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Dynamic Behaviour Comparison of a Switch Panel with Independently Modelled Rails Versus Combined Rails

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Abstract

This paper shows a comparison analysing the differences in multi-body simulation analysis between treating the switch panel as separate entities for the switch and stock rails versus treating them as a single entity. By conducting this comparison, the aim is to clarify the effects of these modelling approaches on the accuracy and efficiency of the analysis. Specifically, it investigates how these different modelling approaches influence the dynamic behaviour and contact calculations within the multi-body simulation framework, shedding light on their relative advantages and limitations for railway track analysis.

Acquiring rail sections needs the scanning of rail profiles, a process to obtain comprehensive geometric data for subsequent analysis. Scanning rails in a switch panel is easier when the switch and stock rail are a single entity. Analysing these bodies separately may not only require more complex post-processing but could also result in the scanning process itself being more costly and complicated.

The vehicle used in the study is a model of a simplified metro train. The cases under analysis will be similar to those described in the switches and crossings benchmark. Additionally, the rail profiles used are also sourced from the switches and crossings benchmark. In order to make a proper comparison, interpolation is used to combine stock and switch rail sections creating a single entity. The multi-body system simulation and analysis are conducted using Simpack software.

Keywords: railway, turnout, switch rails, multibody simulation, train derailments, modelling.

1 Introduction

A railway turnout is a crucial element of railway infrastructure, comprising a switch panel and a crossing panel, all movable longitudinally along the rails. The switch panel primarily directs a vehicle onto either the main line or a diverging line, facilitating the transfer of the vehicle's wheel between the stock rail and switch rail [1]

The switch rails within a railway turnout endure significant loads, creating a harsh operating environment. Fluctuations in rail profile along the switch panel result in undesirable differences in rolling radius and contact conditions. Consequently, this leads to elevated levels of dynamic forces at the wheel-rail interface, resulting in plastic deformation, wear, and rolling contact fatigue [2]

In Figure 1, a schematic of a turnout can be seen. It distinguishes the two most important parts: the switch panel and the crossing panel.

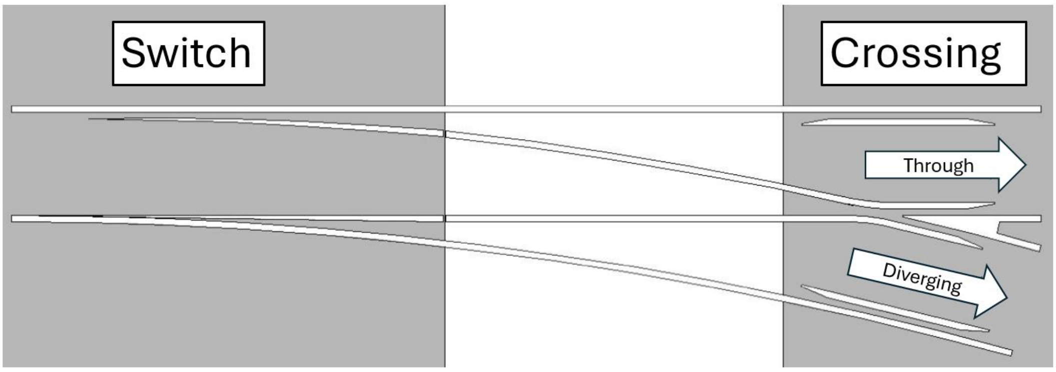


Figure 1:Description of a turnout.

Typically, researchers address these dynamic impact issues through experimental and/or numerical methodologies [3]. Both avenues prove invaluable in enhancing comprehension of dynamic impact phenomena. Nonetheless, experimental investigations tend to be time-intensive and financially demanding. Consequently, the use of numerical simulations is much preferable.

In order to comprehensively analyse system performance, it is crucial to account for the dynamics of vehicle/turnout interaction. Multi-body simulation (MBS) has emerged as a prevalent technique for modelling both vehicle/track interaction dynamics and vehicle/turnout dynamics [4]. A co-running track approach is typically employed in MBS modelling, wherein a simplified multi-body track model is placed beneath the wheelset and moves in tandem with the vehicle at the same velocity. This track model, often characterized by a limited number of degrees of freedom (DOFs), serves to represent essential track components such as rails and sleepers.

In MBS simulations, all vehicle and track components are typically simplified or modelled as rigid bodies. The contact models utilized in these simulations commonly rely on assumptions of an elastic half-space, where material plasticity is not taken into account [3].

Among the advanced numerical methods, finite element method (FEM) is frequently used. While such methods may approximate reality or enable the definition of more detailed models, their computational cost often renders them inefficient.

In order to analyze the condition of a specific railway track, it is essential to obtain precise geometric data of the track rails. In an MBS analysis, acquiring the actual rail profiles is crucial for accurate contact calculations. This necessitates scanning the rails to obtain real profiles, a process facilitated by various types of scanners. The dynamic behavior and contact will depend on the condition of the track.

In the operation of the switch panel, the switch blades undergo lateral movement, joining and separating from the stock rails. Consequently, one of the switch rails will always be positioned adjacent to a stock rail. As a result, separately scanning the profiles of these rails may present challenges.

In 2021 a Benchmark of railway multibody dynamics software application to switches and crossings (S&C) was presented [5], comparing all major commercially available software and a few independent codes. The vehicle used in the analysis was the Manchester Benchmark [6] passenger vehicle. The results of the analysis were that all of those examined software offer a reliable and efficient way to understand the kinematic and dynamic forces between the wheels and track elements.


(a) (b)

Figure 2: Switch blades in closed position (a) and open(b).

The most challenging part was the modelling and combination of multiple rails in simultaneous contact for a single wheel [7]. For example, in the switch panel, at the switch-stock contact or in the crossing panel, at check-stock contact.

Two different representative S&C were examined and analysed in the Benchmark. The rail geometry was provided as sets of transversal rail cross-sections with specified positions along the track.

In this paper, an equivalent analysis to the benchmark will be conducted, assuming that the scanning of the rails in the switch panel has been performed as a single block of rails, rather than two separate rails.

2 Methods

In this study, two types of turnouts are utilized. One is the 56E1 turnout, which belongs to a design from the United Kingdom, while the other is a 60E1 rail from Sweden. In both cases, a study will be conducted with the train traveling on the through route and on the diverging route. In total, four different scenarios will be studied. For each scenario, a specific velocity is assigned.

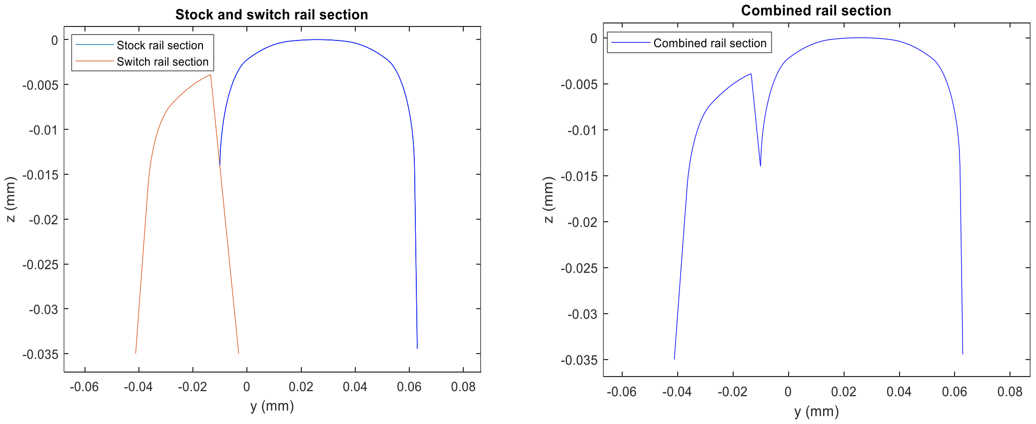
The velocities to be used in the study are described in Table 1. These speeds correspond to those of the actual vehicle. Analysis at higher speeds is not possible as the vehicle is not designed for it. The radius of the diverging route will be 245 and 760 meters.

Number	Rail	Route	Velocity (km/h)	Radius (m)
1	56E1	Through	80	-
2	56E1	Diverging	40	245
3	60E1	Through	80	-
4	60E1	Diverging	40	760

Table 1: Description of routes and velocities.

For each of the previously described scenarios, a comparison will be made between the system with the switch rails treated as separate entities and the system with the switch rails treated as a single unit.

For the scenario where the switch and stock rails are treated as a single entity, the initial geometries of the sections will be derived from the cases where the switch rail and the stock rail are considered separate entities. In Figure 3 both cases of rail sections are shown.



(a) (b)

Figure 3:Rail sections in both cases: independent(a) and combined (b).

It is possible to merge sections located at the same longitudinal position. This process combines previously independent rail sections into a single unified profile. As a result, where there were originally two separate rail sections, a single entity profile is obtained. This method results in more complex rail sections for analysis yet simplifies the overall system by eliminating one entire body.

To accurately merge sections, it is crucial to identify sections belonging to the same longitudinal point. Lateral interpolation is performed, discarding the part where sections intersect.

The vehicle used in the study is a model of a metro train. This vehicle model is relatively more complex than the one used in the Manchester Benchmark. It is considered to be more realistic, and the responses will be more accurate to reality. Table *2* describes the model and details the most relevant data.

Masses (kg)	
Body	21420
Bogies	3113
Wheelsets	1775
Primary suspension stiffness (N/m)	
Stiffness in x	4160995
Stiffness in y	4160995
Stiffness in z	1493426
Primary suspension damping (Ns/m)	
Damping in x	3646
Damping in y	3646
Damping in z	2184
Secondary suspension stiffness (N/m)	
Stiffness in x	281068
Stiffness in y	281068
Stiffness in z	571155
Secondary suspension damping (Ns/m)	
Damping in x	0
Damping in y	0
Damping in z	0
Vehicle Dimensions (mm)	
Bogie semi pivot spacing	5785
Bogie semi wheelbase	1000
Height above rail level of bogie cg	600
Height above rail level of body cg	1700

Table 2: Parameters of vehicle.

The wheel profile utilized is an S1002. The nominal wheel radius is 0,42 m and the spacing of active faces 1425 mm. Additionally, the flange width is 32,5 mm.

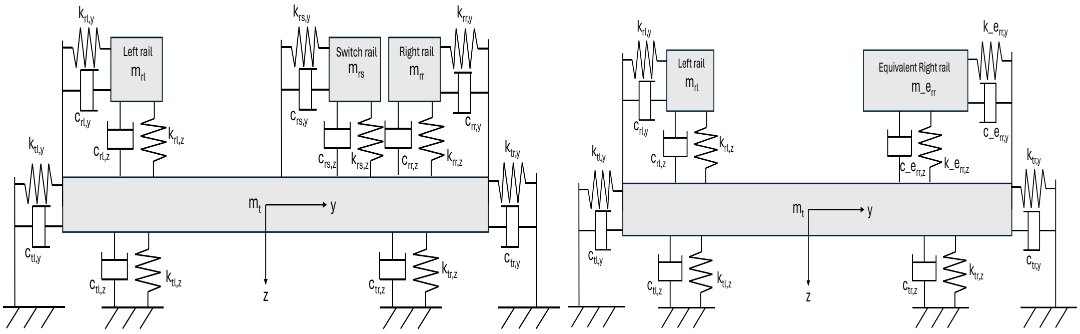
Regarding the track topology, the differences between the two cases under analysis are evident. In the "double entity" scenario, the switch rail and the stock rail are entirely independent. As such, each of these bodies, along with the left rail, will have its own connection to the sleeper, with individual lateral and vertical stiffness and damping properties. Conversely, in the "single entity" scenario, only two bodies will be present above the sleeper, each with its own connections.

In the MBS software Simpack, the definition of wheel/rail contacts will vary between the two cases. In the first case, 12 wheel/rail pairs are defined, one for each rail body and for each wheelset. In the second case, only 8 wheel/rail pairs are defined. However, Simpack defines more than one contact point for each wheel/rail pair. The definition of the sleeper is carried out in the same manner in both cases, ensuring uniform characteristics across all scenarios under analysis.

In all cases analysed in this study, the variable rail sections are consistently located on the right side. Thus, the switch rail is always positioned on the right side, simplifying the cases under examination. In Table 3, the entire track is described. Figure 4 provides a graphical description of the track.

Data	**Symbol**	**Independent**	**Equivalent**
Masses (kg)			
Sleeper mass	m_t	1400	1400
Left rail mass	m_{rl}	60	60
Right rail mass	m_{rr}	60	120
Switch rail mass	m_{rs}	60	-
Stiffness (kN/m)			
Lateral left rail-sleeper	$k_{rl,y}$	30000	30000
Vertical left rail-sleeper	$k_{rl,z}$	150000	150000
Lateral right rail-sleeper	$k_{rr,y}$	30000	60000
Vertical right rail-sleeper	$k_{rr,z}$	150000	300000
Lateral switch rail-sleeper	$k_{rs,y}$	30000	-
Vertical switch rail-sleeper	$k_{rs,z}$	150000	-
Lateral left and right sleeper-ballast	$k_{tl,y}$	70000	140000
Vertical left and right sleeper-ballast	$k_{tl,z}$	140000	280000
Damping (kNs/m)			
Lateral left rail-sleeper	$c_{rl,y}$	150	150
Vertical left rail-sleeper	$c_{rl,z}$	100	100
Lateral right rail-sleeper	$c_{rr,y}$	150	300
Vertical right rail-sleeper	$c_{rr,z}$	100	200
Lateral switch rail-sleeper	$c_{rs,y}$	150	-
Vertical switch rail-sleeper	$c_{rs,z}$	100	-
Lateral left and right sleeper-ballast	$c_{tl,y}$	350	700
Vertical left and right sleeper-ballast	$c_{tl,z}$	1400	2800

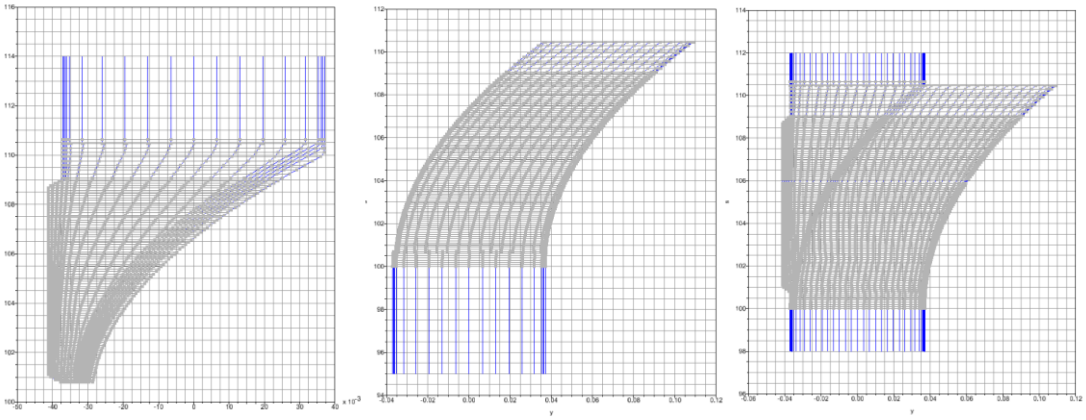
Table 3: Track parameters.



(a) (b)

Figure 4: Graphical description of track: independent switch and stock (a) and equivalent rail (b).

Simpack interpolates variable rail profiles between the various profiles in longitudinal direction by means of Bézier curves. This ensures a smooth transition from one profile to the next one. The number of profiles is not limited, and the profiles do not need to be equidistant along the track. Bézier curves never fit the original profile points exactly. In Figure 5 the rail top view is shown for the switch rail (a), the stock rail (b) and the combined rail (c).



(a) (b) (c)

Figure 5:Bézier curves: independent switch (a), stock (b) and equivalent rail (c).

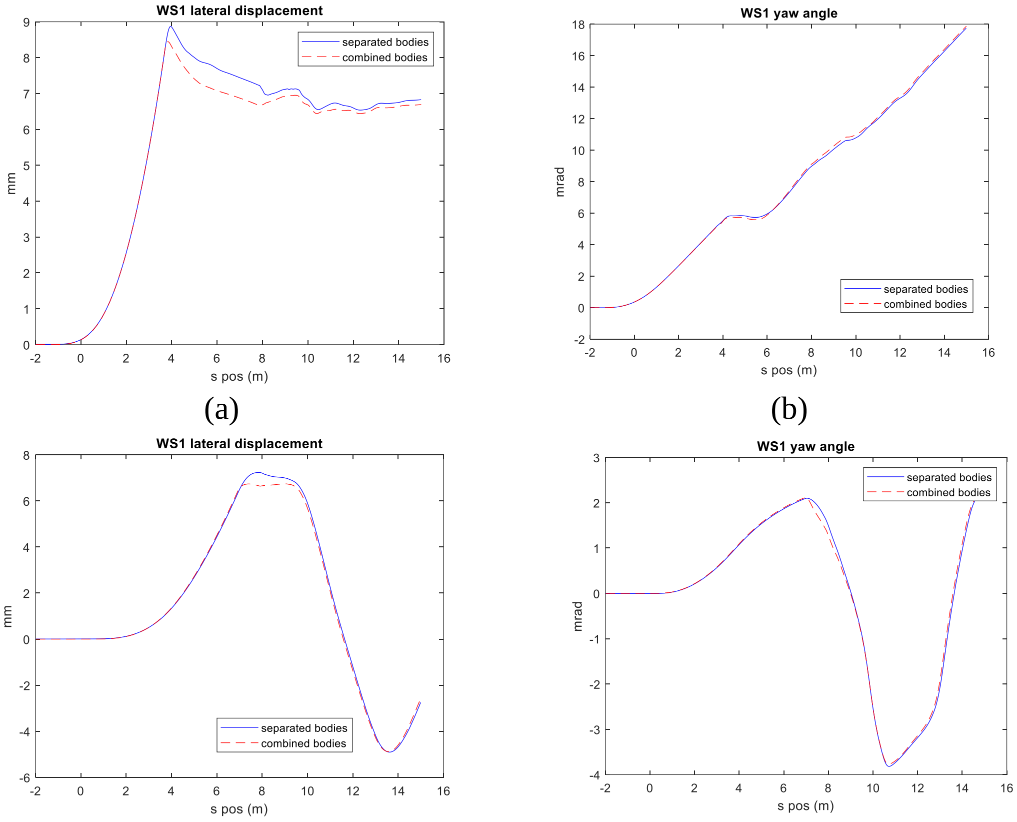
The image illustrates that, as the rail sections change, the longitudinal interpolation performed by Simpack can vary, thereby altering the contact conditions. This may modify the positions at which contacts occur.

3 Results

In the results section, the dynamic response of the model will be analysed. This section shows the differences between the results obtained using two different approaches are presented. One approach involves modelling the switch rail and the stock rail as a single entity, while the other approach models them separately.

Only the results of two out of the four analysed cases will be shown and discussed, as explaining all the cases would not clarify anything new. Only cases with 60E1 rails will be analysed, leaving aside the 56E1 rails. The results of the 'through' and 'diverging' cases will be taken into account, corresponding to cases 3 and 4 described in Table 1.

In the results section, responses are arranged such that at the point s=0, it marks the beginning of the switch rail, with 's' representing the longitudinal position of the first wheelset. This arrangement enables a simpler comparison across all cases. The comparison will be conducted for the right wheel of the first wheelset since it is on the right side where the geometry of the section changes.



(a) (b)

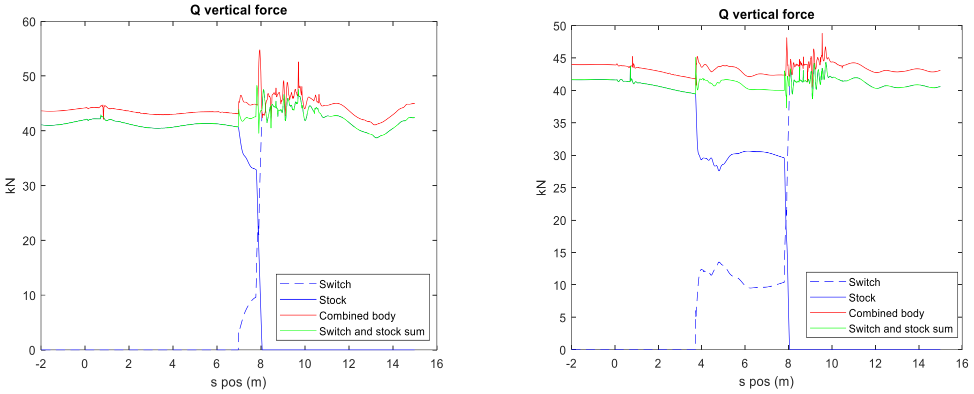
(c) (d)

Figure 6:Overall movement of the train. Lateral displacement (a) and yaw angle (b) in ‘through’ case. Lateral displacement (c) and yaw angle (d) in ‘diverging’ case.

To compare the results obtained for the 'through' and ‘diverging’ case, it will be important to analyse the overall movement of the train. For this purpose, the lateral displacements and yaw rotations of the first wheelset are examined. Figure *6* illustrates the behaviour in both cases.

In the 'through' case, it can be observed that the responses are very similar both in lateral displacement (a) and yaw angle (b). In the ‘diverging’ case, it can be observed that the displacements and the yaw angle are similar in both cases, although there are some differences. The lateral displacements (c) of the wheelsets differ by up to 8%. However, once the displacement stabilizes, the differences are much smaller.

To make a comparison between the vertical forces, it's important to consider that the number of bodies is different: the load, which is distributed between the switch and stock rails in one case, will not be distributed in the same way in the other case. To compare them, it will be important to compare the vertical force on the equivalent rail with the sum of the forces on the switch and stock rails. Figure 7 illustrates the differences between the vertical forces.

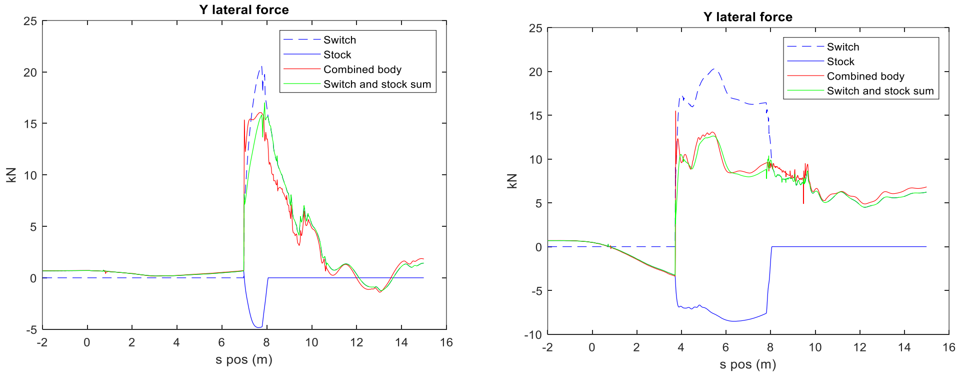


(a) (b)

Figure 7: Vertical force comparison in 'through' case (a) and 'diverging' case (b).

The image shows that the vertical forces are very similar in both cases. However, there is a consistent offset observed throughout the graphs. This error could be due to the equilibrium; the difference in bodies when balancing and applying preloads results in an unequal distribution of forces between the left and right sides. Since this offset is consistent from the beginning and remains constant (2-3 kN), it may be attributed to preloads and equilibrium. However, it can be stated that the comparison between the two types of modeling is positive, as the results are very similar in both cases.

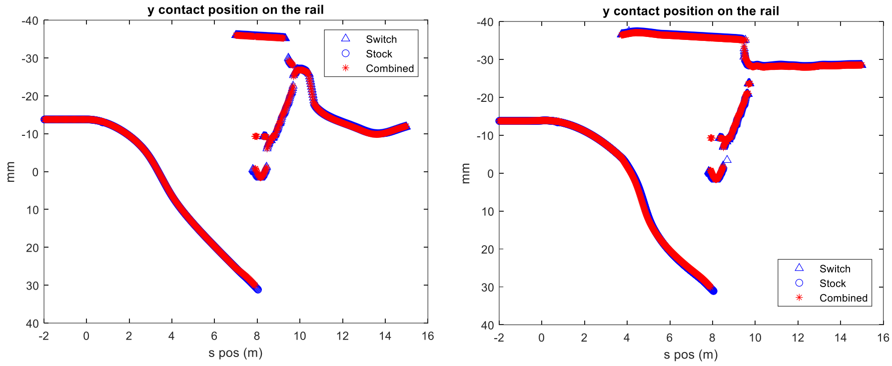
In the comparison of lateral forces, similar to vertical forces, it will be necessary to sum the switch and stock rail forces to compare with the combined rail cases. The Figure 8 illustrates the differences in lateral forces.



(a) (b)

Figure 8:Lateral force comparison in ‘through’ case (a) and ‘diverging’ case (b).

In this case, the results are very similar and it can be assured that the response in both modeling methods is equal or equivalent. Although the obtained values are very close, it can be seen that in the combined rail case, the peak force occurs slightly earlier than in the other case. To better understand this, Figure 9 depicts the lateral position of the contacts at each moment.

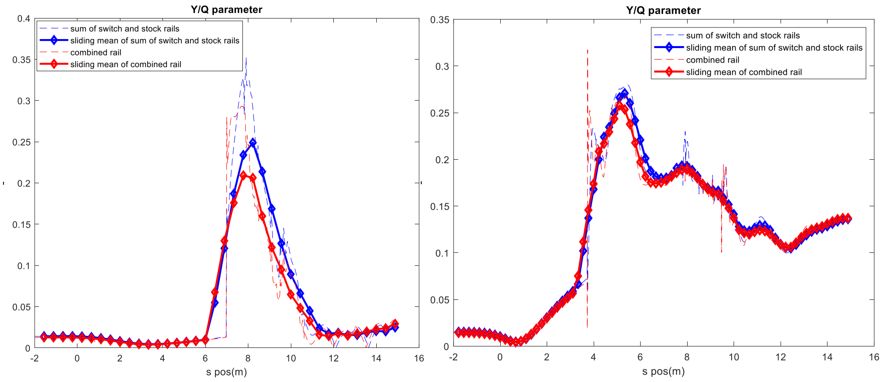


(a) (b)

Figure 9: Lateral contact position in ‘through’ case (a) and ‘diverging’ case (b).

The largest lateral forces occur when the contact abruptly changes position. In the image, it can be observed that the transition from stock rail to switch rail occurs slightly earlier in the combined rail case. This could be the reason why the highest lateral forces occur earlier in the combined rail case. The reason for this could be the Bezier curves used by Simpack when joining all sections. When there is a significant change in geometry, there may be differences in the behaviour.

One of the most commonly used parameters for studying the risk of derailment is the Y/Q parameter. This parameter is the division of lateral force by vertical force. The higher the value of this parameter, the higher the risk of derailment. According to EN14363 standard [8], to analyse the risk of derailment, it is necessary to perform a sliding mean average of the values with a sliding window length of 2 meters and a step length of less than 0.5 meters. Figure 10 shows the values of the Y/Q parameter. In each analysed case (a) and (b), the original values of Y/Q for each track modelling are shown, and also the values of the sliding mean average in all cases are displayed.



(a) (b)

Figure 10: Y/Q parameter in ‘through’ case (a) and ‘diverging’ case (b).

It can be observed that, generally, in cases where the rails are modelled independently, the Y/Q coefficient is higher. This implies that the perception of derailment risk will be lower if the rail is modelled as a single body. Thus, this method can give a sense of safety when in reality the risk of derailment is higher than simulated.

4 Conclusions and Contributions

In this paper, a comparison has been made between two methods of analysing the switch section using multi-body simulation. In the first case, the study was conducted with the switch and stock bodies being completely independent entities. In the second case, an equivalent rail is created that includes both the switch and stock rails. This simplifies the model by defining only one body for the right rail instead of two. For this purpose, in this study, the stock and switch profiles have been merged into a single rail.

Conducting the analysis with the switch and stock defined in the same body offers several benefits:

- Scanning becomes simpler as instead of scanning two independent rails, only one needs to be scanned.
- The model in the multibody software becomes simpler and easier to use as the number of contacts to define reduces from 3 for each wheelset to 2 for each wheelset.
- Those multi-body simulation software that do not allow the definition of more than one rail body for each wheel/rail contact will be able to perform vehicle-turnout interaction calculations.
- After analysing the responses and conducting comparisons, one can conclude that the responses and dynamic behaviour in both cases are very similar.

However, the use of this method presents some challenges that make this modelling less appealing:

- Loss of information for each rail. We won't be able to discern the stress distribution between the stock rail and the switch rail. This can be a significant issue as knowing the stress distribution in each rail can help prevent wear and provide better insight into the condition of turnouts.
- If, as a rule, lower Y/Q coefficients are measured in cases where modelling is done with a single body, there is a risk that we may not realize the true risk of derailment. Therefore, a security coefficient should be applied to obtained results.

In conclusion, it can be stated that, if necessary, a simplification like this can be made in future analyses, although it is important to recognize that valuable information is lost. Therefore, whenever possible, analysing the turnout with the stock and switch rails defined independently will be preferable.

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