Modelling high speed turnouts

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Abstract

The Spanish Railway Infrastructure Administration (ADIF) requires a design of the new turnouts focused on two aspects, among other ones:

- The obtaining of an adequate value of the track stiffness in the turnout (established by ADIF between 40 and 60 kN/mm).

- The developing of an improved geometry of the turnout in order to achieve a better interaction between vehicles and track.

In this paper, the authors present two models that they used to study these questions in a new improved design of Spanish turnout.

1. Introduction

Spain, like other countries in the world, is building new high speed railway lines in order to set up a network integrated in the high speed Transeuropean network. Most of these lines are planned for mixed traffic. In these conditions, it is necessary to maintain the reliability and safety of the operation with moderate maintenance costs. This requirement means, among other actions, to reduce the dynamical forces originated by the movement of the trains over the turnout by means of its design.

The Spanish Railway Infrastructure Administration (ADIF) requires a design of the new turnouts focused on two aspects, among other ones:

- The obtaining of an adequate value of the track stiffness in the turnout (established by ADIF between 40 and 60 kN/mm).

- The developing of an improved geometry of the turnout in order to achieve a better interaction between vehicles and track.

The authors have developed two different models to study these questions in the proposed geometry of a new high speed turnout design, by Felguera Melt, S.A.

2. Track stiffness and turnouts

Track stiffness is of great importance in the case of turnouts, particularly in those designed for High Speed operations. On one hand, the fact that the length of the bearers in a turnout increases from the switches to the crossing, and on the other the existence of zones with inertia that are very different from those of plain tracks (especially in the deflecting device and the common crossing), lead to a marked alteration in the vertical stiffness of the track, increasing it, and as a result favouring the rapid degradation of the ballast.

In this respect, some authors maintain that lower track stiffness values should be achieved in the turnout, leading to lesser requirements on the ballast, and as a result a decrease in its degradation, meaning that maintenance cycles can be longer [7].

However, as in the case of open track, it is necessary to determine an optimum value for the vertical stiffness of the track in the turnout area, as if this value descends too much, there is an increase in the vertical deflection that the rail has to support, which may lead to the obligation to physically limit the possibility of differential movements between the moving and fixed parts of the turnouts (switch rail-stock rail and wing rails-point frog), which both complicates the design of the turnout and makes it more expensive. Furthermore, it is necessary to limit the range of transversal rail movements.

2.1. Example of application

The Spanish Railway Infrastructure Administration (ADIF) requires that the vertical static stiffness range for the High Speed turnouts must be between 40 and 60 kN/mm, whereas on general track a vertical static stiffness of 100 kN/mm is required. This concept is considered as the quotient, for each point of the turnout, between the applied static force and the absolute vertical movement of the rail. In calculating this, it is important to consider the variation in length of the turnout in the stiffness of its component parts. It is therefore necessary to consider the elastic characteristics of the subsidence levels that are identical to those of the rest of the track.

The authors have studied compliance with these requirements in the design of a High Speed turnout for speeds of 350 km/h on main track and 220 km/h on diverted track, with widths of 1435 mm, and a tangent of 0.0215. The approximate length of the turnout is 213 m. Due to the difference in deflection of the rail in open track and turnouts, the ADIF compelled to include a transition stretch with a length of 50m. The track zone to be modelled therefore has a length in excess of 300m.

The total stiffness may be obtained by considering a series of elastic elements (elastic pads under the rail, under the tieplate, under the sleepers, mats under the ballast, etc.). However, it was considered that the most important effect should fall on the rail pads. These must always be of the same thickness, although their elastic qualities must vary according to their position in the turnout. It is recommended that three or four different types of pads be used, with the aim of reducing storage requirements and the possibility of confusion arising during assembly, meaning that they should also be easily identifiable.

2.2. Model designed

The turnout and the adjacent transition tracks has been discretized using a bar grid (figure 1), designed using a commercial programme for finite elements.

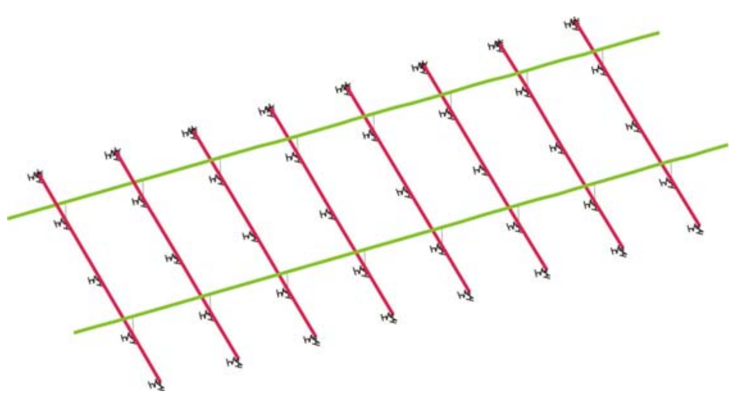

Figure 1.

The rails are discretized as beam elements, made of steel, with 4 elements between two contiguous sleepers.

The sleepers and bearers are also beam elements, made of C50-60 concrete, with different sections according to the geometry. The sleepers are modelled using 4 elements, and the bearers, of variable length, by using a number of elements in proportion to their length. The maximum length of the elements is around 3.5 times more than the sleeper edge, which according to studies carried out in this particular case is sufficient to obtain the necessary precision in the value of the vertical deflection of the rail under a load. There is always a node in the sleepers found in the vertical of each rail.

The behaviour of subsidence levels has been represented using a Winkler model, in which the stiffness of the spring bands is calculated using the ballast module of the ballast-formation system (C_{bp}). Assignment of stiffness to each spring band is carried out according to its tributary area in the sleeper or bearer.

With regard to fastenings, the model designed by the authors makes it possible to establish multilineal stress-deformation functions, for displacements and turns, for each of the three spatial directions. In the example presented later, these curves have been simplified by including the value (amongst others) corresponding to the secant static stiffness of the elastic base plate [5].

The final model contains 10439 bars. In it, the stiffness of 1834 rail pads has to be determined. The effect of the subsidence levels is reproduced through 4433 spring bands of different stiffness.

2.3. Results

In accordance with the requirements of the ADIF, the load applied corresponds to two forces of 100 kN, simulating the passage of a railway axle of 200 kN. Overloading due to dynamic effects was not taken into account, as the ADIF only requires a static analysis of the behaviour of the turnout [6]. The loads were considered when applied to the vertical of the sleeper, although the model makes it possible to study the application of loads in the gap between sleepers or bearers.

The first two figures show the results obtained with the turnout in which elastic base plates were fitted under the rail of 100 kN/mm. The foundation modulus C_{bp} is 0.2 N/mm_3. In figure 2, the axle of 200 kN is circulating on the straight track, whereas figure 3 shows the stiffness graphs calculated for inner and outer rails when the axle circulates on the diverted track.

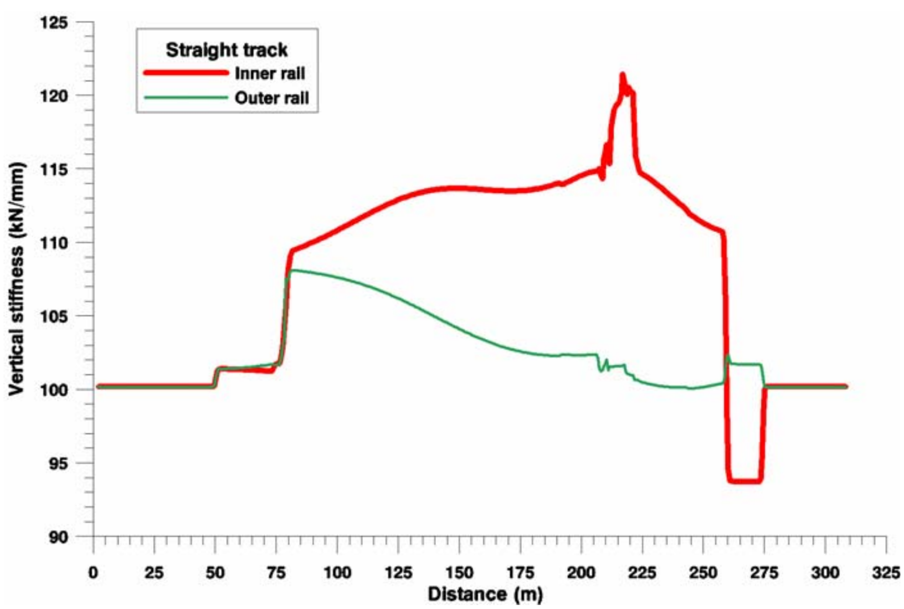

Figure 2. Static track stiffness for straight track.

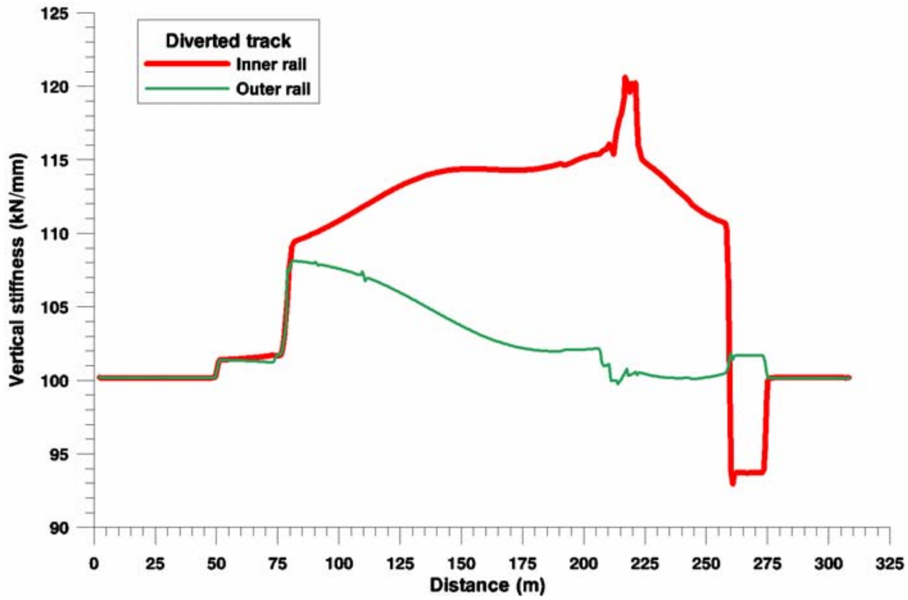

Figure 3. Static track stiffness for diverted track.

The following observations may be made based on an analysis of the previous results:

- The values for the vertical track stiffness obtained for inner rails when the turnout is crossed on straight and diverted track are very similar in this case. The study could therefore be reduced to passage of the load axis on straight track.

- The sudden modification of the track components in areas like the transition from sleeper to bearer or in the frog area, generate peak values of the vertical stiffness.

- The variation of the vertical stiffness of the track is greater in the inner rails of straight and diverted tracks, particularly in the frog area, which would be expected.

- The significant changes in the vertical stiffness of the switch-end of the crossing are due to the asymmetrical cut-off of the sleepers in this area. This design was reconsidered in subsequent stages of the design.

After analysing the data obtained and taking into account the recommendations of the ADIF, the following distribution is considered for rail pads in order to achieve a vertical stiffness in the turnout of between 40 and 60 kN/mm:

- Throughout the whole length of the turnout, rail pads are fitted that make it possible to achieve a vertical stiffness of 17.5 kN/mm.

- The transition from values for vertical track stiffness in open track (100 kN/mm) to those required in the turnout (between 40 and 60 kN/mm), is achieved with two new rail pads with a static stiffness of 25 and 50 kN/mm. These pads are mainly situated in the transition stretches 50 m to either side of the turnout.

With this rail pad configuration, the results for the same turnout are shown in figure 4 (circulation on straight track) and figure 5 (circulation on diverted track). Here may be seen the two levels of stiffness in the transition stretches (corresponding to the rail pads of 25 and 50 kN/mm). The static rail pads of 17.5 kN/mm make it possible for the vertical stiffness values throughout the turnout to remain within the required values.

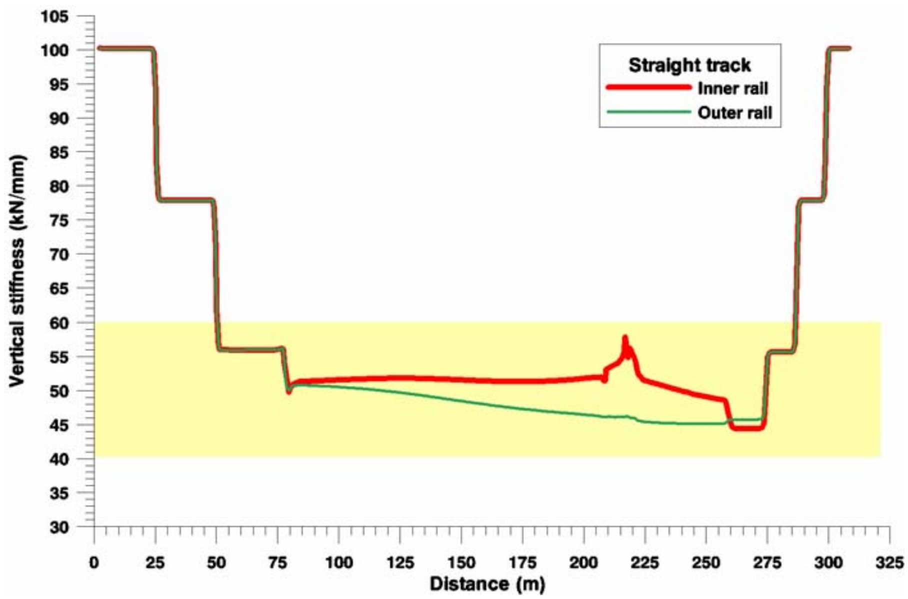

Figure 4. Static track stiffness for straight track with modified rail pads.

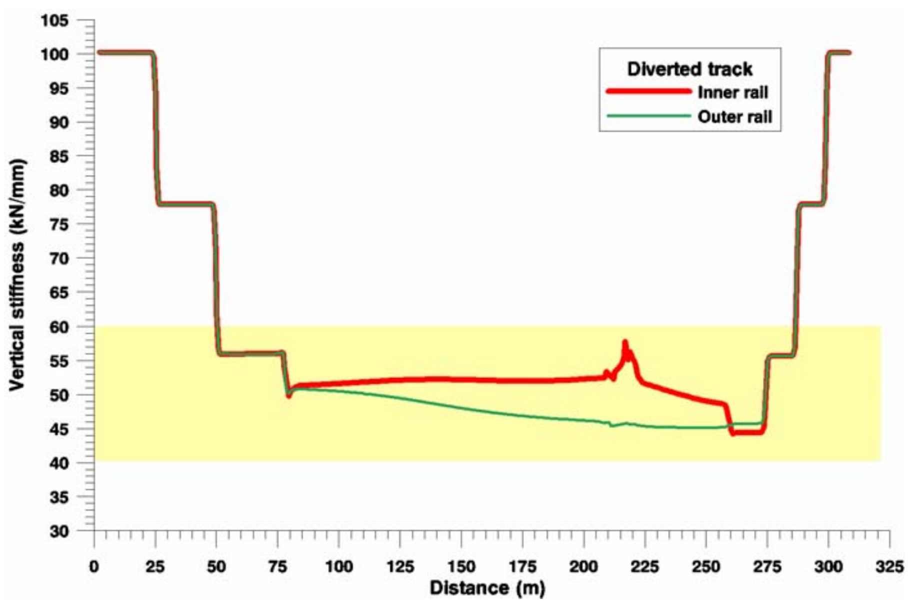

Figure 5. Static track stiffness for diverted track with modified rail pads.

3. Improving the turnout geometry

The mathematical modelling of the phenomenon of vehicles running over turnouts is highly complex, due in part to the geometrical variation of the rails (and in particular the switch rails and the frog area), the range of railway vehicles which may run across a certain deflecting device and the question of discontinuity, both material (a classic example of discontinuity in the wheel support when they change from the wing rail to the point frog) and dynamic (such as the sudden appearance of the diverted track, generally without a transition curve or superelevation, resulting in acceleration and jerk).

Numerous trials (for example, [3]) and mathematical models have been drawn up using the appropriate simplifications (conical wheel, rail turnout features assimilated to line edges, etc.), have permitted the analysis of the problem in question (for example, [4]). Generally speaking, and as far as the models which analyse movement over a turnout deflecting device are concerned, most attempt to simulate vehicle behaviour, normally represented by its guiding axle or the front bogie, taking the diverted track. This is due to the very obvious aggression suffered by the curve switch rail on a diverted track caused by a composition engine. However, the straight switch rail on a main track High Speed turnout does not appear to receive any type of lateral guiding force apart from that produced on standard track.

3.1. A quasistatic approach to the problem

The figure 6 shows a variety of deflecting device sections, corresponding to those which would come into contact with an axle, whose wheel has been drawn [1]. The t_0 section indicates the start of the switch rail; the t_6 shows the switch rail at the moment when it has sufficient thickness to permit the wheel support contact without the intervention of the stock rail. We will later show how this parameter has a considerable influence on vehicle movement. Sections t_2 to t_5 correspond to intermediate stages.

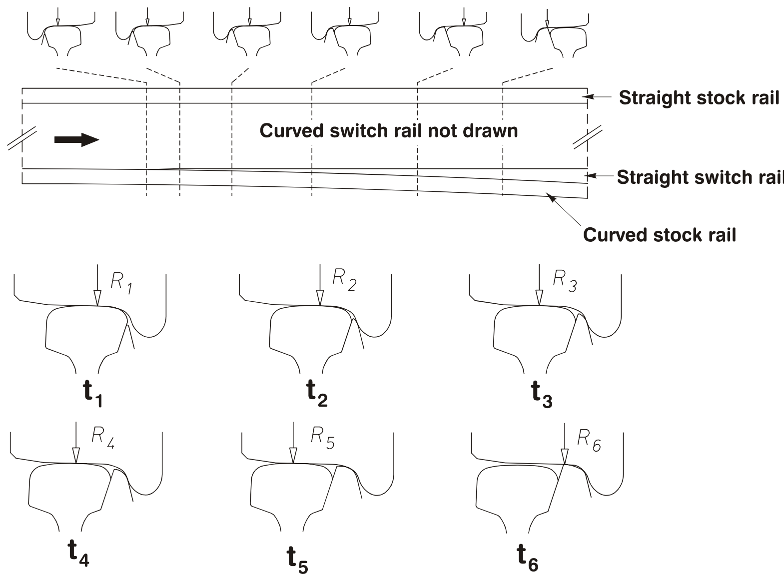

Figure 6.

At t_2, the curve stock rail begins to move towards the exterior part of the track, as the straight switch rail section increases. The displacement of the stock rail also implies the movement of the wheel-stock rail contact point towards the outer part of the wheel.

For each Δt, the stock rail moves further towards the outer part of the wheel, and the contact point moves with it, occasionally at an even faster rate, due to the non-linearity associated with wheel-rail contact. As the wheel shape is approximately conic, with external track vertexes, the instantaneous radius on that wheel is less. This means that the instantaneous rotation centre comes closer to the bicone, situated on the same side as the stock rail curve. This brings about two consequences that are described below:

1. Since it is located on the same side as the curve stock rail, the elemental movement of the wheelset is directed against the straight switch rail.

2. The radiuses of the elemental trajectories will become increasingly smaller; in other words, the tendency to impact the switch rail (i.e. the force) will increase, as the hunting angle becomes larger.

From the above synopsis we can conclude that axle behaviour is characterised by its centre looking for the track axle. Consequently, when variations occur in the gauge, as is the case of the initial deflecting device zones, where the track appears to widen in terms of the rail axles (the contact points of each axle wheel become further apart as the deflecting device advances), the axle moves in an attempt to situate itself in the true centre of the track. If it meets the straight switch rail during the course of this lateral displacement (that is, if the track clearance finishes), the corresponding wheel flange comes into contact with the straight rail. Once the wheel rolls over the straight rail (situation t_6), the track has suddenly become narrower for the axle and, depending on its relative position in accordance with this new situation, will once more tend to situate itself centred on the track.

This disturbance in vehicle movement causes an impact on the deflecting device elements, and explains the wear on the straight switch rail. The consequences of this are described below:

- Wear of the deflecting device elements, especially the straight switch rail.
- Impact on the deflecting device elements and vehicle running gear.
- Higher maintenance costs, especially for deflecting device elements.
- Noise
- Reduced comfort.

3.2. The dynamic numerical model

The quasistatic model described in the preceding section does not enable the simulation of the real behaviour of an axle bearing mass, connected by a set of elastic links to the bogie frame of a vehicle running at high speed. The aim of these paragraphs is to provide an explanation, which is as intuitive as possible, for the complex phenomenon occurring to the movement of railway vehicles on track deflecting devices.

A detailed explanation of a specific dynamic model to simulate the front bogie behaviour of a High Speed train, such as the TGV or AVE, with 7 grades of freedom (lateral displacement and hunting turn on each axle; lateral displacement, hunting turn and bogie frame rolling) is presented in [2].

In order to improve the original design of the High Speed turnout developed by Felguera Melt S.A., the authors developed a model based in Simpack® software. This improved model simulates the behaviour of the Renfe's AVE s. 100 emu, and it allows:

- To simulate the circulation of a High Speed rail vehicle along the complete turnout with enough precision.
- To analyze the variables those intervene in this phenomenon: speed, lateral displacements of wheelsets, yaw angles, wheel profile, rail profiles (in both cases, new and wear ones) and the sense of circulation (from deflecting device to crossing or viceversa).
- To act on the 3D geometry of the deflecting device and in the frog area to avoid or to reduce the local problems that could appear.

In the results shown in this paper, a Renfe's AVE s. 100 emu with NF01-112 wheel profile was modellized.

3.3. Results without geometry optimization

The figure 7 shows the typical results obtained for the lateral displacement of the guiding wheelset. The discontinuities in the rolling contact and their effect in the dynamics of vehicle movements can be observed. The deflection device of this turnout is at right hand.

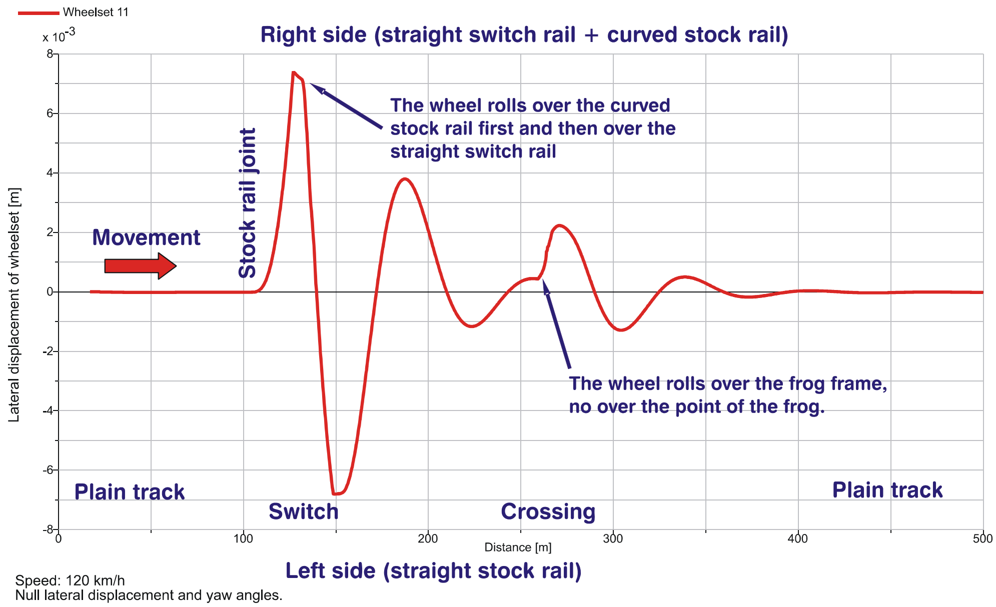

Figure 7.

The figure 8 shows the trajectory of the guiding wheelset across the referred turnout at different speeds (between 120 and 400 km/h). As the speed increases, the influence of the discontinuities in the rolling contact of the wheels in the dynamic behaviour of the vehicle is smaller.

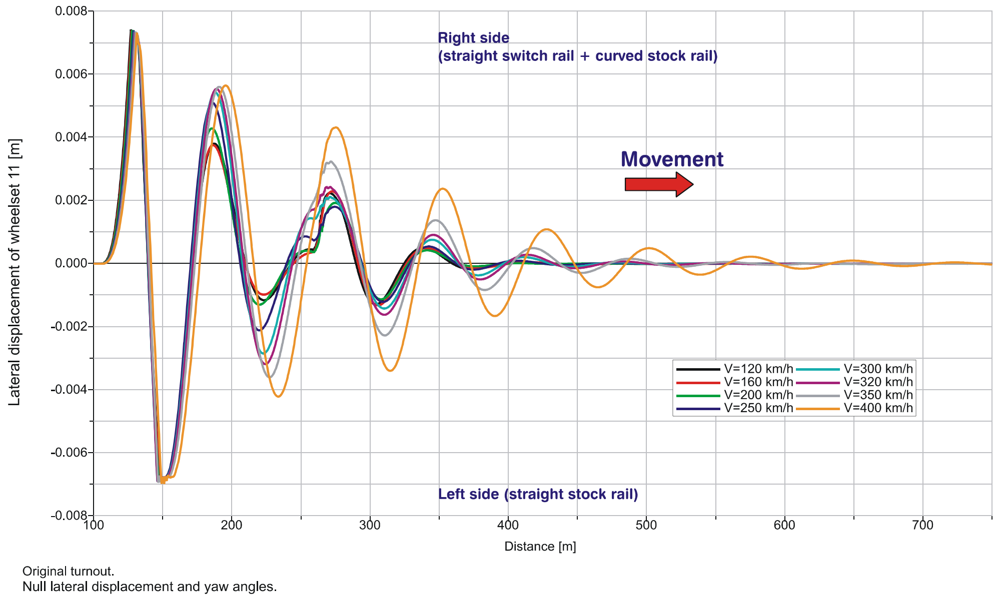

Figure 8.

The figure 9 shows the lateral forces (Y) produced in each wheel of the guiding wheelset when the vehicle runs at 300 km/h. The disruption when the wheel rolls over the curved stock rail (vehicle running in the straight branch) is clearly visible.

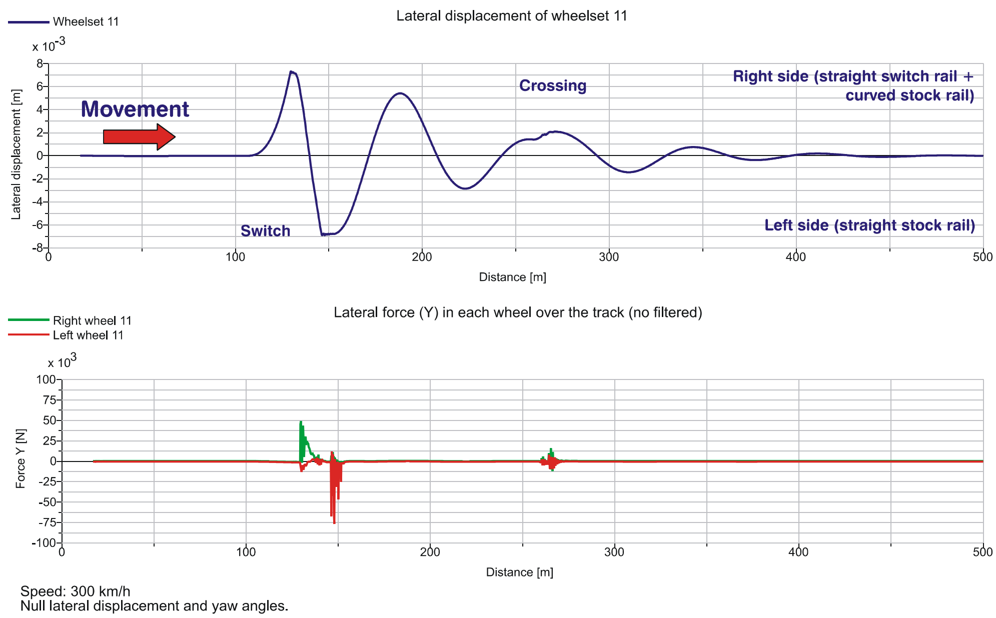

Figure 9.

3.4. Results with geometry optimization

With the aim to reduce the disruptions in the movement of the vehicles due to the variation of the rolling contacts in the turnout, two technologies had been developed to date:

- FAKOP® system (Fahrkinematische Optimierung; at the moment also well-known as KGO®, Kinematic Gauge Optimisation). It is based on displacing the straight stock rail to offset the movement of the wheel-rail contact in the opposite wheel.

- CATFERSAN system. It is based on machining the upper surface of the straight stock rail to offset the same objective than in the previous paragraph. This solution is inspired in the asymmetrical machining of rails [8].

These technologies are described in reference [2].

The figure 10 shows the lateral displacement of the guiding wheelset of the vehicle running across the turnout (switch - crossing direction) with CATFERSAN and FAKOP® technologies, compared with the original result. The achieved improvement is clear.

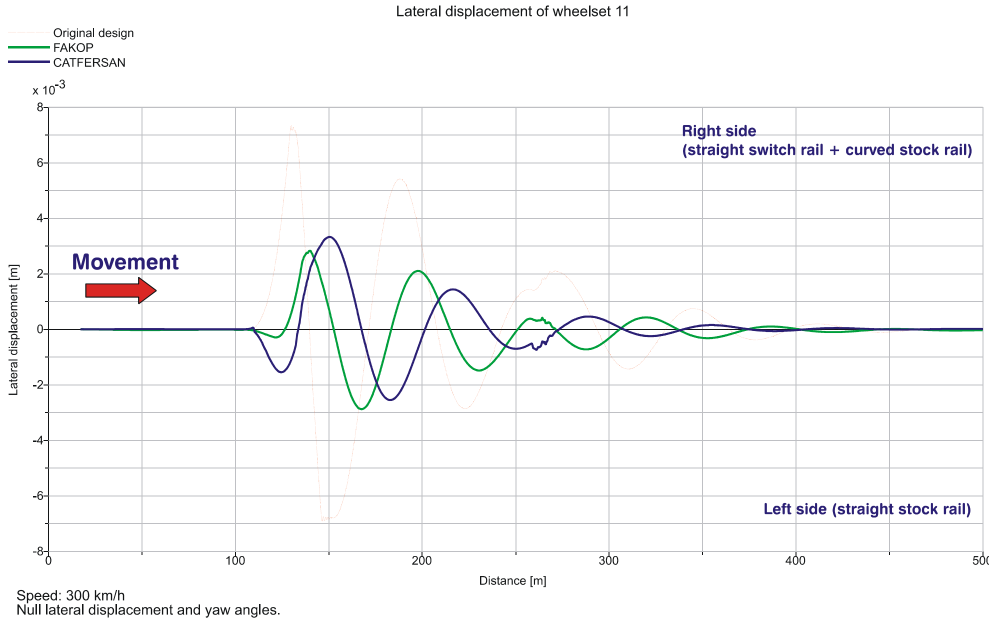

Figure 10.

The benefits of the application of these techniques to the design of the turnouts are patent when the lateral efforts in each wheel (Y) are observed (figure 11). The disruption in the frog area has been practically eliminated by means of a modification of the original machined profile.

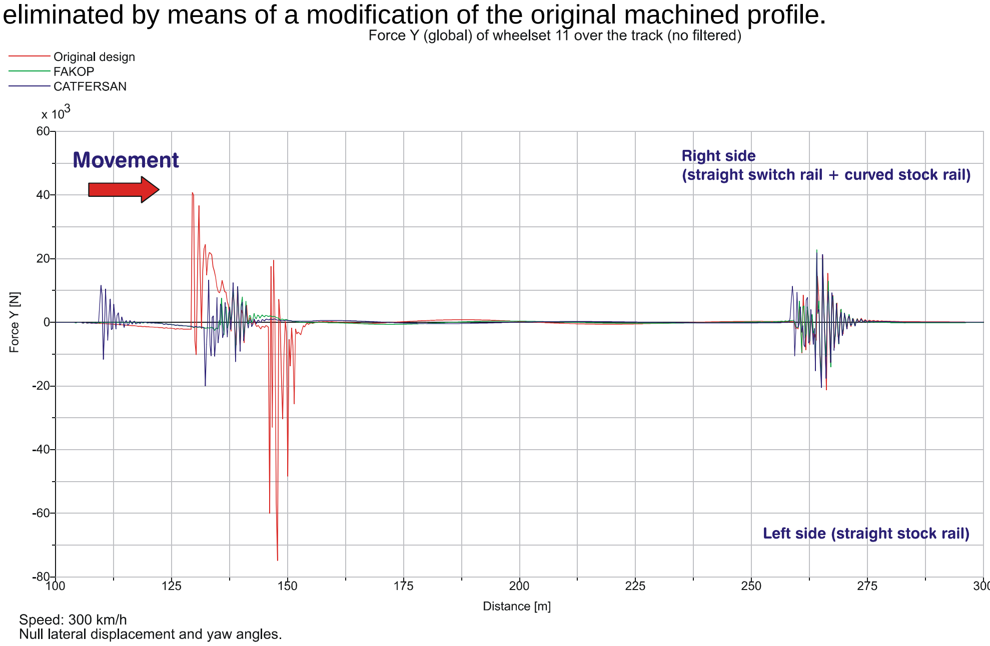

Figure 11.

ACKNOWLEDGEMENTS

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