Characterization of Train-Track Interactions based on Axle Box Acceleration Measurements for Normal Track and Turnout Passages

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ABSTRACT: Railway turnouts feature a switch panel and a crossing panel that are connected by a closure panel. Dynamic interaction between the vehicle and the track is more complex when compared to tangent or curved tracks. Large wheel-rail impact loads with significant contributions from high-frequency vehicle-track interactions are generated when the nominal wheel-rail contact conditions are disturbed at various locations, i.e. during wheel transfer from wing rail to crossing nose [1]. The impact load at the crossing nose may be a significant contributor to ground vibration and noise. The switch panel and the crossing panel are the most important locations for the generation of ground vibrations as they feature changes in the rail cross-section, the track curvature, track irregularities and track stiffness variations [2]. Vibration mitigation measures such as soft under sleeper pads are assumed to influence the train-track interaction significantly. Optimization and improvement of current turnout designs requires an advanced understanding of the dynamic interaction mechanisms between vehicle and track. In the present paper, the influence of the rail profiles on the excitation of vibrations is investigated for different turnout geometries by evaluating axle box acceleration measurements. Measurements show the influence of the crossing panel geometry on the acceleration of the wheel sets. Axle box acceleration measurements at representative turnout crossing speeds are analyzed to quantify the influence of under sleeper pads on the wheel-rail interaction.

KEY WORDS: Train-Track Interactions, Axle Box Acceleration Measurements, Turnouts

1 INTRODUCTION

In railway networks, crossings and turnouts are key components to guarantee modularity and flexibility. Turnouts are used to allow two rail tracks to intersect at the same level. These turnouts contain a switch panel and a crossing panel, connected by a closure panel, as illustrated in Figure 1. The dynamic interaction between vehicle and track is more complex during a turnout crossing when compared to a normal track. Wheel-rail interactions with large amplitudes and significant contributions from high-frequency vehicle-track interactions are generated when the nominal wheel-rail contact conditions are disturbed by a turnout, i.e. during wheel transfer from wing rail to crossing nose. The impact load at the crossing nose is assumed to be a significant source for ground vibration and noise [1].

The switch panel and the crossing panel are assumed to be the most important sources for ground vibration as they feature changes in the rail cross-section, in the track curvature and in the track stiffness [2]. At the point of the intersection, the change in wheel-rail contact is quite abrupt, leading to impacts and jumps of the wheels. In contrast to normal track superstructure, where train-induced vibrations are mainly related to track irregularities, the vibratory response of turnouts is dominated by impacts [4].

2 OBJECTIVE & MOTIVATION

Turnout maintenance, involving safety aspects, is one of the main concerns of railway infrastructure owners besides the strong need of reducing their vibro-acoustic impact.

The objective of the present work is to use existing axle box acceleration measurement data to investigate the vehicle-track interactions for different turnout locations, turnout types (i.e. radius, with/without under sleeper pads) and the influence of maintenance work and wear on the resulting axle box acceleration levels during a turnout crossing. The influence of maintenance work and the degradation of the turnout over time on the resulting acceleration levels of the wheel-set are examined using data acquired from 2009 to 2013 by the Swiss Railways, SBB, for different turnout geometries.

It is assumed that higher vehicle-track interaction amplitudes result in stronger turnout wear and higher ground vibration levels. Therefore, the optimization and improvement of current turnout designs towards lower vibro-acoustic
impacts and reduced maintenance costs requires an advanced understanding of the dynamic interaction mechanisms between vehicle and track.

3 EXPERIMENTAL SET-UP

The Swiss Railways, SBB, is performing track condition monitoring measurements twice a year since 2007 on a significant part of the railway infrastructure. These regular tests also consist of lateral and vertical axle box as well as lateral bogie acceleration measurements. For the measurements, an EW Type IV wagon is equipped with Setra Model 131A acceleration sensors with a range of ±30g, as illustrated in Figure 2. The acceleration sensor signals are filtered with an anti-aliasing filter (Butterworth low-pass at 200Hz) and acquired with a sampling rate of 1kHz.

Figure 2: Axle box equipped with lateral and vertical acceleration sensors [5]

Two different axle boxes (front and rear bogie) on the measurement wagon are instrumented with acceleration sensors on both sides of the axle. Sensors signals are used for the investigation on the side of the axle where the crossing nose is touched.

4 RESULTS

4.1 Investigated Turnout Types

In the present paper, resulting axle box acceleration amplitudes for two different turnout geometries, one with a radius of 900m (EW VI-900-G/B-1:19-F/Be,R(T) type) and one with a radius of 300m (EW VI-300-G-1:19-F/Be,R(T) type) are discussed. For the turnout with a radius of 900m the effect of under sleeper pads, USP, on the resulting dynamic vehicle-track interaction is investigated.

4.2 Crossing Speed Dependent Spatial Resolution

With a constant sampling rate of 1kHz and a varying turnout crossing speed the resulting spatial/temporal resolution of the dynamic vehicle-track interaction differs depending on the actual train speed. Therefore, in the present paper only measurements at a turnout crossing speed of 150km/h – 160km/h are used to compare the dynamic vehicle-track interaction levels for the different turnout types and turnout conditions.

Typical acceleration sensor raw signals are shown in Figure 3. The measured acceleration amplitudes on axle box 1 (front bogie) and axle box 2 (rear bogie) on the same wagon are plotted relative to time. The corresponding frequency content from a turnout crossing is shown in Figure 4. The interaction of the wheel-set with the turnout does not show up as a sharp impulse type excitation in the acceleration sensor raw signal followed by a decay of the acceleration amplitude. There is an interaction event with high acceleration amplitudes over a certain time interval instead.

Figure 3: Acceleration Sensor Signal in Vertical Direction

In the corresponding frequency content the effect of the Butterworth low-pass filter at 200Hz is clearly observable in the acceleration sensor signal spectrum. The peak amplitude is visible at 122.6Hz caused by the excitation of the axle box by the crossing panel. The content from about 65Hz to about 70Hz in the spectrum is assumed to be the result of the nearly periodic excitation due to stiffness variations of the rail supported by a sleeper versus unsupported rail between the sleepers (parametric excitation). A nominal turnout crossing speed of 160km/h and a response at 70Hz leads to a realistic sleeper spacing of about 0.6m.

Figure 4: Frequency Content in the Acceleration Sensor Signal in Vertical Direction

4.3 Peak Acceleration Amplitudes

In a first comparison, peak acceleration amplitudes are used to draw conclusions about the effect of the turnout radius and the installation of soft under sleeper pads on the resulting axle box acceleration amplitudes, both in vertical and lateral directions.

In Figure 5 the measured peak acceleration amplitude in vertical direction for measurements performed from 2009 till 2012 are shown. The measured maximum acceleration amplitudes in vertical direction for four EW VI-900-G/B-1:19-F/BE,R(T) turnout types with a radius of 900m are illustrated for the acceleration sensor on the axle box side where the wheel touches the crossing nose. The peak acceleration amplitudes were normalized with the measured maximum acceleration amplitude in vertical direction throughout the whole measurement period and for all the different turnout types. Turnouts 1 and 2 represented in Figure 5 are equipped with under sleeper pads whereas turnouts 3 and 4 are not equipped with under sleeper pads. The four turnouts are all installed at the same location in the railway.
network. The turnouts were all installed in 2005 and show the same maintenance state. In 2010 rail grinding was performed but as can be seen in Figure 5 the rail grinding has no significant effect on the dynamic vehicle-turnout interaction levels. However, there is an effect of rail grinding on the acceleration amplitudes on normal track in the close vicinity to the turnouts [5]. The root mean square values of the acceleration amplitudes in vertical direction on normal track decreased after rail grinding was applied.

Figure 5: Measured Maximum Acceleration Amplitudes, Vertical Direction, EW VI-900-G/B-1:19-F/Be,R(T) Type

The resulting maximum acceleration amplitudes in vertical direction on the axle box for the measurements performed in January 2009 and in May 2009 show a tendency for higher acceleration amplitudes for turnouts 1 and 2 equipped with under sleeper pads and lower amplitudes for turnouts 3 and 4 without under sleeper pads. However, this trend is not found for the whole series of measurement results. In the three data sets from April 2011 to July 2012 turnout 1 tends to show highest amplitudes and the crossing of turnout 3 resulted in lowest peak acceleration amplitudes. In June 2010 two measurements were performed. The results show deviations in peak acceleration amplitudes of more than 15% except for turnout 1. This difference should be interpreted as a measurement uncertainty and needs to be taken into account in the comparison of different turnout types.

Figure 6: Averaged Maximum Acceleration Amplitudes, Vertical Direction, EW VI-900-G/B-1:19-F/Be,R(T) Type

In Figure 6 the averaged normalized maximum acceleration amplitude in vertical direction is plotted including the standard deviation represented as error bars. Although the number of samples for the statistical analysis is rather low with 8 measurements the results show the trend to higher peak acceleration amplitudes by a factor of about 1.15 for turnouts 1 and 2 with under sleeper pads when compared to turnouts 3 and 4. The standard deviation was found to be in a comparable range for the four data sets (5.6% - 7.8%).

The effect of the turnout radius on the resulting vertical axle box acceleration amplitudes is illustrated in Figure 7. Turnout 5 and turnout 7 are of the EW VI-900-G/B-1:19-F/Be,R(T) type with a radius of 900m whereas turnout 6 is a EW VI-300-G-1:19-F/Be,R(T) type with a radius of 300m. Turnout 5 was replaced by a newly developed turnout equipped with under sleeper pads in November 2012. All the three turnouts are installed at the same location in the railway network.

Figure 7: Measured Maximum Acceleration Amplitudes, Vertical Direction, EW VI-900-G/B-1:19-F/Be,R(T) and EW VI-300-G-1:19-F/Be,R(T) Type

In general, turnout 6 with a radius of 300m tends to result in highest peak acceleration amplitudes in vertical direction. Turnout 6 with a radius of 300m shows the trend to a decrease of the maximum acceleration amplitudes in vertical direction from 2009 to 2013. An explanation for this trend could not yet be found. Turnouts 5 and 7 show stable interaction levels for the same time period. From 2010 to 2012 turnout 7 caused higher acceleration amplitudes when compared to turnout 5. This tendency changes for the measurements done in 2013 where turnout 5 was replaced by a newly developed turnout with under sleeper pads. The measurements performed in January 2013 show higher and the measurements performed in July 2013 show similar peak acceleration amplitudes for the newly installed turnout when compared to turnout 7.

Figure 8: Averaged Maximum Acceleration Amplitudes, Vertical Direction, EW VI-900-G/B-1:19-F/Be,R(T) and EW VI-300-G-1:19-F/Be,R(T) Type
Turnouts 5 and 7 with a radius of 900m show the tendency to result in comparable acceleration amplitudes in average. Overall, turnout 5 causes 6% - 7% lower averaged maximum acceleration amplitudes in vertical direction when compared to turnout 7. The two measurements performed in June 2010 show a deviation between the two data sets of up to 10%. Again this variation of the measured maximum acceleration amplitudes on the axle box can be interpreted as measurement uncertainty, strongly influenced by the spatial and temporal resolution due to the data acquisition system with a sampling rate of 1kHz.

This is confirmed in Figure 8 where the averaged normalized maximum acceleration amplitudes are plotted for turnout 5, turnout 6 and turnout 7. The standard deviation for turnout 6 with a radius of 300m is larger than 23%. The reason for the large standard deviation can be seen in Figure 7 where the peak acceleration amplitudes were higher by more than 60% for the measurements performed in January and May 2009 in comparison to the measurements performed from 2010 to 2013.

The effect of the turnout radius on the resulting lateral axle box acceleration amplitudes is done based on measurement results for turnouts 5, 6 and 7. Figure 9 illustrates the averaged maximum acceleration amplitudes in lateral direction for turnout crossings from 2009 till 2013 for the non-deviated turnout pass-by.

4.4 Impulse Energy

A comparison based on peak amplitudes significantly depends on the temporal resolution of the impulse type interaction between axle box and track structure. With the measurement set-up used for the present data set the detailed sampling of the interaction event is limited by the sampling rate of the data acquisition system of 1kHz. At a nominal turnout crossing speed of 160km/h the spatial resolution is limited to approximately 0.044m.

An approach to partially compensate for the risk of missing the peak acceleration amplitude during data acquisition (due to the limited sampling rate) is a comparison of the interaction levels of the axle box with the track structure based on acceleration signal energy levels. This approach also takes into account for slightly varying turnout crossing speeds for the different measurement runs.

The vehicle-track interaction for normal track conditions is quantified by the root mean square value in the close vicinity of the turnout. However, the resulting rms-value strongly depends on the combined wheel-rail roughness level. Therefore, the resulting acceleration amplitudes for normal track conditions are influenced by the wheel condition of the measurement wagon. The measurement wagon is used for track condition monitoring purposes only and the wheel condition can be assumed to be constant for the measurement period from 2009 to 2013.

In Figure 10 the normalized root mean square acceleration values for tracks with under sleeper pads and for tracks without under sleeper pads are presented.

The acceleration amplitudes are normalized with the same reference value as for the amplitudes in vertical direction. Therefore, the results can be directly compared to the results presented for the vertical direction.

Turnout 6 with a radius of 300m resulted in higher maximum acceleration amplitudes in lateral direction by a factor of 1.34 and 1.44 respectively. Turnout 5 shows 7.5% higher averaged peak acceleration amplitudes in lateral direction when compared to turnout 7. However, this difference in the averaged maximum acceleration amplitude in lateral direction is within the standard deviation for the present data set. In general, maximum acceleration amplitudes in vertical direction are larger by a factor of 4.5 to 6.8 when compared to the amplitudes measured in lateral direction.

Normal tracks with under sleeper pads tend to result in higher root mean square acceleration values in vertical direction in the close vicinity of the turnouts. The rms-values for normal tracks without under sleeper pads were lower by about 44%.

These root mean square values could be interpreted as background noise levels for the evaluation of the impulse energies presented in Figure 13. However, for the impact type interaction of the turnout geometry with the railway vehicle this background noise level is assumed to be negligible due to the comparably small amplitudes. As illustrated in Figure 4, the impulse excitation from the turnout is present at about
120Hz whereas the relevant contributions on the normal track are present at about 72Hz in the spectrum presented in Figure 11 for the track with under sleeper pads and at about 74Hz for the track without under sleeper pads (see Figure 12). The two peaks in the rms-spectrum correspond to the sleeper spacing of about 0.6m. This quasi-periodic excitation of the axle box on normal track results in acceleration amplitudes in the order of 6m/s² to 7m/s². The response caused by the quasi-periodic sleeper excitation tends to be larger for the normal track with under sleeper pads as shown in Figure 11 and in Figure 12. The amplitudes at other frequencies than the quasi-periodic excitation frequency are assumed to be insignificant as their values are below 1.2m/s². Another reason for neglecting the root mean square value in the calculation of the acceleration signal energy is the frequency dependent dynamic stiffness of the under sleeper pads. It is assumed that the stiffness of the under sleeper pads increases significantly for an impulse type dynamic loading of the pads.

In the calculation of the impulse energy in the accelerometer signal, the contribution from the combined wheel-rail roughness and from the quasi-periodic excitation of the axle box by the sleepers is therefore not separately accounted for.

The calculation of the impulse energy in the acceleration signal during the turnout crossing is done based on limits set by the rms-values before and after the crossing nose. The calculation is started at the location where the root mean square acceleration value over a window of 1m exceed the rms-values upstream of the crossing nose and are finished where the root mean square acceleration values over a window of 1m have decreased below the rms-value downstream of the crossing nose. The identification of the relevant stretch is illustrated in Figure 13.

![Figure 13: Relevant Stretch for Calculation of Signal Energy in Acceleration Signal](image)

In Figure 14 the resulting signal energy for the accelerometer signals in vertical direction is shown for turnouts 1 to 4 with a radius of 900m for measurements performed from 2009 to 2012. Turnouts 1 and 2 are equipped with under sleeper pads.

![Figure 14: Normalized Acceleration Signal Energy, Vertical Direction, EW VI-900-G/B-1:19-F/Be,R(T) Type](image)

In contrast to the comparison of peak acceleration amplitudes for the turnouts 1 to 4 in Figure 5, Figure 14 shows that for the signal energy, turnouts 1, 2 and 3 tend to result in higher energy content and turnout 4 in lowest energy levels in the accelerometer signal in vertical direction. This is also represented by mean acceleration signal energy levels, illustrated in Figure 15, for turnouts 1 to 4 for the measurements performed from 2009 to 2012. This trend is in good agreement with acceleration measurements at the turnout frog reported in [1]. The comparison of averaged peak acceleration amplitudes in vertical direction (see Figure 6) showed comparable amplitudes for turnouts 3 and 4. Furthermore, the standard deviation presented in Figure 15 is slightly reduced by using acceleration signal energy levels for the comparison of different turnout configurations instead of
peak acceleration amplitudes that strongly depend on the spatial/temporal sampling of the accelerometer signal.

In the averaged acceleration signal energy there is no effect from the under sleeper pads observable in the resulting energy levels. In the comparison of maximum acceleration amplitudes in vertical direction, turnouts 1 and 2 equipped with under sleeper pads resulted in higher peak amplitudes.

**Figure 15:** Averaged Normalized Acceleration Signal Energy, Vertical Direction, EW VI-900-G/B-1:19-F/Be,R(T) Type

In Figure 16 the normalized acceleration signal energy levels are shown for turnouts 5 to 7. Turnouts 5 and 7 have a radius of 900m whereas turnout 6 has a radius of 300m. In terms of acceleration signal energy, turnout 6 with a small turnout radius resulted in highest energy levels. This tendency corresponds well with the comparison of peak acceleration amplitudes presented in Figure 7, where highest acceleration amplitudes were observed for the turnout with a radius of 300m.

**Figure 16:** Normalized Acceleration Signal Energy, Vertical Direction, EW VI-900-G/B-1:19-F/Be,R(T) and EW VI-300-G-1:19-F/Be,R(T) Type

The averaged normalized acceleration signal energy levels for turnouts 5 to 7 in Figure 17 show comparable levels for turnout 5 and for turnout 7.

However, the comparison with turnouts 1 to 4 with the same turnout radius of 900m but installed at a different location in the railway network shows a significant difference in the resulting acceleration signal energy levels as well as in the resulting maximum acceleration amplitudes in vertical direction. The turnouts investigated in the present work are installed on the same line in the railway network. Therefore, the traffic mix can be assumed to be the same for all the turnouts. The significant difference in the resulting peak acceleration amplitudes and signal energy levels for similar types of turnouts but installed at different locations might be related to geological properties leading to different vehicle-turnout interactions as well as geometric tolerances after the installation of the turnout modules.

**Figure 17:** Averaged Normalized Acceleration Signal Energy, Vertical Direction, EW VI-900-G/B-1:19-F/Be,R(T) and EW VI-300-G-1:19-F/Be,R(T) Type

5 SUMMARY & CONCLUSIONS

Existing axle box acceleration measurement data for turnout crossings from 2009 to 2013 were analyzed to investigate the vehicle-track interactions for different turnouts (geometry, under sleeper pads). Higher vehicle-track interaction levels are assumed to result in stronger turnout wear and in higher noise and ground vibration levels. The optimization and improvement of current turnout designs towards lower maintenance costs and lower vibro-acoustic impacts requires an advanced understanding of the dynamic interaction between vehicle and track structure.

Axle box acceleration amplitudes were measured using a measurement wagon equipped with accelerometers on two different axle boxes on the front and on the rear bogie. The comparison of peak acceleration amplitudes in vertical direction on the axle box showed the trend to higher peak amplitudes by a factor of about 1.15 for turnouts with under sleeper pads when compared to turnouts without under sleeper pads.

Rail grinding was found to have no significant effect on the resulting dynamic interaction between turnout and axle box. However, rail grinding reduced the root mean square acceleration values on the track near the crossing nose. Normal tracks near the turnout equipped with under sleeper pads showed higher root mean square acceleration values in vertical direction. The root mean square acceleration values on the axle box in vertical direction for normal tracks without under sleeper pads were lower by about 44%.

A turnout with a radius of 300m caused higher peak acceleration amplitudes on the axle box in vertical as well as in lateral directions when compared to the amplitudes caused by a turnout with 900m at the same location in the railway network (all measurements with a non-deviation pass-by).
Axle box acceleration amplitudes in lateral direction were larger by a factor of about 1.34 – 1.44 for a turnout with a radius of 300m in comparison to a turnout with a radius of 900m.

In general, peak acceleration amplitudes on the axle box are larger by a factor of about 4.5 – 6.8 in vertical direction when compared to the acceleration amplitudes on the axle box in lateral direction.

The limited sampling rate of 1kHz of the data acquisition system used for the axle box acceleration measurements resulted in a spatial resolution of about 0.044m at a nominal turnout crossing speed of 160km/h. Therefore, the investigation of the dynamic vehicle-turnout interaction was also done based on acceleration sensor signal energy levels. This approach partially compensates for the rather low spatial resolution and decreases the risk of missing the peak acceleration amplitude on the axle box caused by the impulse type excitation from the turnout.

The comparison of vertical axle box acceleration amplitudes based on acceleration signal energies was in good agreement with frog acceleration measurements performed at the same turnouts. The comparison of the axle box acceleration amplitudes based on peak values did not match with the turnout frog acceleration measurements.

The acceleration signal energy in vertical direction averaged over several turnout crossings does not show any effect of the under sleeper pads on the resulting energy level when compared to the same turnout geometry at the same location in the railway network but without under sleeper pads. The frequency dependent dynamic stiffness of the under sleeper pads and their influence on the dynamic vehicle-turnout interaction needs to be further investigated.

The comparison of axle box acceleration amplitudes for similar turnout geometries but installed at different locations in the railway network showed significant differences in the peak acceleration amplitudes as well as in the acceleration signal energy during turnout crossing. The turnouts are installed at different locations in the railway network but on the same line. Therefore, the traffic mix is assumed to be same for the two turnout locations. Differences in the dynamic vehicle-turnout interaction levels may be found in the geological properties leading to different vehicle-turnout interactions as well as in geometric tolerances after the installation of the turnout.

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