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Multibody Study of a Repoint Stub Switch with Passive Locking Feature

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Abstract

The Repoint joint is an innovative switch actuation concept where the rail design is inspired from the expansion joint. Because it is intended to operate as a track switch, the rails are redesigned to accommodate easy movement and a passive locking feature. This article examines the multi-body dynamic simulation model of this joint together with its foundation. The interlock feature of this joint is implemented in two variants, which differ only in its theoretical detail. The Manchester benchmark vehicle is run through it. The forces and displacement of the rail elements are similar in both variants.

Keywords: multi-body dynamic simulation, track switch, expansion joint, REPOINT switch, vehicle-track interaction, SIMPACK

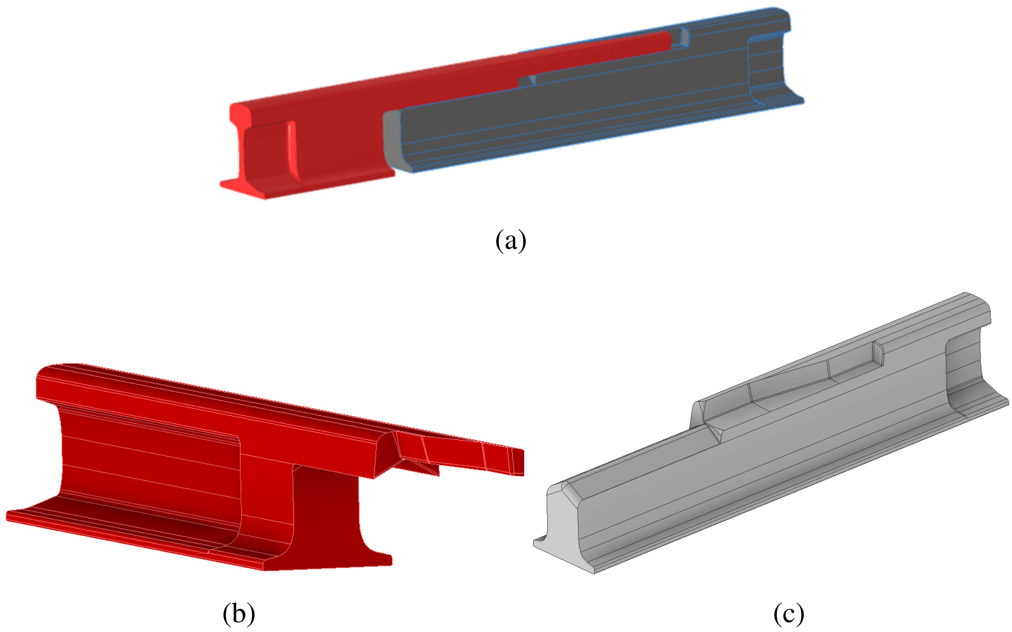


Figure 1: CAD diagram of (a) complete Repoint joint (b) switch/movable rail (c) stock/fixed rail

1 Introduction

The Repoint joint is conceptualised in [1] and [2] presenting an unique lift and drop motion for switching tracks for a train. The design of the Repoint stub switch joint which is shown in figure 1 is engineered to present a rail profile to the passing vehicle that closely resembles existing expansion joints. The length of the Repoint joint is about 0.4m including the expansion gaps. Similar to an expansion joint there is a lateral overlap of rails to compensate for axial deflection of rails due to thermal effects. Apart from it the Repoint design have a modified rail web section. At this rail web, a passive locking feature is introduced. This leads to an additional vertical overlap of the rails.

There were many simulation studies on this joint in the past with various focal points. The actuation system and control loops were studied, where the rails are simulated as elastic elements [3] using co-simulation techniques. In another study on vehicle-track interaction [4], the mechanical properties of the switch were analysed subjected to vehicle forces. However these investigations did not account for the track foundation, assuming an infinitely stiff track bed.

This article presents the effect of the passive lock as this special feature of this rail joint was not studied earlier. It imposes unique contact conditions between the rails which are important to understand the behaviour of track dynamics.

The Repoint joint design and modelling conditions in section 2 are followed by the experimental setup in simulation is described in section 3. The results of the

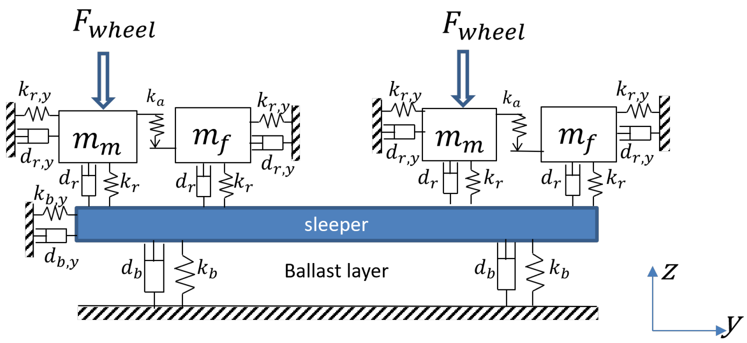


Figure 2: Multi-body dynamic model of track foundation with Repoint joint

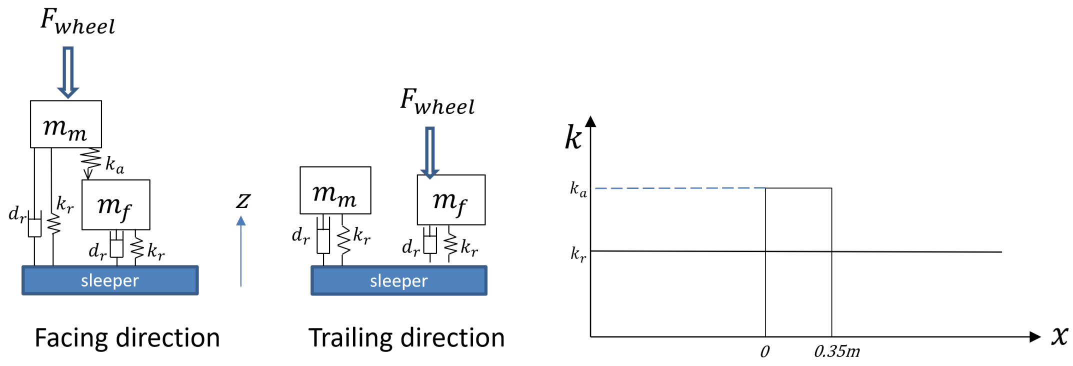
vehicle-track interaction in section 4. The observations are discussed and concluded in section 5.

2 Repoint joint design

Referring to the leading running direction of the wheel in figure 4, the rail on the left side is the movable element (figure 1b). As the wheelset rolls it transfers the load to the fixed rail (figure 1c). In between the wheel is in contact with both the rails where it gradually transfers the load from one to other. Another characteristic of this joint is the connection between these rail bodies, which overlap laterally and vertically. In a conventional expansion joint, the switch and stock rail bodies laterally overlap each other. However in the Repoint joint the web of the rail sections accommodate an interlocking mechanism as a V-shaped groove. This passive locking prevents lateral movement of the rail bodies when the actuators and rails are in the down and locked position.

The multi-body dynamic model is shown in figure 2 whose parameters are taken from [5]. In plain track conditions, the force on the wheel acts solely on either of the rail masses (either m_f or m_m). The subscript f refers to fixed or stock rail and m denotes movable or switch rail. The railpads connect the rails to the sleepers, identified as elements k_r and d_r. The sleepers are suspended to the ground via the ballast elements, labeled as elements k_b and d_b. Both the railpads and ballast elements are modeled as suspension elements, possessing properties in both vertical and lateral degrees of freedom. The sleeper itself has one rotational degree of freedom, rendering the track foundation a model with seven degrees of freedom outside the switch joint.

In the running direction as denoted in figure 4, the wheel load is exerted on to the movable rail and because of the vertical overlap the load passes on to the fixed rail. However in the trailing direction the wheel's load on the fixed rail would not pull the movable rail down in the overlapping section. To model this feature within the joint, a stiffness k_a is introduced. This denotes the bending load passed on from only the



(a) Method 1

(b) Method 2

Figure 3: Approaches to model the rail joint in presence of passive lock

	k_a modelled as	Limits
Method 1	Another element within co-running track model. As shown in figure 3a, it is defined only in running but not in trailing direction.	The effective stiffness supporting the movable rail is greater than that of the fixed rail. This results in a reduced vertical deflection of 70μm in the fixed rails, as shown in figure 5. Considering the magnitude of this deflection, it is not considered a significant concern.
Method 2	It is defined only between $[D \quad D + 0.35m]$, while the other foundation elements are included as part of the co-running track model as shown in figure 3b. Here D refers to the location of switch toe.	The higher stiffness defined only within a small segment induces sudden, discontinuous changes in force behaviour as the wheel approaches the joint as shown in figure 6.

Table 1: Description of modelling rail joint approaches

movable rail m_m to the fixed rail m_f in the vertical direction and not vice-versa. The special rail mass interaction at the joint through the stiffness k_a is modelled by two approaches as described by table 1 with help of figure 3.

3 Simulation setup

In the current study, a vehicle traverses a track segment containing the Repoint switch. The selected vehicle for this study is the Manchester benchmark vehicle [6]. Widely recognized as a standard representation for simulation studies within the railway industry, this passenger vehicle has been modelled in Simpack, a multibody software equipped with a dedicated module for Rail. The vehicle travels on a straight route

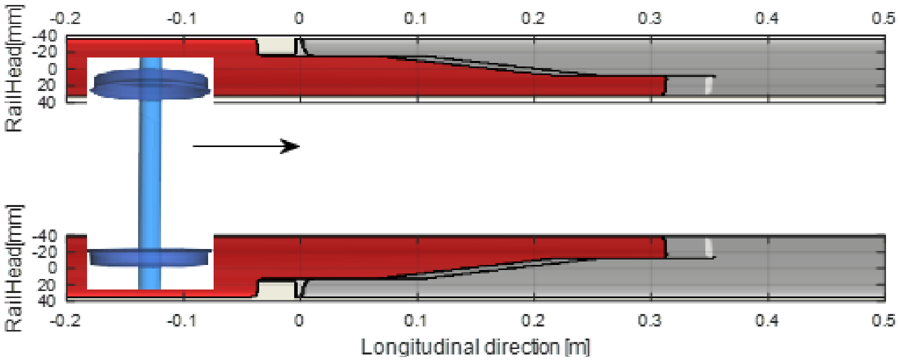


Figure 4: Overview of wheelset in facing direction over the Repoint joint

at a cruising speed of $160kmph$. It is ensured in simulation that the vehicle achieves steady state motion prior to passing through the switch joint.

The switch layout showing a wheelset running through it is shown in figure 4. The wheelset has P8 wheels running over UIC60E2 rails whose cross-sections are adjusted to accommodate the shape of the expansion joint. The rails are inclined at a ratio of 1:20, which is standard practice for this wheel and rail profile combination. The switch is represented as a sequence of cross sections and integrated into Simpack. As a reference scenario, a ballasted track bed is modelled and there are no track irregularities or defects.

4 Vehicle-track interaction results

The results when the vehicle runs in facing and trailing directions are shown in figures 5 until 7. The left and right column of the plots show quantities in vertical and lateral directions respectively. From the top of each figure the plots show wheel-rail contact forces, wheelset displacement, rail body deflection and sleeper deflection. The force transition between the movable and fixed rails occur between 0.1 to $0.2m$ from the switch toe.

The overall observations are the forces smoothly reduce on the movable rail. There are dynamics observed at the fixed rail, with peak forces reaching up to 80% more than the steady-state value. In the lateral direction, forces and deflections are negligible due to the symmetry of the expansion joint on both sides of the track. Because of the short length of the joint, the dynamics settle down sooner compared to the traditional track switch. In the following paragraphs the differences due to the two modelling variants of k_a are described.

When utilizing method 1, a discrepancy of $70\mu m$ is observed in the wheelset and rail vertical deflection as seen in the steady-state values of the plots in figure 5. They appear to drop down after the switch joint by that amount. This discrepancy arises from the co-running track model. Because of the presence of k_a, the effective stiffness

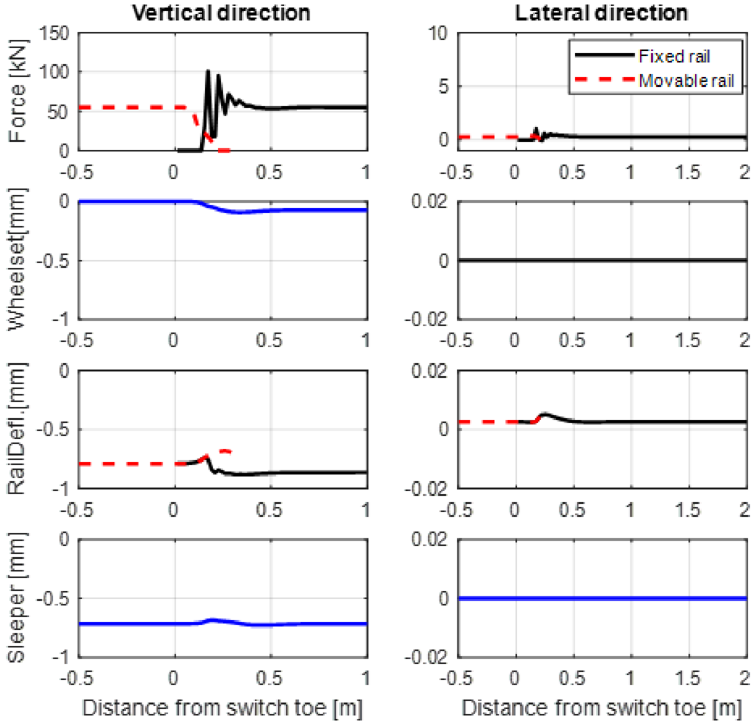


Figure 5: Mechanical quantities on the Repoint joint using modelling method 1

is higher on the movable rail counteracting the wheel forces, results in lesser deflection compared to the fixed rail.

Figure 6 depicts the results from the wheel-track interaction using method 2 to model the bending stiffness between the rails of the joint. In contrast to the earlier method, dynamic forces at the fixed rail peak up to 150% more than the steady-state value. Additionally, the settling time for the forces is extended by $0.3m$ in the spatial direction compared to the earlier method. This is attributed to the activation of bending stiffness at a discrete location. The discontinuous appearance of a new stiffness element k_a in the track model also introduces more fluctuations in the forces. However, unlike the previous case, there is no drop in the steady-state deflections on either side of the switch joint.

The effect of wheelset running in trailing direction is presented in figure 7, with method 1 to model the lock. As the train approaches from the trailing direction, the graphs should be read in decreasing direction of x-axis. Initially the wheel loads are on fixed rails followed by the movable rails as the wheel approaches the switch toe. In comparison to the corresponding baseline case in figure 5, the movable rails exhibit higher peak forces upon receiving the load. This difference is attributed to the free deflection of the movable rail ends until the wheels make contact with them. In contrast, in the baseline case, the fixed rails are already displaced downward before the wheels come into contact. The behavior of other signals, such as sleeper displacement, remains consistent with those observed in the baseline case.

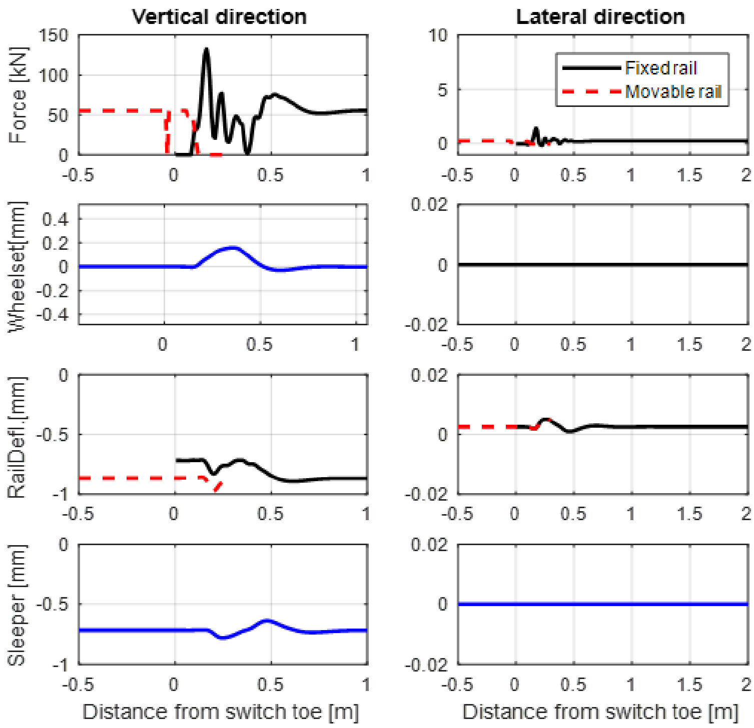


Figure 6: Mechanical quantities on the Repoint joint using modelling method 2

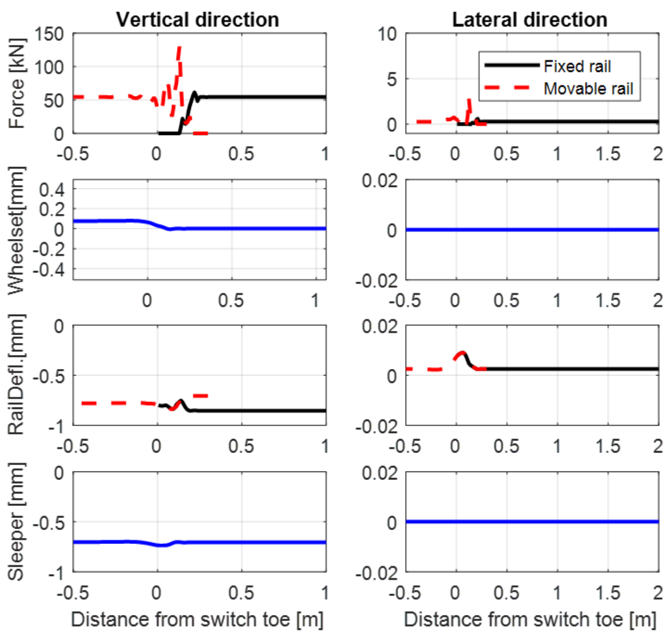


Figure 7: Effect of wheelset running in trailing direction

5 Conclusion

This paper shows the Repoint joint with its multi-body dynamic simulation model. The contact between the switch and stock rails which are arranged with a vertical and lateral overlap is modelled by a spring element, and two variants to simulate it are described. When a vehicle runs over this joint, the difference between the two variants are negligible compared to the overall dimensions of the vehicle and track. The Repoint joint presents a smooth wheel load transfer between the rails and the relative movement of bodies in the track are reduced very much. The symmetry of the joint on the right and left rails ensure that lateral forces and deflections are also reduced compared to a conventional track switch.

Acknowledgements

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