

## Numerical and experimental study of wheel-rail impact vibration and noise generated at an insulated rail joint

Yang, Zhen; Boogaard, Anthonie; Chen, Rong; Dollevoet, Rolf; Li, Zili

**DOI**

[10.1016/j.ijimpeng.2017.11.008](https://doi.org/10.1016/j.ijimpeng.2017.11.008)

**Publication date**

2018

**Document Version**

Accepted author manuscript

**Published in**

International Journal of Impact Engineering

**Citation (APA)**

Yang, Z., Boogaard, A., Chen, R., Dollevoet, R., & Li, Z. (2018). Numerical and experimental study of wheel-rail impact vibration and noise generated at an insulated rail joint. *International Journal of Impact Engineering*, 113, 29-39. <https://doi.org/10.1016/j.ijimpeng.2017.11.008>

**Important note**

To cite this publication, please use the final published version (if applicable). Please check the document version above.

**Copyright**

Other than for strictly personal use, it is not permitted to download, forward or distribute the text or part of it, without the consent of the author(s) and/or copyright holder(s), unless the work is under an open content license such as Creative Commons.

**Takedown policy**

Please contact us and provide details if you believe this document breaches copyrights. We will remove access to the work immediately and investigate your claim.

# 1 Numerical and experimental study of wheel-rail impact vibration 2 and noise generated at an insulated rail joint

3 Zhen Yang<sup>1,2</sup>, Anthonie Boogaard<sup>1</sup>, Rong Chen<sup>2</sup>, Rolf Dollevoet<sup>1</sup>, Zili Li<sup>1</sup>

4 1. Delft University of Technology, Section of Railway Engineering, Stevinweg 1, 2628 CN,  
5 Delft, the Netherlands

6 2. MOE Key Laboratory of High-Speed Railway Engineering, Southwest Jiaotong University,  
7 Chengdu, Sichuan, China

## 8 9 Abstract

10 As essential track components for rail connections and signal transmission and control,  
11 insulated rail joints (IRJs) have been widely used in traditional and high-speed railways.  
12 However, the IRJ is considered as one of the weakest parts of railway track structures due to  
13 the significant discontinuities in stiffness and geometry. When a train runs over an IRJ, a  
14 wheel-rail impact occurs and it increases with train speed. The impact consequently leads to  
15 vibration and noise and accelerates track deterioration in the vicinity of the IRJ. This paper  
16 establishes an explicit finite element wheel-IRJ dynamic interaction model to simulate high-  
17 frequency impact vibration and noise generated at a typical IRJ in the Dutch railway network,  
18 and validates the model against a comprehensive hammer test and a pass-by measurement.  
19 Good agreements between the simulation and measurements indicate that the proposed model  
20 can effectively reproduce high-frequency impact vibration and noise up to 10 kHz. This paper  
21 also connects the dominant frequencies of wheel-IRJ impact vibration and noise with the  
22 dynamic behaviour of the target IRJ, which may contribute to the mitigation of impact  
23 vibration and noise at IRJs as well as to train-borne detection of deterioration types of IRJs.

24

25 **Keywords:** insulated rail joint (IRJ); explicit FEM; hammer test; pass-by; impact vibration  
26 and noise

27

## 28 **1 Introduction**

29 As essential track components for rail connections and signal transmission and control,  
30 insulated rail joints (IRJs) have been widely used in both traditional and high-speed railways.  
31 The performance of IRJs directly influences railway transportation safety. However, due to  
32 the significant stiffness and geometric discontinuities, the IRJ is considered as one of the  
33 weakest parts of railway track structures. When a train runs over an IRJ, wheel-rail impact  
34 occurs. This impact consequently leads to vibration and noise and accelerates track  
35 deterioration in the vicinity of the IRJ.

36

37 Numerous modelling work has been carried out to simulate wheel-rail impacts generated by  
38 IRJs. Because impact problems are inherently non-linear, the simulation models are required  
39 to be solved in the time domain. Computationally efficient analytical models have been  
40 applied to calculate wheel-rail interactions when material complexity and detailed contact  
41 solutions are of less concern [1-4]. Considering a reduction of bending stiffness at a joint,  
42 Kerr et al. [1] simplified the joint of a track model as a gap and connected rails and fishplates  
43 with Winkler-type springs, whose stiffness was calibrated by a static load measurement.  
44 Mandal et al. [2] idealised the dipped rail joint with a sinusoidal profile to analyse the impact  
45 forces considering both track design and operational parameters. Wu and Thompson [3]  
46 treated the joint of their model as a pin between two semi-infinite Timoshenko beams and  
47 calculated the wheel-IRJ interaction by a relative displacement excitation model proposed by  
48 Grassie et al. [5]. Kitagawa et al. [4] modified Wu and Thompson's model by replacing the  
49 pin with a complex spring to represent the joint, whose vibratory behaviour level was

50 validated against a field test. However, these analytical models fail to address the issues of the  
51 complex wheel-IRJ impact contact solutions and high-frequency impact dynamics over 5 kHz.

52

53 With the development of computer technology, the numerical finite element method (FEM)  
54 has increasingly been used to predict degradations of IRJs due to complex wheel-IRJ impact  
55 contact [6-12]. Owing to the capability of coping with non-linear material properties and  
56 arbitrary discontinuous contact geometries, finite element (FE) contact models can provide  
57 more accurate and detailed contact solutions. Nevertheless, many FE IRJ models tend to apply  
58 prescribed wheel loads as their excitations: either static [6-9] or pre-calculated by a simplified  
59 wheel-rail interaction model [10]; hence high-frequency dynamic effects caused by impact  
60 contact cannot be fully considered [11].

61

62 To take account of complex impact contact and high-frequency wheel/rail dynamic effects,  
63 this study simulates a wheel-rail impact at a typical IRJ in the Dutch railway network by an  
64 explicit FEM, which has been proven to be effective for solving impact contact problems [12-  
65 15] and high-frequency wheel/rail dynamics [16, 17]. Performing the integration in the time  
66 domain with an explicit central difference scheme, the explicit FEM manages to reproduce  
67 high-frequency dynamic effects by detailed modelling of the structure of wheel/track system  
68 and employing a small time step. Moreover, the calculations of high-frequency wheel/rail  
69 dynamic responses and contact forces can be automatically coupled, as the wheel/rail dynamic  
70 responses calculated in each time step rely on the contact forces obtained in the previous time  
71 step and in return affect the contact forces updated in the next time step.

72

73 This paper first establishes a three-dimensional (3D) FE wheel-IRJ interaction model to  
74 simulate impact vibration and noise excited by the target IRJ, and then validates the model

75 against a comprehensive hammer test and a pass-by measurement. Compared to the explicit  
76 FE impact models proposed in the previous research [12-17], this model is more sophisticated  
77 in track dynamics and comprehensively validated. The condition of the target IRJ and wheel-  
78 IRJ interaction model are described in Section 2. In Section 3, the track sub-model with an  
79 IRJ is calibrated and its dynamic behaviour is validated against a hammer test; the dynamic  
80 behaviour of the wheel sub-model is validated by a measurement reported in the literature  
81 [18]. Section 4 reports a pass-by measurement to validate the high-frequency (up to 10 kHz)  
82 impact vibration and noise predicted by the FE impact model. The main conclusions are  
83 drawn in Section 5.

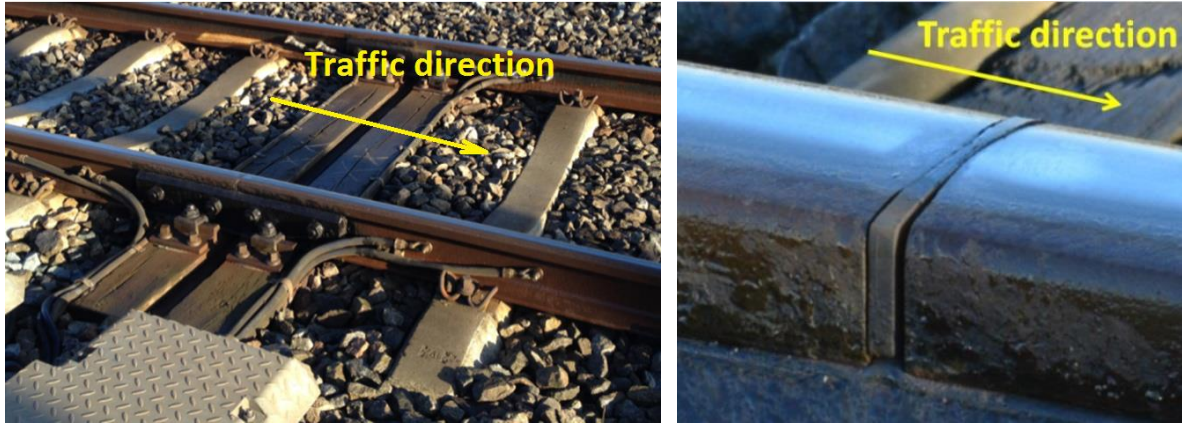
84

## 85 **2 Finite element model**

### 86 **2.1 Condition of the target IRJ**

87 A typical Dutch IRJ without visible damage was selected as the study target in the trunk line  
88 Amsterdam-Utrecht of the Dutch railway network. The IRJ locates on a straight track with a  
89 maximum one-directional train speed of 130 km/h. In the track, UIC54 rails with an  
90 inclination of 1/40 are supported by NS90 sleepers every 0.6 m except in the proximity of the  
91 IRJ, where a pair of adjacent timber sleepers with a distance of 0.24 m are employed to reduce  
92 the deflection of the joint and absorb vibration caused by wheel-IRJ impacts. Fig. 1 shows the  
93 in-situ condition of the target IRJ.

94



(a) General condition

(b) Close-up of the end-post

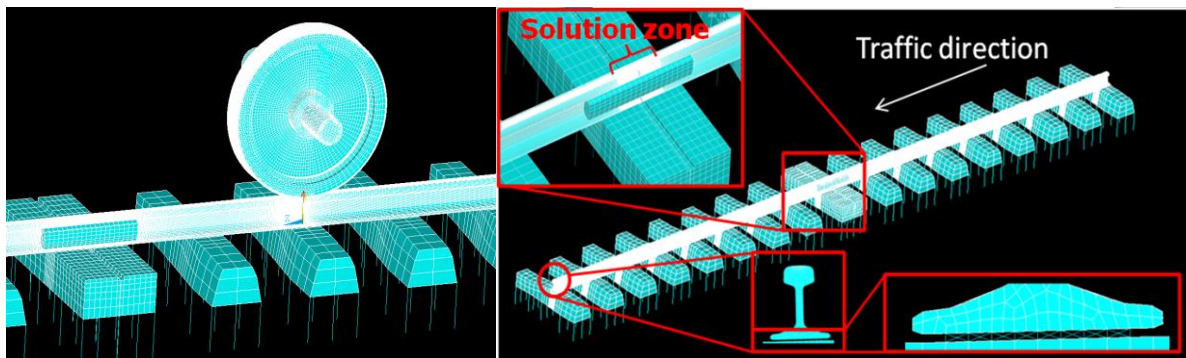
Figure 1: In-situ condition of the target IRJ

## 2.2 Wheel-IRJ interaction model

A 3D FE wheel-IRJ dynamic interaction model, as shown in Fig. 2 (a), was established in this study. The model includes a 10-m length of half-track with an IRJ in the middle and a half-wheelset with the sprung mass of the car body and bogie. The wheel geometry corresponds to that of a passenger car wheel of the Dutch railway with the standard profile of S1002. The IRJ, composed of 2 fishplates, 4 pairs of bolts and a 6 mm gap, was modelled in detail with fine meshes (see close-up at upper left corner of Fig. 2 (b)). Since the value of elastic modulus of the end-post (insulation layer between two rail ends) is much lower than those of the rails and the presence of air gap (shown in Fig. 1 (b)) may result in free rail-end [19], the end-post layer was omitted in the model and simplified as a gap. Free boundaries were used on the rail ends at the joint, whereas non-reflecting boundaries were defined at the far ends of the rails. The wheel, rail and sleepers were modelled using 8-node solid elements. To achieve accurate solutions with a reasonable model size, non-uniform meshing was used and regular discretization was allocated at the wheel-rail contact area. The mesh size is 1 mm around the initial position of wheel-rail contact and within the 0.2-m length of solution zone (see close-up at upper left corner of Fig. 2 (b)). The car body and bogie were modelled as mass elements

115 connected to the wheelset by the primary suspension of the vehicle with parallel linear springs  
116 and viscous dampers. The two neighbouring timber sleepers beneath the IRJ and the concrete  
117 sleepers modelled elsewhere were differentiated by their geometries and material properties.  
118 The baseplate fastening system used on the timber sleepers and the Vossloh fastening system  
119 on the concrete sleepers were modelled by crossed spring and damper elements (see close-up  
120 at lower right corner of Fig. 2 (b)), and adopted different linear stiffness and damping  
121 parameters (see table 1). The ballast was simplified as vertical spring and damper elements,  
122 with the displacements constrained in the lateral and longitudinal directions. Since the  
123 stiffness and damping parameters used to model the fastenings and ballast can hardly be  
124 measured directly in the field, these parameters were calibrated in this study by fitting the  
125 simulated frequency response functions (FRF) to the measurement results, which will be  
126 illustrated in detail in Section 3. Bi-linear elastoplastic material properties were applied to the  
127 wheel and rail models. The calibrated stiffness and damping parameters, as well as the  
128 nominal material properties applied in the model, are listed in table 1.

129



130

131 (a) Wheel-IRJ interaction model

(b) Track model with IRJ (IRJ sub-model)

132 Figure 2: Finite element wheel-IRJ dynamic interaction model

133

134

135

| Parameters                    |                 | Nominal values         | Parameters          |            |            | Calibrated values |
|-------------------------------|-----------------|------------------------|---------------------|------------|------------|-------------------|
| Rails and fishplates material | Young's modulus | 210 GPa                | Baseplate fastening | Vertical   | Stiffness  | 86.7 MN/m         |
|                               | Poisson's ratio | 0.3                    |                     |            | Damping    | 45000 Ns/m        |
|                               | Density         | 7800 kg/m <sup>3</sup> | Vossloh fastening   | Lateral    | Stiffness  | 150 MN/m          |
|                               | Yield stress    | 500 MPa                |                     |            | Damping    | 40000 Ns/m        |
|                               | Tangent modulus | 21 GPa                 |                     |            | Vertical   | Stiffness         |
| Timber sleeper material       | Young's modulus | 20 GPa                 | Damping             | 67500 Ns/m |            |                   |
|                               | Poisson's ratio | 0.3                    |                     | Lateral    | Stiffness  | 100 MN/m          |
|                               | Density         | 1300 kg/m <sup>3</sup> | Damping             |            | 40000 Ns/m |                   |
| Concrete sleeper material     | Young's modulus | 38.4 GPa               | Ballast             | Vertical   | Stiffness  | 45 MN/m           |
|                               | Poisson's ratio | 0.2                    |                     |            | Damping    | 32000 Ns/m        |
|                               | Density         | 2520 kg/m <sup>3</sup> | bolt pretension     |            |            | 12.5 kN           |

136

137

Table 1: The values of parameters used in the model

138

### 139 **3 Calibration and validation of sub-models**

140 In order to accurately predict wheel-IRJ impact interaction and the consequent vibration and

141 noise, the dynamic behaviour of the track and wheel sub-models is validated in this section.

142 The sub-model of the track with an IRJ, IRJ sub-model for short, was established

143 corresponding to the target IRJ of Fig. 1. As shown in Fig. 2 (b), the IRJ sub-model excludes

144 the wheel-sub model and car bodies presented in Fig. 2 (a).

145

146 The dynamic behaviour of track structures is generally characterised by frequency response

147 functions (FRFs) due to the infinite structural nature [18]. Typical resonant frequencies of a

148 track can be deduced according to its FRFs [20]. Hammer test is a widespread method for the



149 identification of the FRFs of track structures [10, 21-23], but its application specifically to  
150 IRJs is limited. Recently, Oregui et al. [24] performed a systematic hammer measurement on  
151 IRJs to assess their health conditions, but their work only investigated dynamic behaviour of  
152 the IRJs in the vertical direction.

153

154 This study conducted a more comprehensive hammer test to measure the FRFs, in terms of  
155 accelerances, of the target IRJ not only in the vertical direction but in the lateral direction as  
156 well, since both of them are responsible for the noise radiation from the track. In this paper,  
157 the measured direct accelerances (response and excitation are measured in the same direction  
158 and location [25]) are used to calibrate the stiffness and damping parameters involved in the  
159 FE IRJ sub-model by fitting their levels and resonant frequencies to the simulation results, as  
160 described in [10, 23, 26, 27]; the measured transfer accelerances (response and excitation are  
161 measured in the same direction but different locations [25]), cross-accelerances (response and  
162 excitation are measured in different directions [25]) and decay rate of the track are then used  
163 to validate the dynamic behaviour of the calibrated track sub-model.

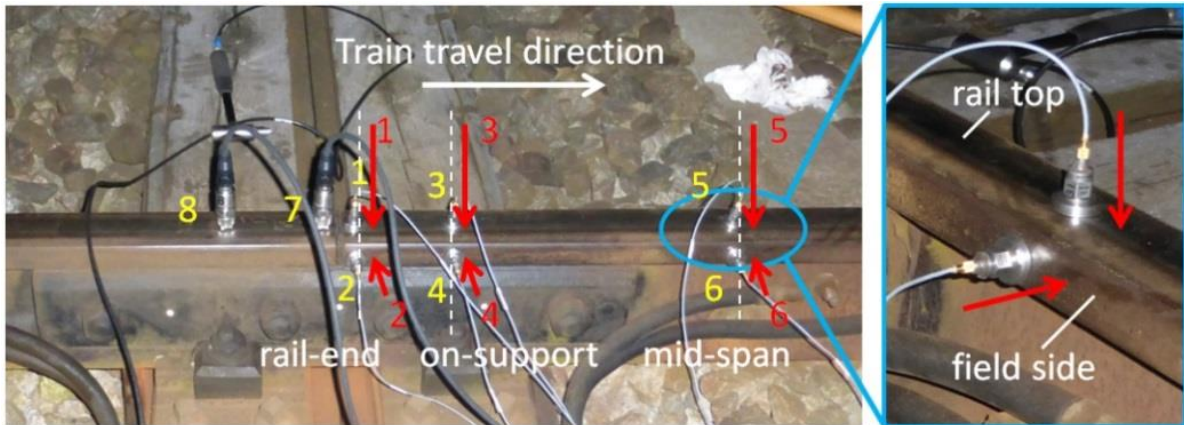
164

### 165 **3.1 Set-up of the hammer test**

166 Fig. 3 shows the set-up of the hammer test employed to identify the accelerances of the target  
167 IRJ. The FRFs of tracks without IRJs are normally measured at two track sections: on-support  
168 and mid-span [10, 20-24], where different responses are expected. This study added a third  
169 section around the rail-end just after the IRJ along the train travel direction, where a  
170 significant dynamic impact can be expected during a train pass-by. Six accelerometers (B&K  
171 4514, denoted as 1-6 in Fig. 3) were placed at the three sections: three on the rail top and the  
172 other three on the field side of the rail head (see the close-up of Fig. 3), respectively, used to  
173 measure the vertical and lateral responses. Two accelerometers (IMI623C00, ensuring electric

174 insulation between two rails in the measurement, denoted as 7 and 8 in Fig. 3) were placed on  
175 the rail top before the joint, respectively at the sections of rail-end and on-support, to detect  
176 vibration transmission to the other side of the joint.

177



178

179 Figure 3: Distribution of accelerometers and force excitation points

179

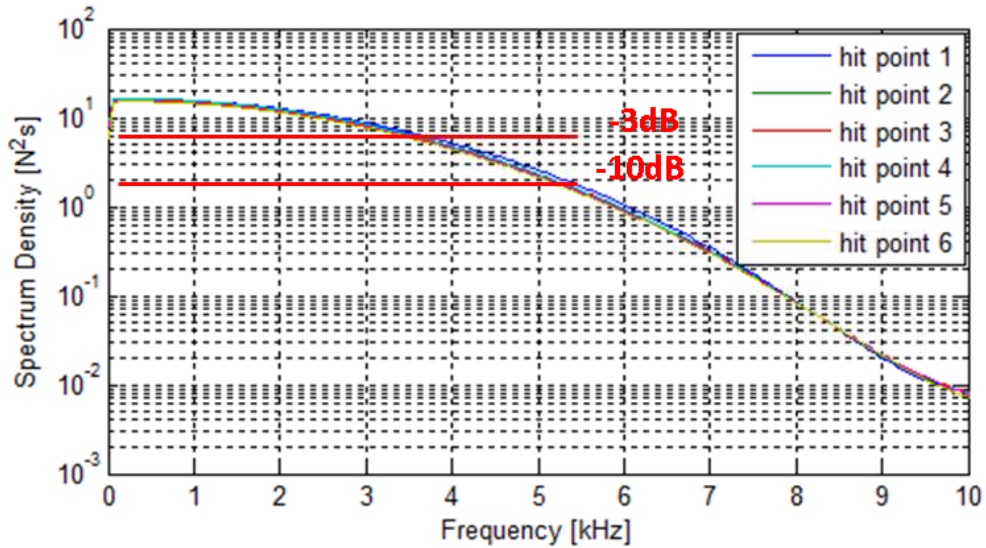
180 (The yellow numbers 1-8 denote the accelerometers placed at the IRJ and the red arrows 1-6  
181 indicate the positions and directions of the hammer excitations)

181

182

183 A hammer (PCB 086D05) with a hard metal tip (PCB 084B03) was used in the measurement  
184 to obtain high-frequency excitations. The hammer excitations were conducted as close as  
185 possible to the accelerometers 1-6, indicated by the red arrows 1-6 in Fig. 3. The excitation  
186 and response signals of 10 impacts were recorded at each hitting position. The spectra of the  
187 excitation loads at the six hitting positions, produced by averaging 10 impacts, are depicted in  
188 Fig. 4. The 3 dB drop (limit of very reliable range [23]) and 10 dB drop (limit of sufficiently  
189 reliable range [23]) occur at about 3.5 kHz and 5 kHz, respectively, indicating that the valid  
190 frequency range of the hammer-excited signals can reach up to 5 kHz.

191



192

193

Figure 4: Valid frequency range determined by the spectra of excitation forces

194

(within 3 dB drop: very reliable range; within 10 dB drop: reliable range [23])

195

196

The reliability of hammer test can be assessed by the coherence of the response signals

197

excited by hammer impacts. Since the dynamic behaviour of the target IRJ under wheel-rail

198

impacts is of particular interest in this study and wheel-rail impacts occur more or less at the

199

rail-end section after the joint, the coherence of the response signals under the vertical

200

hammer excitations at the position 1 (see Fig. 3) is presented here. The coherence curves of

201

the response signals of the eight measurement points (denoted as 1-8 in Fig. 3) excited by the

202

10 hammer impacts at the position 1 are depicted in Fig. 5 (a) and (b). Excellent coherence

203

over 0.9 in the whole frequency range of interest was obtained for the vertical responses

204

(positions 1, 3, 5, 7, 8), whereas for the lateral responses (positions 2, 4, 6) lower coherence

205

was observed at some frequency bands: around 360 Hz, 1160 Hz and over 3 kHz, probably

206

due to that the cross-acceleration is more sensitive to the positions and directions of

207

excitations and responses [22]. The good coherence of the lateral responses excited by the

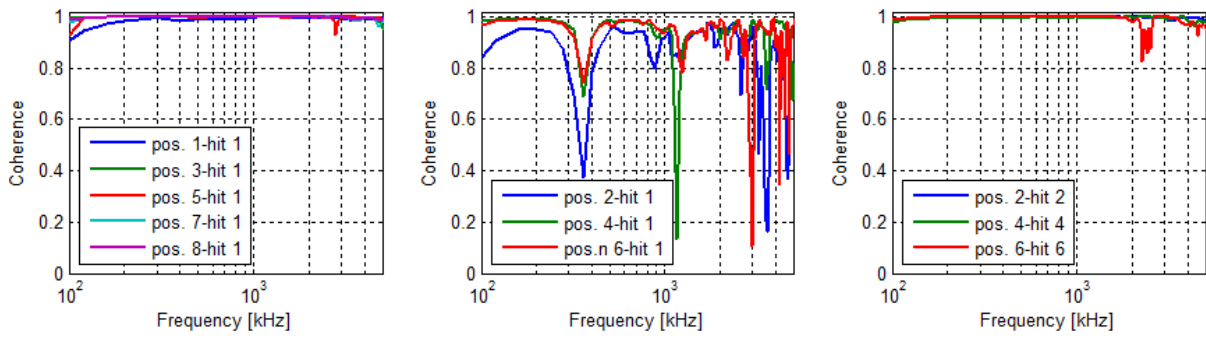
208

hammer in the lateral direction in Fig. 5 (c) indicates the testing signals in the lateral

209

directions are also reliable.

210



211

(a) vertical responses

(b) lateral responses

(c) lateral responses

212 by the vertical excitation at 1

by the vertical excitation at 1

by lateral excitations

213 Figure 5: Coherence curves of the responses at the eight measurement points

214

### 215 3.2 IRJ sub-model calibration

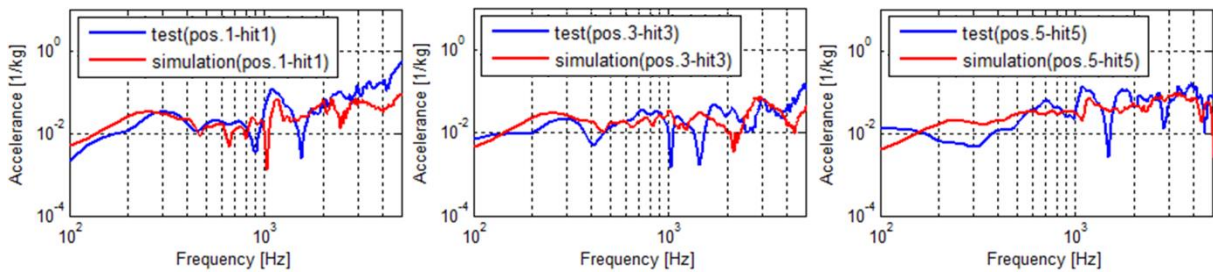
216 As indicated by the coherence analysis, the direct accelerances and transfer accelerances were  
 217 more reliably measured than the cross-accelerances. This study thus applied the measured  
 218 direct accelerances as the reference to calibrate the stiffness and damping parameters involved  
 219 in the FE IRJ sub-model. The accelerances of the IRJ sub-model were calculated by  
 220 reproducing the hammer test with an explicit time integration method, as illustrated in [27].  
 221 The simulated excitations were applied in both the vertical and lateral directions at the three  
 222 sections of the IRJ sub-model, respectively, as the measurement.

223

224 The closest fits of the simulated and measured vertical and lateral direct accelerances are  
 225 depicted in Fig. 6 and 7, respectively. The measurement results shown here are the averages  
 226 of the 10 times hammer-excited signals. Fig. 6 shows that the tendencies and resonant  
 227 frequencies of the simulated vertical accelerances match the measurement. The measured  
 228 typical track (anti-) resonances of the  $f_2$  (rail mass on the fastening stiffness [22]),  $f_a$  (sleepers  
 229 vibrate on the ballast and pad stiffness [22]), 1<sup>st</sup> and 2<sup>nd</sup> order vertical pinned-pinned of the  
 230

231 target IRJ are 280 Hz, 440 Hz, 1050 Hz and 2750 Hz, respectively, whereas those from the  
 232 simulation are 260 Hz, 460 Hz, 1100 Hz and 2550 Hz, respectively. The vertical acceleration  
 233 of the rail-end section is similar to that of the on-support section between 200 and 600 Hz, in  
 234 which frequency range a rail behaves like a mass and its dynamic behaviour is mainly  
 235 dominated by supports [18]. Over 4 kHz, the response of the rail-end section is evidently  
 236 higher than the other two sections, presenting a high-intensity dynamic feature. That is  
 237 probably related to the reduction of the vertical structural stiffness at the joint, which is  
 238 similar to the free end of a cantilever beam [4].

239



240

241 (a) Rail-end (position 1) (b) On-support (position 3) (c) Mid-span (position 5)

241

242 Figure 6: Fitting vertical accelerances (red line: simulation; blue line: test)

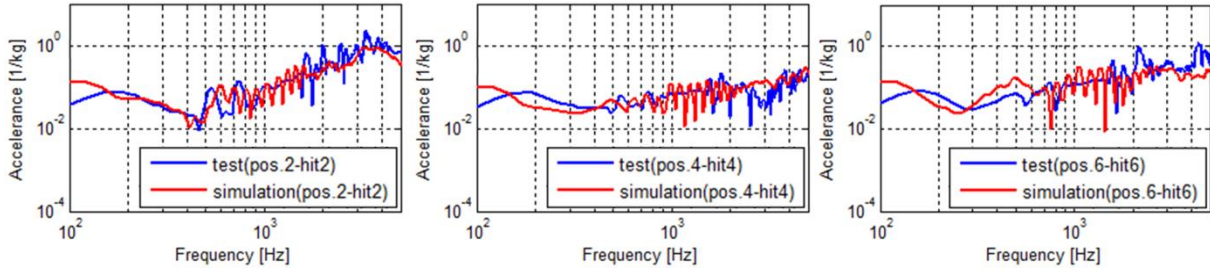
242

243

244 The general levels of the lateral direct accelerances shown in Fig. 7 are higher than the  
 245 vertical ones shown in Fig 6. The tendencies and levels of the measured and simulated lateral  
 246 direct accelerances are well fitted in Fig. 7. The lateral pinned-pinned resonance, less  
 247 pronounced than the vertical one, occurs at about 550 Hz in the measurement and 600 Hz in  
 248 the simulation. More oscillations can be observed in the simulation results between 600-1300  
 249 Hz, which could be influenced by the modelling of the fastening systems. Although an  
 250 improved fastening model represented by crossed linear spring and damper elements managed  
 251 to take into account the lateral dynamics, it failed to fully reproduce the modes of real  
 252 fastenings comprised of rail pads and clamps [26]. The deviation at the low frequency below

253 200 Hz is probably related to the simplification of the ballast model, whose displacements  
254 were constrained in the lateral and longitudinal directions.

255



256

257 (a) Rail-end (position 2) (b) On-support (position 4) (c) Mid-span (position 6)

258

Figure 7: Fitting lateral accelerances (red line: simulation; blue line: test)

259

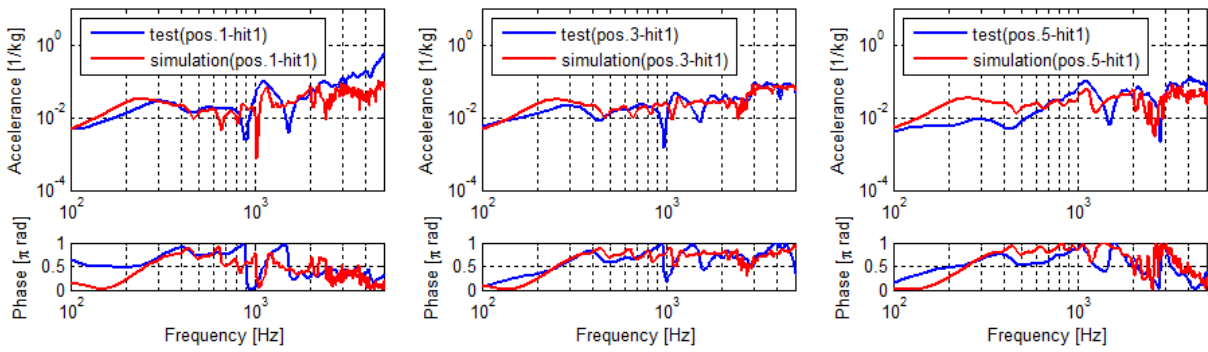
260 It can be concluded from the analysis of the direct accelerances that the dynamic behaviour of  
261 the rail-end section has a high-intensity feature due to the reduction of bending stiffness. The  
262 stiffness and damping parameters calibrated by the direct accelerances are then adopted in the  
263 simulations of the transfer accelerances and cross-accelerances, track decay rate and wheel-  
264 rail impact at the target IRJ.

265

### 266 3.3 Validation of transfer and cross-accelerances

267 The vibration responses of the IRJ under the vertical hammer excitation at the rail-end section  
268 (position 1 in Fig. 3) are studied for the IRJ sub-model validation. The transfer and cross-  
269 accelerances, including phases, simulated by the calibrated IRJ sub-model are compared with  
270 the measurement in Fig. 8. Good agreements for the transfer accelerances and reasonable  
271 agreements for the cross-accelerances are achieved. It is believed that the deviations of the  
272 cross-accelerances at 120 Hz and 500-1300 Hz are respectively caused by the simplification  
273 of the ballast model and the linear spring/damper representations of the rail pads models, as  
274 analysed in the lateral direct accelerances.

275

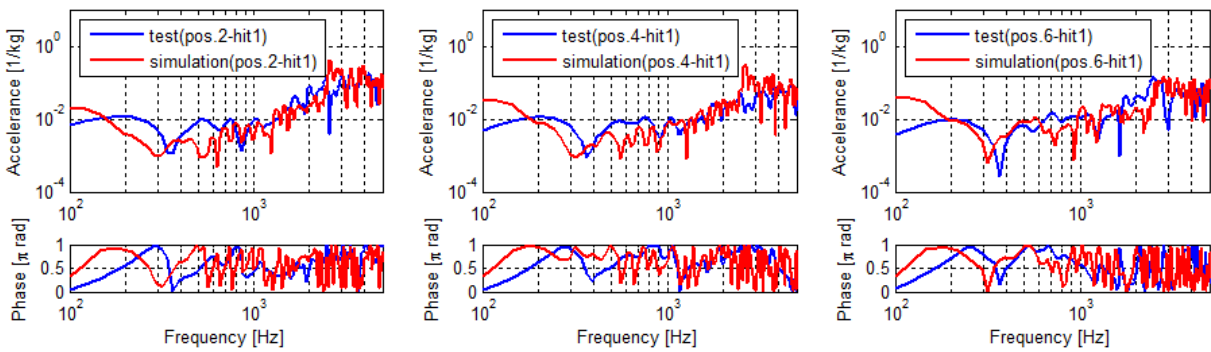


276

277

(a) Rail-end (position 1) (b) On-support (position 3) (c) Mid-span (position 5)

278

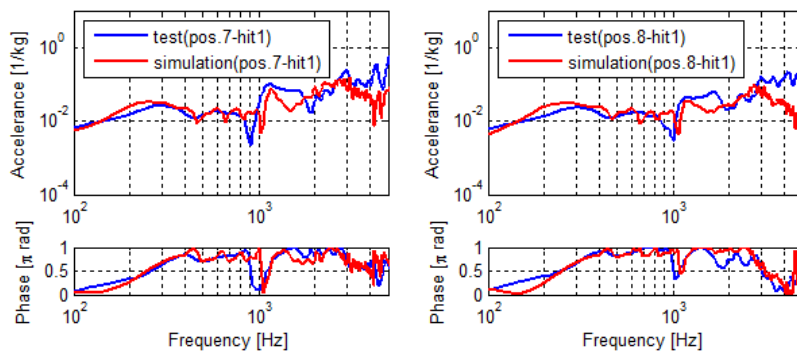


279

280

(d) Rail-end (position 2) (e) On-support (position 4) (f) Mid-span (position 6)

281



282

283

(g) Rail-end (position 7) (h) On-support (position 8)

284

Figure 8: Validations of transfer and cross-accelerances (red line: simulation; blue line: test.

285

transfer accelerances: (a)-(c) and (g)-(h); cross-accelerances: (d)-(f)

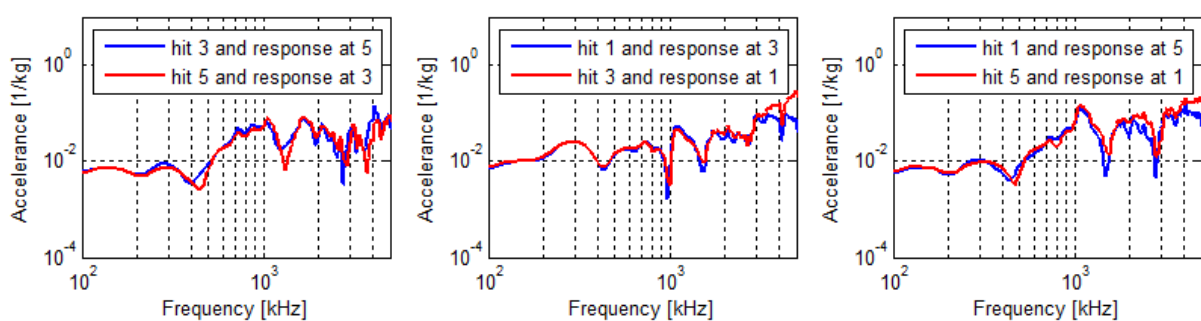
286

287

### 3.4 Validation of decay rate

288 The validation of track decay rate is also important for the accurate prediction of pass-by  
 289 vibration and noise [28]. Owing to the reciprocity, the decay rate of a track with periodical  
 290 structural characteristics is normally measured by a roving hammer method for operational  
 291 convenience [29]. The comparison of the transfer accelerances measured at the on-support  
 292 (position 3) and mid-span (position 5) sections shown in Fig. 9 (a) indicates the reciprocity  
 293 can be basically obeyed there, whereas Fig. 9 (b) and (c) show that the reciprocity can hardly  
 294 be satisfied at the rail-end section in the high-frequency range over 3 kHz. Therefore, this  
 295 study employed a roving sensor approach to measuring the decay rate of the target IRJ. The  
 296 hammer excitations were exerted at the rail-end section after the joint (position 1 in Fig. 3),  
 297 where wheel-IRJ impacts are expected to occur; Nine accelerometers (B&K 4514, denoted as  
 298 1-9 in Fig. 10) were employed in the measurement and a 5-meter section of the track (8  
 299 sleeper spans) just after the IRJ could be covered by roving the sensors once. Referring to the  
 300 hammer excitation positions suggested in the roving hammer test of decay rate [29], the  
 301 sensor distribution schemes adopted in this study are shown in Fig. 10 (a), and the  
 302 corresponding in-situ conditions are shown in Fig. 10 (b) and (c).

303



304

(a) Position 3 & 5

(b) position 1 & 3

(c) position 1 & 5

305

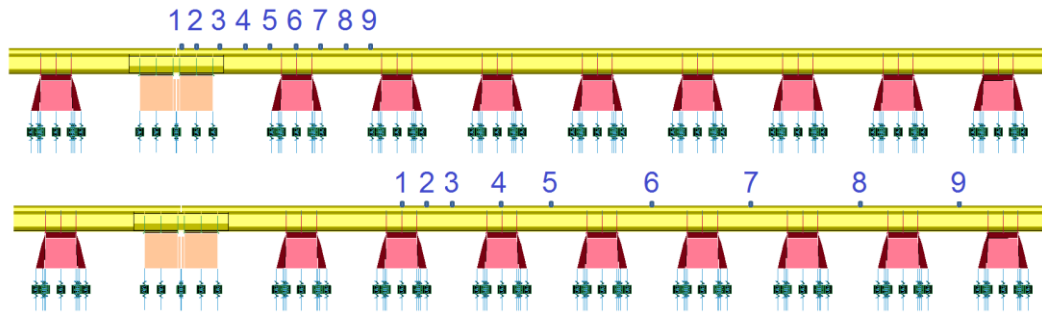
Figure 9: Comparisons of measured transfer accelerances

306

307

308



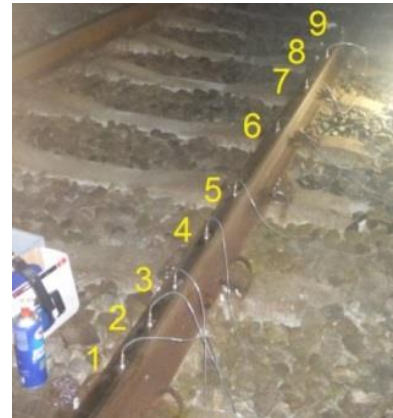


309

310 (a) Sketch of sensor distributions before (upper graph) and after (lower graph) roving



311



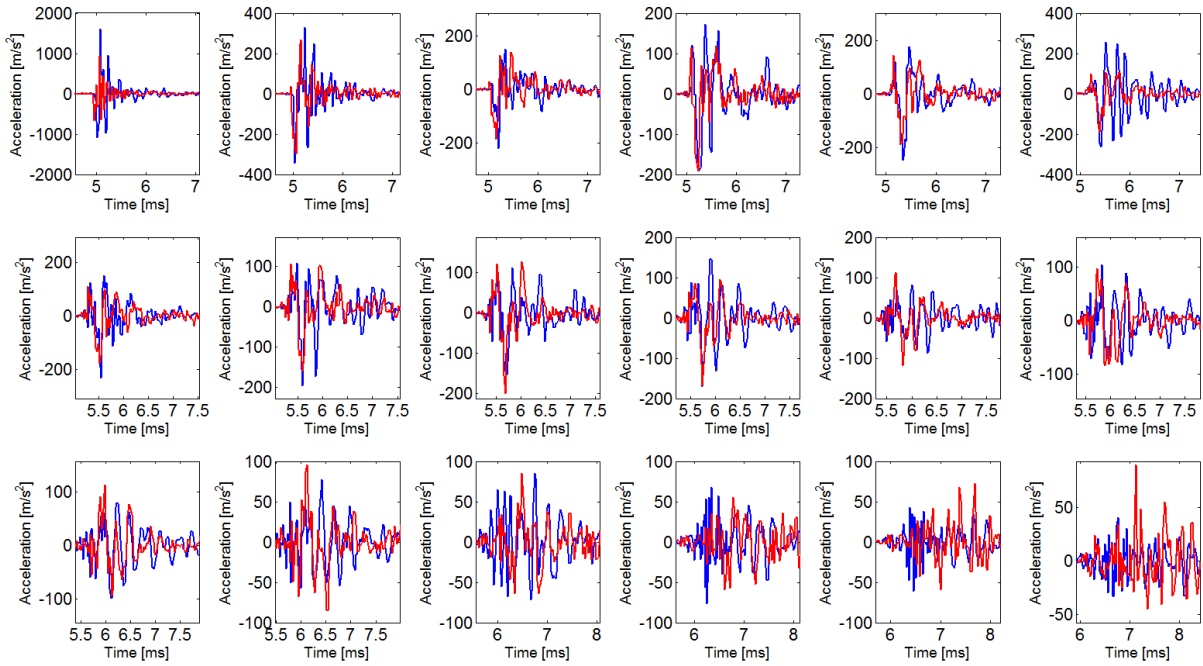
312 (b) Sensor distribution before roving (c) Sensor distribution after roving

313 Figure 10: Sensor distribution schemes in the decay rate test

314

315 The hammer test of the decay rate was also reproduced by the proposed FE IRJ sub-model  
 316 with an explicit time integration method. Fig. 11 compares the time histories of the simulated  
 317 vibration of the 18 response positions (9 sensors  $\times$  2 rounds) with those obtained by the  
 318 measurements. The measurement results shown here are also the averages of the 10 times  
 319 excitations, as the accelerances analysed above. Good agreements can be observed in the vast  
 320 majority of the comparisons. The simulated and measured vertical decay rates of the target  
 321 IRJ are calculated based on these vibration responses [30] and compared in Fig. 12. Good  
 322 agreements in terms of the peak, trough and average level in the comparison of the decay rate  
 323 again validate the dynamic behaviour of the presented FE IRJ sub-model. The decay rate  
 324 peaks at about 3 kHz and troughs at about 900 Hz and 4 kHz.

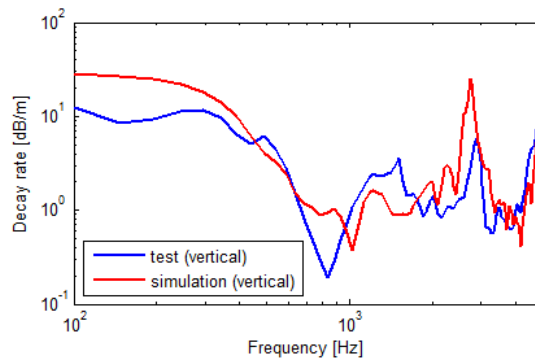
325



326

327 Figure 11: Comparison of acceleration responses (red line: simulation; blue line: test)

328



329

330 Figure 12: Validation of decay rate (red line: simulation; blue line: test)

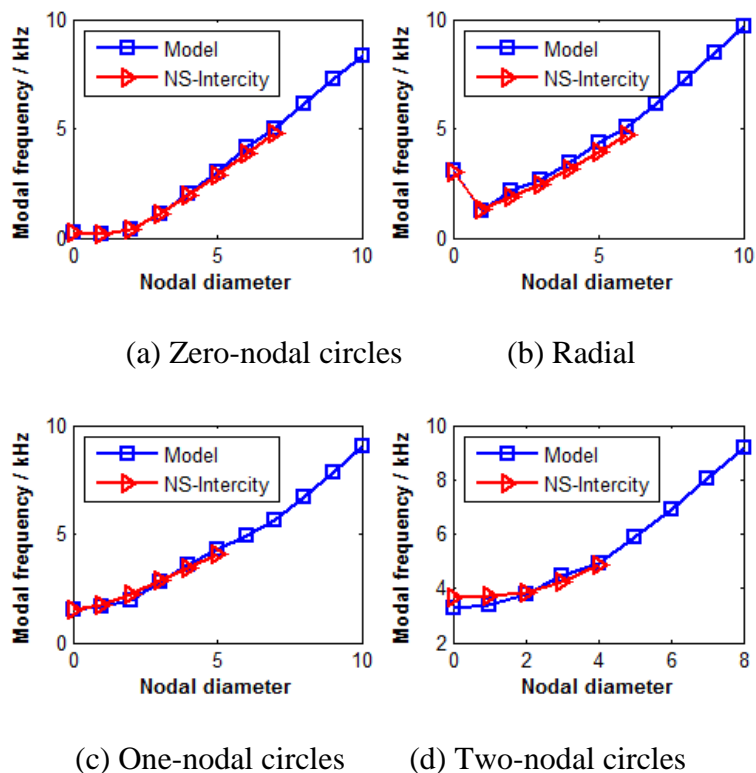
331

### 332 3.5 Validation of the wheel sub-model

333 Although the dynamic behaviour of wheels may also be characterised with FRFs [31], this  
334 study characterised the dynamic behaviour of the FE wheel sub-mode with the modes and the  
335 corresponding natural frequencies and validated them against a measurement reported in [18],  
336 because the material damping of a wheel is generally very low and the exact value of the

337 wheel modal damping is not critical for rolling noise prediction [18]. The frequency range of  
 338 interest is 10 kHz in this study. All modes of the wheel sub-model within the frequency range  
 339 of interest were identified by a modal analysis. For this, the same half-wheelset model as the  
 340 explicit FE model shown in Fig. 2 (a) was used with the inner edge of its hub clamped. The  
 341 calculated wheel modes under such a boundary condition can adequately represent the wheel  
 342 dynamics under contact with the rail [32]. The identified natural frequencies of the axial and  
 343 radial wheel modes are plotted in Fig. 13. Good agreement is observed when comparing the  
 344 calculated results with the experimental wheel natural frequencies from [18], in which an NS-  
 345 intercity wheel is measured for up to 5 kHz. All of these physical modes can be naturally  
 346 included in the transient dynamic simulation when using the full FE model and a small time  
 347 step [33].

348



349

350

351

352

353 Figure 13 Wheel modes calculated by a finite element modal analysis (blue) compared with

354 the measured results (red, from [18])

355

## 356 **4 Impact vibration and noise during pass-by**

357 The validations of the IRJ and wheel sub-models enable the FE dynamic interaction model for  
358 a transient impact simulation. The simulated impact vibration and noise are compared to the  
359 measurement results from a pass-by measurement in this section.

360

### 361 **4.1 Set-up of the pass-by measurement**

362 A pass-by measurement with a travelling speed of about 100 km/h was performed on the  
363 target IRJ on 19<sup>th</sup> February 2015. The vertical and lateral impact vibration, collected on the  
364 rail foot just after the joint (see Fig. 14 (a)), were measured by accelerometers PCB 352A60  
365 (measurement range:  $\pm 4950 \text{ m/s}^2$ ; frequency range ( $\pm 3 \text{ dB}$ ): 5 Hz-60 kHz) and B&K 4514  
366 (measurement range:  $\pm 980 \text{ m/s}^2$ ; frequency range ( $\pm 6 \text{ dB}$ ): 5 Hz-10 kHz), respectively. Two  
367 microphones B&K 4958 (measurement range: 28 -140 dB; frequency range ( $\pm 1 \text{ dB}$ ): 28 Hz-  
368 20 kHz) were placed 1.5 m after the joint in the train running direction, and respectively 2.5  
369 cm (Fig. 14 (b)) and 7.5 m (Fig. 14 (c)) away from the rail to record the near-field and far-  
370 field impact noise.

371



372

373

(a) Accelerometers used to measure impact vibration



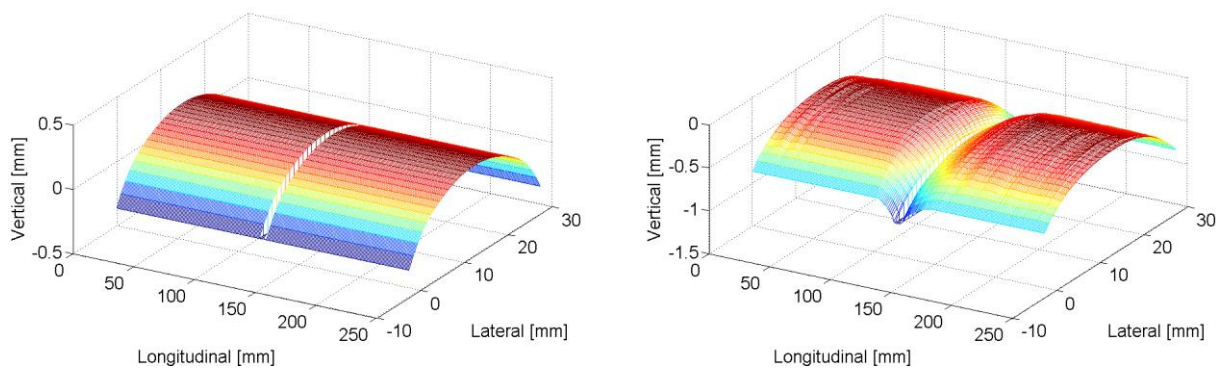
(b) Near-field microphone      (c) Far-field microphone

Figure 14: Set-up of sensors in the pass-by measurement

## 4.2 Impact vibration

The pass-by measurement conducted at the target IRJ was reproduced by the 3D FE wheel-IRJ dynamic interaction model shown in Fig. 2(a) with an explicit time integration. A tiny time step (49 ns) was employed to meet Courant stability condition [34]. This, together with the fine meshing scheme applied in the model, guaranteed that high-frequency dynamic effects up to 10 kHz or higher can be reproduced. The transient dynamic simulation first employed a dynamic relaxation to make the wheel-track system reach an equilibrium state under gravity. The initial position of the wheel model is 1.32 m away from the joint, as shown in Fig. 2. A forward translation velocity and a corresponding rotational velocity were defined as initial conditions applied to all the nodes of the wheel model. The forward translation velocity, which is also the rolling velocity of the wheel, initially equalled to the wheel circumferential velocity, i.e. the product of the rotational velocity and the wheel radius. A driving torque was subsequently applied to the wheel axle as a load boundary condition, due to which the wheel rotational velocity increased so that the wheel circumferential velocity exceeded the rolling velocity. Consequently, creepage and traction force were generated between the wheel and rail when the wheel rolled along the rail from the initial position towards the joint. As the measurement, the simulated pass-by speed was 100 km/h and the impact vibration responses were calculated at the rail foot just after the joint in the model. The measured geometry of the target IRJ (Fig. 1 (b)) by a HandySCAN 3D laser scanner was

397 applied to the FE model. Fig. 15 (a) and (b) show the rail top surface in the proximity to the  
 398 joint model before and after applying the measured geometry, respectively. The measured  
 399 geometric irregularities at 100-150 mm in the longitudinal direction, closer to the joint, is  
 400 significantly larger than those elsewhere in Fig. 15 (b). We applied the measured geometry  
 401 only to the vicinity of the joint rather than to the whole modelled rail surface, because the  
 402 impact vibration and noise studied in this paper occurs transiently just when the wheel rolls  
 403 over and hits the joint.



404  
 405 (a) Nominal geometry at an IRJ (b) Measured geometry at the target IRJ

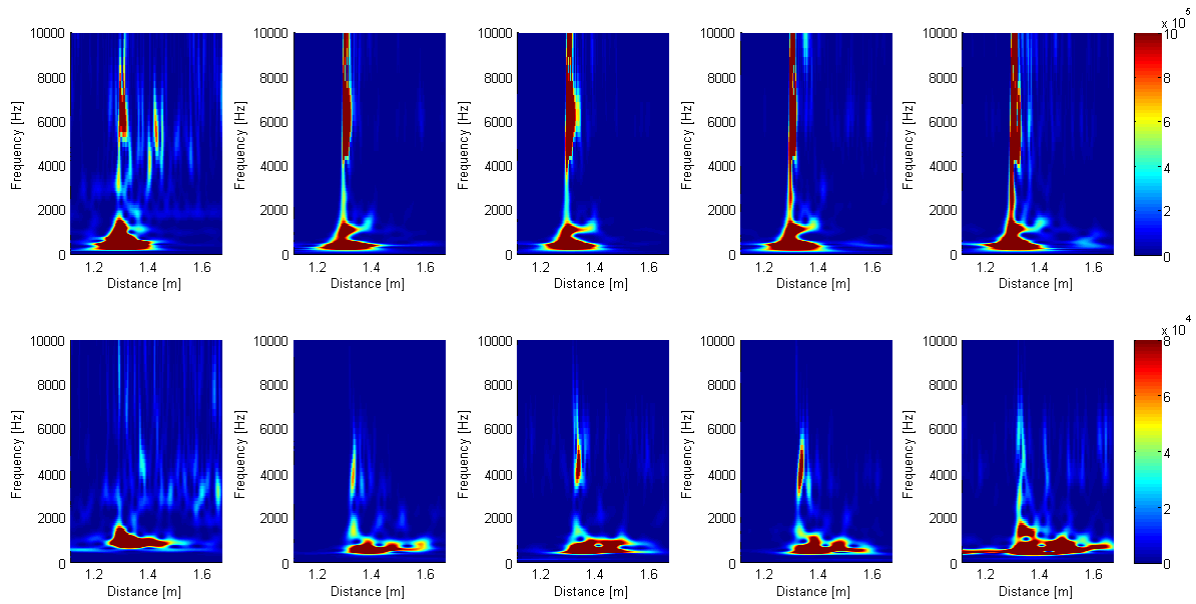
406 Figure 15: Applying the realistic geometry to the IRJ (size of irregularity exaggerated)

407 The wheel/rail dynamic responses were obtained by averaging the motions of the elements  
 408 attached to the wheel/rail volumes, while the contact forces were calculated by adding up the  
 409 nodal forces within the wheel-rail contact patch. The wheel/rail dynamic responses calculated  
 410 in each time step relied on the contact forces obtained in the previous time step, and in return  
 411 affected the contact forces updated in the next time step; therefore, the full coupled solutions  
 412 of wheel-rail dynamics and contact were obtained with a single simulation. The effects of  
 413 transient wheel rotation were also included inherently.

414

415 Wavelet power spectrum (WPS) is considered appropriate for the investigation of non-  
 416 stationary signals with local changes in the frequency components [17]. The WPSs of the  
 417 simulated and measured impact vibration up to 10 kHz are compared on the same scale in Fig.

418 16. The results of the four pass-bys shown in Fig. 16 were successively measured when the  
419 four wheelsets of a coach passed the target IRJ. It can be seen that the simulated impact  
420 vibration agrees well with the pass-by measurement results in both the time domain and  
421 frequency domain. The good agreement of the impact vibration is based on the results of the  
422 sub-models validation and also demonstrates the accuracy of them. Both the measurement and  
423 simulation indicate that energy of the vertical impact vibration (upper row in Fig. 16) mainly  
424 concentrate on around 300 Hz and 1 kHz, corresponding to the  $f_2$  and 1<sup>st</sup> order pinned-pinned  
425 resonance, respectively. Typical impact vibration feature is shown by the WPSs of the vertical  
426 rail vibration: a prominent high-frequency energy concentration with a broadband from 4 kHz  
427 to 10 kHz or higher occurs transiently just at the wheel-IRJ impact. The WPSs of the lateral  
428 impact vibration (lower row in Fig. 16) imply that the dominant frequencies of the simulated  
429 lateral impact vibration range from 600-1200 Hz, slightly higher than the measurement results  
430 of 550-1000 Hz. These dominant frequency ranges may be associated with the lateral pinned-  
431 pinned resonance, as the simulated and measured lateral pinned-pinned resonances occur at  
432 approximate 600 Hz and 550 Hz, respectively, in line with the lower boundaries of the  
433 dominant frequency ranges of the lateral impact vibration. High-frequency components in the  
434 lateral impact vibration are much less pronounced than the vertical ones and vary to some  
435 extent in frequency range. This variation can be attributed to the randomness of the traffic,  
436 such as differences in wheel (worn) profile, suspension condition, hunting motion etc.  
437



438

439 (a) Simulation (b) 1<sup>st</sup> pass-by (c) 2<sup>nd</sup> pass-by (d) 3<sup>rd</sup> pass-by (e) 4<sup>th</sup> pass-by

440

Figure 16: WPSs of the simulated and measured pass-by impact vibration

441

(upper row: vertical impact vibration; lower row: lateral impact vibration.

442

The unit of the colour bar is  $\text{m/s}^2$ )

443

#### 444 4.3 Impact noise

445

The near-field and far-field impact noise were calculated by an efficient frequency-domain

446

Rayleigh method [35] on the basis of the vibration velocities of the surface nodes of the wheel

447

and rail pre-calculated in the explicit time integration. The collection positions of the near-

448

field and far-field noise in the simulation corresponded to those in the measurement. A 100

449

Hz high-pass filter was applied to the noise signals to eliminate the influence of wind noise

450

introduced by train pass-bys. The one-third octaves of the simulated and measured impact

451

noise compared in Fig. 17 indicate that the predictions of the near-field and far-field impact

452

noise are in good and reasonable agreements with the measurements in the frequency range of

453

interest, respectively with deviations of less than 5 dB and 15 dB. The underestimation of

454

both the near-field and far-field impact noise at around 500-600 Hz can be related to the

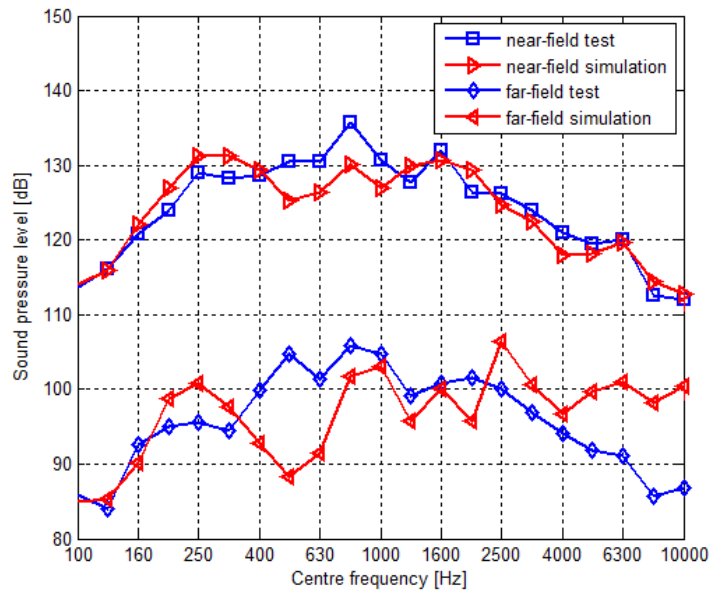
455

deviations between the simulated and measured lateral track dynamics and impact vibration.



456 The omission of noise radiated by sleepers in the simulation may to some extent have  
 457 contributed to the underestimation of the far-field noise at 400-500 Hz. The underestimation  
 458 of the near-field noise around 800 Hz and 4 kHz may correspond to the two troughs of the  
 459 decay rates at these frequencies in Fig. 12, which probably stem from the use of the track  
 460 model with a length of 10 m: the noise contribution by the rail beyond the 10-m model cannot  
 461 be included, resulting in the underestimation of the simulated noise especially when the decay  
 462 rates are low. The overestimation of the far-field noise in the high-frequency range over 4  
 463 kHz might be caused by the limitation of the Rayleigh method, which assumes a radiating  
 464 structure as a plane surface and is considered more accurate for near-field acoustic predictions  
 465 [35]. A 30 dB noise reduction is observed from the near-field collection point just beside the  
 466 rail web to the far-field one 7.5 meters away.

467



468

469 Figure 17: Comparisons of one-third octaves of impact noise

470

471 By comparing the impact vibration (Fig. 16) and noise (Fig. 17) to the dynamic behaviour  
 472 (Fig. 6-7) of the target IRJ, we may deduce that high-frequency contents play more important

473 roles in the impact vibration than impact noise. In addition, the  $f_2$  resonance seems to  
474 contribute to the dominant frequency of the vertical impact vibration at about 300 Hz and the  
475 peak of the impact noise at 250 Hz, whereas the 1<sup>st</sup> order pinned-pinned resonance may  
476 determine the dominant frequency of the impact vibration at about 1 kHz and the 800 Hz peak  
477 of the impact noise. Furthermore, although the energy of the vertical impact vibration at the  
478 dominant frequency of 1 kHz is less pronounced and less long-lasting than that at 300 Hz, the  
479 800 Hz peaks of the impact noise are higher than the 250 Hz ones (except the near-field  
480 simulation case). That is probably due to the contribution of the lateral impact vibration with a  
481 dominant frequency of 600-1000 Hz to the noise generation, as well as the low decay rate of  
482 the IRJ at about 900 Hz (troughs in Fig. 12).

483

## 484 **5 Conclusion and future work**

485 This paper presented a 3D transient explicit FE wheel-IRJ interaction model and validated the  
486 model by a comprehensive hammer test and pass-by measurement. The model is able to take  
487 into account complex wheel-rail dynamic interaction with non-linear material properties and  
488 arbitrary contact geometries; it should thus be able to calculate wheel-IRJ impact forces,  
489 vibration and noise directly and more accurately from the wheel-rail contact point of view,  
490 especially in the high-frequency range. The model also provides a possibility to analyse  
491 wheel/rail vibration in the vertical and lateral directions simultaneously.

492

493 By applying the in-situ measured geometries of the railhead to the IRJ model, the simulated  
494 impact vibration in both the time domain and the frequency domain agreed well with the pass-  
495 by measurement results, and the simulated impact noise was in reasonable agreement with the  
496 measurements, indicating that the explicit FEM is capable of reproducing impact vibration  
497 and noise excited by an IRJ up to 10 kHz.

498

499 The hammer test reveals that typical track resonances of the  $f_2$  and  $f_a$  as well as the 1<sup>st</sup> and 2<sup>nd</sup>  
500 order vertical pinned-pinned of the target IRJ are around 280 Hz, 450 Hz, 1 kHz and 2.8 kHz,  
501 respectively; the rail-end section in the vicinity of the joint presents a high-intensity dynamic  
502 feature and reciprocity cannot be satisfied there in the high-frequency range over 3kHz; the  
503 decay rate of the target IRJ peaks at about 3 kHz and troughs at about 900 Hz and 4 kHz.

504

505 The dynamic behaviour of the target IRJ was not only used to validate the model but also  
506 connected with the dominant frequencies of the wheel-IRJ impact vibration and noise: the  $f_2$   
507 resonance and the 1<sup>st</sup> order pinned-pinned resonance contribute significantly to the dominant  
508 frequencies of the impact vibration and noise; the vertical and lateral impact vibration with  
509 respective dominant frequencies of 1 kHz and 600-1000 Hz as well as the low decay rate of  
510 the target IRJ at about 900 Hz result in that the 1/3 octave spectrum of the impact noise peaks  
511 at around 800 Hz. These results indicate that the impact vibration and noise may be mitigated  
512 by controlling or re-designing the dynamic behaviour of IRJs: e.g. by damping the  $f_2$  and 1<sup>st</sup>  
513 pinned-pinned resonances or by a new design of the IRJ or support structures. The influence  
514 of dynamic behaviour of IRJs on impact vibration and noise deserves further investigations,  
515 and may contribute to the mitigation of impact vibration and noise generated by IRJs, as well  
516 as to train-borne detection of deterioration types of IRJs.

517

518 The main restriction of the presented model is the representation of the fastenings and ballast  
519 by the linear spring and viscous damper elements. When calibrating the stiffness and damping  
520 parameters, efforts were mainly taken to achieve the closest fit of the vertical accelerances  
521 because the vertical dynamics play more significant roles in the wheel-IRJ impact. The  
522 deviations of the lateral dynamics consequently affect the accuracies of the predictions of the

523 lateral impact vibration and noise to some extent. The oscillations of the simulated lateral  
524 accelerances between 600-1300 Hz deserves a careful investigation in the future. A full FE  
525 representation of rail-pads with proper material parameters proposed in [26] could be an  
526 option to be employed to improve the accuracy of the lateral dynamics simulation. In addition,  
527 the effects of the track model length need to be further examined and the noise radiated by  
528 sleepers should be included in the future to improve the accuracies of the acoustic predictions.  
529 More train pass-bys may be measured to validate the model in a statistical way.

530

## 531 **Acknowledgements**

532 This work was supported by the China Scholarship Council; the Dutch railway infrastructure  
533 manager ProRail; and the open research fund of MOE Key Laboratory of High-speed Railway  
534 Engineering, Southwest Jiaotong University.

535

## 536 **References**

- 537 [1] A.D. Kerr, J.E. Cox, Analysis and tests of bonded insulated rail joints subjected to vertical wheel  
538 loads, *Int J Mech Sci*, 41 (1999) 1253-1272.
- 539 [2] N.K. Mandal, M. Dhanasekar, Y.Q. Sun, Impact forces at dipped rail joints, *P I Mech Eng F-J Rai*,  
540 230 (2016) 271-282.
- 541 [3] T.X. Wu, D.J. Thompson, On the impact noise generation due to a wheel passing over rail joints,  
542 *Journal of Sound and Vibration*, 267 (2003) 485-496.
- 543 [4] T. Kitagawa, K. Murata, T. Kawaguchi, S. Tanaka, K. Nagakura, Experimental and Theoretical  
544 Studies on Impact Noise Generation due to Rail Joints, *Notes Numer Fluid Me*, 126 (2015) 55-62.
- 545 [5] S.L. Grassie, R.W. Gregory, D. Harrison, K.L. Johnson, The Dynamic-Response of Railway Track to  
546 High-Frequency Vertical Excitation, *J Mech Eng Sci*, 24 (1982) 77-90.
- 547 [6] Y.C. Chen, L.W. Chen, Effects of insulated rail joint on the wheel/rail contact stresses under the  
548 condition of partial slip, *Wear*, 260 (2006) 1267-1273.
- 549 [7] N.K. Mandal, M. Dhanasekar, Sub-modelling for the ratchetting failure of insulated rail joints,  
550 *International Journal of Mechanical Sciences*, 75 (2013) 110-122.
- 551 [8] J. Sandstrom, A. Ekberg, Numerical study of the mechanical deterioration of insulated rail joints, *P*  
552 *I Mech Eng F-J Rai*, 223 (2009) 265-273.
- 553 [9] A.K. Himebaugh, R.H. Plaut, D.A. Dillard, Finite element analysis of bonded insulated rail joints, *Int*  
554 *J Adhes Adhes*, 28 (2008) 142-150.
- 555 [10] E. Kabo, J.C.O. Nielsen, A. Ekberg, Prediction of dynamic train-track interaction and subsequent  
556 material deterioration in the presence of insulated rail joints, *Vehicle System Dynamics*, 44 (2006)  
557 718-729.

558 [11] Z. Yang, Z.L. Li, R. Dollevoet, Modelling of non-steady-state transition from single-point to two-  
559 point rolling contact, *Tribol Int*, 101 (2016) 152-163.

560 [12] Z.F. Wen, X.S. Jin, W.H. Zhang, Contact-impact stress analysis of rail joint region using the  
561 dynamic finite element method, *Wear*, 258 (2005) 1301-1309.

562 [13] M. Pletz, W. Daves, H. Ossberger, A wheel set/crossing model regarding impact, sliding and  
563 deformation-Explicit finite element approach, *Wear*, 294 (2012) 446-456.

564 [14] W. Cai, Z.F. Wen, X.S. Jin, W.M. Zhai, Dynamic stress analysis of rail joint with height difference  
565 defect using finite element method, *Eng Fail Anal*, 14 (2007) 1488-1499.

566 [15] Z.L. Li, X. Zhao, R. Dollevoet, M. Molodova, Differential wear and plastic deformation as causes  
567 of squat at track local stiffness change combined with other track short defects, *Vehicle Syst Dyn*, 46  
568 (2008) 237-246.

569 [16] Z. Yang, Z. Li, R.P.B.J. Dollevoet, An Explicit Integration Finite Element Method for Impact Noise  
570 Generation at a Squat, *Notes Numer Fluid Me*, 126 (2015) 63-70.

571 [17] M. Molodova, Z.L. Li, A. Nunez, R. Dollevoet, Validation of a finite element model for axle box  
572 acceleration at squats in the high frequency range, *Comput Struct*, 141 (2014) 84-93.

573 [18] D.J. Thompson, *Railway Noise and Vibration : Mechanisms, Modelling and Means of Control*,  
574 Elsevier, 2009.

575 [19] N. Zong, M. Dhanasekar, Experimental studies on the performance of rail joints with modified  
576 wheel/railhead contact, *P I Mech Eng F-J Rai*, 228 (2014) 857-877.

577 [20] D.J. Thompson, N. Vincent, *Track Dynamic Behaviour at High Frequencies. Part 1: Theoretical*  
578 *Models and Laboratory Measurements*, *Vehicle System Dynamics*, 24 (1995) 86-99.

579 [21] M. Oregui, Z. Li, R. Dollevoet, Identification of characteristic frequencies of damaged railway  
580 tracks using field hammer test measurements, *Mech Syst Signal Pr*, 54-55 (2015) 224-242.

581 [22] N. Vincent, D.J. Thompson, *Track Dynamic Behaviour at High Frequencies. Part 2: Experimental*  
582 *Results and Comparisons with Theory*, *Vehicle System Dynamics*, 24 (1995) 100-114.

583 [23] A.P. De Man, *Dynatrack: A survey of dynamic railway track properties and their quality*, in, TU  
584 Delft, 2002.

585 [24] M. Oregui, M. Molodova, A. Nunez, R. Dollevoet, Z. Li, Experimental Investigation Into the  
586 Condition of Insulated Rail Joints by Impact Excitation, *Exp Mech*, 55 (2015) 1597-1612.

587 [25] N. Maia, J. Silva, *Theoretical and Experimental Modal Analysis*, Research Studies Press LTD,  
588 England, 1998.

589 [26] M. Oregui, Z. Li, R. Dollevoet, An investigation into the modeling of railway fastening, *Int J Mech*  
590 *Sci*, 92 (2015) 1-11.

591 [27] M. Oregui, Z.L. Li, R. Dollevoet, An investigation into the vertical dynamics of tracks with  
592 monoblock sleepers with a 3D finite-element model, *P I Mech Eng F-J Rai*, 230 (2016) 891-908.

593 [28] J.N. Yang, D.J. Thompson, Time-domain prediction of impact noise from wheel flats based on  
594 measured profiles, *Journal of Sound and Vibration*, 333 (2014) 3981-3995.

595 [29] *Railway applications - Noise emission - Characterisation of the dynamic properties of track*  
596 *sections for pass by noise measurements*, in: EN 15461:2008+A1:2010(E), 2010.

597 [30] C.J.C. Jones, D.J. Thompson, R.J. Diehl, The use of decay rates to analyse the performance of  
598 railway track in rolling noise generation, *Journal of Sound and Vibration*, 293 (2006) 485-495.

599 [31] T.X. Wu, D.J. Thompson, A Hybrid Model for the Noise Generation Due to Railway Wheel Flats,  
600 *Journal of Sound and Vibration*, 251 (2002) 115-139.

601 [32] C. Glocker, E. Cataldi-Spinola, R.I. Leine, Curve squealing of trains: Measurement, modelling and  
602 simulation, *Journal of Sound and Vibration*, 324 (2009) 365-386.

603 [33] X. Zhao, Z.L. Li, The solution of frictional wheel-rail rolling contact with a 3D transient finite  
604 element model: Validation and error analysis, *Wear*, 271 (2011) 444-452.

605 [34] R. Courant, K. Friedrichs, H. Lewy, On the Partial Difference Equations of Mathematical Physics,  
606 *IBM Journal of Research and Development*, 11 (1967) 215-234.

607 [35] M. Amdi, M. Souli, J. Hargreaves, F. Erchiqui, Numerical investigation of a vibrationacoustic  
608 analysis with different formulations, *Computer modelling in Engineering and sciences*, 85 (2012) 329-  
609 345.

610 Highlights:

611

612 • High-frequency impact vibration and noise up to 10 kHz are successfully reproduced.

613 • The model is validated by a comprehensive hammer test and a pass-by measurement.

614 • The  $f_2$  and pinned-pinned resonances strongly influence impact vibration and noise.

615 • Impact vibration and track decay rate determine the main frequency of impact noise.

616 • This paper may contribute to the mitigation of impact vibration and noise at IRJs.

617