assessed on the contact forces between vehicle wheel and rail. The derailment safety variable used in this assessment was the derailment coefficient Y/Q of the leading wheelset of Wagon 1 and Wagon 3. Meanwhile, the stability was assessed on the lateral and vertical acceleration of the flatbed above the leading bogie of Wagons 1 and 3. In addition to that, the variable related to the train's load, namely the lateral force acting on the stopper frames, were also recorded. Wagons 1 and 3 were chosen, as they best represented the front end and middle point of the wagon-hauled rail system. The train model was simulated to run at a speed of 15, 20, 25, and 30 km/h through the curve to determine the acceptable speed at which the real haulage should operate later.

Safety Assessment of Long Rail Haulage

Simulation Results

The derailment coefficient of the leading wheelset of Wagons 1 and 3, the vertical and lateral acceleration on the flatbed above the leading bogie of the same wagons, and the lateral force acting on the stopper frames were obtained from simulation. In general, the results showed a good safety level of the freight train-hauled rail model. The results for each simulation case are detailed in the following.

An empty freight train model was used in the first simulation to generate the baseline value of the simulation variables. The simulation showed that the model could run through a 150-m curve safely at 15 km/h. This was attested by the derailment coefficient graph of the leading wheelset of the leading wagon in Figure 6. Derailment coefficient Y/Q is a ratio of the lateral force to the vertical force acting wheel-rail contact. Absolute force values were used in the calculation. A large derailment coefficient represents a higher risk of derailment. The derailment coefficient of the leading wheelset of Wagon 1 rose from around zero to a maximum value of 0.36 for the inner wheel and 0.39 for the outer wheel. These values are below the maximum acceptable value of 1.2 [16]. There was only a minor difference between the derailment coefficient of the outer and the inner wheel. This was due to the freight train model being empty and running at low speed, resulting in no substantial lateral load shift on the freight train.

DOI: 10.5614/j.eng.technol.sci.2023.55.8

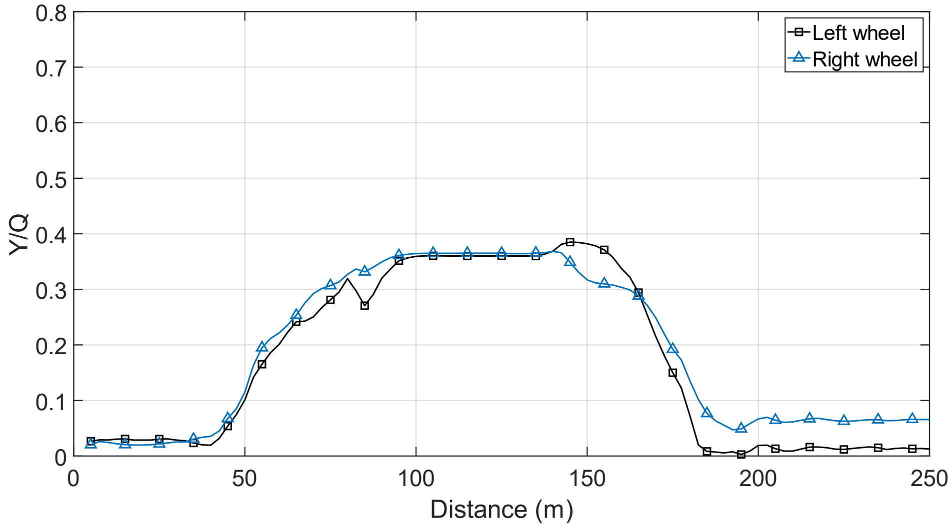


Figure 6 Derailment coefficient of empty freight train model through 150-m curve at 15 km/h.

The stability assessment variables and the lateral and vertical acceleration on the flatbed showed that the freight train model was stable during curving. The lateral acceleration value was recorded to reach a maximum value of 0.81 m/s^2 during the simulation run, lower than the maximum allowable value of 4.91 m/s^2. The vertical acceleration recorded during the simulation run was also within the safety limit, reaching 0.20 m/s^2 with a maximum allowable value of 6.87 m/s^2.

The next simulation case was of a loaded freight train model running through a smooth 150-meter radius curve without track irregularities. This case was simulated to assess how the hauled long rails affect the safety against derailment. During the curving process, the rails bent in conformance to the curve shown in Figure 7. The bend was initiated by the contact force between the rails and the stopper frames along the train. However, the rails were observed trying to maintain their straightness, bending only as far as necessary. The rails can be seen curving with a larger diameter than the train, due to the gap between the rails and the support columns. This translates to the rails pushing the end wagons outward and the middle wagons inward. The maximum value of lateral contact force acting on the leading stopper frames on Wagon 1 and Wagon 3 were observed to be 639.7 kN outward and 358.4 kN inward, respectively.

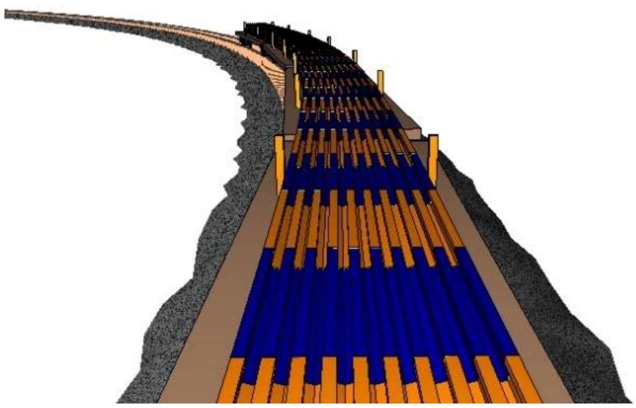


Figure 7 Visualization of freight-train-hauled rail model running through curved track.

The lateral force from the hauled rails working on the wagons shifts the weight balance between the left and the right wheels. During curving, the left (outer) wheels gain weight from the hauled rails, while the right (inner) wheels lose weight, as shown in Figure 8. Its safety against derailment is in turn affected by the shift. However, the simulation result showed that the derailment coefficient of the leading wheelset (Figure 9) and the bogie frame lateral force (Figure 10) were still within the acceptable limit. The maximum value of the derailment coefficient was recorded at 0.44 at the leading outer wheel of Wagon 1.

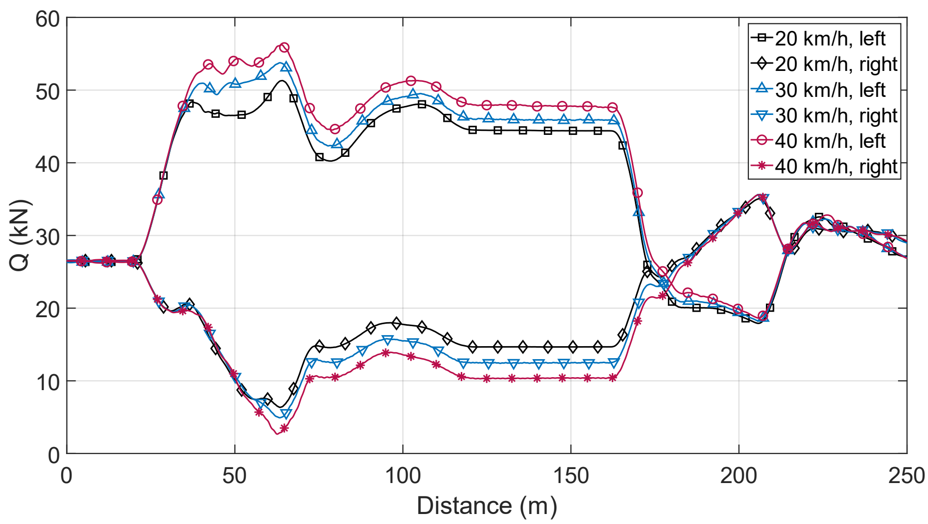


Figure 8 Vertical wheel force of loaded freight train model running through 150-m curve.

The simulations of the haulage scenario, wherein microgeometry defects and track twist were added, showed an expectedly increased value of the derailment safety variables. The 150-m radius curve track run at 30 km/h resulted in a maximum derailment coefficient of 0.70 at the leading outer wheel of Wagon 1. The lateral bogie frame force increased to a maximum value of 18.1 kN. The increase was caused by irregularities along the track and the track twist placed at the exit transition curve, which results in fluctuation of the vertical and lateral wheel forces due to the disrupted wheel-rail contact.

In general, the derailment coefficient and bogie frame force rise as the speed increases, as shown in Figures 9 and 10. However, a vehicle's response to track irregularities is nonlinear and speed-dependent due to the influence of the track irregularity wavelength and the vehicle running gear's geometry and parameters. This may lead to some unexpected results such as the sudden surge around the 200 m mark in Figure 11, where the simulation at 30 km/h yielded a higher derailment coefficient compared to the one at 40 km/h. While the increase means it is potentially more prone to derailment, it is still within an acceptable safety level.

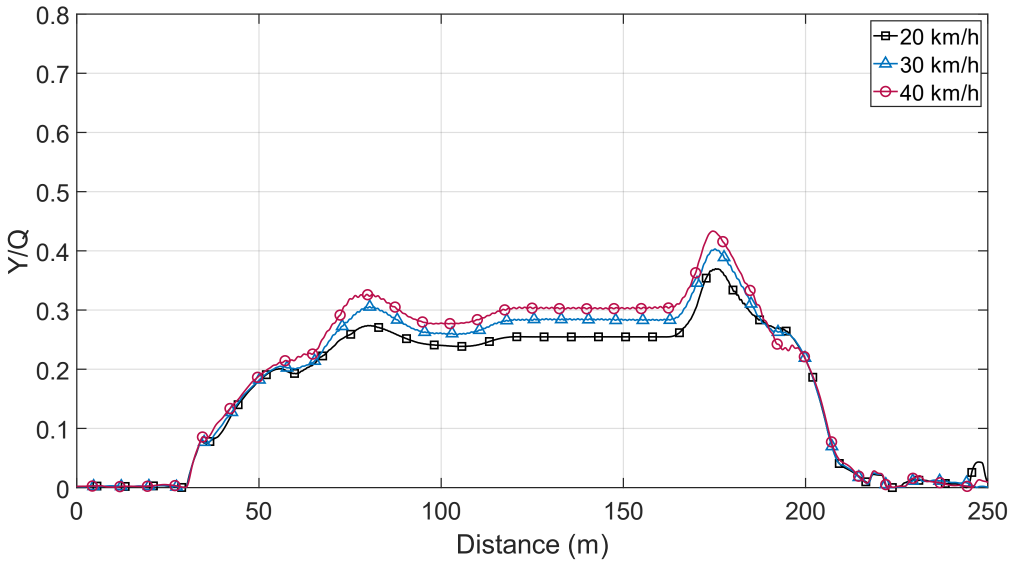


Figure 9 Derailment coefficient of loaded freight train model running through 150-m curve.

DOI: 10.5614/j.eng.technol.sci.2023.55.8

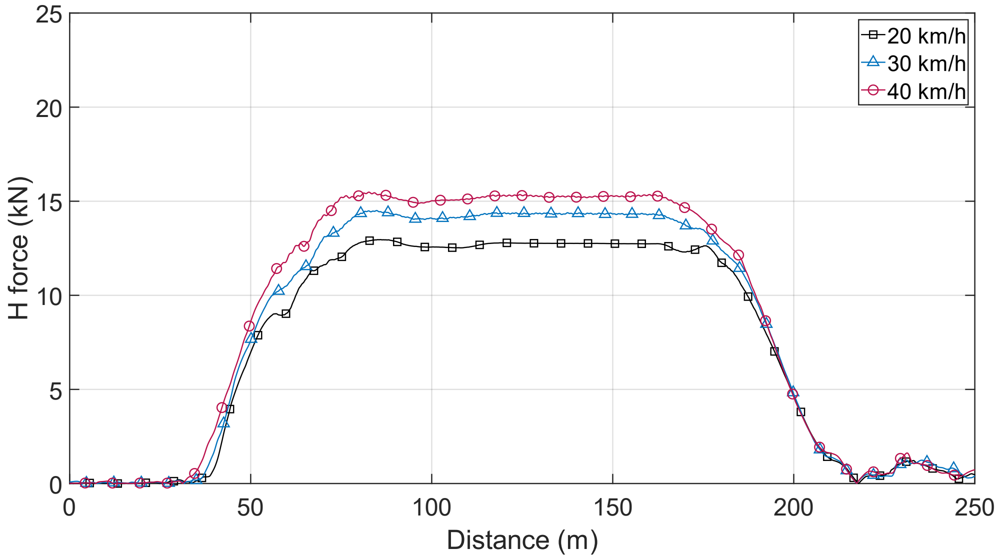


Figure 10 Lateral bogie frame force of loaded freight train model running through 150-m curve.

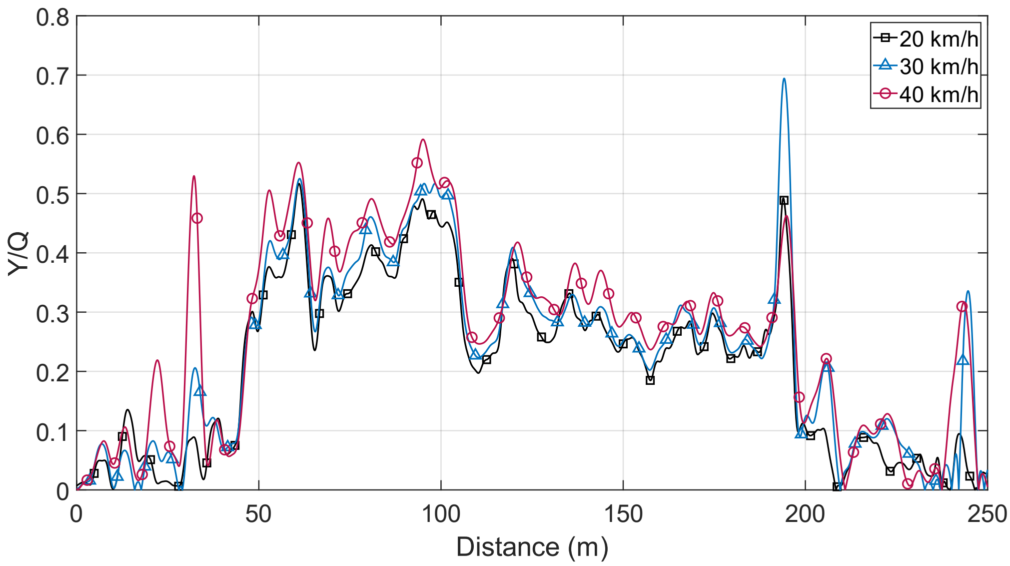


Figure 11 Derailment coefficient of loaded freight train model running through 150-m curve with irregularities.

Comparison between Simulation Result and On-Track Test Measurement

Following the preliminary safety assessment using a multibody simulation, an on-track test run was carried out by Indonesian Railways supported by the China Academy of Railway Sciences Corporation Limited. Ten pieces of 50-meter-long rails were transported using a freight train from Maos Station in Cilacap to Gedebage Station in Bandung. The rails were secured by two stoppers on each wagon along four wagons in total. This configuration was identical to the constructed multibody model. The test route included multiple tight curves, as small as 150 m in radius. A picture of the test train hauling long rails through a small-radius curve is shown in Figure 12, showing the rails bending as predicted by the simulation. The freight train featured instrumented wheelsets located at the first axle of Wagons 1, 3, and 4 as well as accelerometers located on the flatbed above the bogie pivot, measuring lateral and vertical directions. Measurements from the on-track test run were used for comparison with the simulation result of the same running condition. However, the comparison was limited to the values of the variables outlined in the published test report.

Figure 12 On-track test of long rail transport through a small-radius curve.

A comparison was made between the derailment coefficient in the simulation and in the on-track test run. Table 2 contains the comparison results between the simulation results and the on-track test measurements at 30 km/h [17]. The comparison showed that the simulation results were close to the on-track test measurements. The average error across the variables was 17.3%. The error could be a result of uncertainty in the condition of the track geometry and the rail profile along the traversed route, and the wheel profile of the train used. The track and the train used for the on-track test were extensively used and thus potentially worn in certain areas. The error might have also come from the difference between the actual train suspension parameters and the model parameters. Suspension parameters, especially ones related to yaw motion, affect the train's safety against derailment as well as its stability.

Table 2 Simulation results and on-track test measurements at 30 km/h.

Variable	Maximum value	
	Simulation	On-track test
Derailment coefficient, Y/Q	0.70	0.73
Lateral flatbed acceleration (m/s^2)	1.16	1.38
Vertical flatbed acceleration (m/s^2)	1.63	2.39

Conclusion

In this paper, a simplified method for modelling long rails as a multibody system was demonstrated. The group of rails was sectioned along its length into multiple rigid bodies interconnected using rotational joints. Physical properties of the rail were used in defining the rotational joint stiffness of the model, resulting in a good representation of the rail's elasticity in the model. Simulation of the freight train model hauling said long-rail model exhibited comparable results to the on-track test with the same train-load configuration. The simulation results were within 18% of the on-track test results, with the error likely coming from uncalculated differences in actual track irregularities, wheel-rail contact parameters, and component wear and friction.

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Manuscript Received: 18 November 2022
Revised Manuscript Received: 16 March 2023
Accepted Manuscript: 11 May 2023