



Institute of Mechanics,
Materials and Civil Engineering

Railway turnout dynamics: on-track experimental and model-based analysis

Raül Acosta Suñé

Thesis submitted in partial fulfillment
of the requirements for the degree of
Docteur en sciences de l'ingénieur et technologie

Dissertation committee:

Prof. Paul Fisette (UCLouvain, advisor)
Dr. Nicolas Docquier (UCLouvain, advisor)
M. Sylvain Berten (Infrabel, Belgium)
Prof. Georges Kouroussis (Université de Mons, Belgium)
Dr. Christine Funfschilling (SNCF, France)
Prof. Hadrien Rattez (UCLouvain, chair)

January 2026
Louvain-la-Neuve, Belgium

Abstract

Switches and crossings (S&C) are essential systems in railway infrastructure and play a crucial role in distribution of rail traffic as well as in track flexibility and safety. However, due to their geometric complexity and exposure to dynamic loads, they are among the most failure-prone elements of railway infrastructure, especially in dense urban networks like Brussels.

This thesis analyzes the dynamic behavior of a right-hand turnout mounted on wooden sleepers, focusing on the current gap between static measurements (using a digital ruler) and dynamic measurements (by an on-board measurement system on the EM130 measuring train). This thesis, which was carried out as part of AIGUIDYN, a collaborative project between Infrabel and UCLouvain, combines experimental campaigns and multibody simulations to study the influence of train direction, velocity, mass distribution, lateral stiffness and wear on both rail deformation and measurement accuracy.

Two experimental campaigns were conducted: a dynamic campaign using on-board and on-track sensors, and a static campaign using a hydraulic jack to characterize lateral stiffness. In parallel, a multibody model was developed with the ROBOTRAN software to simulate train-turnout interactions, including wheel tread & flange/rail and wheel flange/check rail contact.

Results show that, for operational speeds up to 40 km/h, quasi-static factors that result from the complex guiding mechanism of the train in the turnout — such as train configuration during the passage, the way the vehicle applies lateral loads to the turnout and track curvature — dominate over dynamic effects due to speed and acceleration. This thesis also highlights the importance of sensor location depending on whether the goal is to capture maximal dynamic or geometrical measurements, the influence of

lateral stiffness, and the need to reconsider maintenance thresholds based on dynamic measurements.

These findings help to understand the differences and to contribute to bridging the gap between static and dynamic measurements. Furthermore, they provide practical guidance for improving turnout inspection strategies and enhancing railway safety in the future.

Keywords: switches and crossings, turnout experimental setup, track maintenance, multibody model, wheel/rail interaction, lateral stiffness, dynamic measurements, static measurements.

Résumé

Les appareils de voie (communément appelés aiguillages) sont des systèmes essentiels de l'infrastructure ferroviaire, jouant un rôle crucial dans la distribution du trafic ainsi que dans la flexibilité et la sécurité du réseau. Cependant, en raison de leur complexité géométrique et de leur exposition aux charges dynamiques, ils figurent parmi les éléments les plus sujets aux défaillances, en particulier dans les réseaux urbains denses comme Bruxelles.

Cette thèse analyse le comportement dynamique d'un appareil de type branchement monté sur traverses en bois, en se concentrant sur l'écart actuel entre les mesures statiques (à l'aide d'une règle digitale) et les mesures dynamiques (par un système embarqué sur le train de mesure EM130). Cette thèse, réalisée dans le cadre du projet AIGIUDYN, un projet collaboratif entre Infrabel et l'UCLouvain, combine des campagnes expérimentales et des simulations multicorps afin d'étudier l'influence du sens de circulation du train, de la vitesse, de la répartition des masses, de la raideur latérale et de l'usure sur la déformation du rail et sur la précision des mesures.

Deux campagnes expérimentales ont été menées : une campagne dynamique utilisant des capteurs embarqués et sur voie, et une campagne statique utilisant un vérin hydraulique pour caractériser la rigidité latérale. En parallèle, un modèle multicorps a été développé avec le logiciel ROBO-TRAN pour simuler les interactions train/appareil de voie, y compris les contacts roue-bande de roulement et roue/contre-rail.

Les résultats montrent que, pour des vitesses opérationnelles allant jusqu'à 40 km/h, les facteurs quasi-statiques résultant du mécanisme complexe de guidage du train dans l'aiguillage — tels que la configuration du train lors du passage, la manière dont le véhicule s'appuie sur l'appareil de voie, et la courbure de la voie — dominant sur les effets dynamiques

dus à la vitesse et à l'accélération. Cette thèse met également en évidence l'importance du positionnement des capteurs selon que l'objectif est de capter les effets dynamiques maximaux ou les mesures purement géométriques, l'influence de la rigidité latérale, et la nécessité de reconsidérer les seuils de la norme sur la base des mesures dynamiques.

Ces résultats permettent de mieux comprendre les différences et de contribuer à établir un pont entre les mesures statiques et dynamiques. Ils fournissent également des recommandations pratiques pour améliorer les stratégies d'inspection des aiguillages et renforcer la sécurité ferroviaire à l'avenir.

Mots-clés : appareil de voie, campagne expérimentale, maintenance de la voie, modèle multicorps, interaction roue/rail, rigidité latérale, mesures dynamiques, mesures statiques.

Remerciements

Avant de vous présenter mon travail, je tiens à remercier tous ceux qui ont participé à ma thèse, que ce soit directement par leur précieuse contribution, ou indirectement en apportant leur petit grain de sable.

Mes promoteurs

Je souhaite commencer par remercier mes deux promoteurs, Paul et Nico. Merci de m'avoir donné l'opportunité en 2017 de réaliser mon TFE avec vous, alors que j'étais un "jeune" étudiant Erasmus, et en 2019 de travailler sur le projet CRAMIC, avant de me permettre de commencer ma thèse en 2022.

Si je devais mettre en avant une qualité, ce serait votre grande humanité. Dès mon arrivée en Belgique, vous m'avez reçu avec tant de chaleur et de gentillesse : vous m'avez retrouvé à la gare, mis à disposition un lit, et aidé à aménager ma chambre... Ces gestes de générosité et de bienveillance ne s'oublient pas.

Paul, j'ai beaucoup appris de toi, non seulement sur le plan technique, mais aussi sur le plan humain. Ton énergie positive, qui donnait envie de rester un peu plus longtemps à discuter avec toi à la cafétéria, tes retours toujours constructifs, ta manière d'encourager, ta passion pour ton métier... tout cela est une motivation qui se transmet à ceux qui t'entourent.

Nico, merci pour le suivi très pédagogique de mon TFE, qui m'a permis de découvrir le monde ferroviaire. Ensuite, pendant ma thèse, ta disponibilité, ta gentillesse, le temps que tu prenais pour m'expliquer en détail jusqu'à ce que je comprenne, ainsi que ton excellente expertise technique, m'ont permis de me plonger pleinement dans ce domaine et de l'apprécier.

Mes superviseurs et Infrabel

Un grand merci à mes superviseurs d’Infrabel : Sylvain et Etienne. Merci d’avoir accepté de faire partie de ce projet, pour votre grande disponibilité et pour avoir mis à ma disposition tous les outils possibles pour m’aider : pour la visite à Bascoup, les visites à l’atelier, et pour votre soutien lors de la première campagne expérimentale, qui a été très exigeante sur le plan organisationnel. Merci également de m’avoir permis de réaliser une deuxième campagne, et pour tous vos précieux conseils lors des réunions.

Sylvain, merci aussi de m’avoir accueilli dans votre équipe — appelée, jusqu’à il y a quelques semaines, *Track Data Cell* — et d’avoir accepté de faire partie de mon jury de thèse.

Un remerciement particulier à Rachid, qui m’a accompagné pendant une partie très importante de la thèse : pour sa grande disponibilité, ses fichiers Excel impeccables, sa présence lors des précampagnes et son excellent travail pour la première campagne.

Merci aussi à Robin, Daniel, Sander, Andries, Jean-Sébastien, Karim, aux collègues de l’atelier et à tous les collègues d’Infrabel qui ont rendu possibles les campagnes expérimentales.

Merci également à toute l’équipe *Track Data Cell*, aux collègues du 6ème étage et à ceux du 4ème étage, pour la bonne ambiance et pour avoir toujours été prêts à m’aider chaque fois que j’avais des questions.

Membres du jury

Un grand merci aux membres du jury, le Prof. Georges Kouroussis et la Dr. Christine Funfschilling, en plus des autres membres du jury déjà mentionnés, pour la relecture attentive de ce manuscrit. Merci pour leurs retours constructifs et pour les discussions lors de la défense privée, qui ont permis d’améliorer la qualité de ce travail. Je remercie également le Prof. Hadrien Rattez d’avoir présidé ma défense privée ainsi que ma soutenance.

Innoviris

Je souhaite également remercier Innoviris pour son soutien financier et institutionnel dans le cadre du projet AIGUIDYN (Applied PhD), ainsi que pour la confiance accordée tout au long de ce programme. Votre contribution a été essentielle pour permettre la réalisation des recherches et des expérimentations nécessaires à ma thèse.

UCLouvain

En revenant à l'UCLouvain, la collaboration avec le département LEMSC a été essentielle pour mener à bien une partie clé de ma thèse : les deux campagnes expérimentales. Merci à Alex, Christophe, Pierre, Antoine, Justin, Vincent et à tout le département, pour votre professionnalisme, pour le succès des deux « opérations commando », ainsi que pour la bonne humeur tout au long de ces moments.

Sans la collaboration du CREDEM pour la conception des pièces de la 2ème campagne en un temps record, celle-ci n'aurait pas été possible. Merci à Benoît, Lucas, Vincent, Julien et à tous ceux qui ont collaboré.

Merci à Quentin pour son excellent TFE, dont le modèle FEM a permis d'apporter un complément très intéressant aux analyses de ma thèse.

Je tiens également à remercier Olivier, qui a fait partie de mon séjour au MEED depuis mon TFE. Merci pour ton aide précieuse, tes explications du code Lantsoght++, pour avoir réussi à m'aider à faire fonctionner mon code quand j'étais désespéré, ainsi que pour ta bonne humeur et ton humour (que j'arrivais à comprendre !). Merci également à Marine : que serais-je devenu sans toi et ton modèle ? Merci pour ton formidable travail réalisé lors de ton TFE, qui m'a permis de partir d'un super modèle très développé pour mes simulations, pour tes encouragements durant ma phase finale de thèse et pour notre travail et soutien mutuel pendant les Robotrankathons.

Merci à ceux qui ont fait partie de l'équipe Robotran : Simon, Sébastien, Alexandre, Louis, Émile, François, et à tous. Un merci particulier à Simon pour les bons moments lors des conférences et summer schools.

Un grand merci à Aurél et Guillaume pour nos petites pauses, pour avoir corrigé mes textes en français, pour nos éclats de rire et nos chouettes moments "canailles" partagés ensemble. Merci à tous mes collègues de bureau, qui, à part Aurélia et Olivier, ont contribué à me faire sentir encore plus à l'aise au travail : Quentin, Carlos, Giovanni, Henri (pendant une semaine, mais ça compte) et Luana.

Merci à Catherine Rasse et à Léa d'avoir consacré de leur temps pour m'aider avec mes analyses statistiques. Je suis également très reconnaissant à Nathalie Sergoigne, Catherine Bauwens, Astrid Leduc et Isabelle Hennau pour leur précieux soutien administratif. Un gracias a parte para Nathalie por nuestras charlas en español et pour notre petite compétition d'apprentissage des prénoms à notre arrivée au MEED. Et un remerciement affectueux à Gianmarco, Martin, Eduardo, Julien, ainsi qu'à tous les collègues et anciens collègues du MEED.

Amis et famille

Núria, gràcies per les trucades plenes de riure i que han "ajudat" molt a la meua productivitat, les pijamas parties quan baixava a Barcelona i per ser tan genial. Judi, gràcies per cuidar de mi durant els dies de visita, per les passejades, els moments a casa de la Cris i per la teva dolçor. Nil, gràcies per la teva visita i per la distracció sent el millor rival del fantasy. El Rai, per ser el primer en venir i per la divertida escapada a Colònia. Els empes i els amics de Barcelona i de tot arreu. Sou un pilar des de sempre.

Thanks to those who are here in Belgium or have been here and gave me your support: Laura (perquè mantenim l'esperit de l'Erasmus), Miguel (por apoyarme "estudiando" a mi lado mientras yo redactaba mi tesis), Gabriel (por tus consejos y tu gran apoyo), Melissa (for your advice on my presentation and for laughing about my train sinking into the ground), Luis (mi doctor de referencia), Ceci (mi querida Ceci), Fer (por estar allí cuando llegué a Bélgica y por seguir estando a pesar de tu vuelta a España), Pol, Juan, all my ex-colocs, Zinnekes (à tous ceux qui sont venus à ma présentation et à tous), Les Wolfs, the Brazilian group, and all of you. You are and have been a big support for me.

Isabelle, Pascal, et la famille Everard Bottu, merci de m'avoir accueilli de la meilleure façon possible, pour la bonne ambiance et pour votre aide (merci d'avoir, avec le bruit de la perceuse, contribué à faire de ce jour l'un des plus productifs de ma thèse).

Merci Brieuc, pour ton aide en relisant des parties du manuscrit, pour avoir pris soin de moi dans les moments les plus stressants, pour m'avoir fait rire, pour ta gentillesse et pour ensoleiller mon quotidien.

Gràcies a la família Suñé Banzo + Acosta Bellés, especialment al meu cosinet Joan, que s'ha hagut de menjar hores de bus per veure'm. Júlia, Carmen i Santi, per haver-vos convertit en la nostra família, per les visites, per totes les experiències i bons moments tots junts.

Gràcies mama, papa, germanets, iaia Maria, iaia Alicia i iaio Bautista per ser la meua inspiració. Els meus germans, Guillem i Victor, gràcies per adaptar-vos per poder passar estona amb mi cada cop que vinc a Barcelona. Maria, gràcies per venir a la presentació i per ser tan bonica. Gràcies papa, per estar allà quan ho necessito i per venir-me a buscar i portar-me a l'aeroport cada cop, i gràcies mama per preocupar-te de mi cada dia des de la distància, el teu missatge diari, per la nostra connexió i les nostres confidències.

Us estimo molt.

Contents

Contents	ix
List of Figures	xiii
Glossary	xxiv

I Introduction

1 Context & Objectives	3
1.1 Switches and crossings in the railway network	6
1.2 Research project and scientific motivation	7
1.3 Research questions	10
1.4 Structure of the thesis	11
2 State-of-the-art	13
2.1 S&C: instrumentation and experimentation	13
2.1.1 <i>On-board systems for dynamic S&C inspection</i>	13
2.1.2 <i>On-track measurements and in-situ testing of S&C</i>	15
2.1.3 <i>Experimental evaluation of lateral track stiffness</i>	18
2.2 S&Cs: computer benchmarks and simulation analysis	23
2.3 Thesis contributions	26

II Experimental methodology

3 Dynamic experimental campaign	31
3.1 Objectives and experimental setup	31

3.2	Instrumentation	34
3.2.1	<i>On-track sensors</i>	34
3.2.2	<i>On-board sensors</i>	38
3.3	Data acquisition procedure	41
4	Static experimental campaign on track stiffness	45
4.1	Objectives and experimental setup	45
4.2	Instrumentation of the track	46
4.3	Data acquisition procedure	51

III Multibody model

5	Train–turnout multibody model	55
5.1	Multibody Modeling Framework	55
5.1.1	<i>Multibody Approach</i>	55
5.1.2	<i>Formalism Used</i>	56
5.1.3	<i>Special Modeling Features</i>	58
5.1.4	<i>Algorithmic Implementation</i>	59
5.2	Wheel-rail contact geometry	60
5.2.1	<i>Introduction</i>	60
5.2.2	<i>Wheel tread/rail contact model</i>	61
5.2.3	<i>Wheel flange/rail contact</i>	70
5.3	Measuring train modeling	77
5.4	Turnout geometry representation	82
5.5	Simulation cases	86
5.6	Limitations of the Model	87

IV Results & discussion

6	Results in the Closure Panel	91
6.1	Results reading and interpretation	92
6.1.1	<i>Reading grid and visualisation</i>	92
6.1.2	<i>Facing move</i>	93
6.1.3	<i>Trailing move</i>	94
6.1.4	<i>Numerical model data</i>	96
6.2	Trajectory influence	97
6.2.1	<i>Experimental results</i>	97
6.2.2	<i>Model results</i>	110

6.2.3	<i>Numerical exploration</i>	117
6.2.4	<i>Dynamic vs static: discussion & analysis</i>	120
6.3	Velocity influence	123
6.3.1	<i>Experimental Results</i>	123
6.3.2	<i>Model results</i>	125
6.3.3	<i>Numerical exploration</i>	127
6.3.4	<i>Dynamic vs static: discussion & analysis</i>	129
6.4	Center of mass impact	131
6.4.1	<i>Experimental results</i>	131
6.4.2	<i>Model results and numerical exploration</i>	131
6.4.3	<i>Dynamic vs static: discussion & analysis</i>	133
6.5	Lateral rail stiffness impact	136
6.5.1	<i>Experimental results</i>	136
6.5.2	<i>Model results</i>	137
6.5.3	<i>Dynamic vs static: discussion & analysis</i>	140
7	Results in the Crossing Panel	143
7.1	Trajectory influence	144
7.1.1	<i>Experimental results</i>	144
7.1.2	<i>Model results</i>	157
7.1.3	<i>Numerical exploration</i>	162
7.1.4	<i>Dynamic vs static: discussion & analysis</i>	163
7.2	Velocity influence	169
7.2.1	<i>Experimental results</i>	169
7.2.2	<i>Model results and numerical exploration</i>	169
7.2.3	<i>Dynamic vs static: discussion & analysis</i>	171
7.3	Center of mass impact	173
7.3.1	<i>Experimental results</i>	173
7.3.2	<i>Model results and numerical exploration</i>	173
7.3.3	<i>Dynamic vs static: discussion & analysis</i>	173
7.4	Lateral rail and check rail stiffness impact	176
7.4.1	<i>Experimental results</i>	176
7.4.2	<i>Model results and numerical exploration</i>	177
7.4.3	<i>Dynamic vs static: discussion & analysis</i>	179
7.5	Impact of a worn check rail	182
7.5.1	<i>Model results and numerical exploration</i>	182
7.5.2	<i>Dynamic vs static: discussion & analysis</i>	183
8	Discussion and synthesis	187

V Conclusion and perspectives

9 General conclusions 203

10 Perspectives 207

10.1 Future experimental campaigns 207

10.2 Model development 208

10.3 Maintenance and norm 208

Appendices

A Lateral centripetal force on a single wheel 213

B Rail gauge analysis across curve radii 215

C Statistical analysis of experimental track measurements 221

D Publications and conference contributions 225

Bibliography 227

List of Figures

1.1	Works of maintenance in Brussels-Midi station over a public holiday weekend: replacement of 170 switch motors to enhance infrastructure reliability and safety <i>Source: Infrabel, Annual Report 2019 [2]</i>	4
1.2	Railway network in the Brussels-Capital Region, showing its turnouts, main stations and selected secondary stations.	5
1.3	Examples of different types of switches and crossings (S&Cs). The one studied in the present work is a right-hand turnout, shown at the top of the figure.	8
1.4	Examples of critical dimensions to be checked in the S&C by the digital ruler and the measuring train.	9
2.1	Measured lateral wheel–rail forces on a curved track by Zhai (2009) [26]. Positive values correspond to the outer rail (force towards the outside of the curve), and negative values to the inner rail (force towards the inside), resulting in a dynamic widening of the gauge.	17
2.2	Displacement versus lateral force curves applied to the rail base for two types of rail pads by Wang et al. (2024) [36]. Three different stages obtained are observed for the type A: shearing, sliding, and collision. For the type B, no sliding stage is observed.	20
2.3	Field-measured rail lateral displacement under increasing lateral wheel loads and a constant vertical load of 178 kN. The graph is built with the data from Chen et al. (2016) [37]. The results illustrate the nonlinear behavior of the rail structure.	22
3.1	Main components of a right-hand turnout.	32

3.2	Crossing gap and flangeway gap in the Crossing Panel.	32
3.3	Location of the sensors installed on the turnout: on-track sensors for both the diverging and through route configurations, and on-board sensors installed on the EM130 train for the experimental setup.	34
3.4	The digital ruler [79] is used for static measurements of a railway track.	35
3.5	LVDT were placed in different S&C zones to measure the rail head lateral displacement and rail web flexion.	36
3.6	Schematic representation of the vertical positioning of the LVDT sensors (in green) relative to the rail. The upper sensor is located 14 mm below the top surface of the rail head and the other is positioned halfway up the rail web.	37
3.7	LVDT installation and monitored dimensions in the Crossing and Closure Panels.	38
3.8	EM130 measuring train during the experimental campaign. . . .	39
3.9	GoPro camera setup and an example of record.	40
3.10	Layout illustrating the train orientation (rear to the left, front to the right), the two types of moves (facing and trailing) and the different routes (main, through and diverging).	41
4.1	3D view of the load application assembly. The numbered components correspond to those listed in Table 4.1.	46
4.2	Zoom view of the contact between the steel ball and the rail. .	47
4.3	Experimental setup showing the hydraulic jack coupled to the rail, with two U-shaped pieces securing the assembly longitudinally and the LVDT sensor positioned to measure rail displacement. .	48
4.4	Top view of the different test configurations, showing the layout of the sensors and of the assembly of the hydraulic jack in each panel.	49
4.5	Cross-sectional view of the test configurations, indicating the relative position of the LVDT sensors on the rail head, web, and foot.	50
4.6	Example of the measurements obtained during Test 3. The top graph shows the evolution of the lateral displacement of the rail measured by the LVDT sensors and the bottom graph displays the force applied by the hydraulic jack. The shape of the graphs reveals the force steps applied to obtain stable displacement values.	52

5.1	Left: tree-like multibody system. Right: constrained multibody system.	56
5.2	Left: virtual sensor. Center: 1D link. Right: 3D link.	58
5.3	Flowchart of the multibody formalism implementation.	59
5.4	Figure from Samin & Fiset (2003) [85], adapted to the present work.	61
5.5	Profile geometry parameters at contact point Q	62
5.6	Pitch angle θ , representing the circumferential displacement of the contact point Q along the wheel perimeter.	64
5.7	Normal contact force between the wheel tread and the rail, representing the first constraint of non-interpenetration between rigid bodies.	65
5.8	Cubic saturation model of Johnson & Vermeulen [94].	69
5.9	Contact between the outer surface of the wheel flange and the rail (on the left) and contact between the inner surface of the wheel flange and the rail (on the right).	70
5.10	Schematic representation of the flange contact model. The contact occurs on a virtual ring (in red) with radius r_b , located at the flange and inclined by the virtual conicity angle β_b . The circumferential position of the contact point Q_b is defined by the pitch angle θ_b	71
5.11	Model of rail compliance and normal force at the second contact point.	73
5.12	Reference frames $[\hat{T}]$ and $[\hat{T}_b]$ used to express the velocity at the first and second contact points.	75
5.13	Schematic of the contact model between wheel flange and check rail, showing the slope β and lateral positions $Y_{Q'_b}$ and Y_{CR}^0	76
5.14	Schematic of the multibody model from Boxus [96].	77
5.15	Schematic representation of the wheelsets (A, B, C, D) and the identification of left and right sides.	80
5.16	Three flangeway gap configurations studied in the analysis of the Crossing Panel. Top: adjusted check rail. Middle: nominal value. Bottom: worn check rail.	84
5.17	4 turnout model configurations studied in this thesis.	85
6.1	Wheelset (WS) order and letter assignment for facing and trailing moves. The train's Switch Measurement System (SMS) is installed at a distance of 870 mm from wheelset D.	93

6.2	Sensor measurement interpretation: facing move - collocated wheelset.	94
6.3	Sensor measurement interpretation: facing move - uncollocated wheelset.	95
6.4	Sensor measurement interpretation: trailing move - uncollocated wheelset.	95
6.5	Facing move: (Top) lateral displacement of the outer rail measured by the displacement sensor (DS) for every velocity regime. Each peak corresponds to the moment when a wheelset – A , B , C or D – aligns with the fixed position of the DS sensor on the track. (Middle) Positioning of the wheelsets when the leading one (A) aligns with the DS longitudinal position: $s_A = s_{DS} = 13.4\text{ m}$. (Bottom) Rail curvature, with the corresponding wheelset's position when $s_A = s_{DS}$. In orange, the measurement obtained by the EM130 train; in blue, the reference curvature from the plan, with the corresponding wheelset positions indicated on the graph.	98
6.6	Trailing move: (Top) lateral displacement of the outer rail measured by the displacement sensor DS for each velocity regime. Each peak corresponds to the moment when a wheelset – D , C , B , or A – aligns with the fixed position of the DS sensor on the track. (Middle) Positioning of the wheelsets when $s_D = s_{DS} = 13.4\text{ m}$. (Bottom) Rail curvature, with the corresponding wheelset's position when $s_D = s_{DS}$. In orange, the measurement obtained by the EM130 train; in blue, the reference curvature from the plan, with the corresponding wheelset positions indicated on the graph.	99
6.7	Sensor layout for the through route configuration. The sensor under analysis in this section is highlighted.	100
6.8	Facing move: displacement of the external rail measured by the LVDT sensor during train passage on the through route. Negative values indicate movement inwards toward the center of the track.	101
6.9	Trailing move: displacement of the external rail measured by the LVDT sensor during train passage on the through route. Negative values indicate movement inwards toward the center of the track.	102

6.10	Rail displacement peaks measured by the DS when the leading wheelset aligns with the displacement sensor position, comparing facing and trailing moves across different velocities. For 25 km/h and 40 km/h, the experiment was conducted twice, and the average value has been taken.	103
6.11	GoPro [®] camera positions relative to the wheelsets.	104
6.12	Camera recordings of the wheel as it passes in front of the DS sensor (wheelset D - left camera) at a velocity of 5 km/h: the left image shows the view during the facing move, where the wheel belongs to the trailing wheelset. In this image, there is no contact between the wheel flange and the rail. The right image shows the view during the trailing move, where the wheel now belongs to the leading wheelset. In this image, there is clear contact between the wheel flange and the rail.	105
6.13	Facing move: (Top) Lateral displacement of the outer rail measured by the displacement sensor (DS) for every velocity regime. Each peak corresponds to the moment when a wheelset – A , B , C , or D – aligns with the fixed position of the DS sensor on the track. (Middle) Positioning of the wheelsets when the SMS (in orange) aligns with the rail sensor, DS (in green): $s_{SMS} = s_{DS} = 13.4$ m. (Bottom) Cross-sectional view of the outer rail displacement when the SMS is at the same longitudinal position.	106
6.14	Trailing move: (Top) Lateral displacement of the outer rail measured by the displacement sensor (DS) for every velocity regime. Each peak corresponds to the moment when a wheelset – A , B , C , or D – aligns with the fixed position of the DS sensor on the track. (Middle) Positioning of the wheelsets when the SMS (in orange) aligns with the rail sensor, DS (in green): $s_{SMS} = s_{DS} = 13.4$ m. (Bottom) Cross-sectional view of the outer rail displacement when the SMS is at the same longitudinal position.	107
6.15	Gauge values measured by the switch measurement system at different train velocities, compared with the digital ruler and the quasi-static reference, when the SMS is located at the same longitudinal position as the displacement sensor DS in the closure panel. For 25 km/h and 40 km/h, 'x' marks two distinct measurements, and the circle on the same abscissa represents their average.	109

6.16	Facing move: numerical simulation results validating the direction of outer rail displacements at 25 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign	111
6.17	Trailing move: numerical simulation results validating the direction of outer rail displacements at 25 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign	112
6.18	Facing move. Gauge variation at the contact point with each wheelset (A to D) obtained by the model for a train velocity of 25 km/h. The green vertical dashed line indicates the equivalent sensor position during the experimental campaign.	114
6.19	Trailing move. Gauge variation at the contact point with each wheelset (A to D) obtained by the model for a train velocity of 25 km/h. The green vertical dashed line indicates the equivalent sensor position during the experimental campaign.	115
6.20	Comparison of the lateral displacement of the outer rail at the longitudinal position equivalent to the experimental sensor location, during wheel flange/rail contact for the 1st and 3rd wheelsets, for both move directions: facing and trailing.	116
6.21	S&C geometry with a curved section of 500 m radius, considered as an alternative configuration in the numerical model.	117
6.22	Facing move: comparison of the lateral displacement of the outer rail in the Closure Panel during wheel flange/rail contact, for two curve configurations: 215 m and 500 m radius.	118
6.23	Trailing move: comparison of the lateral displacement of the outer rail in the Closure Panel during wheel flange/rail contact, for two curve configurations: 215 m and 500 m radius.	119
6.24	Facing move: schematic representation justifying the highest difference between the on-track sensors and the SMS, based on on-track displacement measurements, model-computed outer and inner rail displacements, and the relative position of the train's SMS.	121
6.25	Experimental lateral peak measurements of the outer rail of the curve as a function of train velocity. The peaks are ordered according to the sequence of measurements, not by increasing speed.	124

6.26 Real train velocity for the first test at a theoretical velocity of 40 km/h. The locations of the main sensors analysed in this thesis are indicated by vertical lines. 125

6.27 Real train velocity for the second test at a theoretical velocity of 40 km/h. The locations of the main sensors analysed in this thesis are indicated by vertical lines. 126

6.28 Comparison of the lateral displacement of the outer rail at the longitudinal position equivalent to the experimental sensor location, during wheel flange/rail contact for the 1st and 3rd wheelsets, at speeds above 40 km/h, for both move directions: facing and trailing. 128

6.29 Orientation of the EM130 train during the experimental tests. Wheelsets A and B are located on the heavier half of the train (55% of the total mass, containing the motor bogie), whereas wheelsets C and D are on the lighter half (45%, containing the SMS bogie). 131

6.30 Reversed orientation of the EM130 train simulated by the numerical model. 132

6.31 Facing move: comparison of the lateral displacement of the outer rail at the longitudinal position equivalent to the experimental sensor location, for the 1st and 3rd wheelsets, at speeds equal to or below 40 km/h, for both orientations of the train. . . . 133

6.32 Trailing move: comparison of the lateral displacement of the outer rail at the longitudinal position equivalent to the experimental sensor location, for the 1st and 3rd wheelsets, at speeds equal to or below 40 km/h, for both orientations of the train. . . . 134

6.33 Location of the sensors and the hydraulic jack on the Closure Panel, as well as before the Switch Panel. C and D indicate the head and web of the curved closure rail (outer rail), respectively. A and B indicate the head and web of the straight closure rail or the inner rail. 137

6.34 Curves of applied force for the Closure Panel rails versus lateral rail displacement measured by the LVDT sensors. Blue and cyan correspond to the head and web of the curved closure rail (outer rail), respectively; red and orange to the head and web of the straight closure rail. 138

6.35 Curves of applied force for the rails before the Switch Panel rails versus lateral rail displacement measured by the LVDT sensors. Blue and cyan correspond to the head and web of the curved closure rail (outer rail), respectively; red and orange to the head and web of the inner rail. 139

6.36 Influence of vertical load on lateral displacement of the track. Result extracted from Quentin Quoidbach's thesis (2025). It illustrates the interaction between the vertical and lateral components of the load, used in the AIGUIDYN project. 140

6.37 Gauge increase for both facing and trailing moves, corresponding to the passage of the 1st and 3rd wheelsets (WS) when they reach the longitudinal position equivalent to the sensor location in the experimental setup. For higher rail lateral stiffness values, gauge increases are lower, following a decreasing exponential curve. A and B correspond to the motor bogie (heavier half), while C and D correspond to the SMS bogie (lighter half). . . . 141

7.1 Facing move at 30 km/h: (1st) Evolution of the flangeway gap and lateral displacements of the check rail (DS, green) and the nearest running rail (DS, red). (2nd) Wheelsets position when the leading wheelset is in the crossing gap. (3rd) Rail curvature with the corresponding wheelsets position marked on the graph. (4th) Cross-sectional view of the rail displacements. 146

7.2 Trailing move at 30 km/h: (1st) Evolution of the flangeway gap and lateral displacements of the check rail (DS, green) and the nearest running rail (DS, red). (2nd) Wheelsets position when the leading wheelset is in the crossing gap. (3rd) Rail curvature with the corresponding wheelsets position marked on the graph. (4th) Cross-sectional view of the rail displacements. 147

7.3 Check rail displacement measured by the DS when the leading wheelsets of each bogie — the 1st and 3rd WS — are aligned with the sensors in the frog area, comparing facing and trailing moves at different speeds. 148

7.4 Flangeway gap variation peaks measured by the DS when the leading wheelsets of each bogie — the 1st and 3rd WS — are in the crossing gap of the frog, comparing facing and trailing moves at different speeds. 149

7.5 Camera recordings of wheelset D at a velocity of 30 km/h, as the train passes over the crossing nose. The top images correspond to the facing move, where the wheels are those of a trailing wheelset D. The bottom images correspond to the trailing move, where the wheels are those of a leading wheelset D. In the trailing move, the wheels are slightly more displaced to the left, toward the check rail (right wheel) and the crossing nose (left wheel). 151

7.6 Sensor layout for the through route configuration. The sensor under analysis in this section is indicated in the figure. 152

7.7 Facing move: displacement of the check rail measured by the LVDT sensor during train passage on the through route. Negative values correspond to a movement towards the flangeway, indicating a tendency to reduce the flangeway gap. 152

7.8 Trailing move: displacement of the check rail measured by the LVDT sensor during train passage on the through route. Negative values correspond to a movement towards the flangeway, indicating a tendency to reduce the flangeway gap. 153

7.9 Facing move at 30 km/h: (Top) Evolution of the flangeway gap and lateral displacements of the check rail (DS, green) and the nearest running rail (DS, red). (Middle) Wheelset positions when the SMS aligns with the DS location. (Bottom) Cross-sectional view of the running rail and check rail displacements when the SMS is at the same longitudinal position. 154

7.10 Trailing move at 30 km/h: (Top) Evolution of the flangeway gap and lateral displacements of the check rail (DS, green) and the nearest running rail (DS, red). (Middle) Wheelset positions when the SMS aligns with the DS location. (Bottom) Cross-sectional view of the running rail and check rail displacements when the SMS is at the same longitudinal position. 155

7.11 Flangeway gap values measured by the train's Switch Measurement System at different train velocities, compared with the digital ruler, when the SMS is located at the same longitudinal position as the DS in the Crossing Panel. For 25 km/h and 40 km/h, 'x' marks individual measurements, and the point on the same abscissa represents their average. 156

7.12 Turnout geometry with two curved sections of 215 m and 209 m radius, defined as the base configuration in the numerical model. 157

7.13	Facing move: numerical simulation results validating the contact between the check rail and the 1st and 3rd wheelsets (left) and representing the flangeway gap variation (right) obtained at a train velocity of 30 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign. The purple zone highlights the check rail area. . . .	158
7.14	Trailing move: numerical simulation results obtaining a contact between the check rail and the 1st and 3rd wheelsets (left) and representing the flangeway gap variation (right) at a train velocity of 30 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign. The purple zone highlights the check rail area. As it is the trailing move, note that the graphs must be read from right to left. . .	159
7.15	Comparison of the lateral displacement of the check at the longitudinal position equivalent to the experimental sensor location, during wheel flange/check rail contact for the 1st and 3rd wheelsets, for both move directions: facing and trailing.	160
7.16	Comparison of the flangeway gap variation at the longitudinal position equivalent to the experimental sensor location, during wheel flange/check rail contact for the 1st and 3rd wheelsets, for both move directions: facing and trailing.	161
7.17	Turnout curvature measured by the EM130 measuring train, curvature from the plan used for the model and curvature with clothoid used for the alternative model. The area in green marks the zone where a radius of 209 m ($4,78 \text{ km}^{-1}$ of curvature) is expected in the Crossing Panel.	162
7.18	Turnout geometry with a curved section of 215 m radius, followed by a clothoid of 3,90 m length, was chosen as an alternative configuration in the numerical model.	163
7.19	Trailing move: numerical simulation results obtaining a contact between the check rail and the 1st and 3rd wheelsets for the theoretical base model (left) and the model with clothoid (right) at a train velocity of 30 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign. The purple zone highlights the check rail area. As it is the trailing move, note that the graphs must be read from right to left.	164

7.20 Model of turnout with clothoid: comparison of the lateral displacement of the check rail at the longitudinal position equivalent to the experimental sensor location, during wheel flange/check rail contact for the 1st and 3rd wheelsets, for both move directions: facing and trailing. 165

7.21 Schematic representation of the rail and check rail displacements during some phases of train passage for the facing move. . . . 166

7.22 Schematic representation of the rail and check rail displacements during some phases of train passage for the trailing move. . . . 167

7.23 Model comparison of the lateral displacement of the check rail — averaged around the longitudinal position corresponding to the experimental sensor location — during wheel flange/check rail contact for the 1st and 3rd WS, for both facing and trailing moves. 170

7.24 Model comparison of the flangeway gap variation at the longitudinal position corresponding to the experimental sensor location, during wheel flange/check rail contact for the 1st and 3rd wheelsets, for velocities up to 70 km/h and for both move directions: facing and trailing. 171

7.25 Facing move: comparison of the lateral displacement of the check rail at the longitudinal position equivalent to the experimental sensor location, for the 1st and 3rd wheelsets, at speeds between 5 km/h and 40 km/h, for both orientations of the train. . . . 174

7.26 Trailing move: comparison of the lateral displacement of the check rail at the longitudinal position equivalent to the experimental sensor location, for the 1st and 3rd wheelsets, at speeds between 5 km/h and 40 km/h, for both orientations of the train. 175

7.27 Location of the sensors and the hydraulic jack on the Crossing Panel, next to the frog and the check rail. C and D indicate the head and web of the outer rail, respectively. A and B indicate the head and web of the inner rail, respectively. 177

7.28 Curves of applied force for the Crossing Panel rails versus lateral rail displacement measured by the LVDT sensors. Blue and cyan correspond to the head and web of the curved closure rail (outer rail), respectively; red and orange to the head and web of the straight closure rail (inner rail). 178

7.29 Dissipated energy as a function of check rail stiffness for facing and trailing moves, for a train velocity of 30 km/h. 180

7.30	Flangeway gap variation as a function of check rail stiffness for facing and trailing moves, for a train velocity of 30 km/h. . . .	181
7.31	Facing move: total dissipated energy for one train passage as a function of velocity, for a tight check rail (38 mm flangeway gap), nominal check rail (40 mm), and worn check rail (42 mm). The energy corresponds to the sum of all energy dissipated by the wheelsets that come into contact with the check rail during the passage.	183
7.32	Trailing move: total dissipated energy for one train passage as a function of velocity, for three check rail conditions.	184
7.33	Facing vs trailing move: total dissipated energy for one train passage as a function of velocity, for a nominal check rail (40 mm). The energy corresponds to the sum of all energy dissipated by the wheelsets that come into contact with the check rail during the passage.	185
8.1	Wheelset (WS) order and letter assignment for facing (in blue) and trailing (in red) moves. The Switch Measurement System (SMS) is located at a distance of 870 mm from wheelset D. . .	188
8.2	Hypothetical SMS positions evaluated along the train. (Top) SMS installed close to each wheelset. (Middle) SMS at its actual position. (Bottom) SMS installed at the midpoints between C and D, between A and B, and at the center of the train. . . .	190
8.3	Facing move: rail displacement estimated from the lateral forces computed in the multibody model (MBS). The leading wheelset A first reaches the Closure Panel (top figure), followed by the Crossing Panel (bottom figure), at a velocity of 30 km/h. . . .	199
8.4	Trailing move: rail displacement estimated from the lateral forces computed in the multibody model (MBS). The leading wheelset D first reaches the Crossing Panel (top figure), followed by the Closure Panel (bottom figure), at a velocity of 30 km/h. . . .	199

Glossary

COM	Center of Mass
DS	Displacement Sensor, which includes LVDT and laser sensors
EM130	Track geometry measuring train
IMU	Inertial Measurement Unit
LEMSC	Mechanical Testing, Structures, and Civil Engineering Laboratory
LVDT	Linear Variable Differential Transformer
MBS	Multibody System
MEED	Mechatronic, Electrical Energy, and Dynamic systems
S&C	Switch & Crossing
SMS	Train's Switch Measurement System
WS	Wheelset

PART I

Introduction

1

Context & Objectives

WHEN traveling by train, as passengers, one of our main priorities is to travel on a train that is comfortable, safe, and as fast as possible. Ensuring these objectives is a major technical challenge, and even more so at high speeds, where track inspection and maintenance operations play a key role. The difficulty of these operations increases significantly in complex elements of the railway infrastructure that allow trains to change from one track to another: switches and crossings, also known as turnouts.

These mechanical systems — little known and sometimes visually invisible to passenger, yet perceptible through irregularities in the ride — are crucial for the safety and flexibility of rail operations. Their good functioning makes it possible for trains to be efficiently distributed throughout the network, particularly in densely populated city centers like Brussels. However, because of their complexity and their exposure to dynamic loading — vertical, but also lateral —, switches and crossings are probably the most failure-prone components of the track. A failure can impact several related routes, causing cascading delays, expensive maintenance interventions, and even possible safety risks.

A concrete example of this infrastructure complexity can be observed at Brussels-Midi, the busiest station in Brussels and in the whole of Belgium [1]. The impressive number of connected and intertwined tracks at this station is a good illustration of the operational difficulties related to switches and

crossings. Maintenance operations are of great importance there, such as the large-scale maintenance operation carried out by Infrabel in October 2019. During two public holiday weekends, Infrabel replaced 170 switch motors. Due to the complexity and scope of the project, some tracks were cut, and some trains were canceled, diverted or rerouted. Such large-scale interventions are necessary to guarantee long-term safety and performance, particularly in high-density areas where tiny failures of the infrastructure can have significant repercussions.



Fig. 1.1 Works of maintenance in Brussels-Midi station over a public holiday weekend: replacement of 170 switch motors to enhance infrastructure reliability and safety *Source: Infrabel, Annual Report 2019 [2].*

These infrastructural challenges are not only technical but also strategic, as they have a direct impact on urban mobility policies. In the Brussels-Capital Region, the Good Move plan [3] — the regional mobility plan for 2020–2030 — encourages the valorisation and the use of trains. Railways’

role is considered a key to complement urban public transport for medium-distance trips. The mobility plan aims to multiply by five the proportion of trips made by train within Brussels by 2030. To achieve this, it suggests increasing the frequency of trains, extending operating hours and reinforcing east-west connections, in particular between the European Quarter, the North District, Jette and Berchem-Sainte-Agathe (Fig. 1.2).

Moreover, the federal project approved in 2022 called Belgium's Rail Vision 2040 [4], also highlights the strategic importance of railways in Brussels. Since the city is the European capital and home to many international institutions, the plan seeks to increase direct connections with other major European cities, supporting both medium- and long-distance lines.

The objectives of the regional and federal plans discussed above will inevitably place greater demands on the railway infrastructure, especially on switches and crossings, which must be able to handle increased traffic with high reliability. This increased demand could lead to a greater need for more effective and less invasive maintenance strategies, in order to minimize disruption to traffic.

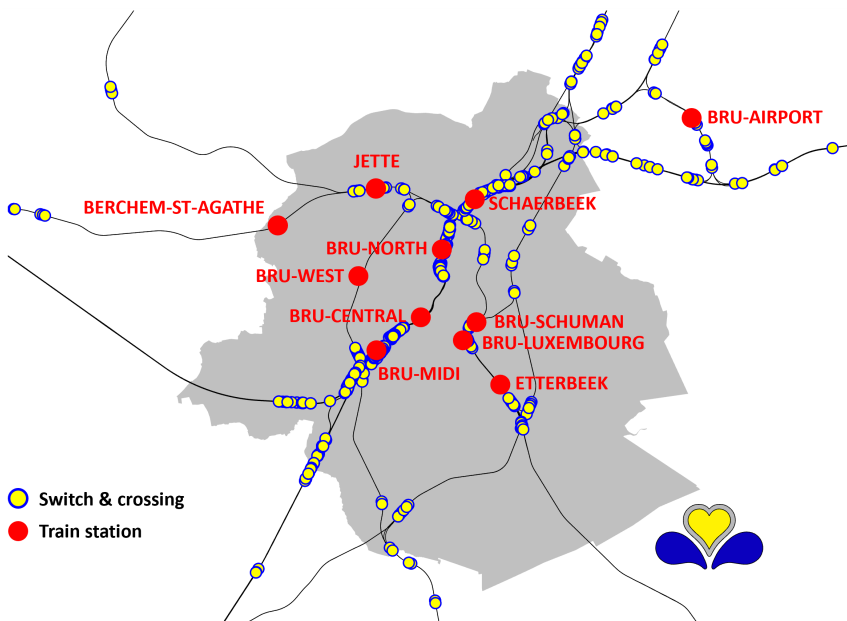


Fig. 1.2 Railway network in the Brussels-Capital Region, showing its turnouts, main stations and selected secondary stations.

1.1 Switches and crossings in the railway network

Switches and crossings (S&Cs) denote the elements of the railway infrastructure that allow trains to cross a track and to move from one to another. They facilitate complex operations such as overtaking, routing and access to stations, depots or sidings. Although they constitute a limited proportion of the track layout, their operational role is critical, particularly in complex junctions and high-density areas.

The Brussels-Capital Region is a clear example of such high-density areas, as illustrated by the number of stations, turnouts, and rail lines shown in Fig. 1.2. The ratio of all types of S&Cs per kilometre of track in this region is at least twice as high as in other parts of Belgium, as shown in Table 1.1. This concentration therefore implies increased maintenance complexity for the railway network in the capital.

Region	Track type	Track length [km]	N° of S&C	Ratio [S&C/km]
Brussels-Capital	Main	316	472	2.00
	Secondary	124	412	
Flanders	Main	3289	2016	1.06
	Secondary	1018	2561	
Wallonia	Main	2946	1509	0.81
	Secondary	556	1325	
Total		8249	8295	1.01

Table 1.1 Track length and number of switches and crossings (S&C) by track type and region in Belgium, including the total S&C density per region, in 2024. *Source: Infrabel, Open Data* [5].

A typical S&C generally has movable rails (denoted as switch rails) that guide the train from one route to another, and crossings (or frogs) where wheels pass through intersecting rails. Due to the dynamic interaction with passing trains, these components suffer intense mechanical stress including not only vertical loads but also significant lateral forces, especially during route changes. As a result, they represent some of the most maintenance-intensive and failure-prone parts of the railway infrastructure.

There are different types of S&Cs (Fig 1.3) depending on the number of tracks to connect, the desired trajectories, the available space and the cost — i.e., complex S&C such as double slips are more expensive than a right-hand turnout. They can be mounted on wooden or concrete sleepers.

The general preference is to install the simplest configurations — such as right-hand, left-hand or Y turnouts — whenever possible, due to their lower cost, easier maintenance and lower complexity. When the space is limited but multiple trajectories are required, then more complex configurations may be considered, for example: crossover, diamond crossing, single slip, 3-way turnout or double slip. All these configurations are illustrated in Fig. 1.3.

This work focuses on a right-hand turnout (shown at the top of the figure) mounted on wooden sleepers. It allows the train to move between the main route and either the through route or the diverging route, which is determined by the positioning of the switch rails. It consists of three main zones — the Switch, the Closure, and the Crossing Panel — whose components will be described in detail in Chapter 3.

1.2 Research project and scientific motivation

As previously mentioned, Brussels has a dense railway network that constitutes a critical mobility link in the surrounding area and even throughout Belgium. The reliability of its numerous switches and crossings (S&C) is essential to maintaining smooth railway traffic. They therefore require constant attention to ensure that they are in good working order: maintenance of their actuators, checking of their geometry, etc.

Infrabel, the Belgian railway infrastructure manager, must regularly check the geometric dimensions of the S&C to ensure that the vehicle wheelsets can pass over them safely. Traditionally, these dimensions are checked manually, using a specific ruler limited to static measurements and requiring staff to work directly on the tracks. This means that measurements can only be performed when no trains are passing through the section being inspected.

Recently, Infrabel developed an on-board system on their measuring train, the EM130 [6], that measures the above-mentioned dimensions dynamically, at speeds of up to 40 km/h. This measuring train operates within regular traffic, offering a safer alternative to manual inspections and enabling the acquisition with high frequency.

These dimensions (Fig. 1.4) must be checked to ensure that they remain within the tolerances and do not surpass the Immediate Action Limit (IAL). This threshold represents the values beyond which immediate track intervention is required.

The problem is that, quite often, a measurement taken with the digital

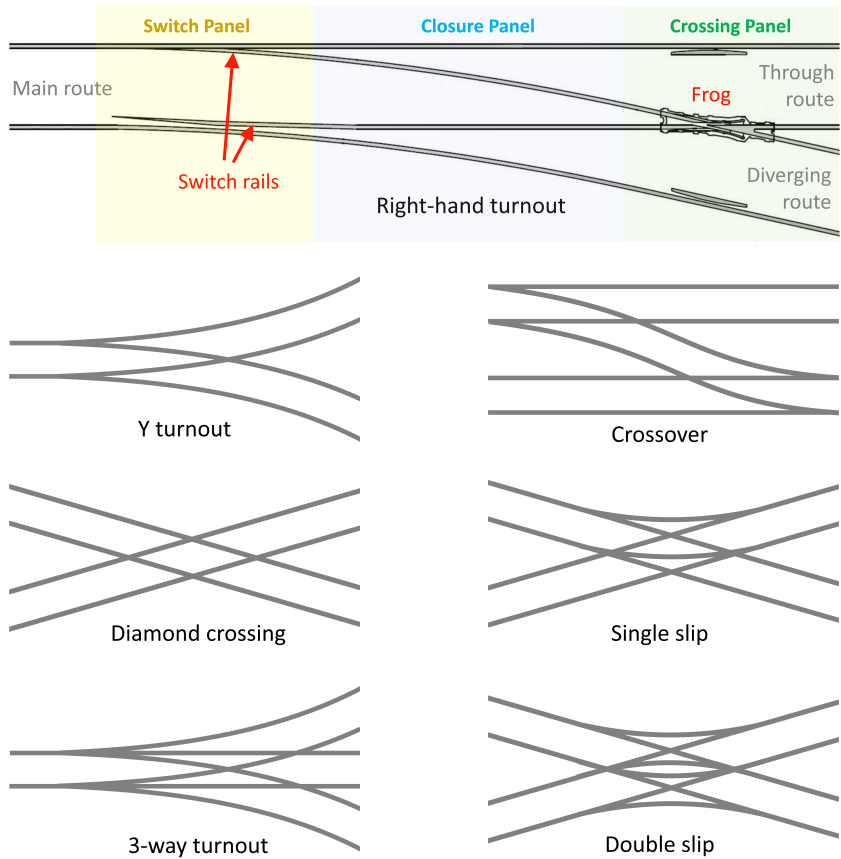


Fig. 1.3 Examples of different types of switches and crossings (S&Cs). The one studied in the present work is a right-hand turnout, shown at the top of the figure.

ruler is within tolerance, while the same measurement performed with the measuring train indicates that the geometry has exceeded the maintenance threshold. In other words, EM130 measurements sometimes suggest the need for maintenance — to avoid reaching the IAL — even when the static measurements do not exceed any threshold.

Why does this happen? That is the question Infrabel is addressing. An evident dynamic effect occurs when the measuring train passes through the switches and crossings: due to loading forces — from weight or lateral forces

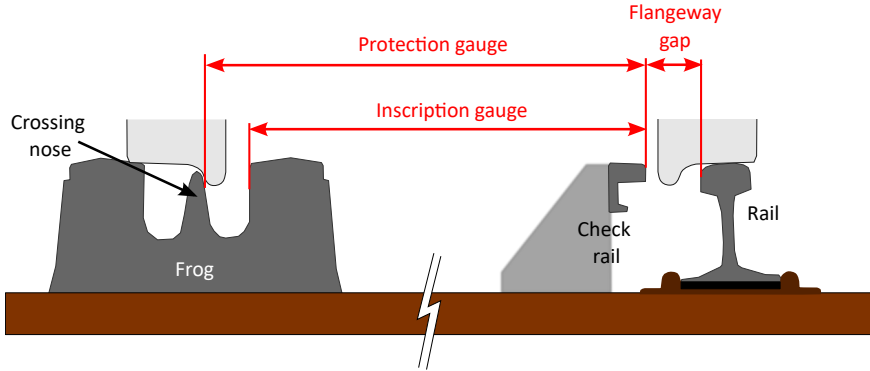


Fig. 1.4 Examples of critical dimensions to be checked in the S&C by the digital ruler and the measuring train.

linked to the train changing direction — the track undergoes deformations that are not yet fully understood. Under both vertical and lateral loads, the stiffness of the track plays a key role, as it governs how the rails and their support system respond under dynamic conditions and thus influences the resulting deformations. These deformations are captured by the measuring train, creating a clear gap between the static measurements (digital ruler) and the dynamic measurements (measuring train).

This research constitutes the AIGUIDYN¹ project, a collaboration between Infrabel and UCLouvain. The project aims to combine and compare dynamic measurements from on-board train sensors and from sensors installed on the track (on-track sensors) and static measurements (manual ruler), with a view to better quantifying the risks of derailment and adapting the maintenance procedures of S&Cs by promoting the use of more effective and safer measurement techniques for track technicians.

From a scientific point of view, the long-term motivation is to better understand and quantify the interaction between a vehicle and a S&C subject to dynamic motions (train) and deformations (track), using *in situ* experimental measurements and computer modeling tools.

¹Applied PhD, funded by the Brussels-Capital Region through Innoviris, reference 2021-PHD-8, duration 4 years.

1.3 Research questions

The present research aims to **bridge the gap** between static and dynamic measurements of S&Cs by conducting a comprehensive experimental campaign, developing a computer multibody model of the system (vehicle + turnout) and analyzing the results in light of the research questions of this work, which emerged at the beginning and during the early phase of the project, based on our interactions with the industrial partner, Infrabel.

To set the context regarding the requirements for onboard measurement systems, standard EN13848-2 (2020) [7], which regulates the minimum requirements for measurement systems on board track inspection vehicles, does not define anywhere where the measurement system should be ideally installed in the train (e.g., between bogies, near the wheel-rail contact, etc.). It only implies a general recommendation stating that “the sensors placed under the vehicle’s frame shall be as close as possible to one of the vehicle’s loaded axles to respect measurement conditions indicated in EN 13848-1 (2019) [8]”, leaving some freedom for interpretation and variability in the installation.

The standard EN13848-1 (2019) further emphasizes that, “in order to reproduce the dynamic effects of vehicles, all the geometric parameters should preferably be measured on a loaded track”. It also warns that differences in track geometry parameter values can occur depending on whether measurements are taken under loaded or unloaded conditions, or under static or dynamic conditions. These differences must be considered when comparing measurements obtained under different conditions, including direction of travel and speed.

This raises the question of how the positioning of the measurement system, as well as other track or train configuration parameters, may influence the recorded data. Accordingly, the research questions are listed as follows:

- **Q1) Influence of the location of the onboard measuring system on the recorded measurements:** how does the longitudinal location on the train of the measurement device affect the recorded geometry, especially in relation to the actual wheel-rail contact local deformation?
- **Q2) Influence of train running direction, velocity and orientation on the rail deformation:** how do these operational parameters impact the dynamic response of the turnout and the accuracy of the geometrical measurements?

- **Q3) Impact of lateral stiffness on rail track dynamics:** to what extent does the lateral flexibility of the rail and its support system contribute to the observed turnout geometry?

Assuming these questions will provide practical guidance for future reinstallation of the onboard measuring system, as well as for planning measurement campaigns or interpreting data collected under varying operational conditions (e.g., train orientation, running direction and velocity).

The core innovation of the project lies in the integration of dynamic components into both the measurement process (capturing deformations, lateral movements and impacts) and the modeling approach: a multibody simulation model is developed to quantify at best the dynamic interactions between the vehicle and the S&C. This work will focus on a specific type of turnout — for example, a right-hand turnout — to illustrate and justify the general problem.

From Infrabel's perspective, the implementation of these experimental protocols and modeling tools for their own S&C designs (switches, crossings, fastenings) represents a significant innovation. Previous projects have focused more on prospective technologies for future switch systems, whereas AIGUIDYN [9] addresses current infrastructure with immediate applicability.

1.4 Structure of the thesis

The thesis is structured into five main parts:

- **Part I** introduces the research project. Following the context already presented in Chapter 1, Chapter 2 provides a state-of-the-art review of the experimental procedures carried out and the numerical models of railway vehicles, with a particular focus on switches and crossings (S&C).
- **Part II** is divided into two chapters aiming at presenting experimental campaign carried out during the project. The first one, Chapter 3, explains the experimental methodology used for the dynamic tests performed during train passages over the S&C under different conditions of velocity, trajectory and route type. The second one, Chapter 4, describes the methodology followed to quantify and discuss the lateral stiffness of the rails as a function of applied load and for different types of rail supports.

- **Part III** deals with the numerical model based on a multibody approach. The only Chapter 5 in this part firstly introduces the multibody formalism used in the research. Secondly, it discusses the different types of wheel/rail contact and the corresponding contact laws. Thirdly, it presents the measuring train model, followed by the description of the turnout geometry. Finally, all simulation cases considered in the numerical model are described, including input parameters and expected outputs for each scenario.
- **Part IV** provides a detailed analysis of the results obtained from both the experimental campaigns and the numerical simulations for the Closure (Chapter 6) and for the Crossing Panels (Chapter 7). It deals with the experimental validation of the model, a discussion of its assumptions and limitations and a comparative analysis of the two zones studied: the Closure Panel and the Crossing Panel. A synthesis and discussion of these results, and answers to the research questions defined in Section 1.3 are presented in Chapter 8.
- **Part V** concludes the manuscript. Chapter 9 presents the main conclusions of the thesis, bringing the key contributions of the work and Chapter 10 some future perspectives in the field.

2

State-of-the-art

Although the scientific literature is not abundant in the field of S&Cs, the present Section highlights scientific contributions structured around two distinct themes: experimentation and instrumentation on the one hand, computer simulation and benchmarks on the other. The following subsections are structured along these themes, even though some of the works are dealing with both fields, as it should be.

2.1 S&C: instrumentation and experimentation

2.1.1 On-board systems for dynamic S&C inspection

In the literature, several techniques are used for the dynamic inspection of switches & crossings (S&C) using on-board systems, with the objective of capturing either the geometry or the deformation of the track. On one hand, methods such as laser triangulation sensors, high-definition cameras, photogrammetry, and the chord method are mainly focused on reconstructing the geometry of the track, measuring parameters such as rail profile, gauge, or curvature. On the other hand, systems like IMUs and multisensor configurations embedded in trains aim at continuous monitoring to detect rail deformations and vibrations, which can be used to assess the track condition or the evolution of its stiffness.

Sometimes, some of these systems are installed on in-service trains, but there are also specialized measuring trains that allow for an automated inspection of the geometry of the railway track, including switches and crossings (S&C). These trains are often instrumented with laser sensors, cameras, and data acquisition systems which contribute to preventive maintenance and improved safety.

This study uses the EM130 measurement train, operated by Infrabel, that combines laser sensors and high-definition cameras to measure key geometric parameters, such as track gauge at the closure panel and the critical flangeway gap — that is, the clearance between the running rail and the check rail — at the Crossing Panel.

There are some research works, e.g., that of Minbashi et al. (2016) [10], which report the application of similar optical technologies as those employed in the train considered in our work to generate three-dimensional profiles of the rail and to detect wear or cracking. For example, SIM (Switch Inspection and Measurement) [11] and ASIV (Automated Switch Inspection Vehicle) [12] systems utilize laser sensors and machine vision technologies to perform automatic inspection at operational speeds.

The research by Zschesche (2024) [13] and Izvoltova (2013) [14] emphasizes the application of laser scanning and photogrammetry for identifying rail geometric deviations and creating three-dimensional models in global reference systems, which are useful for tracking long-term deformations.

Research in this area has resulted in patents like EP2165915A2 (2010) [15], created by VolkerRail Nederland. This patent proposes an innovative architecture with laser triangulation sensors and vision modules mounted obliquely to acquire the rail profile with high accuracy and identify deformations in components like switches and frogs.

Following this line of research, Hauck and Gniado (2024) [16] present an on-board system divided into two modules: one for rail geometry measurement (by employing the chord method to estimate track curvature) and one for infrastructure inspection (crossings, fastenings, ballast). The system makes use of triangulation laser scanners and a compression algorithm, enabling the effective storage of image format data.

Alongside optical systems focusing on geometric profiling, inertial measurement unit IMU-based systems offer an alternative for continuous monitoring. Farkas (2020) [17] presents a detailed review of these systems, although limited in spatial accuracy by the need to integrate accelerations. Along these lines, Yeo (2017) [18] implements a system on board a British commuter train, demonstrating that an IMU mounted on the bogie can

achieve self-consistent repeatability within 0.2 mm, comparable to that of dedicated vehicles such as the New Measurement Train (NMT) [19], and proposes methods for estimating track stiffness from multiple passes.

Complementary, Weston et al. (2015) [20] highlights that sensors embedded in trains in service allow for a much higher measurement frequency, which facilitates the monitoring of degradation and the evaluation of maintenance. Nevertheless, they caution that accurately reconstructing the geometry requires careful compensation for the dynamic effects of the vehicle.

Although these systems are technologically advanced, none of the cases analyzed clearly describe the exact location of the measurement system on the train or specify the optimal relative position with respect to a reference on the train. This lack of information limits direct comparison with the EM130, whose measurement system is installed between two wheelsets on the same bogie. This configuration allows capturing geometric parameter in an area theoretically considered representative of the vehicle's dynamic behavior.

As previously discussed, EN13848-2 (2020) [7] does not explicitly describe the exact location of on-board measurement systems on track inspection vehicles, thus allowing for interpretation and variability in practice.

2.1.2 On-track measurements and in-situ testing of S&C

Track instrumentation and field testing play an important role in understanding the dynamic behavior of S&Cs. Reliable and precise measurement techniques are required to monitor their geometry and ensure proper performance and safety. A review of the state of the art is therefore necessary, as the present work includes two experimental campaigns in which a S&C is instrumented with sensors. Several research initiatives have addressed this issue from complementary experimental perspectives.

Large European projects such as S-CODE [21], TURNOUTS [22], IN2-TRACK [23] and INNOTRACK [24, 25] provide good examples of the integration of field measurements into innovation strategies for S&Cs. The TURNOUTS project (2003–2007) studied new urban rail turnout design concepts. Laboratory tests with loads that simulated wheels were carried out to validate structural resistance and stiffness. Then, they were followed by in-situ installations where the dynamic response of innovative turnouts was compared to conventional ones. Additionally, the S-CODE project (2016–2019) explored new switch concepts and focused on dynamic and fault monitoring. Optical sensors and accelerometers were used in field

tests to determine performance and durability. New materials, such as self-healing composites, were also tested under operational conditions. The IN2TRACK project (2016–2019) developed whole-system models for S&Cs, investigated rolling contact fatigue and implemented sensor demonstrators for remote condition monitoring. The earlier INNOTRACK project (2006–2009) also contributed to the on-field research of S&C by conducting field tests on optimized switch geometries and stiffness configurations. In this project, simulation models were validated and design improvements were proposed to reduce lifecycle costs.

Zhai et al. (2009) [26] analyzed the experimental results obtained in China [27] to validate their three-dimensional coupled vehicle–track model. The campaigns consisted of an acceleration test and a high-speed test in sharp curves. The measurements were done by sensors installed on the rails and vehicles to record wheel/rail contact forces, accelerations, and displacements. For instance, on a curve with a radius of 287 m and wooden sleepers, the passage of freight trains generated lateral rail displacements of up to 5 mm and dynamic track gauge widening of up to 9 mm. These data were essential for calibrating the numerical model, which reproduced the phenomenon of gauge widening under simulated conditions. Relevant results from this experiment, such as the opposite direction of the contact forces on each rail (Fig. 2.1), the lateral displacement, and the dynamic widening of the track gauge, constitute a precious reference for this thesis — also dealing with both experimental data and computer simulation — and demonstrate the importance of analyzing the track/vehicle interaction in track devices, like S&Cs.

The closest example of an extensive field test related to this thesis is that conducted by Kassa (2008) [28], performed on a right-hand S&C in Sweden. In that study, only the wheels of the train were instrumented with strain gauges to measure lateral and vertical contact forces between the wheel and the rail. The test train, with an axle load of 25 tonnes, passed through the S&C on both the main and diverging routes. The results indicate that the maximum lateral and vertical contact forces were significantly influenced by: train velocity, moving direction (facing or trailing), and route (direct or diverging). Some interesting remarks are useful for the present work, like for example, the lateral contact force remain similar for velocities up to 40 km/h, but increase sharply at higher velocities (from 60 km/h), especially in the diverging route. These parameters will be examined later in this thesis.

As with Kassa, Drożdziel et al. (1999) [29] analysed vehicle-track inter-

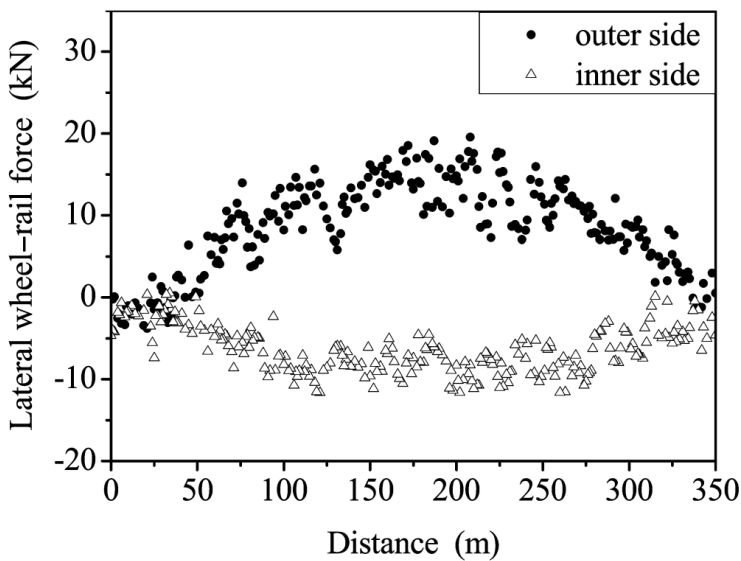


Fig. 2.1 Measured lateral wheel–rail forces on a curved track by Zhai (2009) [26]. Positive values correspond to the outer rail (force towards the outside of the curve), and negative values to the inner rail (force towards the inside), resulting in a dynamic widening of the gauge.

action and showed that even geometric deviations within tolerance could lead to lateral forces of up to 40 kN in the crossing area, necessary to guide the wheel, with opposite forces between inner and outer wheels on curves. These results emphasized the limitations of traditional monitoring and encouraged the implementation of more precise measurement methods. This force magnitude in the crossing as well as the sign of the forces in each rail will serve as reference for the present study.

Subsequent research focused on multisensor experimental setups and model validation. Shih et al. combined laboratory and field approaches: in 2018 [30] they simulated a “run-through” event on a non-trailable turnout using a hydraulic actuator, showing that forces up to 35 kN did not damage drive components but could deform switches, thereby validating a 3D FEM model. In a later field campaign in 2022 [31], accelerometers, geophones, and strain gauges were used to study the crossing zone. Results showed that lateral accelerations are not enough to determine degradation, whereas shear strain gauges permitted the estimation of impact forces more effectively. An index based on shakedown theory was suggested for the prediction of fatigue cracks, supporting the advantage of multisensor approaches to S&C monitoring.

Continuing this direction of multisensor development, Liu et al. (2018) [32] put forward two complementary devices for railway crossing condition monitoring: ESAH-M [33], coupling accelerometers and inductive sensors for the detection of impact accelerations and irregular flange contacts, and a Video Gauge System (VGS), relying on digital image correlation for measuring rail and sleeper displacements. Field tests proved them capable of identifying worn, damaged, and repaired crossings, showing the promise of this kind of multisensor systems for predictive maintenance of S&Cs.

In summary, the development from full-scale projects through to on-field studies and multisensor strategies demonstrates a logical progression: an understanding of geometry and load effects upon S&C forces has directly informed the advancement of novel monitoring techniques and predictive maintenance strategies, providing the basis for the experimental methodology followed in this thesis.

2.1.3 Experimental evaluation of lateral track stiffness

Lateral stiffness of the track plays an important role in the maintenance of the track geometry, especially in curves and in switches & crossings, where the lateral forces are more important, due in part to wheel flange/rail

contact. If the rail suffers excessive displacements, the rails can lead to gauge widening or changes in other critical dimensions like the flangeway gap that could lead to safety issues. Several experimental studies have searched for the quantification of this stiffness by field or laboratory experiments. They tested different load and fastening conditions, with the objective to improve the predictive models and the maintenance strategies.

An interesting reference is the full-scale tests conducted by Liu et al. (2014) [34]. As in the current Aiguidyn project, they used a hydraulic jack that applied incremental force between the two rails. The objective was to test three types of WJ-8 fastenings [35] and thus determine the lateral stiffness. The results identified the nonlinear relationship between force and displacement, with an "turning point" at which static friction gives way to rail slip. During this slip phase, the average lateral stiffness values ranged from 38 to 55 kN/mm.

More recently, Wang et al. (2024) [36] conducted laboratory tests also using hydraulic jacks. The objective was to characterize the nonlinear lateral stiffness of fastening systems for urban railway applications. Intervals of 2 kN were applied to the rail base, up to a maximum of 30 kN and its lateral displacement was recorded. This test provided load-displacement curves that evaluated the nonlinear behavior of the stiffness. The results showed significant differences between two types of elastic pads (Type A and Type B in Fig. 2.2). In the case of type A, three distinct stages of stiffness were identified: shearing between rail pad and baseplate, sliding between baseplate and baseplate pad, and collision between the plate and the anchoring bolts. In contrast, type B did not present a sliding phase. Its behavior went directly from the shear stage to the collision stage. These differences are mainly attributed to the coefficient of friction between the base pad and the base plate, which is higher for type B, directly influencing the transmission of lateral forces and the occurrence of a sliding phenomena. In this thesis, the force and displacement applied to the rail head may not be directly comparable, since the stiffness values correspond to the rail base — that is, the displacement of the base under a lateral force applied at the base —, and a greater flexion of the rail occurs when a force is applied on the head instead of the base. However, the stages in the stiffness curves are an interesting context for interpreting the lateral stiffness measurements that we obtained, especially regarding the transition between static friction and sliding.

While vertical loads were not the main focus in the experimental studies previously mentioned, Chen et al. (2016) [37] present an in-depth analysis

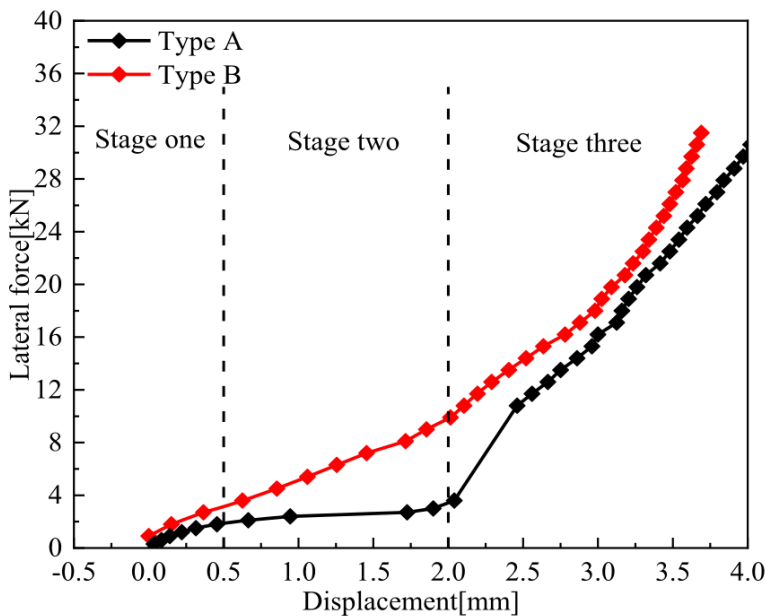


Fig. 2.2 Displacement versus lateral force curves applied to the rail base for two types of rail pads by Wang et al. (2024) [36]. Three different stages obtained are observed for the type A: shearing, sliding, and collision. For the type B, no sliding stage is observed.

of the dynamic response of railway S&Cs under both lateral and vertical wheel impact loads. Their work integrates field-measured rail profiles into multibody and FEM simulations to compare stress distributions and identify zones sensitive to failure. Building on from early research investigating the correlation between wheel-rail interaction and rail seat degradation — such as those by Zarembski (1977) [38], Choros et al. (2007) [39], Marquis et al. (2011) [40], and Wu and Kerchof (2014) [41] — Chen and Andrawes (2016) [37] worked on a model to quantify the lateral rail head deflection as a function of the fastening system design. Their results show that under increasing lateral wheel loads and a constant vertical force, the effective shear area of the rail pad is reduced due to rail base rotation. This results in a nonlinear reduction in lateral stiffness (see field measurements in Fig. 2.3), as also observed by Liu et al. and Wang. These observations serve as valuable guidance for the experimental results of the present work, as they highlight (i) the importance of considering the coupled effects of vertical and lateral loading in the definition of lateral stiffness, as well as (ii) the need to take into account rotational effects and nonlinear stiffness behavior when interpreting measurements under increasing lateral forces.

Also considering vertical loads, Zakeri and Barati (2015) [42] investigated the effect of these on lateral resistance. They conducted tests on track with standard and frictional sleepers (with a ribbed underside). While they measured lateral displacement of the sleeper instead of the rail - as in the present thesis - where the force was applied, their findings provide hints regarding how vertical load can enhance lateral stiffness of the track infrastructure, particularly in sleepers with a grooved base that enhance friction with the ballast.

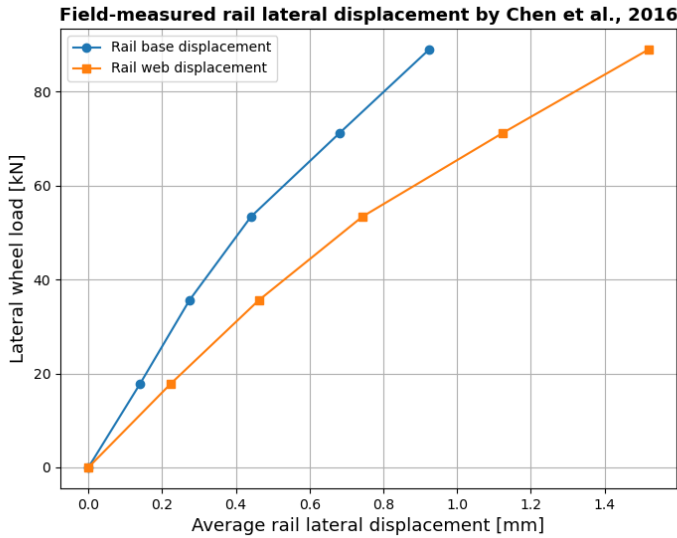


Fig. 2.3 Field-measured rail lateral displacement under increasing lateral wheel loads and a constant vertical load of 178 kN. The graph is built with the data from Chen et al. (2016) [37]. The results illustrate the nonlinear behavior of the rail structure.

Regarding the influence of rail stiffness on track degradation, Berggren et al. (2009, 2014) [43, 44] developed a method for continuous measurement of vertical track stiffness and deflection using a track recording car. Their results also showed that stiffness variability along the track is a good indicator of areas susceptible to accelerated degradation. Moreover, Sadri and Steenbergen (2018) [45] also used energy dissipation under dynamic stress to construct a theoretical model. Although strictly theoretical, Sadri and Steenbergen's are consistent with the experimental interpretation in that rail bending stiffness, sleeper spacing, and rail pad stiffness are significant design elements that affect degradation rates.

Malekjafarian et al. (2021) [46] have studied the identification of local reduction in track stiffness with a new method called Bogie Filtered Displacement (BFD). This method double-integrates bogie vertical acceleration and applies a band-pass filter to estimate track stiffness variation. It captures key features of the loaded track profile and enables detection of defects like hanging sleepers using data from regular train when it is in service.

This study showed that repeated measurements can indicate more precisely possible losses in track stiffness.

In summary, experimental research has shown that lateral track stiffness is highly sensitive to fastening conditions, sleeper design, vertical load, and changes in subgrade stiffness. Nonlinear behavior and friction effects must be considered for an analytical study of track dynamics. Repeatability is also important to consider for an analysis of track stiffness evolution. This knowledge serves as the basis for defining the experimental methodology adopted in this thesis and its subsequent modeling analysis, especially in the context of track devices, where lateral forces are critical.

2.2 S&Cs: computer benchmarks and simulation analysis

Numerical models and simulations of S&Cs can complement experimental tests to study dynamic scenarios that are difficult to realize in practice. Therefore, simulation benchmarks provide a standardized framework to validate models and compare approaches. Scientific contributions regarding simulations and computational benchmarks for railways, in particular for S&Cs, are discussed below. Specific results from these studies that can be compared to those obtained in this work are highlighted.

A reference benchmark for simulating dynamic vehicle-track interactions in S&Cs is presented by Bezin (2021) [47]. The vehicle is the passenger train from the Manchester Benchmarks (Iwniki, 1998) [48]. Some modeling challenges as abrupt changes in rail profiles, simultaneous wheel-rail contact points and varying track characteristics are considered. Different geometric difficulties are highlighted by evaluating two S&C designs: a Swedish 60E1-R760-1:15 and a British 56E1-R245-1:9.25. The purpose of the benchmark is to allow for replication and compare results (Bezín, 2023) [49], highlighting key differences in software performance, particularly regarding rail geometry representation, contact detection and dynamic load prediction, which all are critical for a S&C model accuracy.

Following Bezin's benchmark, Pålsson et al. (2021) [50] compare various track model formulations for vehicle-track interaction modeling of S&Cs. On the basis of a FEM model of a 60E1-760-1:15 turnout [51], they compare co-running, modal superposition, finite element integration within multi-body simulation (MBS), and FE-MBS co-simulation methods. Their results demonstrate good agreement among the models in wheel-rail forces and rail displacements, with more advanced formulations successfully reproducing higher-frequency responses. This research offers a helpful guidance

for choosing the most appropriate modeling technique based on particular simulation goals.

One of the most influential contributions in the field of dynamics has been Zhai with his theory of Vehicle–Track Coupled Dynamics (2020) [52]. Zhai studied the dynamic interaction between vehicle and track as an integrated system rather than treating the two subsystems separately. As noted in Zhai (2009) [26], more than 10 studies deal with track dynamics and wheel–rail dynamic interactions [53–65], however, few on the lateral system dynamics [54, 57, 62, 65]. Zhai’s coupled models allow us to study gauge widening on closed curves, lateral rail displacement, and the impact of stiffness on vertical and lateral forces. His models have been validated through experimental campaigns, such as the one described in [27] and discussed in the previous section.

Pletz et al. (2009–2012) [66–68] also used a FEM that they developed to analyze the passage of a wheel over the crossing panel, considering both facing and trailing moves. Vertical contact forces and von Mises stress levels are evaluated according to the direction of the train’s passage, its speed, and the nature of the material in the crossing. It is observed that in the facing move, the impact, that occurs on the crossing nose, is higher than the one in the trailing move, where the impact occurs on the wing rail. A transition zone is also identified in which the wheel is simultaneously in contact with the wing rail and the crossing nose. This causes sliding and oscillations that modify local stiffness and wear. The Aiguidyn model used in this project focuses on lateral impacts with the S&C and less on vertical impacts. It is important to bear in mind the influence of vertical loads on lateral stiffness in order to understand the potential limitations of the present model.

The effects of track irregularities on the dynamics of a train have been investigated over the past years in the literature. Andersson and Dahlberg [69] considered the effects of irregularities in crossings, whereas Alfi and Bruni [70] included profile variations and longitudinal stiffness in their coupled model. Xu et al. [71] assessed the wear of movable rails and its influence on dynamic loads through multibody and FEM simulations. In accordance with these strategies, Bosso et al. [72] investigated the effects of track irregularities in S&Cs on vehicle dynamics through multibody simulations. They identified critical defect combinations affecting safety and stability, highlighting that widening track gauge and reducing rail pad stiffness can decrease impact loads and wear.

The state-of-the-art review of Bruni (2020) [73] discusses the developments in multibody simulation methods for railway systems, including

S&Cs. It targets wheel-rail contact modeling, flexible track model integration, and co-simulation methods, emphasizing trends such as hybrid simulations and virtual certification to connect numerical simulations and physical testing.

While the previous studies provide valuable context on simulation methodologies and modeling approaches of S&C, below are some references where the results can be more directly related to those of our work.

Hiensch and Burgelman (2017) [74] carry out multibody simulations to analyze the influence of operating factors on lateral wear in S&C. Switch rail lateral wear in the diverging route of a turnout is found to be significantly influenced by wheel-rail friction, while the direction of travel (facing or trailing) had a lesser effect. Other investigated parameters, such as vehicle speed and gauge widening, had a small influence on lateral wear. It is important to point out, as an interest for the present work, that for the leading wheel of each bogie, contact with the flange occurs almost immediately after entering the switch, resulting in a sharp increase in the lateral wheel force, as will be observed later in this thesis. Throughout the switch and crossing panels, the wheels remain in flange contact, showing a rather irregular behavior during the first 8 meters. Unsurprisingly, the leading wheel of the front bogie suffers a higher lateral force than the leading one from the second bogie in the Switch & Crossing zone.

Lagos et al. (2012) [75] evaluate the impact of turnout designs and wheel profiles on vehicle dynamics using the Simpack software [76]. Comparing a conventional and a modified turnout, they show that geometric modifications, like varying the geometry of the switch and including moveable crossing vees, reduce rail impacts and improve dynamics, while wheel profile variations have limited influence. Another interesting observation for the present work is that the lateral forces generated when circulating on divergent tracks have a higher variation in the Switch Panel and in the Crossing Panel, where the rail geometry varies the most.

As discussed in the previous section, Kassa (2006-2008) [77] [28] conducted field experiments whose measured contact forces are compared with calculated forces for a validation of two numerical models. The results show that velocity, direction, and S&C route influence both lateral and vertical forces, with higher lateral forces obviously observed at higher speeds in diverging routes. This information provides valuable insights, since the estimated parameters are in line with several of those explored in this thesis.

Ren et al.[78] established a coupled vehicle-turnout dynamic model to study lateral and vertical behavior while a vehicle passes through the

diverging route of a S&C. The results of this model show that the peaks of the lateral force are produced in the Switch Panel, while vertical forces are highest at the nose area. In the curved section, the lateral force on the outer rail is greater than that on the inner rail. As a result, the outer rail of the curve suffers greater lateral displacement than the inner rail, resulting in gauge widening. In tight curves, the maximum lateral displacement of the bogie occurs due to centrifugal effects, increasing rail wear.

As in the present model it will be necessary to define a lateral stiffness at the contact between the wheel flange and the rail, Table 2.1 summarizes several values used in simulations and models found in the literature.

Source / Author	Rail pad (kN/mm)	Check rail (kN/mm)	Model
Pålsson et al. (2021)	25	120	FEM
Zhai (2020)	20–50	20–50	Simulation Parameters / Literature
Ren (2005)	50	15	Simulation Model (Chinese No.12 turnout)
Bezin et al. (2021)	–	120	Multibody Simulation Benchmark

Table 2.1 Lateral stiffness values of the rail and check rail used in various models and simulations in the literature.

In summary, benchmarks and numerical models have been proven to be complementary tools for understanding the complex dynamics of S&C. Zhai’s coupled models, Bezin’s benchmarks and Hiensch’s multibody simulations showed that elements such as lateral stiffness, S&C geometry, wheel profile and contact conditions have a significant impact on dynamic forces and rail wear. These analyses provide a useful comparative framework for the findings of this thesis, particularly concerning lateral stiffness, behavior on diverging routes and wheel/rail contact in sensitive areas such as the Crossing Panel.

2.3 Thesis contributions

Even if still limited, research in the field of switches and crossings (S&C) has grown in recent years. Typically, studies have focused on validating numerical models or optimizing turnout geometry. However, there are still gaps

in the integration of dynamic and static measurements, limitations in on-board measurement systems and a lack of comprehensive approaches that combine multiple sources — such as numerical models and field experimentation — for assessing real-world infrastructure performance. To address these gaps, this thesis combines static and dynamic measurements — both on-track and on-board — to detect the limitations of current inspection methods and to provide a basis for future improvements in maintenance equipment and strategies for existing switches and crossings.

The main contributions of this work are:

Comprehensive experimental campaigns

- Design and execution of an experimental campaign integrating three types of measurement: on-board dynamic measurements, on-track dynamic measurements, and on-track manual measurements. The campaign uses the EM130 measuring train equipped with on-board sensors that passes through a S&C by high-resolution on-track instrumentation (LVDTs, laser sensors, cameras).
- Design and execution of a static experimental campaign to quantify lateral track stiffness under static conditions.
- Critical analysis of the collected data and validation of the complementarity between the three types of measurements inputs, which will serve as input for the numerical model.

Multibody simulation model

- Adaptation and use of a multibody model of the measuring train, including a detailed S&C geometry and their interactions by the wheel–rail contact laws.
- Simulation of several parametrized scenarios (velocity, direction, center of mass location, stiffness) to replicate and interpret experimental results. They are complemented by numerical exploration of cases that could not be reproduced experimentally.

Integrated analysis of field and model data addressing the research questions

- Q1) Influence of the location of the on-board measuring system on the recorded measurements: evaluation of the limitations of the Switch

Measurement System (SMS) in capturing peak rail displacements due to its positioning.

- Q2) Influence of train direction, velocity, orientation and wear on rail dynamics: analysis of the influence of trajectory, velocity, wheelset position and vehicle mass distribution on rail displacements (for speeds ≤ 40 km/h in the diverging route), to assess the nature of the phenomena, and to determine whether dynamic effects are present.
- Q3) Impact of lateral stiffness on railway dynamics: analysis of the lateral stiffness behavior under lateral loads and its impact on railway dynamics.

Practical insights for infrastructure managers

- Insights for Infrabel regarding the interpretation of dynamic measurements and the planning of maintenance operations, which can later be used to enhance predictive maintenance strategies.

PART II

Experimental methodology

3

Dynamic experimental campaign

THIS chapter outlines the methodology employed for the first experimental phase¹. It describes the objectives of the tests, the instrumentation used both on the track and on the train movement, as well as the different stages developed during the experimental process. The analysis of the results will be presented in Part IV.

3.1 Objectives and experimental setup

The objective of the first experimental campaign is to establish a baseline for studying the dynamic effects that occur when a train passes over a turnout (S&C). The aim is to identify and distinguish the variables that may influence this dynamic behavior. Additionally, this campaign seeks to collect data for a specific right-hand turnout that can later be used and compared with the numerical model presented in the following sections.

The chosen experimental site is located on an operational railway track in Schaerbeek, Brussels. The selected turnout is mounted on wooden sleepers, a configuration identified by Infrabel to highlight differences between static and dynamic measurements during their maintenance operations. This

¹Carried out on the weekend of September 16th and 17th, 2023, in Schaerbeek, Brussels.

3 | Dynamic experimental campaign

site was also chosen for its accessibility, which facilitates the installation of the necessary testing equipment, and for the inherent simplicity of the right-hand turnout, thereby reducing complexity and limiting the number of variables compared to other switch configurations.

The studied turnout includes the following key components and zones (Fig. 3.1):

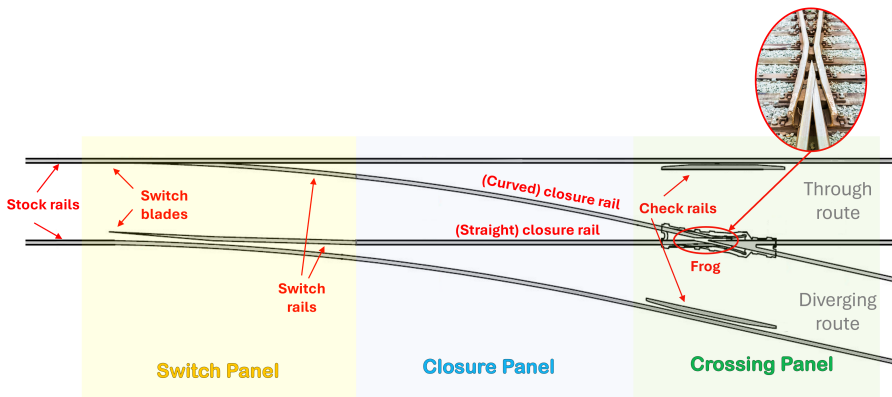


Fig. 3.1 Main components of a right-hand turnout.

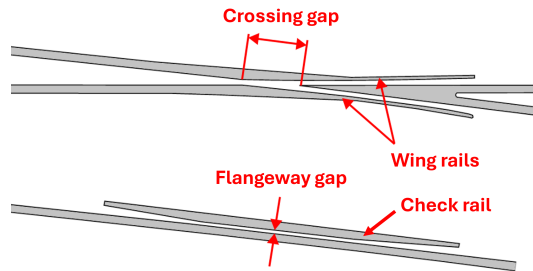


Fig. 3.2 Crossing gap and flangeway gap in the Crossing Panel.

- **Switch Panel:** this zone is crucial for directing trains onto diverging routes. It contains the moving switch rails — that are actuated to guide and transfer wheels from one track to another — and the stock rails against which the switch rails move and rest. The switch blade is

the pointed ends of the switch rail which rest on the stock rails and follow its shape.

- Closure Panel: the closure panel connects the switch panel to the crossing panel. It typically includes a curved section of the track where the two closure rails begin to converge towards the frog.
- Crossing Panel: the salient elements of this Section are as follows:
 - The frog: the zone where the two closure rails intersect. The crossing gap, also known as frog clearance, refers to the specific space defined by the geometric interruption of the two closure rails. The correct dimensions of the crossing gap (Fig. 3.2) are critical to allow the wheels to pass smoothly through the crossing without risk of derailment.
 - The check rail: profiled stop parallel to the running rail that ensures the opposite wheels follow the correct path through the frog.
 - The flangeway gap: it is the space between the check rail and the stock rail that allows for the passage of the wheel flanges through the crossing. Proper measurement and maintenance of the flangeway gap are essential to prevent wheel/rail interference and ensure safe train operation.

3.2 Instrumentation

Measurement systems can be classified into two main groups: on-track sensors, which remain stationary on the track while performing measurements, and on-board sensors, which are in motion, mounted on the train during the measurement (Fig. 3.3).

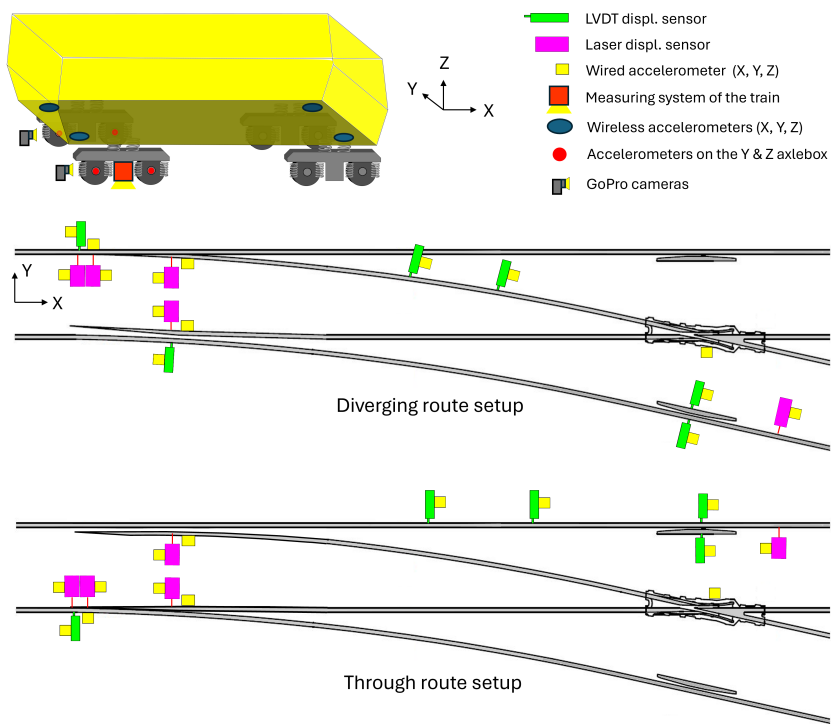


Fig. 3.3 Location of the sensors installed on the turnout: on-track sensors for both the diverging and through route configurations, and on-board sensors installed on the EM130 train for the experimental setup.

3.2.1 On-track sensors

On-track sensors include those installed directly on the infrastructure to measure the track’s physical response in static mode, for example with the digital ruler (Fig. 3.4) when no train is running over the turnout, or in

dynamic mode, during train passages. These sensors provide localized data with high-resolution measurement at critical points of the turnout.

Digital ruler

The digital ruler is a key tool in our experimental setup, since it provides static measurements of the geometry of the track (Fig. 3.4). This highly precise device, with a resolution of $0,1\text{ mm}$, establishes a static reference baseline to compare with the dynamic measurements of the EM130 train and the on-track sensors.

The digital ruler measures parameters such as rail gauge, elevation and twist, providing stable reference dimensions of the track without the influence of any external forces. This reference is essential as it enables us to assess changes in track geometry under dynamic conditions.



Fig. 3.4 The digital ruler [79] is used for static measurements of a railway track.

LVDT displacement sensor

Linear Variable Differential Transformers (LVDTs) are capable of measuring displacements with a high level of accuracy, with a repeatability that does not exceed $1.5\text{ }\mu\text{m}$ for a range of $\pm 10\text{ mm}$ [80]. In our experimental setup, two LVDT maintain direct contact with the rail flange for capturing rail lateral motion and web flexion, respectively (Fig. 3.5). These sensors were sampled at 1000 Hz to capture high dynamic changes during train passages.

The LVDT displacement sensors are strategically placed at locations identified as critical by Infrabel, to measure dynamic rail deflections (Fig. 3.3). These zones are:

- The switch blade, in the Switch Panel, at the point where the switch rail begins to move.



Fig. 3.5 LVDT were placed in different S&C zones to measure the rail head lateral displacement and rail web flexion.

- The open switch, located right after the switch blade and before the Closure Panel, with sensors installed on both sides.
- The Closure Panel, at the intermediate section between the switch panel and the crossing panel, with sensors installed at two different locations.
- The Crossing Panel, at the intersection of the diverging and through routes, where the frog and check rail guide the wheels across the crossing gap.

Generally, LVDT sensors are installed on the rail head and halfway up the web (Fig. 3.6). The sensors on the rail head were positioned 14 mm below the top surface of the rail. This specific height corresponds to the level at which the digital ruler and the laser from the measuring train take their measurements. The second LVDT sensor was installed midway up the web to analyze the potential lateral deflection. For the switch blade and the Crossing Panel, additional sensors were also placed on the side of the sleeper to measure the absolute lateral displacement of the track. This setup allows for a comprehensive analysis of rail movements (lateral and in flexion) in response to dynamic forces.

In the present work, we focus on analyzing the sensors located in the Closure and Crossing Panels, due to scope constraints and the relevance of these zones for capturing critical dynamic behaviors. The objective of the LVDT sensors is to measure variations of dimensions considered critical

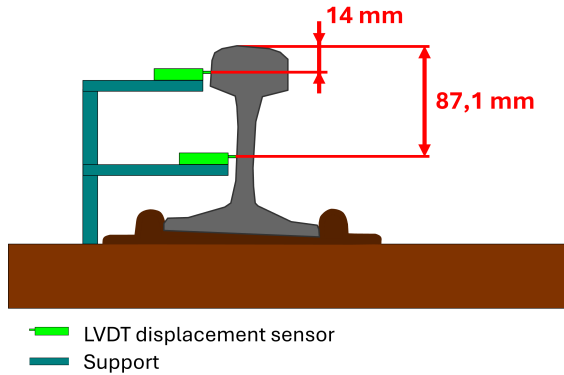


Fig. 3.6 Schematic representation of the vertical positioning of the LVDT sensors (in green) relative to the rail. The upper sensor is located 14 mm below the top surface of the rail head and the other is positioned halfway up the rail web.

by Infrabel. In the Crossing Panel (Fig. 3.7, top), the LVDTs installed next to the check rail and on the running rail head capture variations of the flangeway gap, which can be directly compared with the corresponding SMS measurements. In the Closure Panel (Fig. 3.7, bottom), the sensor setup on the diverging route measures the lateral displacement of the outer rail in the curve, allowing a partial comparison with the gauge measurements from the SMS.

Laser displacement sensor

A Laser sensor (Sick O2 [81]) is essential for measuring lateral displacements that the LVDT sensors could not capture due to their vulnerability to be damaged from the wheels. The laser sensor, also sampled at 1000 Hz, was primarily used to synchronize data from various sensors and triggered to detect the exact location of the train over a precise location of the turnout. This allows us to correlate specific behaviors observed in the data with the exact position of the wheelsets.

To ensure consistency with the LVDT sensors and the laser system on the EM130 measuring train, the laser sensors (in purple in Fig. 3.3) were installed at a height of 14 mm below the top of the rail head — with the exception of the first laser sensor in the turnout directed toward the switch, which was positioned to measure the switch displacement.

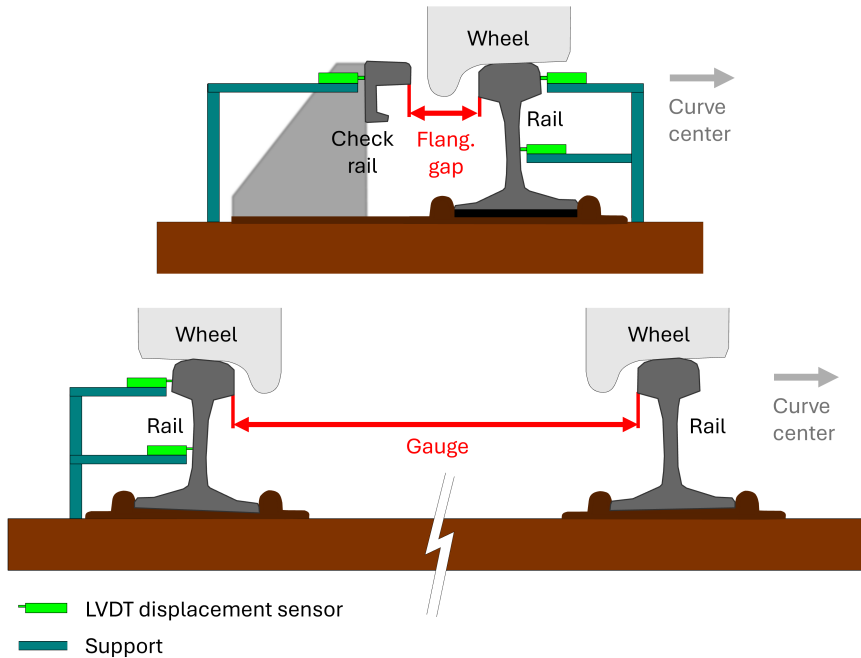


Fig. 3.7 LVDT installation and monitored dimensions in the Crossing and Closure Panels.

On-track accelerometers

Several on-track accelerometers were installed on various parts of the turnout, such as one on the outer side of the stock rail head near the switch, two on the switch rail foot and one near the crossing nose (Fig. 3.3). Further sensors were mounted on the supports of the sensors for verification of correct functioning. These sensors were also sampled at a 1000 Hz frequency. Most of these sensors were excluded from the final scope of the study.

3.2.2 On-board sensors

On-board sensors were mounted on the EM130 measuring train to collect data during motion. These sensors complement the on-track instrumentation by providing continuous measurements along the entire length of the turnout.

EM130 Switch Measurement System (SMS)

The EM130 is a measuring train (Fig. 3.8) especially designed to inspect and analyze railway tracks. The vehicle is equipped with an on-board system, known as the Switch Measurement System (SMS), which employs laser sensors to take continuous measurements of track geometry parameters like alignment, elevation and gauge.

In contrast to on-track sensors that capture data on fixed positions, the SMS collects data dynamically every 2 cm along the turnout as the train moves. As for the on-track sensors, measurements are taken 14 mm below the top of the rail head.

On-board video capture

Two cameras (GoPro) were installed facing the wheels located towards the rear of the EM130 measuring train, one on each side (Fig. 3.3 & Fig. 3.9). The cameras have been positioned to capture at best the contacts between the wheels and the rail, providing video data of their interactions. Although intrinsically qualitative, the data collected provides information on the wheel/rail lateral displacement, the wheel/rail flanges contact and the wheels' passage through critical zones of the S&C, i.e. the switch blades, the frog and the check rail.

On-board accelerometers

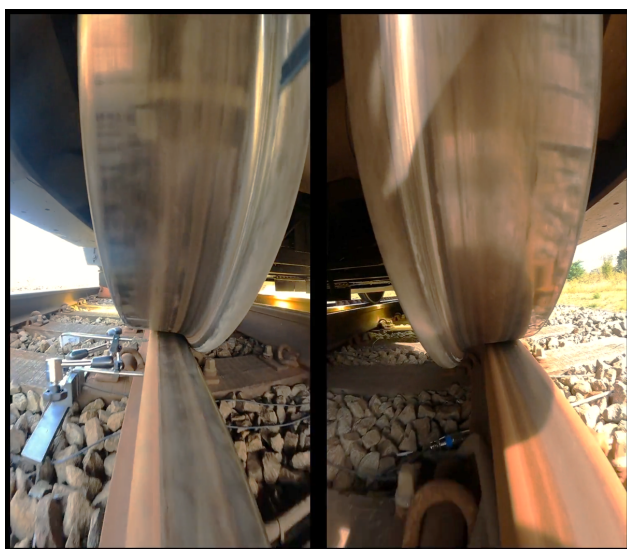
Recovib® [82] accelerometers were installed inside the train to measure acceleration along the three axes: vertical, lateral, and longitudinal. These sensors, with a measurement range of $\pm 2/6$ g and sampled at 1000 Hz, were primarily used to calibrate the numerical model. Additional accelerometers, intrinsically integrated into the train systems, were also available.



Fig. 3.8 EM130 measuring train during the experimental campaign.



(a) GoPro camera with its mounting system



(b) Frame of the video recorded by the cameras

Fig. 3.9 GoPro camera setup and an example of record.

3.3 Data acquisition procedure

The experimental setup was carried out over two days on a weekend, when the line was out of service for regular trains to enable us and the EM130 train to take measurements in complete safety. On the first day the measuring train ran on the diverging route and on the second day, it ran on the through route (Fig. 3.10).

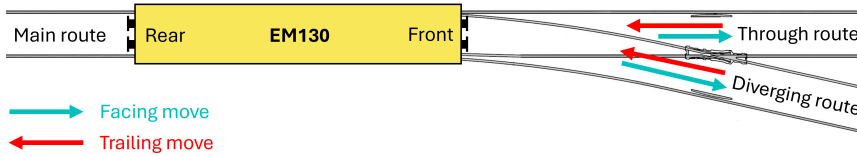


Fig. 3.10 Layout illustrating the train orientation (rear to the left, front to the right), the two types of moves (facing and trailing) and the different routes (main, through and diverging).

A range of passage velocities was defined on the basis of the traffic limit for the two routes: from 0 to 40 km/h for the diverging route (Table 3.1) and from 0 to 90 km/h for the through route (Table 3.2). Instead of a sequential protocol with increasing velocities (e.g., 5 km/h, 10 km/h, 15 km/h, 20 km/h, etc. consecutively), the EM130 passages were performed with mixed velocities (e.g., 5 km/h, 40 km/h, 30 km/h, 25 km/h...). This sequence helps to avoid cumulative effects related to the order of execution, such as localised degradation, track settlement and progressive heating. Furthermore, by breaking the correlation between speed and passage order, it becomes easier to detect the effect of velocity itself, and the statistical robustness of the analysis improves as each passage is seen as an independent event.

Nevertheless, in order to accurately determine the impact of the direction of passage (facing vs. trailing move), the two directions were run at the same speed and close temporal succession. This ensured that track conditions were similar for both moves, reducing the risk of degrading the comparison. Conducting the two tests widely separated in time might have allowed confounding effects from track evolution throughout the test campaign.

For reasons related to the local railway network, the EM130 always maintained the same orientation. The 'rear' part of the train, where the measurement system (SMS) was located, faced the main route. The 'front' part of the train faced either the diverging route or the through route,

3 | Dynamic experimental campaign

depending on the type of experiment being performed.

Passage order	Velocity [km/h]	Move direction
1	5	Trailing
2	5	Facing
3	40	Trailing
4	40	Facing
5	30	Trailing
6	30	Facing
7	25	Trailing
8	25	Facing
9	15	Trailing
10	15	Facing
11	35	Trailing
12	35	Facing
13	20	Trailing
14	20	Facing
15	40	Trailing
16	40	Facing
17	25	Trailing
18	25	Facing

Table 3.1 Diverging route test: passage sequence (velocity + move direction).

The repeatability of the measurements was unfortunately limited due to time constraints. As each measurement was performed only once or twice, this limitation must be considered when analysing the results further (Part IV).

Passage order	Velocity [km/h]	Move direction
1	30	Trailing
2	30	Facing
3	90	Trailing
4	90	Facing
5	50	Trailing
6	50	Facing
7	70	Trailing
8	70	Facing
9	10	Trailing
10	10	Facing
11	90	Trailing
12	90	Facing
13	80	Trailing
14	80	Facing

Table 3.2 Through route tests.

4

Static experimental campaign on track stiffness

THIS chapter describes the methodology followed during the second experimental campaign, conducted one year after the first one¹. It outlines the objectives, the instrumentation used on the track and the procedure carried out. The analysis of the results will be presented in Part IV.

4.1 Objectives and experimental setup

The present campaign was complementary to the first one, but no measuring train was used in this case. The main objectives were to estimate the order of magnitude of the track's lateral stiffness at specific points of the S&C, using a hydraulic jack that spread the rails apart, in order to feed the numerical model. Moreover, the campaign aimed to observe how stiffness evolves as the applied lateral force increases.

Tests were conducted on the same turnout as in the 1st campaign in Schaerbeek, Brussels, and on each zone of the turnout: Switch, Closure and Crossing Panels.

¹Carried out on October 19th, 2024, in Schaerbeek, Brussels.

4.2 Instrumentation of the track

In order to apply a lateral force to spread the rails, and to accurately measure this force, a hydraulic jack was used in combination with a load cell. Various pieces were designed to couple the jack and the load cell with the track. These parts were designed in SolidWorks (Fig. 4.1 and Table 4.1) and manufactured at UCLouvain, in the CREDEM workshop.

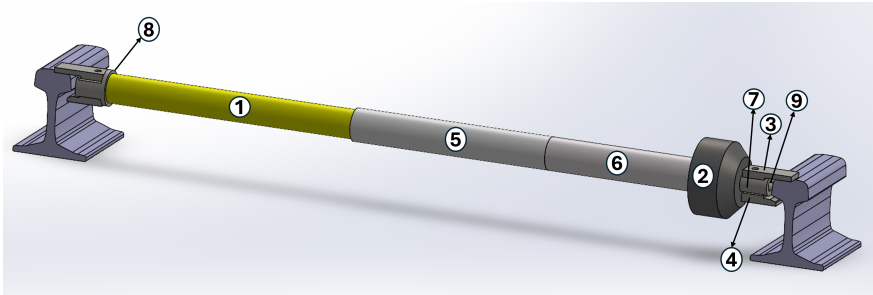


Fig. 4.1 3D view of the load application assembly. The numbered components correspond to those listed in Table 4.1.

N°	Component	Custom-made	Material
1	Hydraulic jack	No	–
2	Load cell	No	–
3	Upper bearing plate	Yes	Steel (42CrMo4)
4	Lower bearing plate	Yes	Steel (42CrMo4)
5	Long rod	Yes	Aluminium
6	Short rod	Yes	Aluminium
7	Ball support attached to the load cell	Yes	Steel (42CrMo4)
8	Ball support attached to the jack	Yes	Steel (42CrMo4)
9	Steel ball	No	Steel

Table 4.1 Components of the load application system.

Two steel balls were inserted at the end of assembly for providing a point contact to the rails (Fig. 4.2), in order to ensure precise alignment along the axis and enabling accurate local stiffness measurements. The middle cylindrical extensions between the jack and the load cell, which were mainly axially loaded, were manufactured from aluminium in order to make the

overall weight lighter and easier to handle. By contrast, the interface pieces between the jack or the load cell and the rail were manufactured from steel (42CrMo4) to provide the necessary mechanical strength. The employed hydraulic jack was an RC-1010 model with a 101 kN maximum capacity and a stroke from 349 mm (collapsed) to 606 mm (extended). The load cell was a pancake-type sensor, having a capacity of 75-100 kN and being suitable for tension and compression.

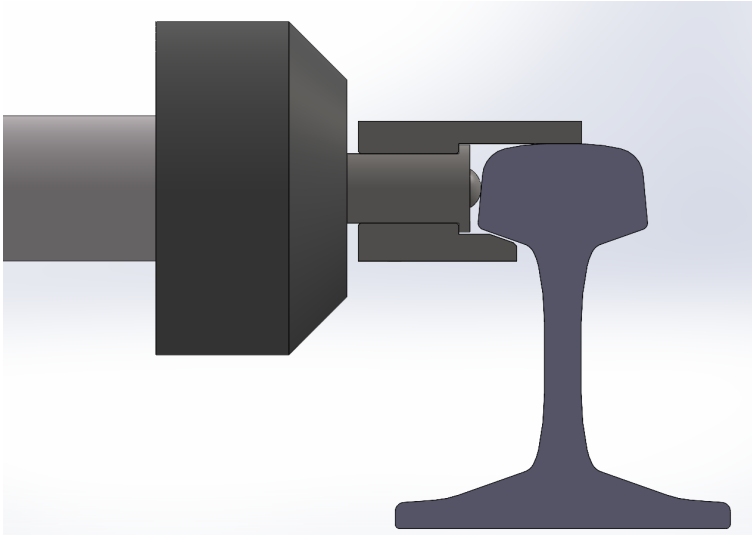


Fig. 4.2 Zoom view of the contact between the steel ball and the rail.

The internal stresses were analyzed through a finite element method (FEM) analysis with SolidWorks, with a special focus on the upper and lower bearing plate that vertically constrain the system. These parts have angular geometries that could be more susceptible to stress concentration and failure. The analysis also aimed to verify the risk of buckling and ensure that all stress levels were lower than the materials' yield strength.

To longitudinally secure the assembly for testing, two extra U-shaped pieces with threaded rods were mounted on the head of the rails (Fig. 4.3). They served as lateral stops for the system, providing stability and preventing slip movement during the application of force.

To measure the lateral displacement of the rails where the hydraulic jack applied the forces, LVDT sensors were installed in the three zones: the Crossing Panel, after the check rail on the diverging route; the Switch Panel,



Fig. 4.3 Experimental setup showing the hydraulic jack coupled to the rail, with two U-shaped pieces securing the assembly longitudinally and the LVDT sensor positioned to measure rail displacement.

2 meters before the switch blades; and the Closure Panel, in the same place where sensors were installed in the 1st experimental campaign (see Fig. 4.5).

The experimental setup generally consisted in measuring the lateral displacement at two positions on each rail where the hydraulic jack applied force. One LVDT sensor was placed 14 mm below the top surface of the rail head and another was positioned at the center of the rail web, 87.1 mm below the top surface, as in the 1st campaign.

Apart from the basic configurations mentioned above, the tests also included some particular configurations. In the Crossing Panel test, all the sensors were placed on the same side of the same rail. Two LVDT sensors — one at the rail head and one at the web — were installed at the location where the force was applied to one of the rails. Another pair was positioned 75 cm further along the diverging route, on the same side (Test 1.2). The

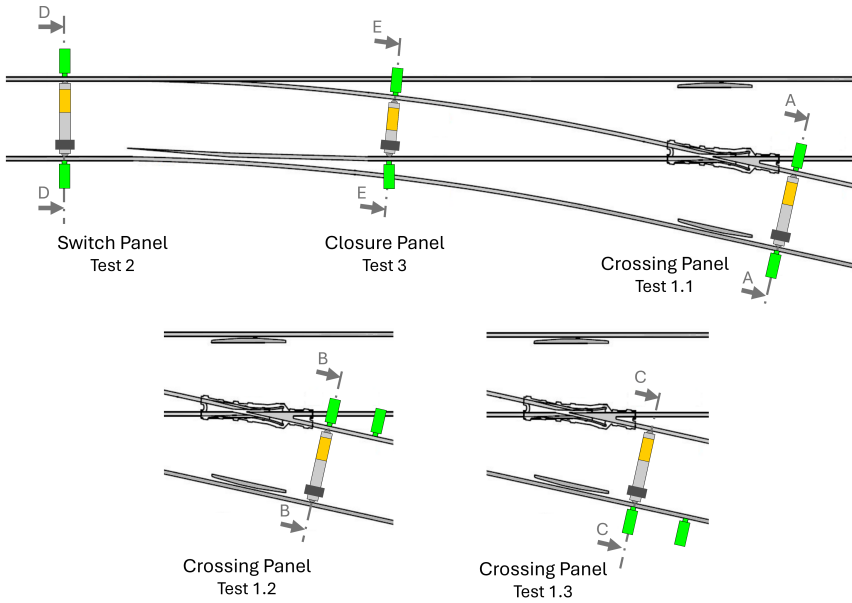


Fig. 4.4 Top view of the different test configurations, showing the layout of the sensors and of the assembly of the hydraulic jack in each panel.

same experiment was conducted on the other rail (Test 1.3). The objective was to evaluate the influence of a lateral force applied 75 cm away from the measurement point.

In the Switch Panel test, one sensor was placed on the side of the rail head, while the other, instead of being positioned near the web, was installed vertically on the foot to measure its vertical displacement. This configuration was designed to analyse rail flexion through the rotation of the rail cross-section under lateral loading.

Finally, in the Closure Panel, the test configuration (Test 3) was not perpendicular to the rails due to the complex, non-orthogonal geometry of the measured track in that area. To approximate the perpendicular direction, the isosceles triangle rule was applied.

4 | Static experimental campaign on track stiffness

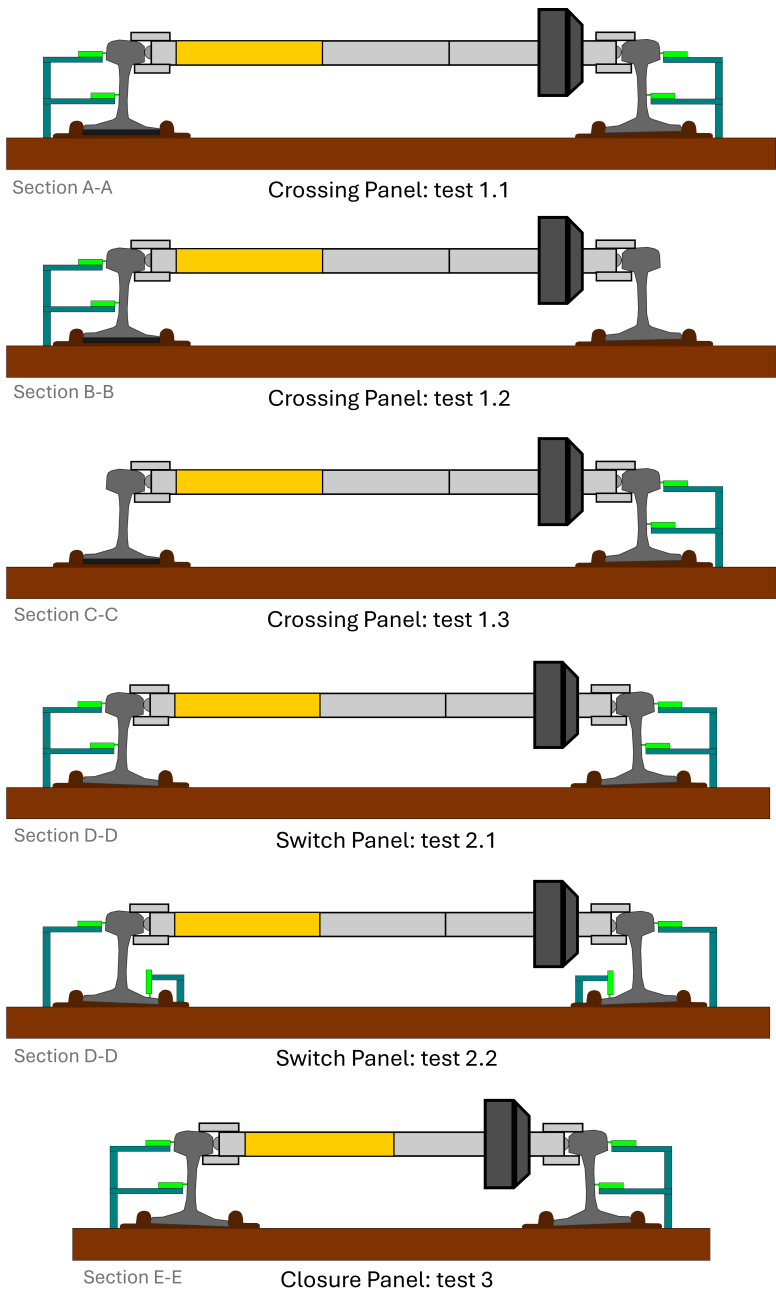


Fig. 4.5 Cross-sectional view of the test configurations, indicating the relative position of the LVDT sensors on the rail head, web, and foot.

4.3 Data acquisition procedure

Prior to the field tests, an initial validation of the hydraulic jack assembly was carried out at the LEMSC laboratory of UCLouvain. This pre-test, performed without sensors, aimed to check the integrity and correct installation of all components: the jack, the mechanical components and the load cell.

The field measurements were conducted on the same S&C of the 1st campaign that was out of service that day, ensuring both safety and the absence of dynamic effects caused by passing trains.

In every test, the lateral force was incremented by means of the hydraulic jack. While data was being recorded continuously, the force was incremented step by step to enable the recording of stable values in quasi-static conditions. At each step, both the applied force and the corresponding rail displacements were recorded (see Fig. 4.6). This data is later used to plot Force vs. Displacement curves for each configuration.

To ensure the safety of the setup and avoid exceeding the elastic limit of the rail, the maximum lateral displacement allowed per rail was limited to 2.1 mm — close to the maximum value observed during the first experimental campaign. Additionally, to maintain a safety margin for the mechanical assembly, the applied force never exceeded 45 kN.

A summary of the order of the tests carried out and their number of repetitions is presented in Table 4.2.

Table 4.2 Summary of test configurations and repetitions.

Order	Test Type	Repetitions
1	Test 1.1	3
2	Test 1.2	1
3	Test 1.3	1
4	Test 2.1	3
5	Test 2.2	1
6	Test 3	3

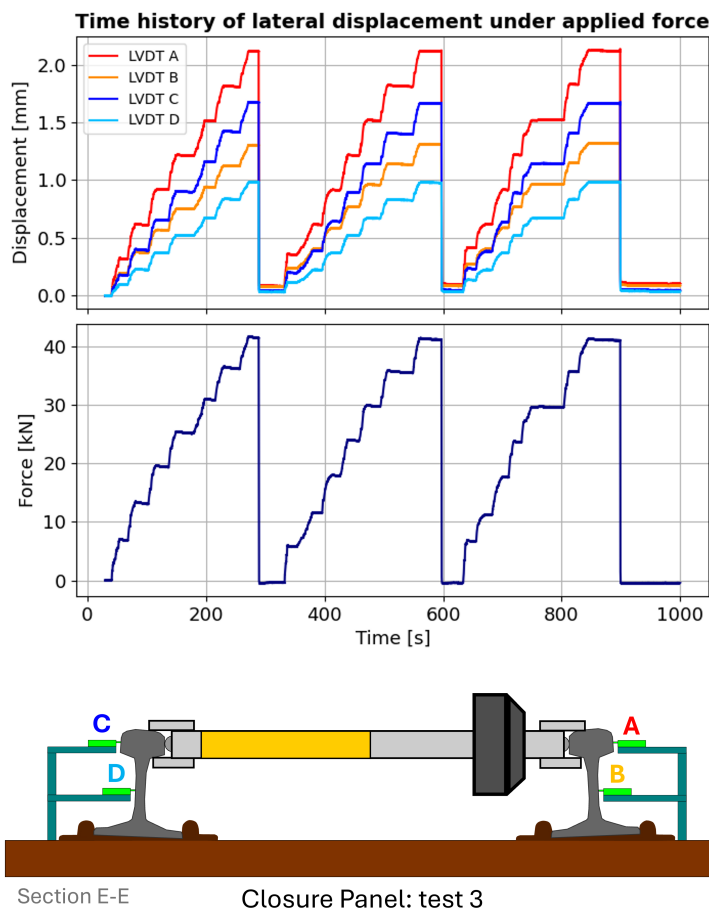


Fig. 4.6 Example of the measurements obtained during Test 3. The top graph shows the evolution of the lateral displacement of the rail measured by the LVDT sensors and the bottom graph displays the force applied by the hydraulic jack. The shape of the graphs reveals the force steps applied to obtain stable displacement values.

PART III

Multibody model

5

Train–turnout multibody model

TO carry out the dynamic study underlying this work, we require a three-dimensional mechanical model of the measurement train EM130 and its interaction with the turnout. Manually writing such a model would be unrealistic: multibody modeling techniques are perfectly suited to this type of vehicle.

5.1 Multibody Modeling Framework

5.1.1 Multibody Approach

The multibody formalism, developed since the 1970s, aims to model and analyze any poly-articulated mechanical system (robot, vehicle, transmission, human body, etc.). A railway vehicle is a typical example of a multibody system, composed of a car body, two bogies, and four wheelsets, articulated through a series of suspension elements.

The goal of the multibody formalism is to derive the equations of motion based on fundamental physical laws, notably the Newton–Euler equations.

5.1.2 Formalism Used

The approach underlying the multibody software ROBOTRAN [83], developed at UCLouvain and used in this work, relies on the use of relative generalized coordinates q , representing displacements (in meters) or angles (in radians) within the system's joints.

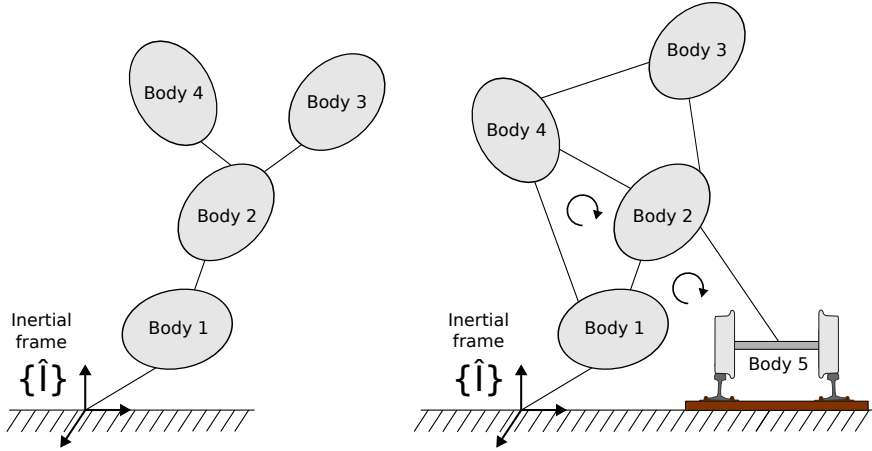


Fig. 5.1 Left: tree-like multibody system. Right: constrained multibody system.

Among multibody systems, we distinguish between tree-like MBS and constrained MBS (see Fig. 5.1). In the former, the relative coordinates are kinematically independent. In the latter, kinematic constraints link these variables, for example when bodies form a loop or, in the railway case, when wheelsets are constrained to follow the track.

To obtain the final equations of motion, several formalisms are available: virtual power principle, Lagrange equations, or recursive methods based on Newton–Euler equations. Regardless of the formalism, the final form can be written as:

$$\mathbf{M}(q)\ddot{q} + \mathbf{c}(q, \dot{q}) = \mathbf{Q}(q, \dot{q}) + \mathbf{J}^T(q)\lambda \quad (5.1)$$

$$\mathbf{h}(q) = 0 \quad (5.2)$$

$$\dot{\mathbf{h}}(q, \dot{q}) = \mathbf{J}(q)\dot{q} = 0 \quad (5.3)$$

$$\ddot{\mathbf{h}}(q, \dot{q}, \ddot{q}) = \mathbf{J}(q)\ddot{q} + \dot{\mathbf{J}}(q, \dot{q})\dot{q} = 0 \quad (5.4)$$

where q , $\dot{q}$, $\ddot{q}$ are the relative generalized coordinates, velocities, and accelerations of the system; $\mathbf{M}(q)$ is the mass matrix; $\mathbf{c}(q, \dot{q})$ represents the Coriolis, centrifugal and gravitational effects; $\mathbf{Q}(q, \dot{q})$ is the vector of generalized forces, $\mathbf{J}(q)$ is the Jacobian matrix of the constraints, and λ are the Lagrange multipliers associated with the constraints.

A key feature of the software used in this thesis is its ability to generate these equations symbolically, i.e., from alphanumeric characters, as one would do manually. The matrices $\mathbf{M}$, $\mathbf{c}$, $\mathbf{J}$, as well as the constraints $\mathbf{h}$, $\dot{\mathbf{h}}$, $\ddot{\mathbf{h}}$ are generated as functions in a given language (C, Python, Matlab...), easily interfaced with numerical analysis modules (integration, optimization, control...).

Since the system of equations (5.1)–(5.4) is a DAE (differential-algebraic equations), a suitable method to make it compatible with ODE numerical integrators is the coordinate partitioning method [84], adapted to the formalism used here [85, 86].

This method partitions the $n + m$ generalized coordinates q into:

$q = \begin{pmatrix} u \\ v \end{pmatrix}$ where u are the n independent variables and v the m dependent variables, based on the m constraints $\mathbf{h}(q) = 0$.

An algebraic process reduces the DAE system to an ODE system, after eliminating the Lagrange multipliers λ and the dependent variables v , $\dot{v}$, $\ddot{v}$, leading to:

$$\mathbf{M}_r(u)\ddot{u} + \mathbf{c}_r(u, \dot{u}) = \mathbf{Q}_r(u, \dot{u}) \quad (5.5)$$

These equations are true ODEs, whose variables are the n independent generalized coordinates u . This system (5.5) can be integrated using standard numerical integration algorithms, such as those available in most scientific computing libraries.

5.1.3 Special Modeling Features

The modeling of a vehicle such as the EM130 relies on several specific multibody features, illustrated in Fig. 5.2:

- **Virtual sensor** (left): this feature allows the computation of the absolute kinematics (position, velocity, acceleration) of any point on any body in the system, based on relative generalized coordinates. For example, it can be used to compute the absolute velocity of the wheel–rail contact point to estimate slip.
- **1D link** (center): this multibody functionality introduces a unidirectional force between two points P and Q located on two distinct bodies, according to the action–reaction principle. The force can represent a spring, damper, friction, etc. For instance, 1D links can model suspension elements such as hydraulic dampers.
- **3D link** (right): a generalization of the 1D link, this feature allows the modeling of three force components and pure torque acting between two bodies (point P , frame $\hat{P}$; point Q , frame $\hat{Q}$) through elements such as shear springs, silent blocks, or bushings.

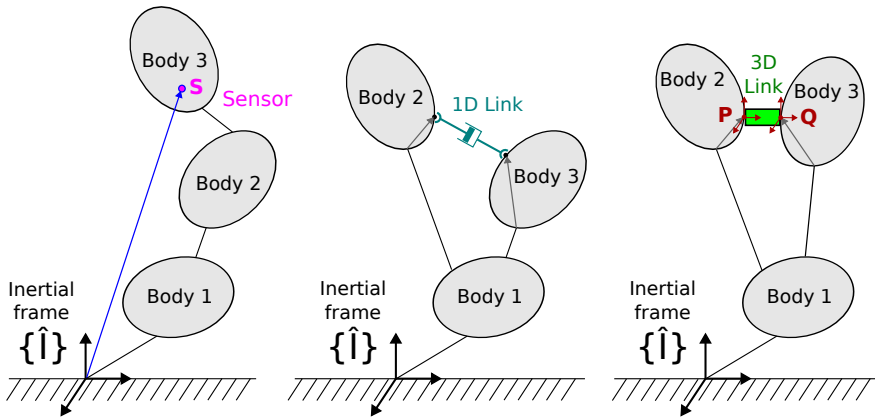


Fig. 5.2 Left: virtual sensor. Center: 1D link. Right: 3D link.

5.1.4 Algorithmic Implementation

The sequential implementation of the process is shown in the flowchart in Fig. 5.3, where we distinguish:

- **Model part:** constraint resolution, computation of applied forces and torques, generation of dynamic equations, reduction via coordinate partitioning, and final computation of independent accelerations $\ddot{u}$.
- **Numerical analysis part:** time integration.¹

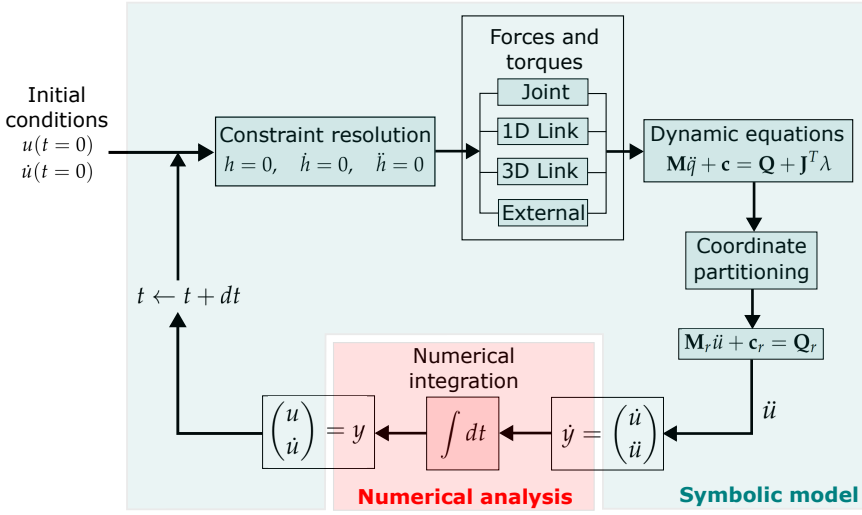


Fig. 5.3 Flowchart of the multibody formalism implementation.

Now that the multibody formalism and its components relevant to this thesis have been explained, we will successively detail: the wheel–rail contact model, the multibody model of the train and the track model.

¹Other analyses such as static equilibrium, modal analysis, or inverse dynamics can also be performed based on the symbolic model.

5.2 Wheel-rail contact geometry

5.2.1 Introduction

The modeling of a railway vehicle based on physical laws requires special attention to accurately capture the interaction forces between wheels and rails.

In particular, the contact model and its level of complexity should be carefully chosen based on the scope of the study and the questions it raises.

For example, when studying the global motion of a train on a perfect track, simple geometries (wheel and rail) and linear contact force models may suffice. However, this is not the case when lateral stability issues or wheel wear are central to the study; in such cases, more advanced geometric and dynamic models are preferred.

In our case, the research questions and field observations (experimental) suggest the use of an intermediate complexity model in order to:

- avoid prohibitive computation times,
- limit the number of input parameters in the model.

Compared to a standard track, Switches and Crossings (S&C) have characteristics that inevitably complicate the contact model. Among others:

- the specific geometry of switch blades and their extremities,
- the presence of check rails and their contact with the wheel flange,
- the presence of the gap in the crossing nose,
- etc.

As previously mentioned, this work will focus more on the dynamics in the diverging track, considering it as a classical curved trajectory (with entry /exit segments for the transition to the straight track), to which two key elements related to the crossing nose will be added: the check rail and the gap.

To introduce the developed and implemented model, we will start from the one elaborated at UCLouvain [85], whose main steps are presented in the next section (5.3).

The generalization of the wheel-rail contact model to the third contact (flange-check rail) will be discussed in the following section (5.2.3).

The specific treatment of the gap will be addressed in the final section (5.4).

- the orientation, i.e., the rotation matrix $R^Y(q)$ such that $[\hat{Y}] = R^Y(q)[\hat{I}]$, the angular velocity $\vec{\omega}$, and the angular acceleration $\vec{\dot{\omega}}$ of the wheel are also available at all times, as functions of $q, \dot{q}, \ddot{q}$.

Kinematic model (contact between the tread and the rail)

One of the first assumptions of the geometric model is to consider the wheel and the rail as rigid bodies between which contact occurs at a single point Q , whose displacement along the wheel tread and the rail depends on the profiles of both surfaces and the positioning of the wheel on the rail.

This assumption naturally leads to the establishment of kinematic constraints, requiring that at every instant the wheel be tangent to and resting on the rail.

Without going into the details of the model (see Samin & Fisette (2003) [85] for full developments), let us first list the relevant parameters and data.

- Profile parameters:

The parameters characterizing the geometry of the profiles are:

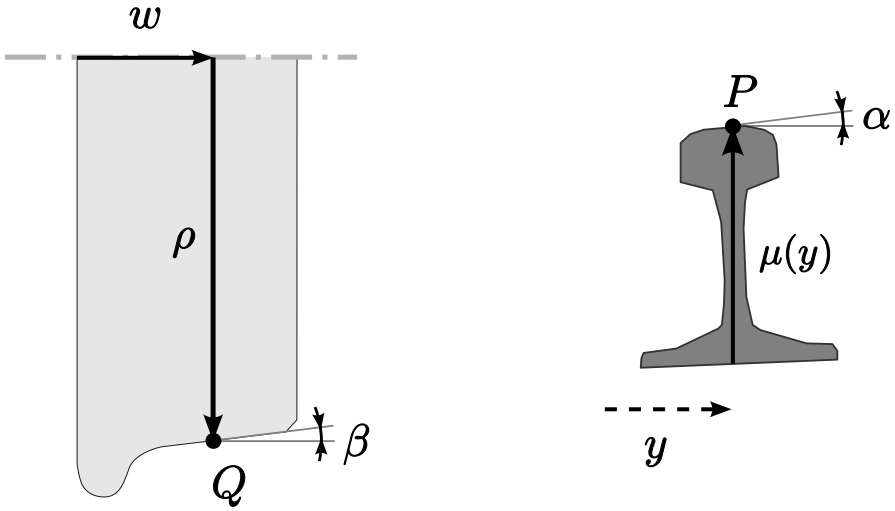


Fig. 5.5 Profile geometry parameters at contact point Q .

- $\rho(w)$: rolling radius at point Q [m].
- β : wheel profile slope at point Q [rad].

- $\mu(y)$: rail profile at point $P = Q$ [m].
 - α : contact angle at point $P = Q$ [rad].
- Vectors (see Fig. 5.4):
- $\vec{x}$: absolute position vector of the wheel's center of mass, G .
 - $\vec{u}$: absolute position vector of the contact point Q on the wheel tread.
 - $\vec{w}$: such that $\vec{x} = \vec{w} + \vec{u}$, vector positioning the contact point Q relative to G .
- Reference frames:
- $[\hat{I}]$: inertial frame. On a straight track, $\hat{I}_1$ is along the track axis, $\hat{I}_3$ is vertically upward, and $\hat{I}_2$ points to the left.
 - $[\hat{Y}]$: multibody frame attached to the wheel, with $\hat{Y}_2$ along its rotation axis. We have $\hat{Y} = R^Y(q)(\hat{I})$.
 - $[\hat{X}]$: obtained from $[\hat{Y}]$ by canceling the wheel's rotation around its axis. It is associated with a so-called geometric wheel that "would not rotate". It facilitates the calculation of the contact point Q . We have $\hat{X} = R^X(q)(\hat{I})$.

The displacement of the contact point Q can be characterized by the lateral motion w of a plane perpendicular to $\hat{X}_2$ moving along $\hat{X}_2 = \hat{Y}_2$, but also by a circumferential displacement, θ , (see Fig. 5.6) along the wheel's perimeter.

The pitch angle θ must be introduced as an additional variable, and allows defining the following frame:

- $[\hat{S}] = R_2^\theta(\hat{X})$; such that point Q is always in the plane $\{\hat{S}_2, \hat{S}_3\}$.
- $[\hat{T}]$: final frame, tangent to the wheel profile (angle β), such that $[\hat{T}] = R_1^\beta[\hat{S}]$ (see Fig. 5.4).
- $[\hat{R}]$: equivalent of $[\hat{T}]$ on the rail side, such that $\hat{R} = R_1^\alpha[\hat{I}]$ for a straight rail.

The previous definitions allow us to write the vectors $\vec{x}$, $\vec{u}$ and $\vec{w}$ as follows:

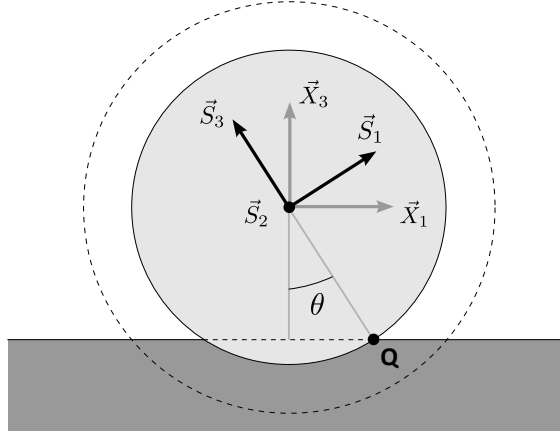


Fig. 5.6 Pitch angle θ , representing the circumferential displacement of the contact point Q along the wheel perimeter.

$$\vec{x}(q) = [\hat{I}]^T x(q)$$

$\vec{x}$ depends only on the multibody kinematics upstream of the wheel.

$$\vec{w}(q, w, \theta) = [\hat{S}]^T \begin{pmatrix} 0 \\ w \\ \rho(w) \end{pmatrix}$$

where ρ denotes the instantaneous contact radius.

$$\vec{u}(q, w, \theta) = [\hat{I}]^T \left(x + R^{xT} R^{\theta T} \begin{pmatrix} 0 \\ w \\ \rho(w) \end{pmatrix} \right) = [\hat{I}]^T \begin{pmatrix} u_1 \\ u_2 \\ u_3 \end{pmatrix}$$

which positions point Q in the inertial frame at any time.

Three kinematic constraints are necessary to enforce the tangential contact between the wheel and the rail. They are written as follows:

$$h_1(q, w, \theta) = u_3 - \mu(u_2) = 0 \quad (5.6)$$

$$h_2(q, w, \theta) = \hat{T}_3 \cdot \hat{R}_1 = 0 \quad (5.7)$$

$$h_3(q, w, \theta) = \hat{T}_3 \cdot \hat{R}_2 = 0 \quad (5.8)$$

From an interpretative point of view, the first constraint, h_1 , ensures that the wheel shares a common point with the rail. The other two (h_2 and h_3) ensure that both profiles are perfectly tangent at the contact point, leaving only the spin around $\hat{T}_3 = \hat{R}_3$.

Once these three constraints are solved using an iterative and/or dichotomic numerical method [85], the exact position of point Q on the wheel is determined (precision $\approx 10^{-9}$ m).

To rigorously account for these three constraints in the final multibody model, they must be differentiated with respect to time.

If we define:

$$h(q, w, \theta) = \begin{pmatrix} h_1(q, w, \theta) \\ h_2(q, w, \theta) \\ h_3(q, w, \theta) \end{pmatrix}$$

we can write:

$$\dot{h}(q, w, \theta) = J(q, w, \theta) \dot{q} = 0 \quad (5.9)$$

$$\ddot{h}(q, w, \theta, \dot{q}, \dot{w}, \dot{\theta}, \ddot{q}, \ddot{w}, \ddot{\theta}) = J(q, w, \theta) \ddot{q} + \dot{J}(q, w, \theta, \dot{q}, \dot{w}, \dot{\theta}) \dot{q} = 0 \quad (5.10)$$

These equations can be solved directly using linear algebra techniques, given their linearity with respect to $\dot{q}$ and $\ddot{q}$, respectively.

Without going into detail, it is shown in [85] that the first constraint (5.6), once differentiated (5.9), can be projected along the normal to the contact (see Fig 5.7):

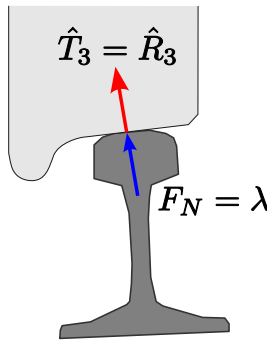


Fig. 5.7 Normal contact force between the wheel tread and the rail, representing the first constraint of non-interpenetration between rigid bodies.

5 | Train–turnout multibody model

and written as follows:

$$h'_1 = \dot{\vec{u}} \cdot \vec{R}_3 = 0 \quad (5.11)$$

whose physical interpretation is clear: at every instant, the absolute velocity of the contact point Q is zero in the direction normal to the rail; in other words, it expresses the non-interpenetration of two rigid bodies.

Moreover, the Lagrange multiplier λ associated with this constraint (written as in (5.11)) corresponds exactly to the normal contact force (5.12) from the rail on the wheel, regardless of the static or dynamic loads applied to the system.

$$\lambda = F_N \quad [\text{N}] \quad (5.12)$$

Furthermore, a negative value of λ indicates a loss of contact and an exit from the model's validity domain.

Dynamic Model

Among all contact force models, those of Prof. Kalker [87] remain to this day a reference in the railway domain. From the simplest (linear) to the most complex (nonlinear, conformal, non-Hertzian contact, etc.), all variants proposed by him or developed subsequently [88–92] are based on two fundamental ingredients whose nature is:

- related to materials and Hertzian theory (elliptical contact):
 - contact curvatures
 - Young's modulus E
 - shear modulus G
 - Poisson's ratio ν
 - normal force F_N
- related to contact kinematics (creepage):
 - along $\hat{T}_1$
 - along $\hat{T}_2$
 - along $\hat{T}_3$

In the context of our problem, it is useful to establish the expression of creepages.

Roughly speaking, it is “simply” a matter of computing an average slip velocity within the contact ellipse (cf. Hertz), a task that Kalker greatly simplifies by defining *creepages* based on the instantaneous velocity of the material point Q , located at the center of the contact ellipse, and the angular velocity of the wheel in the direction normal to that point.

Their computation is given as follows (see Fig. 5.4):

Longitudinal creepage:

$$\nu_x = \frac{\vec{u} \cdot \hat{T}_1}{|v_x^G|} \quad [\%] \quad (5.13)$$

Lateral creepage:

$$\nu_y = \frac{\vec{u} \cdot \hat{T}_2}{|v_x^G|} \quad [\%] \quad (5.14)$$

Spin creepage:

$$\varphi_z = \frac{\vec{\omega}^{[Y]} \cdot \hat{T}_3}{|v_x^G|} \quad [\text{m}^{-1}] \quad (5.15)$$

In the denominator, v_x^G is the longitudinal velocity of the wheel’s center of mass, which ensures that the shear deformation physics is respected independently of speed.

In his linear model — the simplest, yet already capable of characterizing the stability (and its limits) of a train as a function of its speed — Kalker proposes the following three relations:

$$\begin{pmatrix} F_x \\ F_y \\ M_z \end{pmatrix} = - \begin{pmatrix} k_{11} & 0 & 0 \\ 0 & k_{22} & k_{23} \\ 0 & -k_{23} & k_{33} \end{pmatrix} \begin{pmatrix} \nu_x \\ \nu_y \\ \varphi_z \end{pmatrix} \quad (5.16)$$

where F_x , F_y , and M_z represent respectively the longitudinal force (along $\hat{T}_1$), the lateral force (along $\hat{T}_2$), and the spin torque (along $\hat{T}_3$) exerted by the rail on the wheel at point Q .

In this linear model by Kalker, the normal force is automatically taken into account via the Lagrange multiplier λ (5.12), which is involved in the computation of the Kalker coefficients k_{ij} , which also depend on the previously mentioned material parameters.

The linearity of F_x , F_y , and M_z with respect to the creepages ν_x , ν_y , and φ_z is appropriate for rolling conditions where slip is small and the forces

are not saturated: for instance, a train running on a straight track or a wide curve, at constant speed, with contact occurring near the center of the running surface.

Although this is the most common situation in practice, cases involving tight curves, switches & crossings, or worn rails and wheels lead to saturation of the contact forces and/or larger spin torques M_z .

Without going into the details of the literature, several nonlinear models have been developed as extensions to linear models such as Kalker’s (5.16), with varying levels of complexity.

In the context of our research, we will show that wheel–rail interactions in a turnout are far from standard, with intermittent contact on the flange and other nonlinearities due to impacts and component geometry.

As a result, we made the deliberate choice:

- to distinguish between contact on the running surface (model in Section 5.2.2) and contact on the flange (models in Section 5.2.3),
- to use a nonlinear model for the forces F_x and F_y , with saturation based on a hyperbolic tangent law or a cubic law [93], as in the model by Johnson & Vermeulen (1964) [94], summarized below.

The saturation model² is based on the linear Kalker model:

$$F_x = -k_{11}v_x \quad F_y = -k_{22}v_y - k_{23}\varphi_z$$

Since the magnitude of this force cannot physically exceed the friction limit, we impose:

$$F_v = \sqrt{F_x^2 + F_y^2} \leq \mu F_N$$

where μ is the static dry friction coefficient.

We define:

$$k \triangleq \frac{F_v}{\mu F_N}$$

This allows to saturate the force F_v as follows:

²In this nonlinear model, the spin torque M_z is not saturated, which is justified for low wheel conicities (typically between 1/20 and 1/40).

$$F_v^{\text{sat}} = \begin{cases} \mu N \left(k - \frac{1}{3}k^2 + \frac{1}{27}k^3 \right) & \text{if } k \leq 3 \\ \mu N & \text{if } k > 3 \end{cases} \quad (5.17)$$

This law is illustrated in the Fig. 5.8 on the left (cubic model).

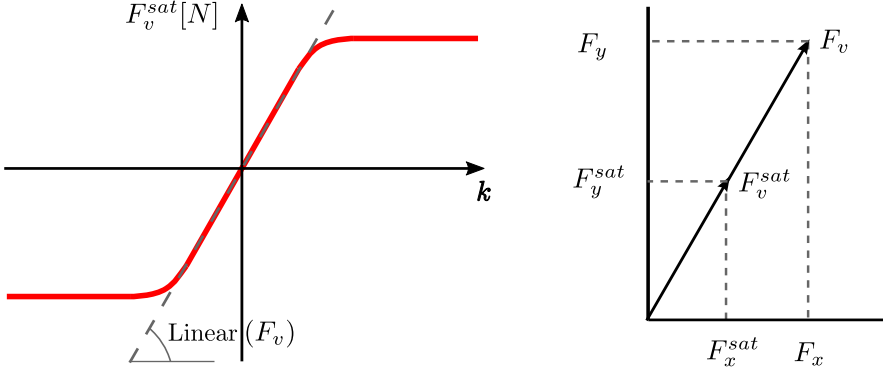


Fig. 5.8 Cubic saturation model of Johnson & Vermeulen [94].

We now need to distribute this saturated force F_v^{sat} in the x and y directions. To do so, we assume that the distribution is equivalent for both the linear and nonlinear models, using the properties of similar triangles (Fig. 5.8 on the right), and write:

$$\frac{F_x^{\text{sat}}}{F_v^{\text{sat}}} = \frac{F_x}{F_v} \quad \frac{F_y^{\text{sat}}}{F_v^{\text{sat}}} = \frac{F_y}{F_v}$$

This allows us to compute the saturated forces as follows:

$$F_x^{\text{sat}} = F_x \cdot \frac{F_v^{\text{sat}}}{F_v} \quad (5.18)$$

$$F_y^{\text{sat}} = F_y \cdot \frac{F_v^{\text{sat}}}{F_v} \quad (5.19)$$

The spin torque M_z remains unsaturated and is computed using the linear model:

$$M_z = k_{23}v_y - k_{33}\varphi_z \quad (5.20)$$

5 | Train–turnout multibody model

These two saturated forces $F_x^{\text{sat}}, F_y^{\text{sat}}$ and the unsaturated spin moment M_z are those we will use in our railway model for the wheel–rail interaction at the main contact point (running surface).

5.2.3 Wheel flange/rail contact

Introduction

Earlier, a distinction was made between contacts on the wheel tread (constrained and assumed to be permanent) and those occurring on the wheel flange (unconstrained and intermittent). This choice, although debatable, allows for an efficient and independent treatment of the two possible flange contact scenarios in the case of switches and crossings (S&C), as illustrated in Fig. 5.9.

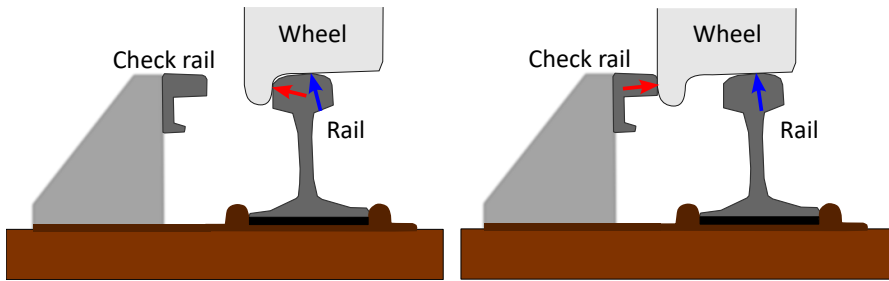


Fig. 5.9 Contact between the outer surface of the wheel flange and the rail (on the left) and contact between the inner surface of the wheel flange and the rail (on the right).

The figure on the left shows the “classic” flange contact between the wheel and the rail, typically encountered in tight curves or in cases of excessive lateral disturbances: the contact occurs on the inner side of the flange and acts as a mechanical stop (safety feature).

The figure on the right shows the case of contact between the outer side of the wheel flange and the check rail of an S&C, whose function (Chapter 3) is to guide the train in the correct direction at the crossing nose.

To avoid making the system hyperstatic in the presence of two contacts, and given the intermittent nature of flange contacts — which may occur simultaneously (up to three contacts) — these are no longer modeled by kinematic constraints between rigid bodies, but rather through a regularization principle that considers:

- a spring-damper law in the normal direction;
- a dry kinetic friction law in the tangential direction, assuming fully saturated friction.

Wheel flange/rail contact model

This model, inspired by [85], is based on the following geometric assumptions:

On the wheel side, the contact — intermittent — occurs on a ring with no thickness, whose radius r_b is greater than that of the tread/rail contact ($\Delta r_b \simeq 100$ mm), located at the actual flange and characterized by a virtual slope corresponding to the flange conicity ($\beta_b \simeq 69^\circ$) (see in red in Fig. 5.10).

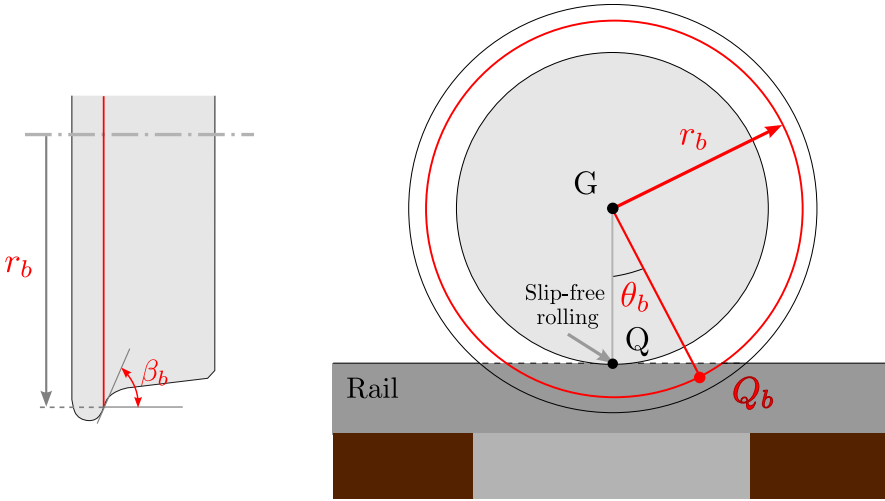


Fig. 5.10 Schematic representation of the flange contact model. The contact occurs on a virtual ring (in red) with radius r_b , located at the flange and inclined by the virtual conicity angle β_b . The circumferential position of the contact point Q_b is defined by the pitch angle θ_b .

The contact point Q_b on this ring may move circumferentially (right-hand figure, pitch angle θ_b), similarly to the contact on the wheel tread (angle θ , see Fig. 5.6).

The equation used to determine this angle θ_b is transcendental and is written as [85]:

$$R_{31}^x \cos \theta_b + R_{11}^x \sin \theta_b = R_{21}^x \tan \beta_b \quad (5.21)$$

whose solution (in terms of $\sin \theta_b$, $\cos \theta_b$) is analytical. Here, R_{i1}^x denote the elements of the first column of the rotation matrix R^x , i.e., the projections of the inertial track direction $\hat{l}_1$ onto the axes of the geometric wheel frame $[\hat{X}]$ (the non-rotating wheel frame used for contact calculations).

From a geometric perspective, this equation shows that θ_b increases with the flange angle β_b and the yaw of the axle (via R^x).

Note: however, due to the significant conicity of the flange ($\beta_b > 1$ rad), this angle will be much larger ($\theta_b \gg \theta$), potentially causing significant unloading of the wheel (or even train derailment) [95].

It is therefore essential to compute it precisely, without exceeding a fixed limit angle (assumed to be $|\theta_b^{\max}| = 0.175$ rad), due to the finite height of the rail.

In dynamic terms, the assumptions are linked to the choice of a regularized contact model.

For the normal contact force between the flange and the rail, denoted F_b^N , it is assumed that, when contact occurs, it behaves as a linear spring-damper system. This modeling is based on the lateral stiffness and damping characteristics of the rail.

For the tangential force, denoted F_b^T , it is assumed to be constantly saturated from the friction point of view:

$$|F_b^T| = \mu |F_b^N|$$

where μ is the coefficient of kinetic friction.

This assumption is based on the following considerations:

- quasi-rolling without slip at contact point Q between the wheel tread and the rail head (creepages from Kalker's model);
- the measurable difference between the two rolling radii, $\Delta r_b = r_b - r$, with r being the instantaneous radius at point Q (and r_b at Q_b), on the order of centimeters.

For a precise calculation of these two components, it is necessary to account for the lateral intermittent gap and the pitch angle θ_b calculated previously.

1) Normal force of the second contact.

The force exerted by the rail is:

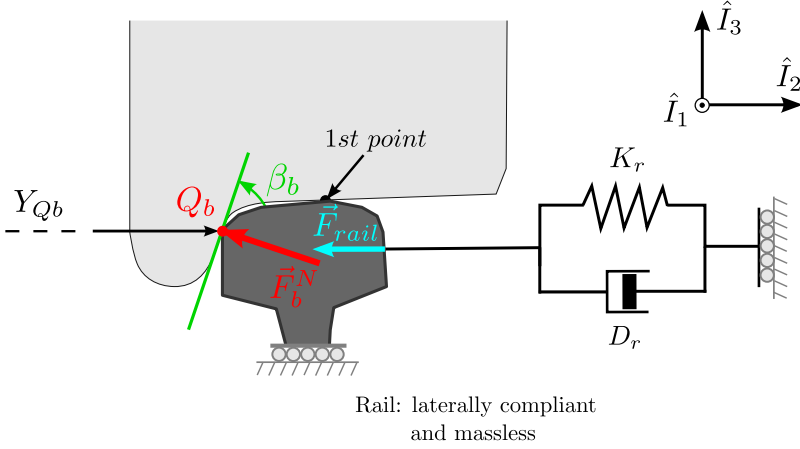


Fig. 5.11 Model of rail compliance and normal force at the second contact point.

$$\vec{F}_{rail} = \begin{cases} (-K_r(Y_{Q_b} - Y_R^0) - D_r\dot{Y}_{Q_b})\hat{I}_2 & \text{if } (Y_{Q_b} - Y_R^0) > 0 \\ 0 & \text{if } (Y_{Q_b} - Y_R^0) \leq 0 \end{cases} \quad (5.22)$$

with:

- Y_R^0 : lateral displacement of point Q_b when the wheel is flush with the rail;
- Y_{Q_b} : lateral displacement of point Q_b ;
- K_r : rail lateral stiffness (including real pivoting in x , ballast stiffness...);
- D_y : lateral damping (from ballast, fasteners...).

The tangential force projected onto the inertial frame is given by:

$$\vec{F}_b^N = F_{rail}\hat{I}_2 - \frac{F_{rail}}{\tan\beta_b}\hat{I}_3 \quad (5.23)$$

$$\vec{F}_b^N = -(K_r(Y_{Q_b} - Y_R^0) + D_r\dot{Y}_{Q_b})\hat{I}_2 + \frac{(K_r(Y_{Q_b} - Y_R^0) + D_r\dot{Y}_{Q_b})}{\tan\beta_b}\hat{I}_3 \quad (5.24)$$

2) Tangential force of the second contact.

According to Coulomb's law:

$$\vec{F}_b^T = -\mu \|F_b^N\| \hat{e} \quad (5.25)$$

and $\hat{e}$:

$$\hat{e} = \frac{\vec{v}_{Q_b}^t}{\|\vec{v}_{Q_b}^t\|} \quad (5.26)$$

- $\hat{e}$ is the unit vector indicating the direction of the force $\vec{F}_b^T$
- μ is the coefficient of friction between flange and rail
- $\vec{v}_{Q_b}^t$ the instantaneous flange tangent velocity of point Q_b

$$\vec{v}_{Q_b} = \vec{v}_{COM} + \vec{\omega}_x^y \times \vec{w}_{Q_b} \quad (5.27)$$

Where $\vec{w}_{Q_b}$ is the position vector of Q_b within the wheel, with respect to G, the center of mass (COM).

In the regularized form, it can be written as:

$$\vec{F}_b^T = -\mu_b \|F_b^N\| \tanh\{\gamma \|\vec{v}_{Q_b}^t\|\} \hat{e} \quad (5.28)$$

The tangential velocity $\vec{v}_{Q_b}^t$ and the unit vector $\hat{e}$ still need to be computed. Let the reference frame of the second contact $\{\hat{T}_b\}$ be defined in the same way as $\{\hat{T}\}$ for the first contact. Let us express the velocity in this reference frame:

$$\vec{v}_{Q_b} = [\hat{T}_b]^T v_{Q_b} = [\hat{T}_b]^T \begin{pmatrix} v_{Q_b x} \\ v_{Q_b y} \\ v_{Q_b z} \end{pmatrix} \quad (5.29)$$

At point Q_b , the tangential velocity is defined by eliminating the normal component of the velocity $\vec{v}_{Q_b}$.

$$\vec{v}_{Q_b}^t = [\hat{T}_b]^T \begin{pmatrix} v_{Q_b x} \\ v_{Q_b y} \\ 0 \end{pmatrix} \quad (5.30)$$

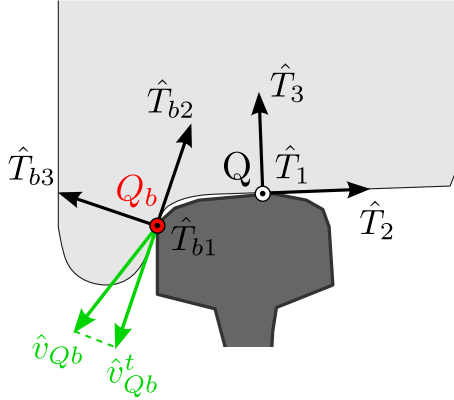


Fig. 5.12 Reference frames $[\hat{T}]$ and $[\hat{T}_b]$ used to express the velocity at the first and second contact points.

Wheel flange/check rail contact model

The principles and assumptions of the contact model between the outer part of the wheel flange and the check rail (Fig. 5.13) — present in switches & crossings — are similar to the previous model:

- Intermittent contact on a disk of radius r'_b and slope β (less than 90°);
- Lateral position of the check rail, Y_{CR}^o , is constant regardless of the position of the contact point Q'_b on the flange;
- Regularized contact force (normal = spring-damper type, tangential = dry friction type).

Given these assumptions, the reasoning and equation developments are strictly equivalent to the *Wheel flange/rail contact model*. The following figures summarize the model schematically.

In this figure:

- $Y_{Q'_b}$: represents the instantaneous lateral position (along the track) of the contact point Q'_b .
- Y_{CR}^o : represents the lateral position of the check rail (assumed constant for any Q'_b on the wheel flange).

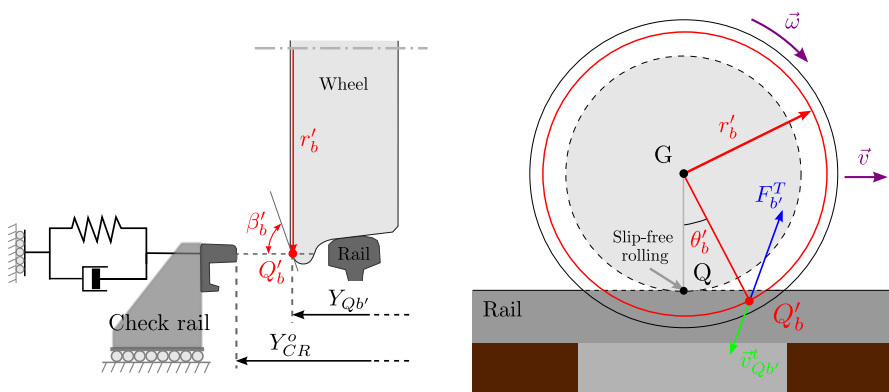


Fig. 5.13 Schematic of the contact model between wheel flange and check rail, showing the slope β and lateral positions $Y_{Q'_b}$ and Y_{CR}^o .

5.3 Measuring train modeling

The track geometry measuring train EM130 [6], used during the dynamic experimental campaign, serves as the reference model for the multibody simulations of train passage over the turnout.

The benchmark implementation described by Iwnicki [48] and used in this thesis was developed by the UCLouvain spin-off Quandt [86]. It is based on a 3D multibody approach implemented in the software Robo-tran [83], developed at UCLouvain. Within the framework of the present thesis, Boxus [96], during her Master's thesis, adapted the benchmark to the EM130 measuring train using theoretical data, dimensional measurements carried out in the workshop, and a measurement campaign conducted on board the train during a test run. Some minor adjustments, such as the recalibration of the carbody center of mass, were subsequently introduced to adapt the model to the analyses carried out in the present work.

The resulting model represents a single wagon passenger coach composed of one carbody, two bogies, four wheelsets, and two suspension stages as shown in Fig. 5.14.

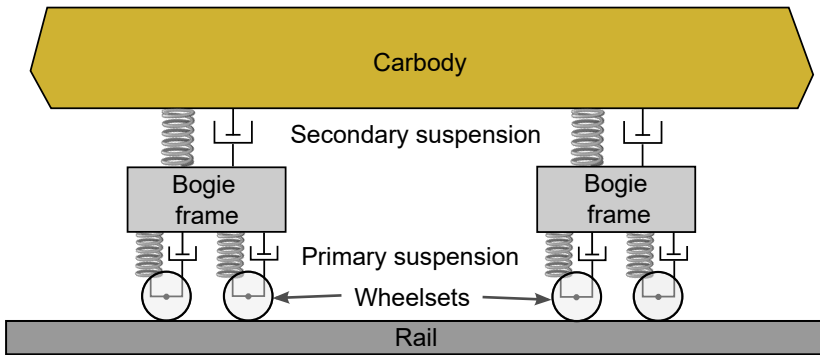


Fig. 5.14 Schematic of the multibody model from Boxus [96].

This 3D model assumes that the carbody, bogies, and wheelsets are rigid bodies interacting with each other through forces applied from one body to another. The suspensions are modeled with spring and damper laws. Other elements, such as bumpstops, a traction rod, and an anti-roll bar, are also considered.

To represent more accurately the train considered in this thesis, the benchmark model from Iwnick (1998) [48] had to be adapted due to the

fact that the bogies of the EM130 have particularities inspired by freight vehicles.

The most important modification concerns damping within the suspension stages. Passenger vehicles usually prefer viscous damping whereas freight vehicles often use friction surfaces. This design choice is due to its ability to adapt to the significant load differences in freight transport. Therefore, all the damping except for the vertical damping of the second suspension is modeled following a numerically friendly version of Coulomb’s friction law, see equation 5.31.

$$F_{friction} = \mu \cdot N \cdot \tanh\left(\frac{V}{\beta}\right) \quad (5.31)$$

Where μ is the friction coefficient of the surfaces, N is the normal force applied, V is the relative velocity between the surfaces, and β is a tradeoff coefficient to adapt the steepness of the friction law.

In addition to this major modification and to ensure that the model fits the EM130, different techniques were used to estimate the model parameters.

Some data was recovered by accessing blueprints from the train’s constructor and owner. This information includes the length, width and height of the carbody, the mass of the train, and the position of the bogies and wheelsets.

The stiffness of each suspension spring was estimated using geometric dimensions, considering that they are helical compression springs with ends squared and ground.

$$k = \frac{d^4 G}{8 D^3 N} \quad (5.32)$$

Where k is the spring rate, d is the diameter of the spring wire, G is the shear modulus of the material, D is the mean diameter of the coil, and N is the number of coils.

The lateral and longitudinal position of the center of mass of the carbody was based on the weight distribution measured under each wheel.

Once the model was parameterized, the next goal was to validate it and ensure that its results are a good representation of the reality. To do this, several methods were employed.

First, the static flexibility coefficient was compared with the value mea-

sured experimentally. The flexibility coefficient is defined as:

$$c_s = \frac{\alpha_{CB} - \alpha_{Track}}{\alpha_{Track}} \quad (5.33)$$

Where α_{CB} is the angle of the carbody and α_{Track} is the cant angle of the track.

Secondly, the frequency response of the train was analyzed. On the model side, the eigenmodes were computed directly through the Robo-tran software. On the experimental side, Fast Fourier Transforms were performed on acceleration signals recorded inside the carbody during a series of experimental campaigns. Similar eigen-frequencies in both cases are a comforting step towards a validated model.

The final validation step was conducted via a sensitivity analysis. Considering the numerous unknown parameters included in the model, identifying the parameters that most significantly impact the desired results can help determine the model's trustworthiness for the intended analysis. In the case of this study, if a parameter has no impact on the wheel-rail contact forces, the inability to estimate it precisely is not a critical problem.

Mass distribution

The calibration of the loads under each wheel was carried out through an iterative process in which the carbody center of mass was adjusted until the desired values were obtained. The objective was to preserve the same mass distribution ratio between the bogies and the center of mass of the train, while approximating as closely as possible the measured values provided by Infrabel under each wheel (Table 5.1). In this way, the bogie located under the heavier half of the train (containing wheelsets A and B) supports 55% of the total mass, whereas the bogie under the lighter half (containing wheelsets C and D) supports 45% (Table 5.2).

Regarding the longitudinal and lateral components of the center of mass, the global values derived from Infrabel's measurements are (1 259, -6.4) mm, while those obtained from the model are (1 262, -6.34) mm. Both sets of coordinates are expressed with respect to the geometric center of the carbody.

The identification of wheelsets and the corresponding left and right sides of the train are shown in Fig. 5.15.

Table 5.1 Mass measurements performed at Infrabel under each wheel.

Wheelset	Left wheel (kg)	Right wheel (kg)	Proportion
A	9170	9070	54.80%
B	8650	9370	
C	7550	7520	45.20%
D	7420	7420	

Table 5.2 Mass under each wheel in the model, measured at low speed (5 km/h) on a straight track.

Wheelset	Left wheel (kg)	Right wheel (kg)	Proportion
A	8978.6	9125.1	54.72%
B	8978.0	9124.4	
C	7418.4	7564.2	45.28%
D	7417.8	7563.6	

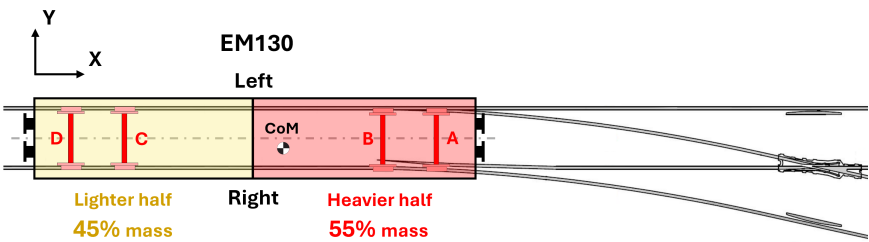


Fig. 5.15 Schematic representation of the wheelsets (A, B, C, D) and the identification of left and right sides.

Model parameters

A summary of the different parameters used in the model can be found in table 5.3.

Table 5.3 List of the parameters used in the measuring train model.

Carbody	Length Width Height Mass	23.8 m 2.8 m 2.5 m 47 970 kg
Carbody center of mass	Longitudinal position Lateral position Height above the carbody floor	1.106 m -0.008 m 0.67 m
Bogies and Wheelsets	Spacing between bogies Spacing between wheelsets of the same bogie Mass of 1 bogie Mass of wheelsets	17 m 2.5 m 4 500 kg 2 300 kg
Primary suspension	Longitudinal stiffness Lateral stiffness Vertical stiffness Damping friction coefficient	26.3 kN/mm 3.253 kN/mm 1.022 kN/mm 0.4
Secondary suspension	Longitudinal stiffness Lateral stiffness Vertical stiffness Damping friction coefficient Vertical damping	5 kN/mm 5 kN/mm 2.12 kN/mm 0.4 5 kNs/m
Wheel profile	Slope of type P profile	1/40

5.4 Turnout geometry representation

Several geometric configurations of railway turnouts are investigated to analyze their influence on train dynamic behavior and wheel-rail interaction, particularly in the Closure Panel and the Crossing Panel. The configurations are based on Infrabel's theoretical design, with modifications inspired by real measurements and hypotheses related to wear.

The main configurations, represented in Fig. 5.17 used in the multibody model are:

- **Configuration 1 (baseline model):** it is based on Infrabel's theoretical plan, with the following geometric sequence: *straight-curve-curve-straight*. The flangeway gap, i.e., the distance between the check rail and its closest running rail is 40 mm. To represent the transition zone of the check rail, a sigmoid shape is applied at both ends. To simplify the model, the track gauge is kept constant at 1435 mm, differing from the original plan.
A crossing gap of 510 mm is defined only for lateral contact, this is to say that within this gap no second contact is considered. However, the primary contact between the wheel tread and the running rail is still active along the entire length of the gap. If the train deviates laterally within the gap zone, flange contact disappears abruptly. As the vehicle continues its trajectory and re-establishes contact, a sudden numerical impact occurs, which makes the system unstable and ultimately leads to a simulation error.
- **Configuration 2 (closer to measured curvature):** a clothoid is introduced in the Crossing Panel, based on curvature measured by the EM130 measuring train. This configuration allows the study of curvature variation effects, especially a smoother transition between the straight section and the curvature of the Closure Panel, on lateral displacements in the Crossing Panel.
- **Configuration 3 (constant higher radius):** the entire curved zone is modeled with a constant radius of 500 m, following the sequence *straight-curve-straight*. This configuration approximates another existing turnout type, enabling comparison with the baseline model and analysis of the radius effect on track gauge in the Closure Panel.
- **Configuration 4 (optimized model):** a 2 cm clothoid is added in the Switch Panel (transition from straight to curved zone of radius

of 215 m) and a 1 cm clothoid in the Crossing Panel (between the curvature of the and the straight section of the diverging route), to improve geometric continuity and avoid potential energy losses due to discontinuities that could affect the model integrator. This configuration is used in the power dissipation studies. A verification with Configuration 1 showed that displacement differences with and without these clothoids were on the order of 0.001 mm, validating the approximation.

In configuration 4, three variants of the flangeway gap at the crossing area (Fig. 5.16) have been studied in order to evaluate the effect of wear and check rail adjustment on the dissipated energy and its relation to the direction of movement:

- **Configuration 4.1 (adjusted check rail):** represents a common practice of tightening the check rail to anticipate for wear, thereby reducing the flangeway gap.
- **Configuration 4.2 (nominal value):** the standard flangeway gap value is used according to specifications, as a reference case.
- **Configuration 4.3 (worn check rail):** the effect of a worn check rail is simulated by increasing the flangeway gap.

These configurations allow for an analysis of how the geometry of the turnout influences the lateral dynamics of the train, specifically the power dissipated in the wheel flange/rail contact, which may translate into wear, as well as the influence of other parameters such as the direction of movement (facing or trailing move) when the flangeway gap is modified.

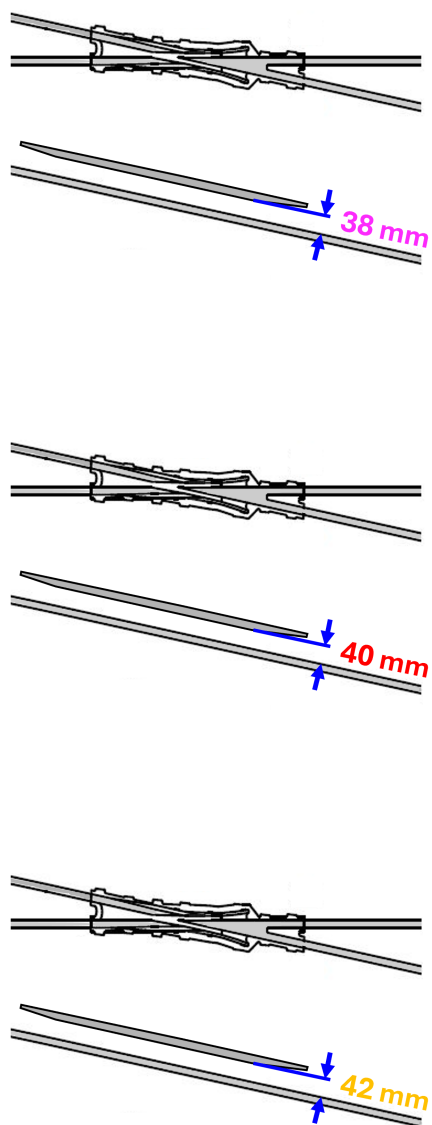


Fig. 5.16 Three flangeway gap configurations studied in the analysis of the Crossing Panel. Top: adjusted check rail. Middle: nominal value. Bottom: worn check rail.

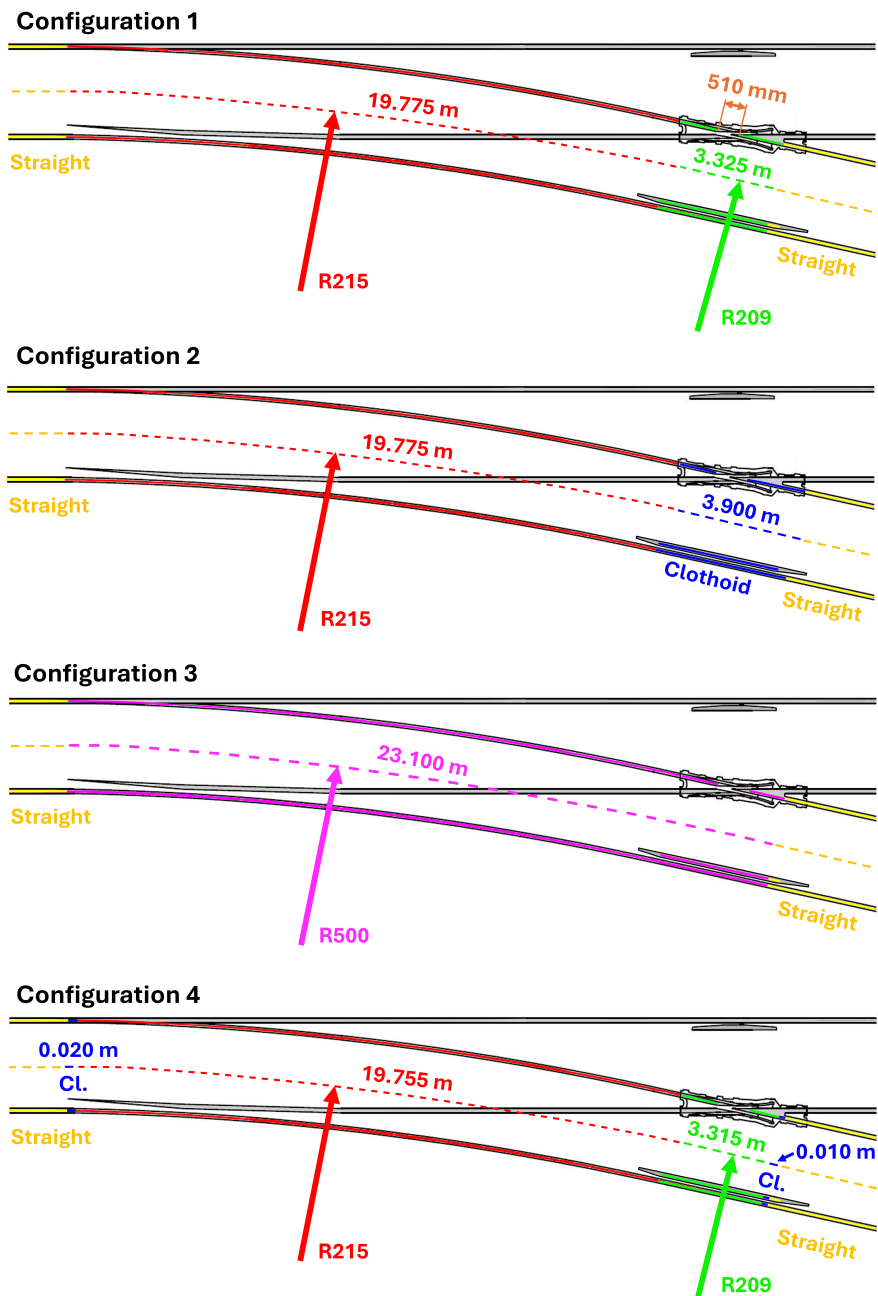


Fig. 5.17 4 turnout model configurations studied in this thesis.

5.5 Simulation cases

Config.	Move	Vel. [km/h]	Main outputs
1	Facing	5–70	- Rail displacement - Gauge - Check rail displacement - Δ flangeway gap
	Trailing	15–80	
	Rev. facing	15–40	
	Rev. trailing	5–40	
2	Facing	5–40	- Rail displacement - Check rail displacement
	Trailing	15–40	
3	Facing	5–70	- Rail displacement - Gauge
	Trailing	15–80	
4	Facing	5–40	- Δ flangeway gap - Dissipated energy (3 flangeway gap config.)
	Trailing	15–40	

Table 5.4 Summary of simulation cases

Table 5.4 shows the different simulation cases performed. The model calculates the contact forces, in particular the Kalker forces at the contact between the tread and the rail head, as well as the penalty contacts on the inner and outer surfaces of the wheel flange with the rail head.

In order to interpret these results and compare them with the displacement measurements from the experimental campaign, most of the graphs and comparisons are presented in terms of **equivalent displacements**. In the case of the main rail (or running rail), lateral displacements are obtained from the sum of the Kalker forces and the contact forces of the wheel flange, taken from a rail reference system and divided by the lateral stiffness defined for it. For the check rail, the displacement is calculated from the lateral forces resulting from the internal contact between the wheel flange and the rail head, divided by the stiffness of the check rail itself.

Different passage scenarios were considered, including facing and trailing moves, as well as reversing the train’s orientation. In some cases at very low speeds (5–10 km/h), numerical integration problems were observed that generated unwanted lateral vibrations, so the corresponding results are not analyzed.

In terms of configurations:

- **Configuration 1** (baseline model): explores a wider range of speeds (up to 70–80 km/h depending on the direction) and includes the reversed orientation of the train.
- **Configuration 2** (closer to measured curvature): performs displacement analyses for the two panels studied, with speed ranges up to 40 km/h.
- **Configuration 3** (larger radius of curvature): similar to Configuration 2, but also includes higher speeds to study the centrifugal effect.
- **Configuration 4** (optimized model): three variants of the flangeway gap in the Crossing Panel are analyzed to study the effect of wear and check rail adjustment on energy dissipation.

5.6 Limitations of the Model

The model has been developed to study the dynamics in the Closure and Crossing Panels, particularly for analyzing lateral forces and deducing the equivalent rail displacements when the train is passing. It also enables sensitivity analyses on several parameters, such as:

- **Curvature:** influence of the radius or the length of the curved zone.
- **Lateral dimensions such as the flangeway gap and track gauge:** assessment of different gap values and track gauge values and their influence on lateral forces and, specifically for the flangeway gap study, on the wear of the check rail, up to a limit value associated with an increased risk of derailment (detectable through lateral contact with the crossing gap).
- **Train passage conditions:** influence of running velocity, the train trajectory and orientation on the dynamic behavior of the track.
- **Lateral stiffness of the track:** the model allows modifying the lateral stiffness depending on the zone of the turnout, to adapt more flexible devices, such as the closure rails, and stiffer ones, such as the check rail.

However, the model has several limitations to study the following aspects. The main ones are listed below:

- **Switch Panel:** the detailed dynamic behavior in this area is not represented. The Switch Panel, for the diverging route, is considered as a perfect track with an imposed change of curvature. In the context of this work, results are only analysed for time instants when no wheelsets are present in the switch area, which therefore no longer influences the track behavior. A specific and more advanced model would be required to investigate this region in detail.
- **Rail deformation represented through an equivalent lateral stiffness:** the model includes the deformation of the track components through an equivalent lateral stiffness at the wheel/rail contact. To study in detail how each individual component of the track deforms, a complementary model — such as a finite element model — would be required, following the direction initiated in the Master thesis of Quoidbach [97].
- **Forces and equivalent induced displacements only present when wheel/rail contact exists:** the multibody model does not provide information on track behavior outside the wheel/rail contact region. A FEM model would be needed to complement the multibody model in order to determine rail deformations beyond the contact regions.
- **Rail vertical irregularities and dynamics:** since the present thesis focuses mainly on lateral behavior, some parameters related to vertical dynamics — such as rail vertical irregularities and track stiffness — are not included. Integrating these aspects into the Crossing Panel would be valuable to assess their potential influence on the lateral wheel/rail interactions that this work focused on.
- **Ideal conditions assumed:** variations in rail temperature, humidity, or contamination of the track components have not been considered. The model could be extended to incorporate such effects by modifying the wheel/rail contact conditions, especially through changes in the friction coefficient at specific locations on the rail and points on the wheels (flange vs. tread contact).

PART IV

Results & discussion

6

Results in the Closure Panel

THIS chapter presents the key findings from the experimental and numerical model data of the AIGUIDYN project in the Closure Panel. The analysis is structured into five complementary sections.

The first section, **Section 6.1**, introduces a reading methodology designed to facilitate the understanding of the experimental data, including the visualization of the sensor output depending on the position of the train during its passage and the distinction between facing and trailing moves through the switch & crossing (S&C). The following sections focus on the analysis of the effects of key variables — such as train trajectory (**Section 6.2**), velocity (**Section 6.3**), center of mass location (**Section 6.4**) and lateral track stiffness (**Section 6.5**) — on the dynamic behavior of the track.

As a reminder, the experimental measurements presented in this chapter are grouped into two main categories:

- **On-track systems:**
 - *LVDTs*: these sensors measure localized lateral rail displacements relative to the sleeper reference, being able to quantify the differences between static geometry and dynamic responses induced by train passage.
 - *Digital ruler*: it provides static baseline measurements of track dimensions without the influence of train dynamics.

- **On-board systems:**

- *Switch Measurement System (SMS)*: it continuously captures dynamic track geometry from the train's perspective, so it obtains measurements of rail geometry under load.
- *On-board camera recordings*: they provide visual observations of switch & crossing behavior during train passage, complementing quantitative data with qualitative evidence.

These systems, which were used during the experimental setup, enable a detailed comparison of rail behaviour by integrating static and dynamic data across facing and trailing moves at various train velocities. Based on this integrated dataset, this chapter aims to combine the experimental measurement data with the data obtained from the developed model to identify behavioral patterns, validate hypotheses and propose guidelines to improve the understanding of switch & crossing dynamics.

6.1 Results reading and interpretation

6.1.1 Reading grid and visualisation

For the sake of clarity, it is useful to explain at first the key to interpreting the graphs and schematics chosen to present the experimental results in the next sections.

Sensors being fixed on the S&C, the measured quantities (e.g. the rail displacement from a LVDT) will preferably be presented as a function of the position of a reference wheelset along the track, rather than of time.

In both the diverging and the through route of the turnout, this position of the wheelset along the rail path is referred to as the curvilinear coordinate s .

To avoid confusion in the interpretation of the graphs, it is important to clarify the order of the wheelsets depending on the move. In the case of a facing move, the first wheelset (1st WS), which is the leading one, corresponds to wheelset A. However, in the case of the trailing move, the leading - 1st WS - is wheelset D. The correspondence of the wheelset positions for each move are illustrated in the Fig 6.1.

The following two sections are therefore only intended to explain how to interpret the results — for both train directions respectively — with a representative sensor located in the closure panel of the S&C.

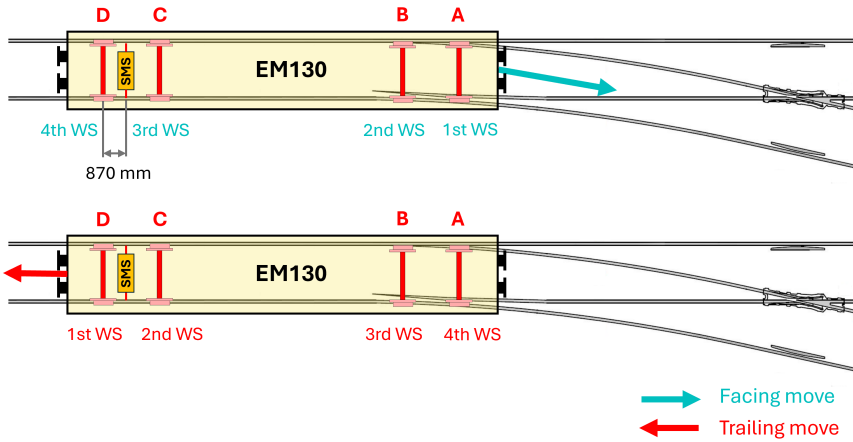


Fig. 6.1 Wheelset (WS) order and letter assignment for facing and trailing moves. The train's Switch Measurement System (SMS) is installed at a distance of 870 mm from wheelset D.

6.1.2 Facing move

The bottom of Fig. 6.2 shows a typical configuration in which the measuring train moves at speed V along the S&C from left to right (facing move). At the curvilinear coordinate $s = 13.4\text{m}$ of the S&C (in the crossing panel area) let us consider a sensor, called DS, which measure the rail lateral displacement (positive = outward the track, negative = inward the track). On the train side, the four wheelsets are designated by the letters **A** (leading), **B**, **C** and **D** (trailing). The instantaneous configuration highlighted by the vertical dashed line as an example in Fig. 6.2 is that where the **A** wheelset is precisely in front of the sensor DS.

An illustrative result of those shown in the following sections is visible at the top of Fig. 6.2: it represents the lateral displacement of the rail measured by the sensor (y-axis) as a function of the curvilinear position s_A of the leading wheelset (x-axis), used as the reference moving body throughout this study (wheelset **A** in this case).

The graph must therefore be read as follows: the peak displacement measured by the sensor (mainly due to wheelset **A**) is approx. 1.1 mm (outward displacement of the rail) when **A** is in front of it, i.e. at 13.4m from the origin ($s = 0$) of the S&C.

To reinforce understanding of the graph, another configuration – a few moments later – is shown in Fig. 6.3: the leading wheelset **A** is 32.9 m from the S&C origin, the 4th wheelset **D** being now just in front of the sensor DS, causing a peak displacement, mainly caused by wheelset **D** of -0.22mm (inward displacement of the rail).

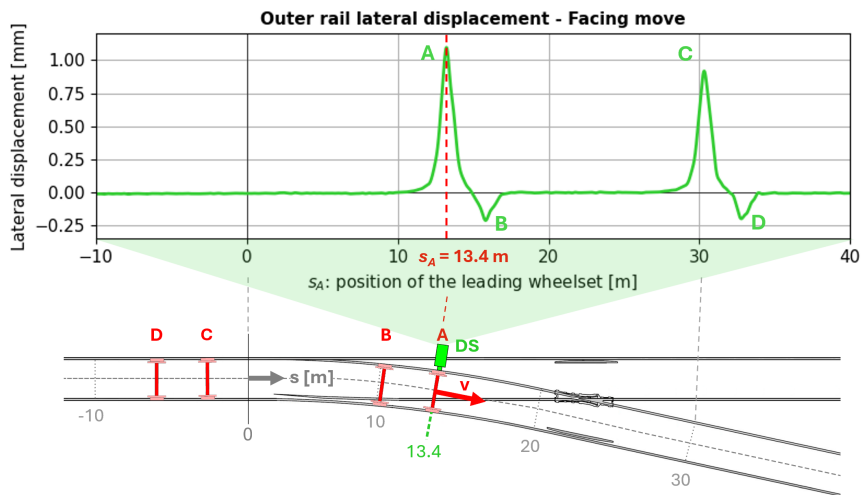


Fig. 6.2 Sensor measurement interpretation: facing move - collocated wheelset.

6.1.3 Trailing move

In the case of the trailing move (Fig. 6.4), the reasoning is similar except that in this case the train – which has not turned around – moves from right to left; so the graph in Fig. 6.4 should now be read from right to left, the reference wheelset being **D**, the leading one. In Fig. 6.4, the rail displacement peak **D** appears when **D** is just in front of the sensor DS.

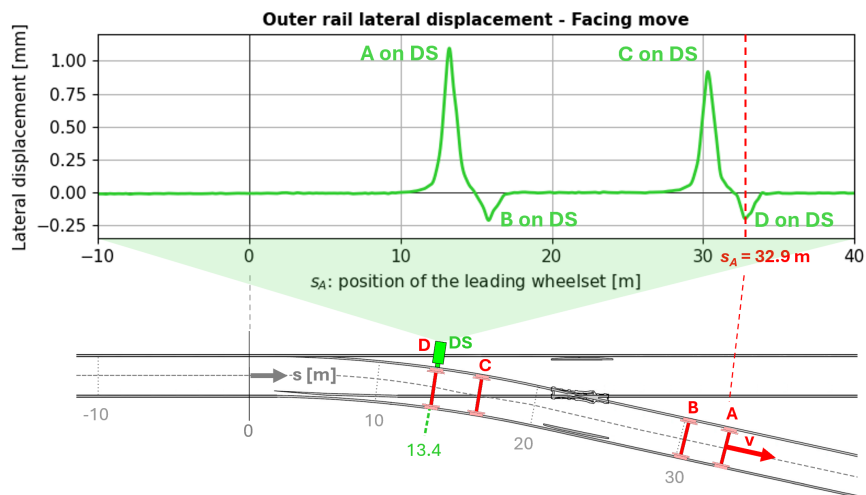


Fig. 6.3 Sensor measurement interpretation: facing move - uncollocated wheelset.

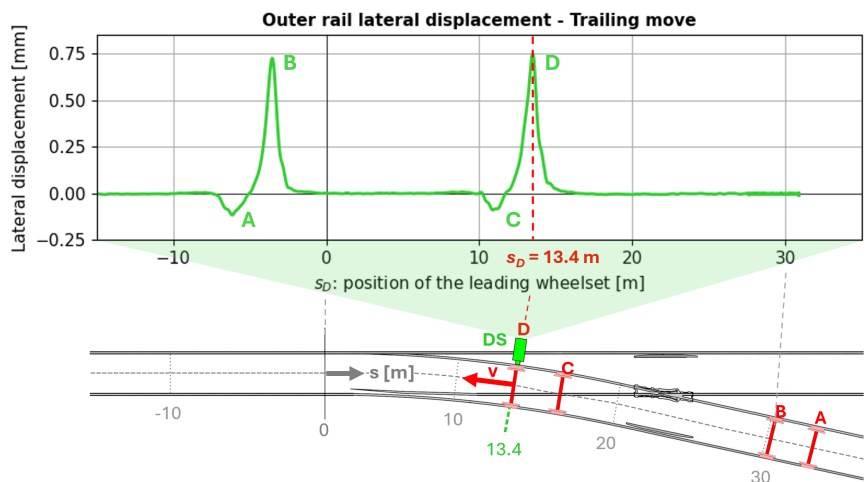


Fig. 6.4 Sensor measurement interpretation: trailing move - uncollocated wheelset.

6.1.4 Numerical model data

To facilitate the reading and comprehension of the results, all graphs representing data obtained from the multibody model are indicated through a legend displayed with a grey background and referenced in the caption. In some cases, this is further reinforced by a graph title labeled *Model*.

In contrast, results obtained from the experimental setup are indicated through a legend with a white background and referenced in the caption. In some cases, this is complemented by labels such as *Experimental* or *Exp.* in the graph title.

6.2 Trajectory influence

This section analyzes the influence of the train trajectory — facing or trailing move — on the Closure Panel. It focuses on how the move direction has an effect on the track geometry and stiffness characteristics, particularly on lateral displacements and rail deformation. The study combines experimental measurements with numerical simulations that allow for analysis beyond the theoretical configuration.

6.2.1 Experimental results

Diverging route

The LVDT sensors placed on the rail head (see Fig. 3.5) measure the evolution of the lateral rail displacements relative to the sleeper (Fig. 6.5). Before the train arrives at the S&C, the displacements are calibrated to zero, the static configuration of the track having been chosen as the reference.

What can we observe?

During the facing move, in the Closure Panel of the diverging route, when the first wheelset **A** reaches the same position ($s = 13.4$ mm) as the displacement sensor DS, the sensor detects an initial positive peak, with values ranging between 0.8 and 1.2 mm across all measured velocities, indicating that the rail moves outward from the curve. As the train continues to move forward and the 2nd wheelset **B** reaches the DS position, a second – smaller – peak is observed but in the opposite direction, indicating an inward movement of the rail.

This pattern repeats with the 3rd **C** and 4th **D** wheelsets, which show similar outward and inward displacements of the rail, respectively, to those observed with the first and second wheelsets. In particular, the shape of these displacement curves remains consistent across all velocity regimes, indicating that the rail response pattern is not affected by the train velocity.

For the trailing move, the train approaches from the right side of the figure, so the top graph in Fig. 6.6 should be read from right to left. As in the previous case, when the leading wheelset **D** aligns with the position of the displacement sensor DS, the rail still experiences an outward displacement from the curve, measuring less than 1.0 mm across all recorded velocities. As the train moves forward and the other wheelsets pass in front of DS, the same displacement pattern observed in the facing move reappears: the rail experiences an outward displacement due to the 1st and 3rd wheel contacts, and a smaller inward displacement for the 2nd and 4th wheel contacts.

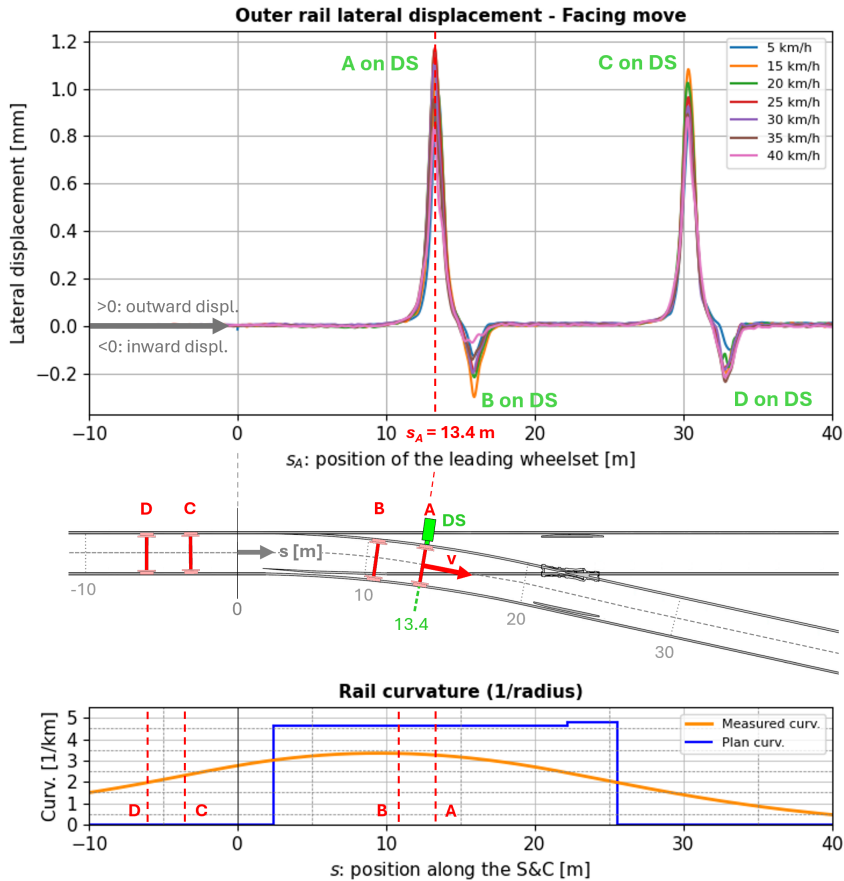


Fig. 6.5 Facing move: (Top) lateral displacement of the outer rail measured by the displacement sensor (DS) for every velocity regime. Each peak corresponds to the moment when a wheelset – A, B, C or D – aligns with the fixed position of the DS sensor on the track. (Middle) Positioning of the wheelsets when the leading one (A) aligns with the DS longitudinal position: $s_A = s_{DS} = 13.4$ m. (Bottom) Rail curvature, with the corresponding wheelset's position when $s_A = s_{DS}$. In orange, the measurement obtained by the EM130 train; in blue, the reference curvature from the plan, with the corresponding wheelset positions indicated on the graph.

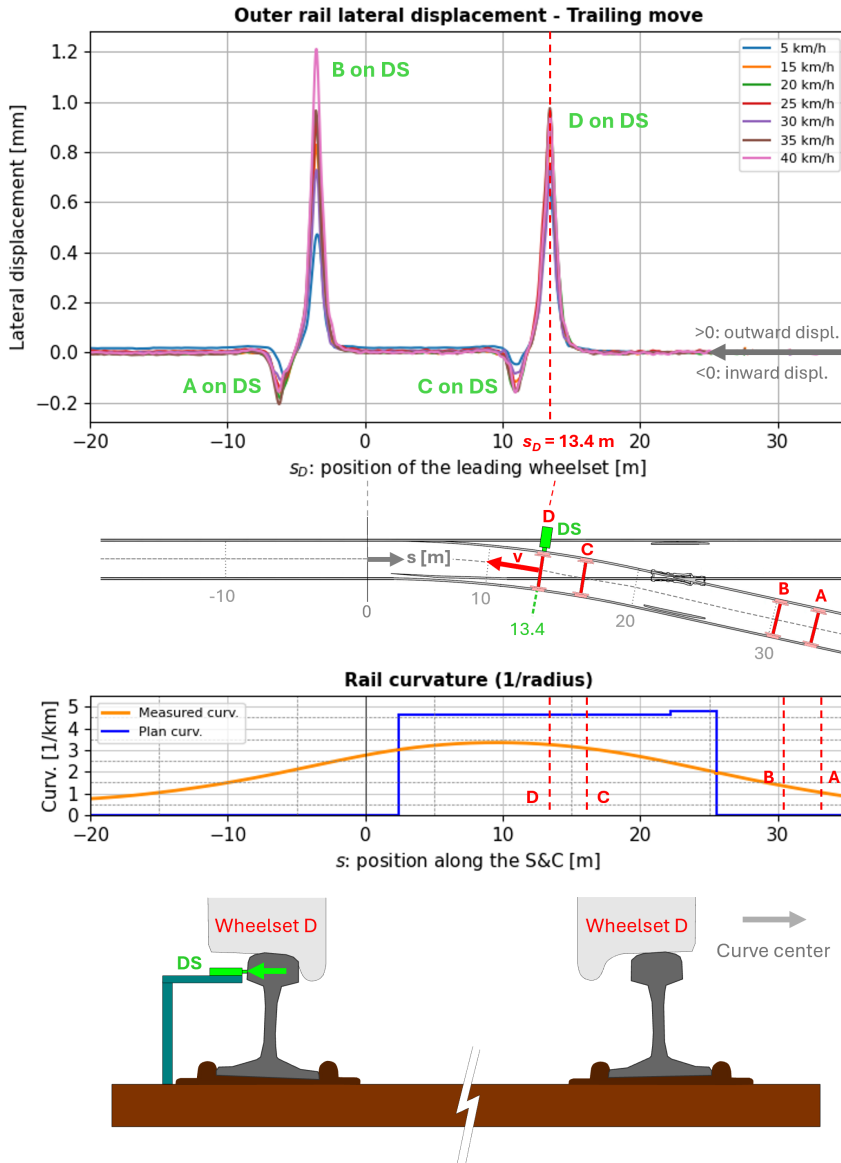


Fig. 6.6 Trailing move: (Top) lateral displacement of the outer rail measured by the displacement sensor DS for each velocity regime. Each peak corresponds to the moment when a wheelset – D, C, B, or A – aligns with the fixed position of the DS sensor on the track. (Middle) Positioning of the wheelsets when $s_D = s_{DS} = 13.4$ m. (Bottom) Rail curvature, with the corresponding wheelset's position when $s_D = s_{DS}$. In orange, the measurement obtained by the EM130 train; in blue, the reference curvature from the plan, with the corresponding wheelset positions indicated on the graph.

Through route

The equivalent LVDT sensors placed at the same longitudinal coordinate as in the diverging route are used to analyze the rail behavior along the through route (Fig. 6.7). These measurements allow a direct comparison between the two trajectories under similar sensor positioning and train passage configurations.

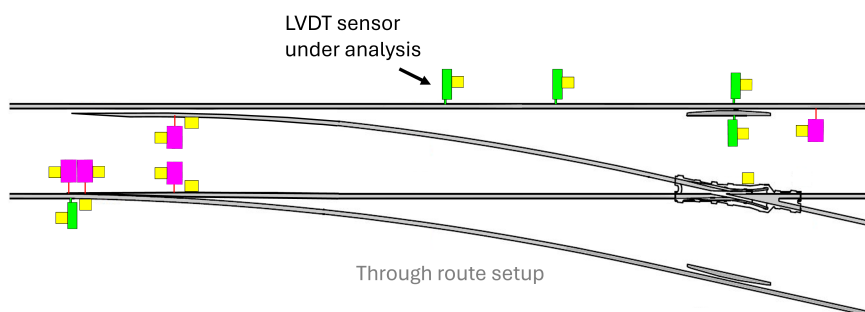


Fig. 6.7 Sensor layout for the through route configuration. The sensor under analysis in this section is highlighted.

During facing (Fig. 6.8) and trailing moves (Fig. 6.9) on the through route, the sensors detect a lateral displacement of the rail inward, i.e. towards the track centerline, at every moment that a wheel is aligned with the sensor. This contrasts with the diverging route, where alternating outward and inward displacements were observed depending on the wheelset alignment.

The displacement curves recorded for all velocity regimes show a similar shape and amplitude. Peak values are generally below 0.5 mm, higher for the facing move than for the trailing move. This suggests that the rail response on the through route is more stable with lower lateral forces induced by the wheelset compared to the diverging route. The absence of direction changes in the displacement pattern further indicates a more uniform contact behavior between the wheel and the rail head. This difference can be attributed to the centrifugal effect: in the diverging route, the train negotiates a curve, generating outward lateral forces due to the change in direction. In contrast, the through route follows a straighter path, minimizing the lateral forces and resulting in a more uniform and inward displacement pattern.

These results highlight the influence of the route on the rail's dynamic

response. The diverging route shows different displacement patterns due to the curvature, whereas the through route maintains a more consistent inward displacement profile at the sensor location.



Fig. 6.8 Facing move: displacement of the external rail measured by the LVDT sensor during train passage on the through route. Negative values indicate movement inwards toward the center of the track.

Comparison of the displacement peaks due to the leading wheels in the diverging route

The displacement peaks of the rails in the diverging route were analyzed when contact with the wheel flange of the leading wheelset occurred. The values were collected for each train velocity and for both moves – facing and trailing – resulting in two distinct curves (Fig. 6.10). Observing the graph, no clear influence of velocity on the displacement was observed. But, it should be remembered that the velocity was limited to 40 km/h due to operational restrictions on the diverging track. However, a clear influence of the train direction can be noticed, with generally higher displacements recorded during the facing move.

One hypothesis to explain the impact of the train direction is related to the trajectory before reaching the sensor area. The peak curvature of the

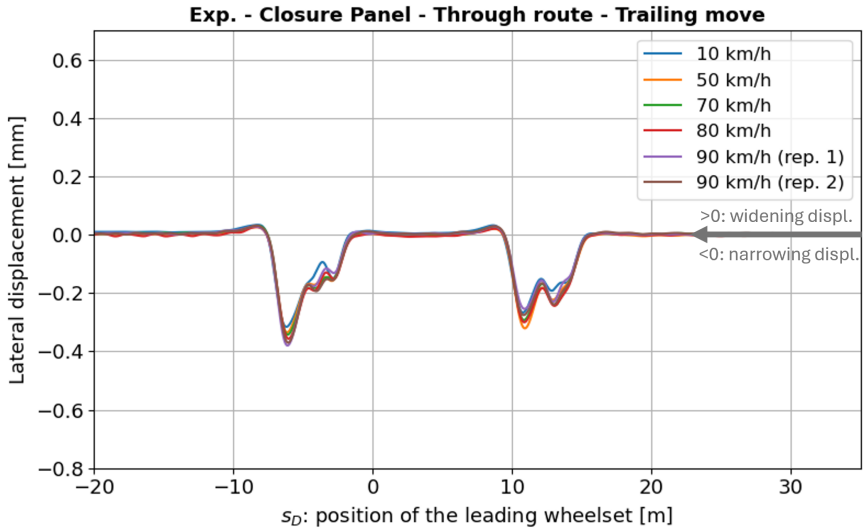


Fig. 6.9 Trailing move: displacement of the external rail measured by the LVDT sensor during train passage on the through route. Negative values indicate movement inwards toward the center of the track.

track is located approximately 3 meters before the sensor position when the train approaches in the facing move (Figure 6.5, *Rail curvature* graph), therefore about 3 meters after the sensor for the trailing move (Figure 6.6, *Rail curvature* graph). So for the facing move, the wheel encounters a higher track curvature just before arriving at the DS position, which could induce higher lateral contact forces. For the same reason, at the moment of contact between the rail and the leading wheel at the DS position, the track curvature experienced for the other wheels is higher when the train comes from the facing move compared to the trailing move.

Therefore, this effect is shown to depend more on the passage configuration and the yaw of the bogie during curve negotiation than on dynamic effects linked to speed, given the low-velocity range of the experiment ($\leq 40 \text{ km/h}$). This interpretation is in line with Fissette (1996) [98], who showed that in low-speed curving conditions ($\approx 25 \text{ km/h}$) the bogie yaw configuration strongly influences lateral and flange forces, while speed was kept constant and not considered a determining factor. This effect of the passage configuration is analyzed using the camera recordings, which

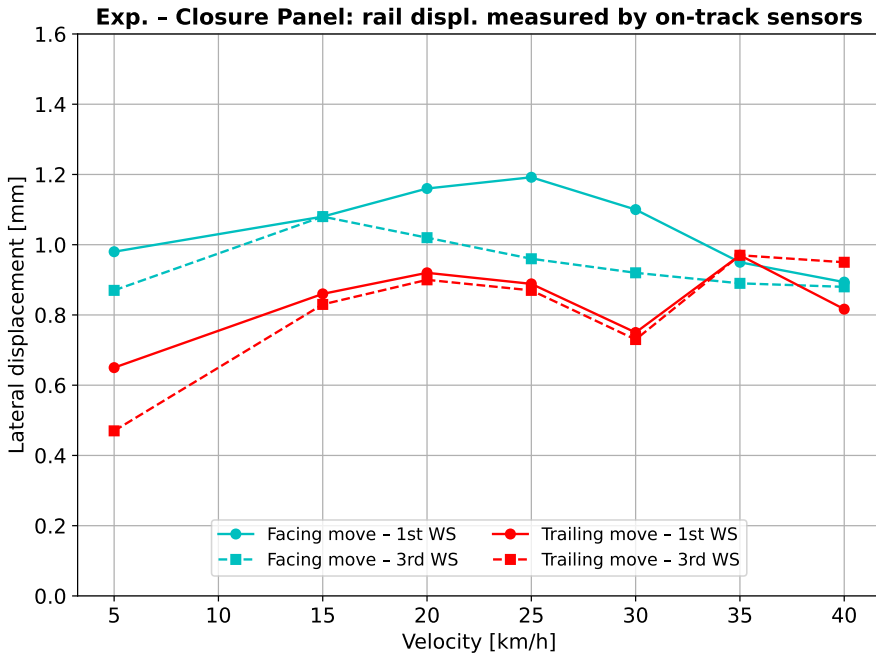


Fig. 6.10 Rail displacement peaks measured by the DS when the leading wheelset aligns with the displacement sensor position, comparing facing and trailing moves across different velocities. For 25 km/h and 40 km/h, the experiment was conducted twice, and the average value has been taken.

captured precise instances of wheel flange/rail interactions.

Additionally, the flexion of the sleepers could also contribute to the lateral displacements of the rail. Although this effect was not measured during the experimental campaign, it is plausible that vertical and lateral forces applied by the train induce some bending of the sleepers, which in turn may induce rail roll deflections and thus rail head lateral response.

Observations from camera recordings

To complement the data obtained from the displacement sensors (DS), two compact action cameras (GoPro®) were positioned flush with the rail in front of the wheels of wheelset **D** (Fig. 6.11) to capture close-up footage of the wheel-rail interactions. In the movie images shown in Fig. 6.12, taken from a video recorded at a low velocity of 5 km/h, the wheels can be seen

in contact with the rail just where the displacement sensor DS is located, allowing for a visual comparison of the wheel-rail contact during both facing and trailing moves.

For the facing move (left image), wheelset **D** acts as the trailing wheelset. At the equivalent moment in Fig. 6.5, a negative peak was recorded by the DS sensor, indicating an inward displacement of the rail (towards the curve center). The camera footage for this case shows no contact between the wheel flange and the rail.

For the trailing move (right image), with wheelset **D** as the leading wheelset, a larger positive peak was observed in the DS sensor data (Fig. 6.6), indicating an outward displacement of the rail (i.e. away from the curve center). Consistently, the video footage shows that a flange-rail contact occurs when the leading wheelset **D** aligns with the DS position (see Fig. 6.12-right), indicating that this outward displacement is most likely due to the wheel flange pushing the rail outward the curve.

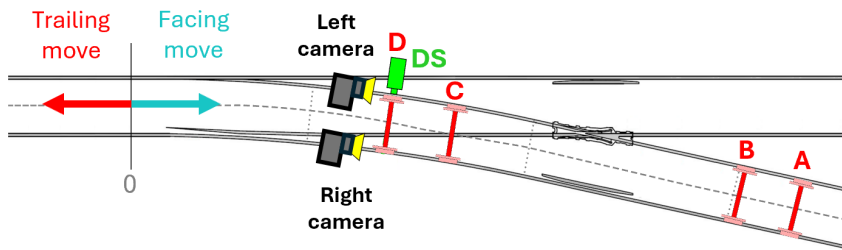


Fig. 6.11 GoPro® camera positions relative to the wheelsets.

Comparison between Track Sensors, Train Sensors and Static Measurement

The train's Switch Measurement System (SMS) is positioned on the bogie between wheelsets **C** and **D**, closer to wheelset **D** at a distance of 870 mm (Fig. 6.13). Consequently, it does not take measurements at the same longitudinal position as the contact point between a wheel and the rail, which the DS sensor does allow.

Thus, comparable measurements can only be achieved when the longitudinal position of the mobile sensor (SMS) coincides with that of the rail sensor (DS). This alignment is illustrated in Fig. 6.13 for the facing move and Fig. 6.14 for the trailing move. In these cases, the train sensor (SMS)



Facing move (4th WS)

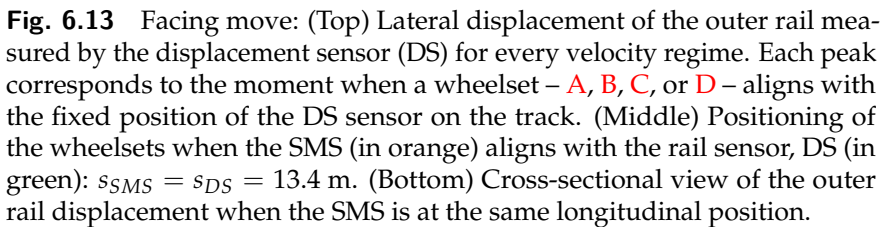


Trailing move (1st WS)

Fig. 6.12 Camera recordings of the wheel as it passes in front of the DS sensor (wheelset **D** - left camera) at a velocity of 5 km/h: the left image shows the view during the facing move, where the wheel belongs to the trailing wheelset. In this image, there is no contact between the wheel flange and the rail. The right image shows the view during the trailing move, where the wheel now belongs to the leading wheelset. In this image, there is clear contact between the wheel flange and the rail.

measures the gauge – i.e. the distance between the inner and the outer rails – not at the moment when the peak is recorded by the displacement sensor but between the two peaks caused by wheelsets **C** and **D**, as indicated by the vertical discontinuous red line in Figs. 6.13 and 6.14.

The measuring train provides a dynamic gauge measurement, along with other measurements mentioned in 3.2.2. To evaluate the deviations of this measurement relative to the static value, it might be compared with the readings obtained using the digital ruler. However, the static value obtained



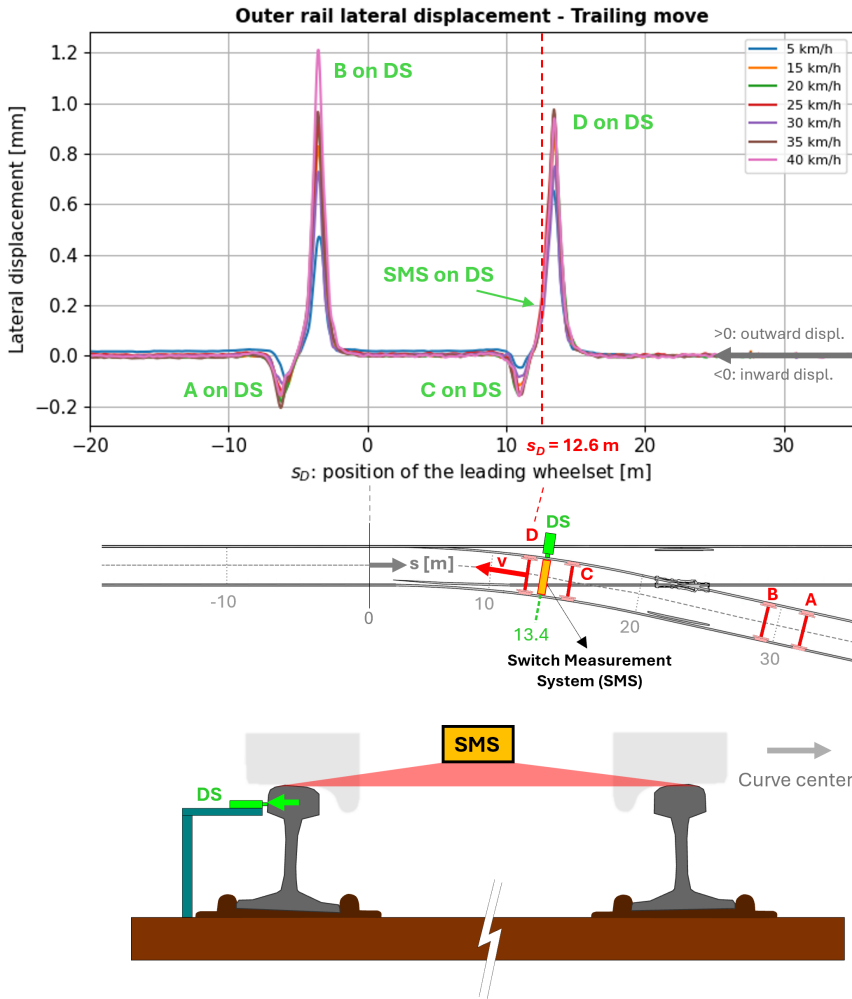


Fig. 6.14 Trailing move: (Top) Lateral displacement of the outer rail measured by the displacement sensor (DS) for every velocity regime. Each peak corresponds to the moment when a wheelset – A, B, C, or D – aligns with the fixed position of the DS sensor on the track. (Middle) Positioning of the wheelsets when the SMS (in orange) aligns with the rail sensor, DS (in green): $s_{SMS} = s_{DS} = 13.4$ m. (Bottom) Cross-sectional view of the outer rail displacement when the SMS is at the same longitudinal position.

with the digital ruler does not take into account the *in-charge static reference*, that is, the lateral displacement of the rails when the train is stopped and exerting load on them. This reference would be more appropriate for analyzing the dynamic effect of the train's passage.

Since this in-charge static reference was not measured during the experimentations, the value recorded during the train's passage at 5 km/h has been taken as a **quasi-static reference**. Although contact between the wheel flange and the rail occurs at all speeds, the centripetal acceleration at 5 km/h is only:

$$a_c = \frac{v^2}{R} = \frac{(1.389)^2}{215} \approx 0.009 \text{ m/s}^2$$

It results in a lateral force of approximately 326 N, as shown in Appendix A. This force is mainly due to the geometric curvature and static loading, with negligible dynamic contributions. In contrast, at 40 km/h, the centripetal acceleration increases to

$$a_c = \frac{(11.111)^2}{215} \approx 0.574 \text{ m/s}^2$$

It produces a lateral force of approximately 20.9 kN. This represents an increase of more than **6 300 %** compared to the quasi-static case. Therefore, the low-speed measurement provides a justified baseline for quantifying the dynamic effect of lateral forces due to train velocity.

At 5 km/h, contact between the wheel flange and the rail still occurs, as in all velocity regimes, so significant lateral forces are present. However, due to the low velocity, these forces can be interpreted as mainly geometric and (quasi-)static in nature, although different from purely static measurements obtained with a ruler, thus providing a reliable baseline for comparison with higher-velocity cases.

Fig. 6.15 shows the gauge values measured by the SMS and the digital ruler at the exact same position where the DS is installed. Two additional horizontal dashed lines are represented to represent the quasi-static reference for both moves.

The DS installed in the closure panel is positioned only on the outer rail, where it records the lateral displacement of the rail relative to the sleeper. As such, it cannot measure the equivalent of the gauge variation, as it would also need to measure the lateral displacement of the inner rail.

Although the inner rail was not directly measured by the displacement sensors, camera recordings confirm the absence of contact between the

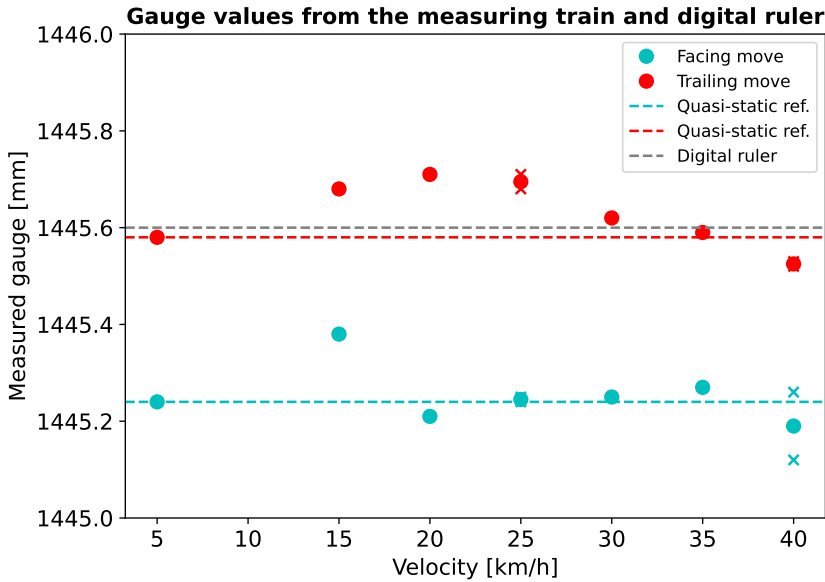


Fig. 6.15 Gauge values measured by the switch measurement system at different train velocities, compared with the digital ruler and the quasi-static reference, when the SMS is located at the same longitudinal position as the displacement sensor DS in the closure panel. For 25 km/h and 40 km/h, 'x' marks two distinct measurements, and the circle on the same abscissa represents their average.

flange of wheelset **D** and the inner rail for both the facing and trailing moves. Consequently, the only contact force acting on the inner rail results from the creep forces between the wheel tread and the rail head.

Regarding the results in Fig. 6.15, for the facing move, the measuring train records gauge values that remain consistent across all velocities and close to the quasi-static reference. The displacement sensor registers lateral displacements close to zero, suggesting that the inner rail also undergoes negligible lateral displacement. For the trailing move, the measuring train records gauge values close to the static reference, with a slight widening of up to 0.2 mm at certain velocities.

The behavior of the inner rail, which was not directly measured, and its influence on the gauge variation, are further analyzed using the numerical model results presented in the following section.

6.2.2 Model results

The model is able to provide the forces and displacements of the rail at the wheel/rail contact points along the entire track. The displacements are represented for the outer rail of the curve, with values extracted at the equivalent position of the LVDT installed during the experimental campaign.

As observed in the experimental data, the model also captures a change in the sign of the rail displacement. For the facing move (see Fig. 6.16), the leading wheelset of the first bogie is **A**, and that of the second bogie is **C**. For the trailing move (see Fig. 6.17) the 1st and 3rd wheelsets (WS) are in this case **D** and **B**, respectively. At the contact points between the rail and these wheelsets, the rail suffers a displacement outward from the track centerline due to flange contact. In contrast, for the contact points with the second and fourth wheelsets — **B** and **D** for the facing move, **C** and **A** for the trailing move —, the rail experiences a displacement in the opposite direction, with a lower absolute value. In these cases, there is no flange contact, only tread-rail interaction.

This trend is consistent with the experimental observations and confirms the presence of flange contact for the first and third wheelsets for both moves: facing and trailing.

In the experimental campaign, only one rail was instrumented in the closure panel. The Switch Measurement System (SMS), installed between wheelsets **D** and **C** — closer to **D** — was able to measure the gauge variation at the same specific location as the sensor on the rail.

Unlike the SMS, which provides a gauge value measured at a longitudinal position between two wheelsets (**D** and **C**) and always from the same relative reference position from the train, the numerical model calculates the lateral displacements of both rails at the exact moments when they are in contact with each wheel.

Fig. 6.20 presents the simulated lateral displacement of the rail in the closure panel, at the same longitudinal position where the displacement sensor (DS) was installed during the experimental campaign (see Fig. 6.10). The results are shown for both the 1st and 3rd wheelsets (WS), and for facing and trailing moves, across a range of velocities.

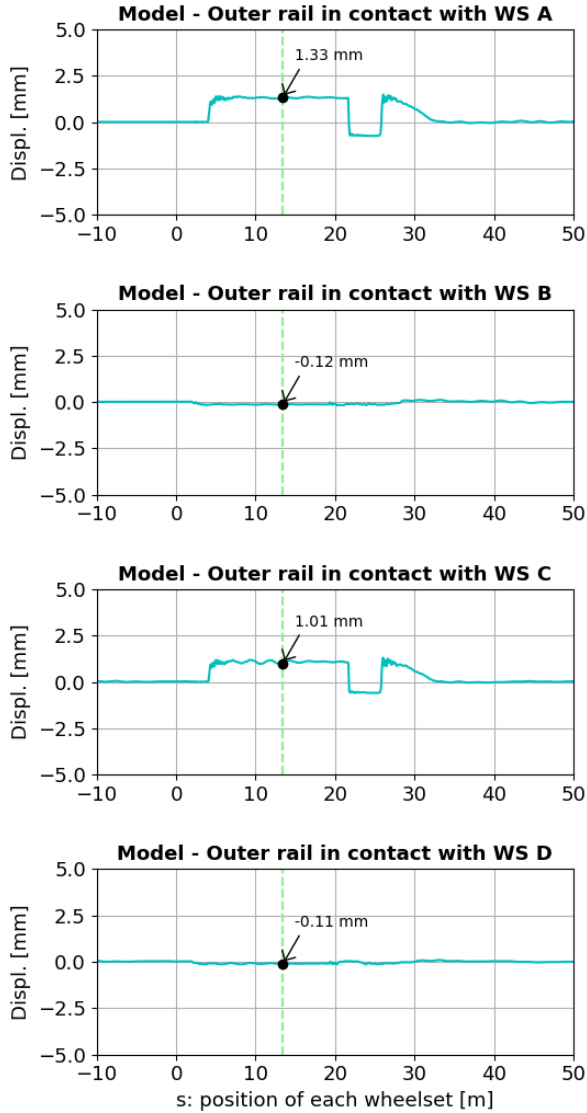


Fig. 6.16 Facing move: numerical simulation results validating the direction of outer rail displacements at 25 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign¹.

¹The model obtains forces as outputs in the wheel/rail contacts. These forces are divided by the lateral stiffness of the rail to obtain equivalent displacements.

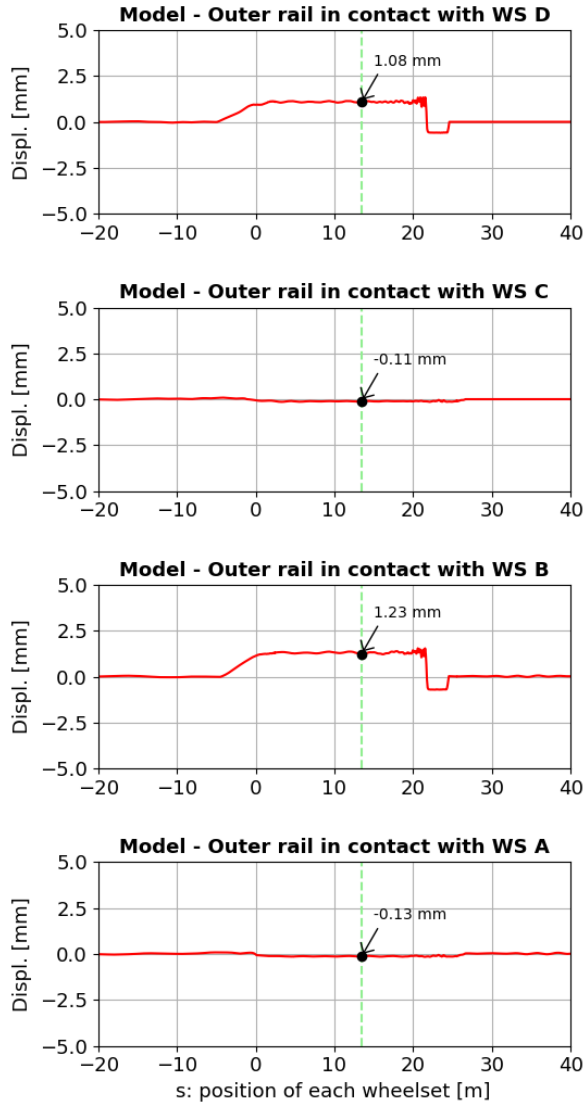


Fig. 6.17 Trailing move: numerical simulation results validating the direction of outer rail displacements at 25 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign².

²The model obtains forces as outputs in the wheel/rail contacts. These forces are divided by the lateral stiffness of the rail to obtain equivalent displacements.

During the curved section, the rail displacement predicted by the model remains relatively stable and consistent throughout most of the curves (Figures 6.16, 6.17, 6.18 and 6.19). Small oscillations are observed, but these were not captured nor analyzed during the experimental phase, and are therefore considered outside the scope of this study. To reduce their influence and obtain a robust value for comparison across all velocities and directions, the displacement was averaged over a window of ± 1.5 m around the position corresponding to the sensor location in the experimental setup. This region lies within the pure curve, where the rail behavior is relatively uniform.

The comparison with experimental data shows that the first peaks recorded by the DS sensor align well with the model predictions, being the curve of the 1st WS of the facing move above the 1st WS of the trailing move with an approximate difference of 0.25 mm.

However, the model suggests that the displacement caused by the 3rd wheelset during a trailing move is similar to that caused by the 1st wheelset during a facing move, and vice versa. This asymmetry indicates that, in the central part of the curve, the influence of wheelset order and movement direction is limited and its influence may mainly come from the location of the center of mass, that is analysed later in Section 6.4.

Based on the values obtained by the averaged displacements, the contribution of the lateral displacement of the outer rail to the total gauge variation was estimated for each simulation (see Appendix B for all the results). At 25 km/h, the model obtains a gauge widening of 2.07 mm, while the lateral displacement of the outer rail alone accounts for 1.31 mm, corresponding to 63,20% of the total gauge variation. This confirms the higher influence on lateral forces of the wheel whose flange interacts with the rail, which can serve as an indication of the locations where dynamic effects are more pronounced.

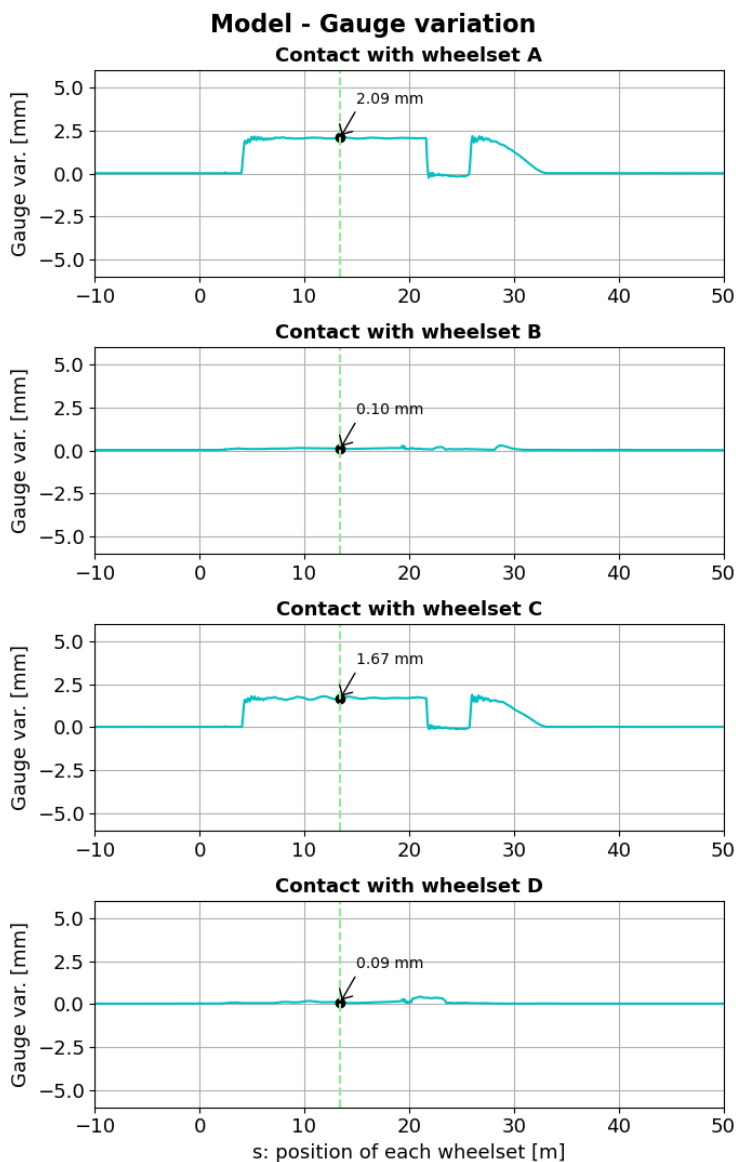


Fig. 6.18 Facing move. Gauge variation at the contact point with each wheelset (A to D) obtained by the model for a train velocity of 25 km/h. The green vertical dashed line indicates the equivalent sensor position during the experimental campaign.

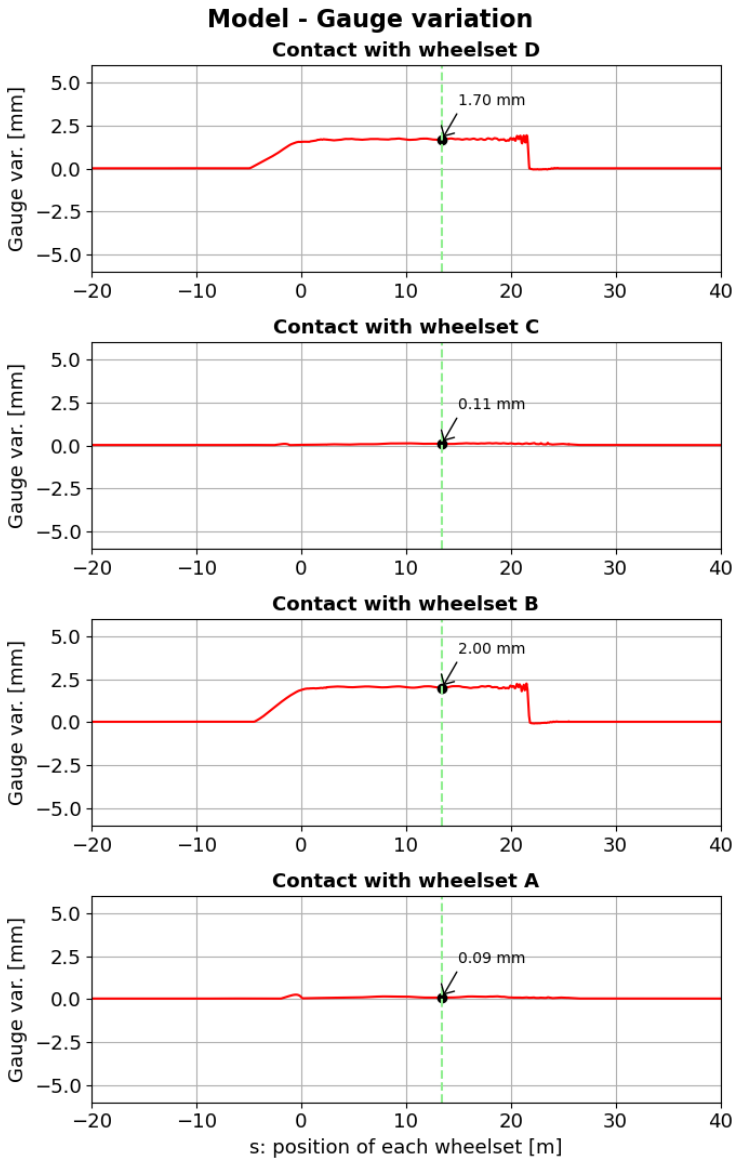


Fig. 6.19 Trailing move. Gauge variation at the contact point with each wheelset (A to D) obtained by the model for a train velocity of 25 km/h. The green vertical dashed line indicates the equivalent sensor position during the experimental campaign.

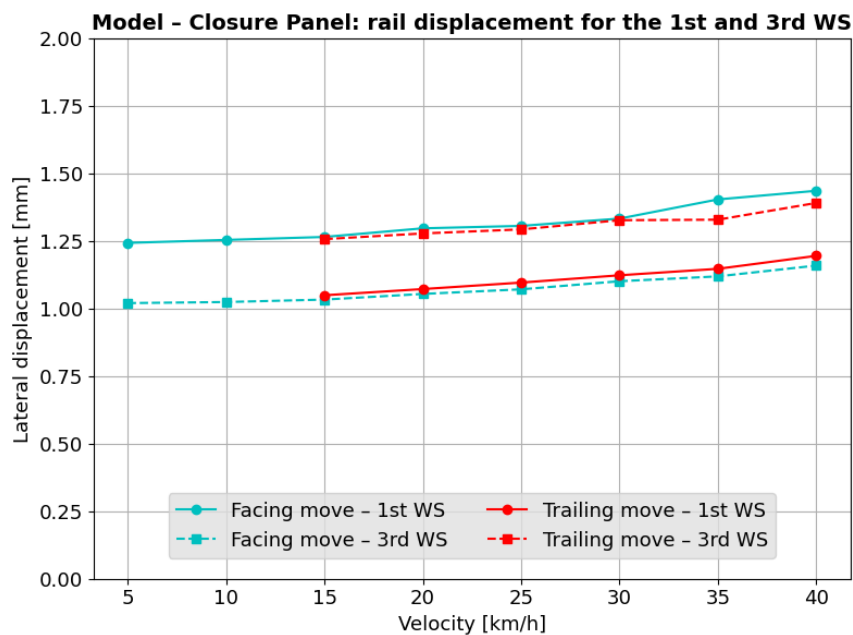


Fig. 6.20 Comparison of the lateral displacement of the outer rail at the longitudinal position equivalent to the experimental sensor location, during wheel flange/rail contact for the 1st and 3rd wheelsets, for both move directions: facing and trailing.

6.2.3 Numerical exploration

The same model used previously, referred to here as **R215**, is employed again, but with a modification to the entire curved section. The original curve (Closure Panel of 215 m and a smaller portion of the Crossing Panel with a 208 m radius) is replaced by a curve with a radius of 500 m — representative of certain S&C configurations installed in Belgium, and referred to here as **R500** (see Fig. 6.21).

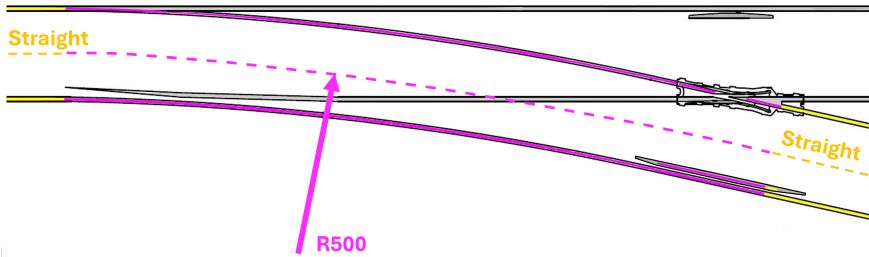


Fig. 6.21 S&C geometry with a curved section of 500 m radius, considered as an alternative configuration in the numerical model.

Observations from Figures 6.22 and 6.23 show more stable values that increase less with speed (see analysis in Section 6.3) and remain lower than those obtained with the previous model, R215.

The flange contact is less pronounced in this configuration. At 25 km/h, the contribution of the outer rail — the one interacting with the wheel flange — to the total gauge variation (Table 6.1) is lower in percentage and also as an absolute value than for the real configuration for both moves.

This suggests that tighter curves, which require more active guidance, tend to induce stronger flange contact. Consequently, this increased interaction result in higher lateral forces and may lead to accelerated wear.

Table 6.1 Contribution of outer rail displacement to total gauge variation at 25 km/h for two type of configurations: radius of 215 m and radius of 500m.

Parameter	R215		R500	
	Facing	Trailing	Facing	Trailing
Outer rail displacement [mm]	1.307	1.294	1.106	1.098
Gauge [mm]	2.068	2.055	1.890	1.883
Outer rail / Gauge	63.20%	62.98%	58.52%	58.33%

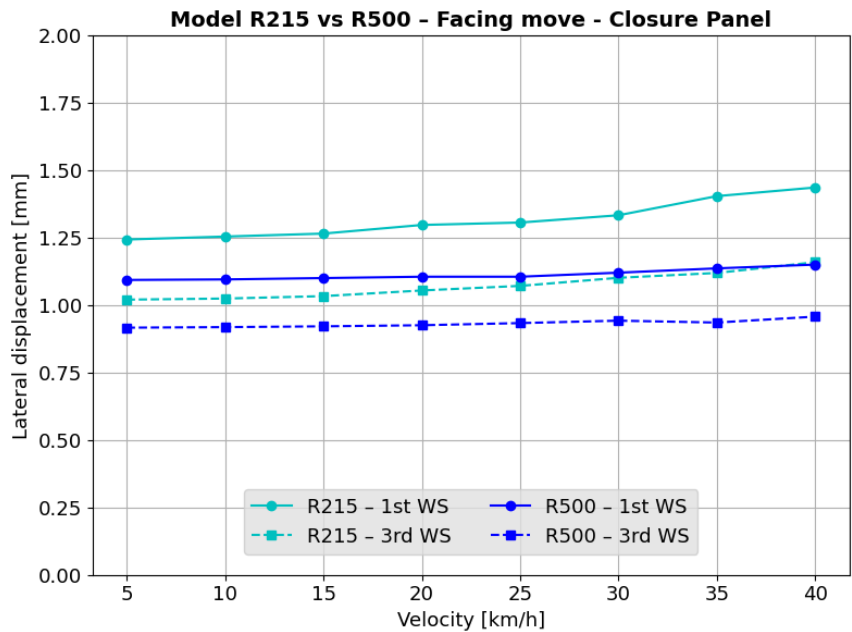


Fig. 6.22 Facing move: comparison of the lateral displacement of the outer rail in the Closure Panel during wheel flange/rail contact, for two curve configurations: 215 m and 500 m radius.

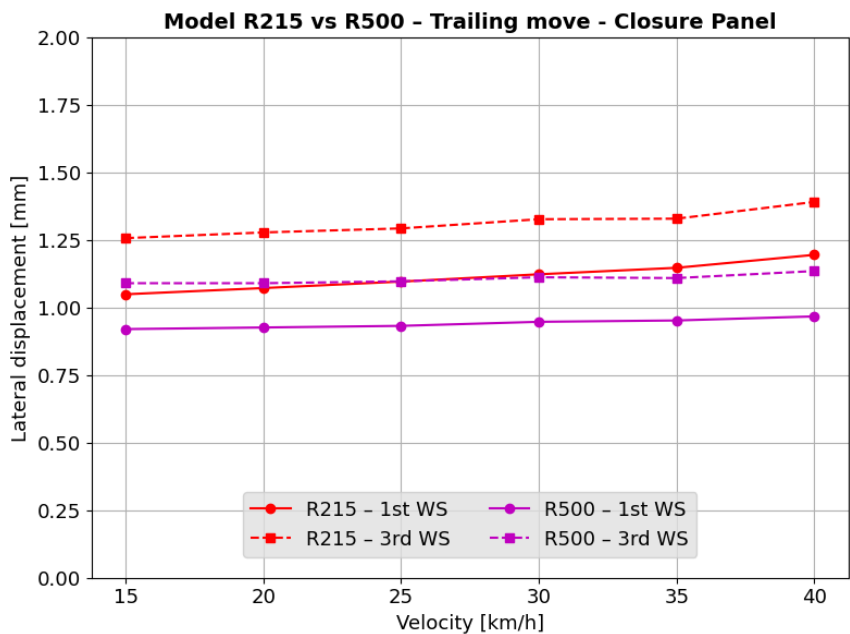


Fig. 6.23 Trailing move: comparison of the lateral displacement of the outer rail in the Closure Panel during wheel flange/rail contact, for two curve configurations: 215 m and 500 m radius.

6.2.4 Dynamic vs static: discussion & analysis

Influence of the trajectory and position of the measurement system

The results obtained show that the perception of geometric variations in track profile is strongly influenced by both the train trajectory and the position of the on-board measurement system (SMS). The SMS is located close to the wheelset D. In the facing move, it means that it is closer to the 4th wheelset (trailing WS), where lateral displacements of the rail are low. In contrast, in the trailing move, the wheelset D becomes the 1st WS (leading WS) where lateral displacements are higher due to contact between the wheel flange and the outer rail. This difference in positioning may explain why the SMS tends to underestimate actual deformations depending on the train's direction.

Based on the model and sensor data, it is estimated that the lateral displacement of the outer rail represents approximately 63% of the total gauge variation in the actual configuration (radius of 215 m) for velocities around 25 km/h. For example:

- In the facing move at 25 km/h, the SMS does not detect the peak displacement of 1.18 mm recorded by the outer rail sensor, representing a difference in the measurement up to 1.87 mm in the gauge measurement, as it can be seen in the Fig. 6.24.
- In the trailing move, the difference between the peak displacement recorded by the sensor and the value measured by this sensor when the SMS is aligned with the sensor position is approximately 0.76 mm, which corresponds to a difference in the measurement up to 1.21 mm in the gauge measurement.

These discrepancies show that the SMS, being located away from the actual wheel-rail contact point, does not capture these peaks. If the objective is to detect the maximum dynamic effect, the measurements would underestimate the true track deformations. On the other hand, if the goal is to measure the geometry while minimizing dynamic effects, the SMS readings would be closer to the static measurements obtained with the digital ruler.

Experimental limitations

The repeatability of the measurements was limited, with only one or two repetitions per speed. Although efforts were made to minimize the effects of passage order, this limitation must be taken into account when interpreting

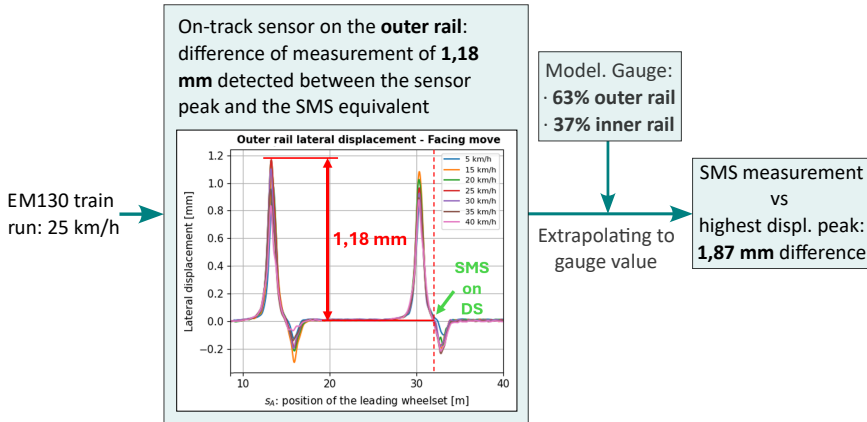


Fig. 6.24 Facing move: schematic representation justifying the highest difference between the on-track sensors and the SMS, based on on-track displacement measurements, model-computed outer and inner rail displacements, and the relative position of the train's SMS.

the results. Statistical analyses (see Appendix C) confirm that, after the first few passes, the system stabilizes and the variations are mainly due to trajectory rather than dynamic effects.

Comparison between dynamic and static data

Thanks to the analysis of the trajectory influence in the Closure Panel by combining the experimental results (digital ruler, sensors, and SMS) and the numerical model, it is possible to explain part of the gap between static and dynamic measurements. In particular, the difference between static measurements and loaded rail measurements depends on whether flange contact occurs:

- **Static vs quasi-static reference without flange contact:** in the absence of contact between the flange and the rail (e.g., on the 2nd and 4th wheelsets — the trailing wheelsets of each bogie), the outer rail tends to move inward, both in facing and trailing movements. This can be verified by the measurement sensors in the through route, where the rails move inward. Sensor data and numerical modeling for the diverging route also show inward movement of the rail for the 2nd and 4th wheelsets. In the facing move on the diverging route, the SMS is close to the 4th (trailing) wheelset and detects a decrease in gauge.

- **Static vs quasi-static reference with flange contact:** in the presence of contact (1st and 3rd wheelsets), the outer rail moves outward, contributing to a widening of the track gauge. This behavior is confirmed by on-track sensors and the numerical model. In the trailing movement, the SMS, being close to the 1st (leading) wheelset, generally tends to record an increase in gauge.

These observations highlight that the dynamic behavior of the rail in the Closure Panel is less influenced by the train's move direction, and more by the relative position of the wheelsets and the presence or absence of flange contact. Therefore, the interpretation of dynamic measurements must take into account the train configuration and the location of the measurement system, especially when comparing them to static references.

6.3 Velocity influence

To study the influence of velocity on the track dynamic behavior in the Closure Panel, all measurements from the experimental campaign are analyzed and compared with the model simulations, which allow extending the analysis to velocity conditions beyond those tested experimentally.

6.3.1 Experimental Results

According to the graph in Fig. 6.10, which shows the lateral displacement peaks of the outer rail for the leading wheelsets of each bogie (1st and 3rd), an increase in peak displacement is observed up to 25 km/h, followed by a decrease up to 40 km/h.

However, when the peaks are ordered by the sequence of measurement rather than by the theoretical train passage speed (Fig. 6.25), a particular behavior appears. For the first five passages (each including a trailing and a facing move, in that order), the lateral displacement peak tends to increase with each passage, particularly for the trailing move. From the sixth passage onward, the curves tend to stabilize, reaching a steady state. A statistical analysis using ANOVA, presented in Appendix C, shows that the measurement order is statistically significant for the initial passages. For the last four passages, however, the order is no longer significant.

Some hypotheses may help explain this phenomenon. The first is a stabilization effect of the track after a transient phase during the first five passages. The diverging route of this turnout (AW15984) is very rarely used, with a train passing approximately once every two days. Therefore, a possible stabilization phase — i.e., due to slight track settlement that increases track stiffness — could explain this behavior.

A second hypothesis is a possible learning effect in the train driving behavior, including how the train enters the turnout curve and adjusts to the target speed. Examples can be seen in Fig. 6.26 for the first passage at 40 km/h and in Fig. 6.27 for the second passage at the same theoretical speed.

During the first 40 km/h passage, the speeds in the Closure Panel — which are below the target value and higher in the facing move — can lead to increased centripetal forces in the facing move and are not constant, particularly in the trailing move where the train is accelerating. This variability may induce longitudinal forces that, in turn, affect the lateral forces when the 1st and 3rd wheelsets are in flange contact and slightly crabbing relative

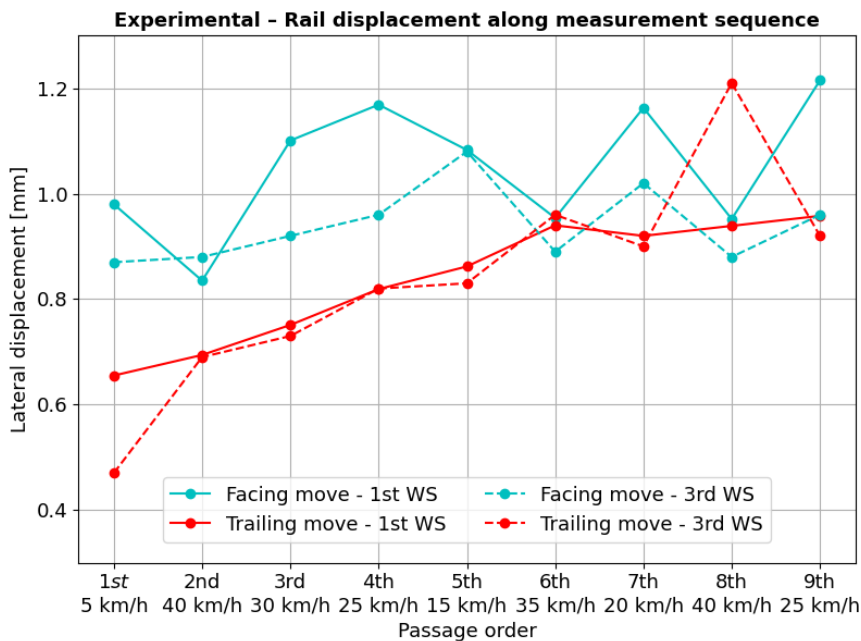


Fig. 6.25 Experimental lateral peak measurements of the outer rail of the curve as a function of train velocity. The peaks are ordered according to the sequence of measurements, not by increasing speed.

to the rail.

In contrast, during the second passage, the facing speed is lower than the trailing speed. While the train is slightly accelerating in the trailing move, it is decelerating in the facing move. Compared to the first 40 km/h passage, the lateral forces induced in the trailing move could be higher in this second run, which may explain why the corresponding displacements in the 2nd 40 km/h run (Fig. 6.25) are larger.

However, due to the limitation in repeatability of the passages, no definitive conclusions can be drawn. A more in-depth study of this phenomenon would require multiple repetitions of each passage.

To sum up, no clear velocity-related factor could be identified through either the sensors or the SMS system.

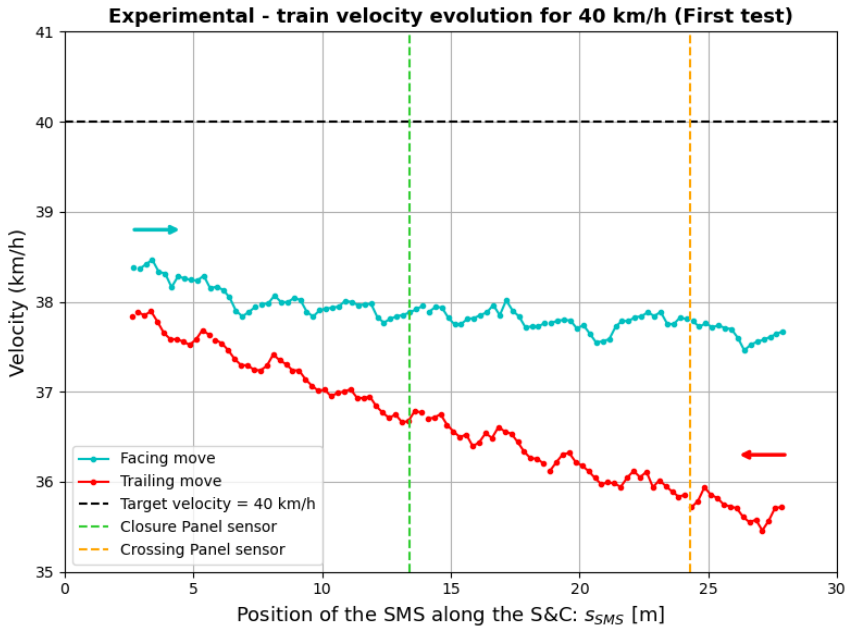


Fig. 6.26 Real train velocity for the first test at a theoretical velocity of 40 km/h. The locations of the main sensors analysed in this thesis are indicated by vertical lines.

6.3.2 Model results

The lateral displacement peak of the outer rail increases with velocity. This increase is primarily attributed to the effect of the centripetal force. Its theoretical values are calculated in Appendix A, based on the assumption that one half of the train carries 55% of the total weight (the half containing wheelsets A and B), while the other half carries 45% (the half with wheelsets C and D).

The lateral forces acting on the wheels of the heavier half located in the curve — the ones from wheelsets A and B — are obtained and presented in Table 6.2.

As described in detail in the appendix, the theoretical centripetal total force on the heavier bogie (the one containing wheelsets A and B) was calculated for 5 km/h and 40 km/h. The resulting values are 0.32 kN and 20.88 kN, respectively — the former representing only 0.16% of the latter.

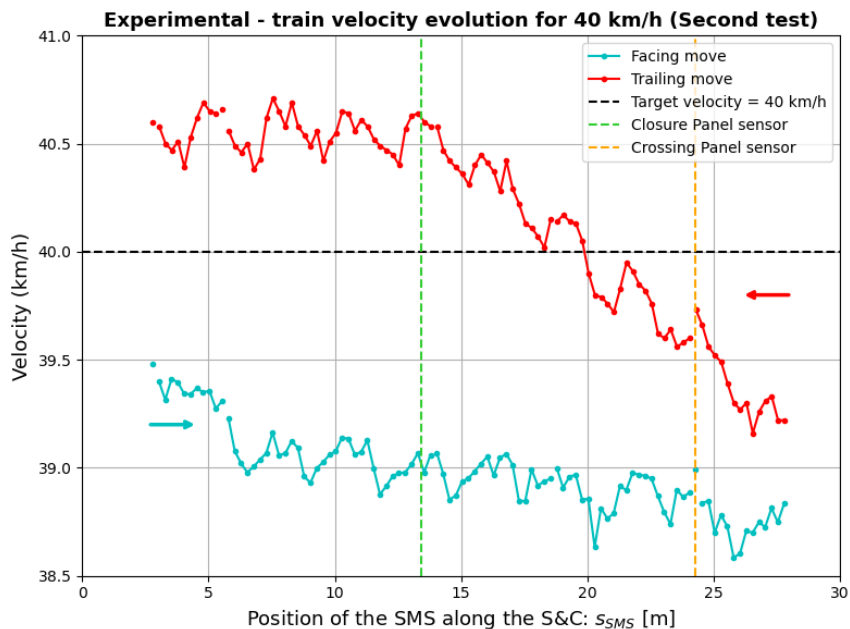


Fig. 6.27 Real train velocity for the second test at a theoretical velocity of 40 km/h. The locations of the main sensors analysed in this thesis are indicated by vertical lines.

Similarly, in the numerical model, the total lateral force from the bogie at 5 km/h (0.19 kN) is significantly lower, corresponding to just 1.1% of the force observed at 40 km/h (17.48 kN). This large difference in force magnitude is directly reflected in the behavior of the rail.

Although no flange contact occurs on the inner rail, a slight decrease in lateral force — and consequently in lateral displacement of the rail — is observed as train speed increases (Appendix B). This behavior is consistent with the redistribution of centripetal load: at higher velocities, the vehicle tends to shift outward in the curve, increasing the load on the outer rail while reducing the lateral pressure on the inner rail.

Quantitatively, the lateral displacement of the outer rail at 40km/h obtained from the simulations is approximately 0.19 mm higher than the quasi-static reference at 5 km/h. However, when considering the gauge — that is, the difference between the outer and inner rail, which is the

Table 6.2 Model lateral force results - Heavier half in the curve (R215, facing move)

Velocity	WS	Outer Rail [kN]	Inner Rail [kN]
5 km/h	1 st (A)	43.53	-26.90
	2 nd (B)	-5.30	-11.14
	Total force	0.19 kN	
40 km/h	1 st (A)	50.29	-24.07
	2 nd (B)	-2.81	-5.93
	Total force	17.48 kN	

parameter of interest for Infrabel — the difference is 0.11 mm higher. This highlights that, although the dynamic effect of velocity up to 40 km/h does exist, its impact on rail deformation remains limited.

6.3.3 Numerical exploration

In a numerical exploration of the train passage, simulations were performed for velocities higher than those used in the experimental campaign — i.e., above 40 km/h — and are shown in Fig. 6.28. Up to speeds of 30 or 40 km/h, the graph exhibits an approximately linear trend. However, for higher velocities, the curve follows an approximately quadratic shape. This behavior can be explained by the effect of the centripetal force, as it is proportional to the square of the velocity and, therefore, it becomes more important at higher speeds.

It is also interesting to note that when the velocity tends toward zero, the displacement values do not converge to zero, which was also observed in the experiment. In the model results, the maximum displacements stabilize at around 1.25 mm for both moves. This value, which depends only marginally on speed since the centripetal acceleration is nearly zero, appears to be quasi-static in nature. Therefore, it results mainly from the way the wheels apply lateral forces on the rails through tread creep and from the lateral support generated by flange contact.

Regarding the redistribution of lateral forces, the influence of the outer rail on the total gauge variation increases in a quasi-quadratic way for velocities above 35 km/h. As observed in the previous section, the displacement of the inner rail also decreases, but more significantly at these higher speeds. For example, at 70–80 km/h, in both moves — facing (Table 6.3) and trailing (Table 6.4) — the outer rail displacement accounts for approximately 80% of

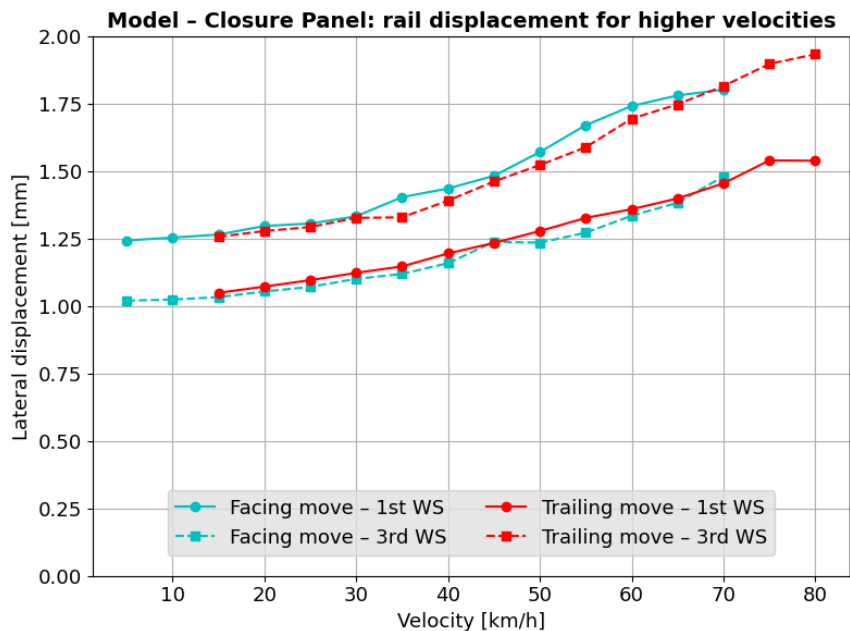


Fig. 6.28 Comparison of the lateral displacement of the outer rail at the longitudinal position equivalent to the experimental sensor location, during wheel flange/rail contact for the 1st and 3rd wheelsets, at speeds above 40 km/h, for both move directions: facing and trailing.

the total gauge variation.

To study the influence of velocity using the S&C model, S&C with a 500 m curve radius — referred to here as R500 — was selected, as was done in the numerical exploration of the trajectory. For this analysis, the outer rail displacement graphs for the facing move (Fig. 6.22) and the trailing move (Fig. 6.23) from Section 6.2.3 are revisited.

It can be observed that between 5 and 40 km/h, the displacement difference for R500 is only 0.06 mm, indicating that the influence of speed is almost negligible compared to the R215 case, which is consistent with the centripetal force being inversely proportional to the curve radius.

Velocity [km/h]	Outer rail displ. [mm]	Inner rail displ. [mm]	Gauge [mm]	Outer/Gauge [%]
5	1.244	0.768	2.012	61.83
15	1.266	0.766	2.032	62.30
35	1.405	0.726	2.131	65.93
70	1.804	0.449	2.253	80.07

Table 6.3 Facing move. Outer and inner rail highest peak displacement as a percentage of gauge at some speeds obtained from the model simulations. All the cases are in the Table B.1 in Annex B. These values correspond to the contact of wheelset A (the 1st one) with the rail and are averaged around the sensor location used in the experimental tests.

Velocity [km/h]	Outer rail displ. [mm]	Inner rail displ. [mm]	Gauge [mm]	Outer/Gauge [%]
15	1.258	0.761	2.019	62.30
35	1.330	0.766	2.096	63.45
70	1.817	0.485	2.302	78.91
80	1.935	0.430	2.365	81.84

Table 6.4 Trailing move. Outer and inner rail peak displacement as a percentage of gauge at different speeds obtained from the model simulations. All the cases are in the Table B.3 in Annex B. These values correspond to the contact of wheelset B (the 3rd one) with the rail and are averaged around the sensor location used in the experimental tests.

6.3.4 Dynamic vs static: discussion & analysis

The influence of velocity on the dynamic behavior of the switch & Crossing has been analyzed from both experimental and numerical perspectives.

Concerning the experimental point of view, the lateral displacement sensors (DS) recorded displacement peaks that increased during the first train passages, regardless of the theoretical speed. This phenomenon suggests the existence of a transitional phase in which the S&C gradually stabilizes under repeated loads. To a lesser extent, a learning effect from the experiment could also have an influence, as the velocities were not completely constant or homogeneous between the move facing and trailing.

From the sixth pass onwards, the values tend to stabilize. It means that the order of the measurements plays a more significant role than velocity itself. This trend is supported by statistical analysis (ANOVA), which reveal a significance of the order of the measurements in the first passes, but not

in the last ones and not in the velocity.

These results indicate that the repeatability of the measurements is limited. More repetitions would be required to validate if, during this stationary stage, there is a real influence of velocity on the measured displacements.

Nevertheless, experimental measurements confirm that a significant rail displacement exists even at low velocities, which does not tend to zero, indicating a quasi-static component in the track response.

In line with experimental observations, the numerical model also shows a significant force — and corresponding displacement — even at low velocities, driven more by the passage configuration than by speed itself. A clear correlation is observed between increasing velocity and lateral forces, especially in the curve of 215 m radius. For speeds greater than 40 km/h, the curve for lateral displacement of the outer rail takes a quasi-quadratic shape, consistent with the centripetal force formula ($F = mv^2 / R$).

However, when comparing these results with those obtained for the R500 configuration (500 m radius), the dynamic effect of velocity is significantly lower for a larger radius. For example, the difference in displacement between 5 km/h and 40 km/h in the R215 configuration is 0.19 mm, whereas in R500 it is only 0.06 mm. This finding is also consistent with the formula for centripetal force, as the radius is inversely proportional to this force. This indicates that, for larger radii, the train could operate at higher speeds because the lateral forces are lower than for smaller radii such as R215, and the *Nadal criterion* [95] — the ratio L/V between the lateral wheel flange force L and the vertical wheel load V — will also be lower, reducing the risk of wheel climb and derailment.

In summary, although the model shows that velocity causes higher lateral forces and displacements in the S&C curve with a radius of 215 m, for the facing move, the gauge difference between 40 km/h and the quasi-static reference (5 km/h) is 0.11 mm, or 5.6%. This reinforces the idea that, when interpreting the dynamic measurements, velocity has a limited effect for speeds equal to or below 40 km/h. The experimental results also suggest that, for the tested velocities (≤ 40 km/h), the differences between static and dynamic measurements are mainly attributable to quasi-static effects rather than purely dynamic ones.

6.4 Center of mass impact

As a reminder, the train consists of four wheelsets distributed across two bogies. The mass distribution is not symmetrical: approximately 55% of the weight is located in the middle of the train where wheelsets A and B are located, while 45% is in the half where axles C and D are located (see Fig. 6.29). In addition, there is a slight lateral shift of the center of mass (CoM) toward the inner rail.

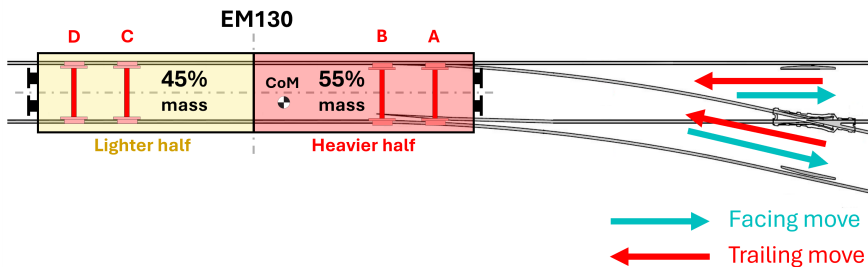


Fig. 6.29 Orientation of the EM130 train during the experimental tests. Wheelsets A and B are located on the heavier half of the train (55% of the total mass, containing the motor bogie), whereas wheelsets C and D are on the lighter half (45%, containing the SMS bogie).

6.4.1 Experimental results

In the experimental results shown in Fig. 6.25, it can be seen that, in the facing move, the 1st WS exhibits greater lateral displacement than the 3rd WS3. This difference coincides with the fact that WS1 is located in the heavier half of the train — motor side. Two factors could explain this difference: the fact that it is the first axle of the train (leadership effect) or that it is located in the heavier half (mass effect).

In contrast, in the trailing move, no significant difference is observed between the 1st and 3rd WS.

6.4.2 Model results and numerical exploration

Previously, it has been shown that the model and the experimental confirm that the leading wheelsets of each bogie (1st and 3rd WS) have flange contact with the outer rail, which generates greater lateral displacements than the

rear wheelsets. In particular, the leading WS of the motor bogie (bogie with WS A and B) shows higher rail displacements. As illustrated in Fig. 6.20, during the facing movement, this corresponds to WS A, which is the 1st wheelset; whereas in the trailing movement, it corresponds to WS B, which is the 3rd wheelset.

A numerical exploration was carried out by reversing the orientation of the train (Fig. 6.30 — an approach not feasible in the experimental setup — and the results are represented in Fig. 6.31 and Fig. 6.32. These simulations show that wheelsets located in the heavier half of the train exhibit higher rail displacements, regardless of whether they are the 1st or 3rd wheelset, or the type of move. In contrast, wheelsets in the lighter half show lower displacements across all configurations. This confirms that mass distribution has a stronger influence on lateral rail displacement than either the train orientation or the position of the leading wheelset within the train.

To better illustrate the equivalence observed among leading wheelsets under different move configurations, Table 6.5 groups the cases according to train mass distribution, move direction, and orientation, indicating which leading wheelsets exhibit the highest rail displacements.

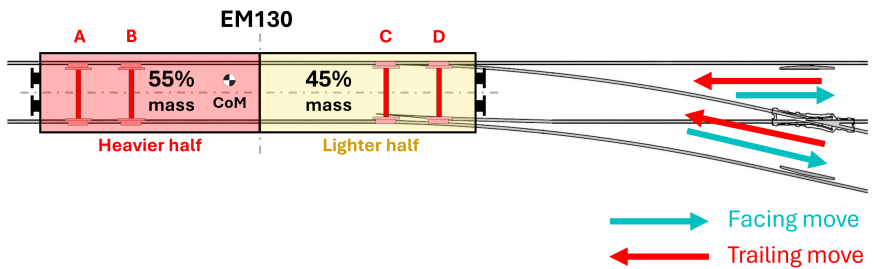


Fig. 6.30 Reversed orientation of the EM130 train simulated by the numerical model.

Additional simulations are performed with the train's center of mass positioned exactly at the geometrical center, both longitudinally and laterally. In Fig. 6.31 and Fig. 6.32, this configuration is represented by the black curves. The results show no significant differences between the wheelsets.

At a low speed of 10 km/h³ — where the dynamic effects of velocity are minimal — the difference between the 1st and 3rd wheelsets is 0.016 mm,

³10 km/h was lower speed that the symmetrical model allowed for the simulations.

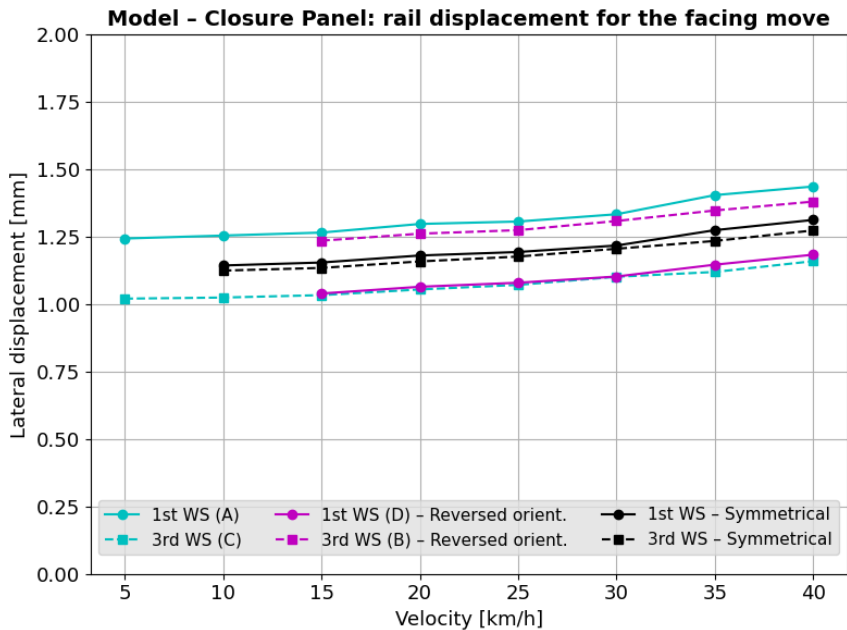


Fig. 6.31 Facing move: comparison of the lateral displacement of the outer rail at the longitudinal position equivalent to the experimental sensor location, for the 1st and 3rd wheelsets, at speeds equal to or below 40 km/h, for both orientations of the train.

which represents only 1.66% of the total displacement of the first wheelset (0.019 mm / 1.144 mm).

In contrast, when using the real center of mass, the difference at 5 km/h — where the dynamic effects of velocity are also minimal — between the leading wheelsets located on the heavier and lighter sides of the train reached 0.223 mm in the facing direction. This corresponds to approximately 18% of the displacement of the first wheelset on the heavier side (0.223/1.244), highlighting the strong influence of mass asymmetry on rail response.

6.4.3 Dynamic vs static: discussion & analysis

The numerical results highlight a strong influence of mass asymmetry on rail lateral response. When the center of mass of the vehicle is longitudinally

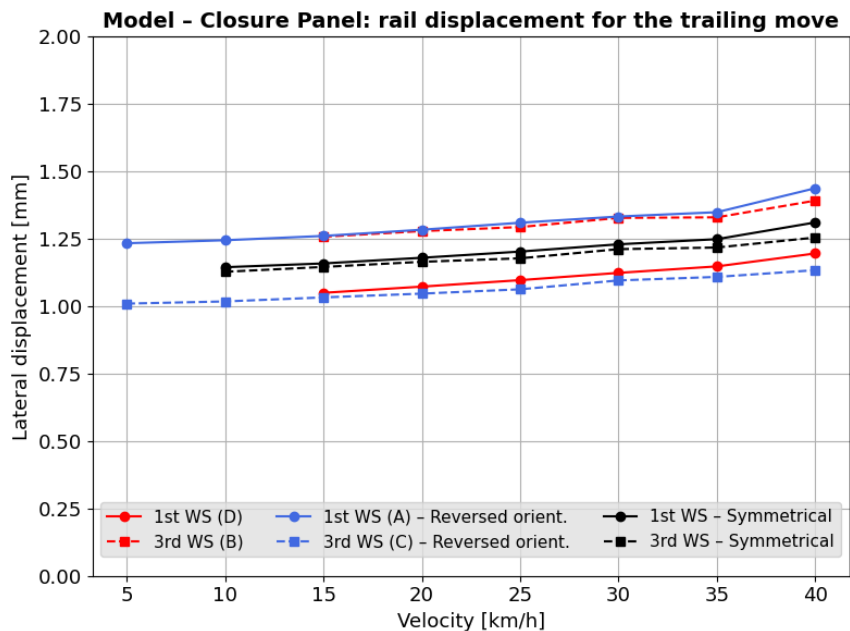


Fig. 6.32 Trailing move: comparison of the lateral displacement of the outer rail at the longitudinal position equivalent to the experimental sensor location, for the 1st and 3rd wheelsets, at speeds equal to or below 40 km/h, for both orientations of the train.

offset, the wheelsets on the heavier side are logically subjected to greater vertical loads. On curved track sections, the leading wheelsets of each bogie (typically the 1st and 3rd wheelsets) are more likely to engage in flange contact, with greater contact forces observed when the wheelset is located on the heavier side.

In the model used, the wheel-rail contact is composed of two different types. The tread contact is based on Kalker's theory [87], which takes into account elastic deformation, contact geometry and friction. The flange contact is modeled as a so-called penalty contact, in which the normal force is proportional to the geometric interpenetration and a contact equivalent stiffness.

The pressure in both contact areas increases when the vertical load increases. In the tread zone, this leads to a larger Hertzian contact patch and

Table 6.5 Leading wheelsets grouped by mass distribution, move direction and rail displacement.

Mass side	Move direction	Wheelset position	Wheelset ID	Rail displacement
Heavier	Facing	1st WS	A	Higher displ.: 1.20-1.45 mm
	Facing - Reversed	3rd WS	B	
	Trailing	3rd WS	B	
	Trailing - Reversed	1st WS	A	
Lighter	Facing	3rd WS	C	Lower displ.: 1.00-1.20 mm
	Facing - Reversed	1st WS	D	
	Trailing	1st WS	D	
	Trailing - Reversed	3rd WS	C	

higher tangential stiffness. According to Kalker’s linear theory, the lateral force F_y is proportional to the lateral creepage ξ_y through the coefficient c_{22} , which increases with vertical load:

$$F_y \approx -c_{22} \xi_y \tag{6.1}$$

In the flange zone, the higher vertical load on the leading wheelset — due the longitudinally offset center of mass — increases the intensity of flange contact. This results in a stronger normal force at the flange-rail contact, which increases the lateral force and causes greater rail displacement in the curve. This effect is confirmed by simulations showing that, at a low velocity of 5 km/h, the difference in rail displacement between leading wheelsets on the heavier and lighter sides reached up to 0.223 mm — approximately 18% of the total displacement. This confirms that, for the studied case, mass distribution has a greater impact on lateral rail displacements than other variables such as wheelset position (1st vs 3rd) or train move direction (facing vs trailing).

6.5 Lateral rail stiffness impact

In this study, the objective is to determine an equivalent lateral stiffness at the wheel/rail contact point, which takes into account the overall response of the railway infrastructure and to study its influence on the rail behavior. The aim is not to break down the individual stiffnesses of each component of the infrastructure (such as seat plates, sleepers, or ballast), but rather to capture the overall behavior as perceived by the train during dynamic passage.

6.5.1 Experimental results

For the experimental analysis of the lateral stiffness of the rails, no train was included in the test setup. Instead, lateral forces were applied using a hydraulic jack.

Curves of applied force for the Closure Panel rails versus lateral rail displacement measured by the LVDT sensors (see location in Fig. 6.33) are shown in Fig. 6.34. The graphs indicate that the rail heads — red and blue — experience greater lateral displacement than their webs — orange and cyan, respectively —, suggesting a rotational behavior or apparent flexion of the rails under lateral loading.

In the case of the sensor placed on the curved closure rail, which corresponds to the outer rail of the curve, the graph shows two different stiffness stages. The first stage is stiffer, up to a lateral displacement of approximately 0.3 mm. Beyond this point, a second stage appears, with reduced stiffness for higher force levels. For the straight closure rail, the stiffness remains approximately constant throughout the test.

Compared to the rails located approximately two meters before the switch — which share the same configuration but lack an elastic pad between the rail and the tie plate (a metal plate positioned between the rail base and the sleeper, serving as a support interface) — different stiffness stages can also be observed (see Fig. 6.35), with a tendency for the slope to decrease for lateral forces greater than 30 kN. This phenomenon could be related to observations in the literature, particularly in Wang (2024) [36] and in Liu (2014) [34] (detailed in Section 2.1). In those studies, different main stiffness stages were identified, mainly a static friction one between the rail or rail pad and the tie plate (stiffer) and sliding one (less stiff). In the study of Wang, a third one is considered: collision between the tie plate and the anchoring bolts (stiffest).

In the present tests, although no rail pad is present, the rail itself may experience a similar mechanical behavior. The curve becomes less rigid for lateral forces greater than 30 kN, which may indicate a sliding phase of the rail on the tie plate. This pattern implies that micro-movements or partial loss of contact may cause the rail-chair interaction to display nonlinear stiffness characteristics even in the absence of an elastic interface.

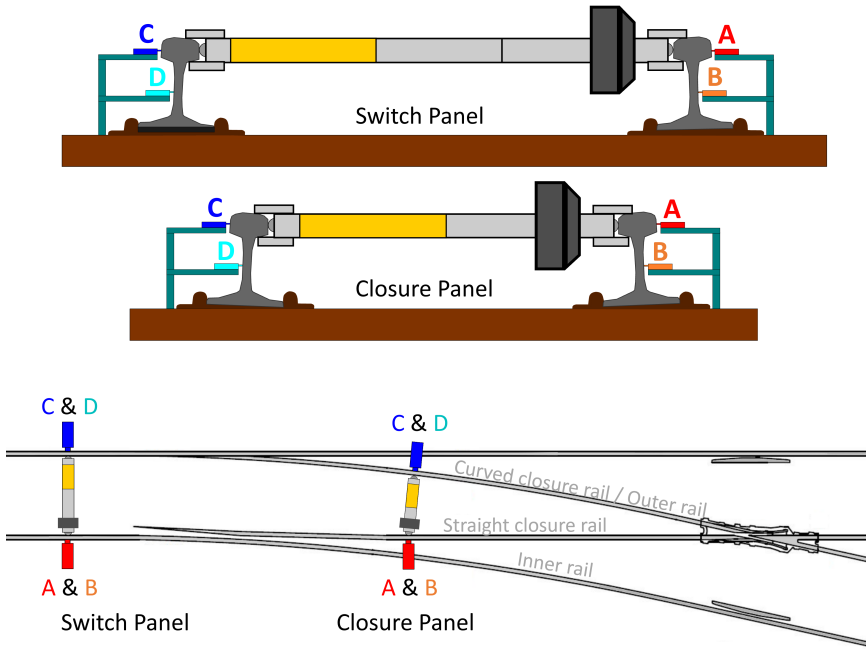


Fig. 6.33 Location of the sensors and the hydraulic jack on the Closure Panel, as well as before the Switch Panel. C and D indicate the head and web of the curved closure rail (outer rail), respectively. A and B indicate the head and web of the straight closure rail or the inner rail.

6.5.2 Model results

The numerical model, when the train passes through a curve, provides maximum lateral force values due to wheel/rail contact, ranging from 43 to 50 kN. Based on the lateral stiffness obtained during the experimental test with the hydraulic jack (Fig. 6.34), this would correspond to the area of the graph showing a stiffness of 22.40 kN/mm.

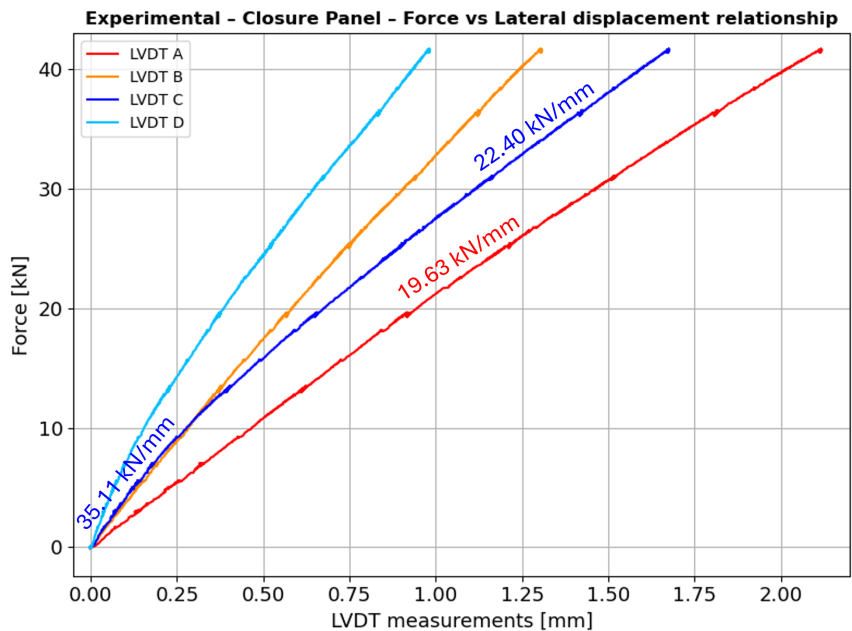


Fig. 6.34 Curves of applied force for the Closure Panel rails versus lateral rail displacement measured by the LVDT sensors. Blue and cyan correspond to the head and web of the curved closure rail (outer rail), respectively; red and orange to the head and web of the straight closure rail.

However, this value does not take into account the influence of vertical loads, since the test was carried out with a jack applying only lateral force. Vertical forces have the effect of stiffening the track laterally [42]. This interaction between the vertical and lateral components of the load was highlighted in Quoidbach’s master thesis [97], as illustrated in Fig. 6.36.

The figure compares the case of rail deformation with only lateral force as a reference (100% deformation) to other cases where, in the presence of identical lateral force, the vertical force increases. It can be seen that the increase in vertical load gradually reduces lateral deformation, particularly for the highest levels of lateral force.

For cases with weaker lateral forces, an increase in vertical force would cause a renewed increase in deformation, but less than in the case of pure lateral force.

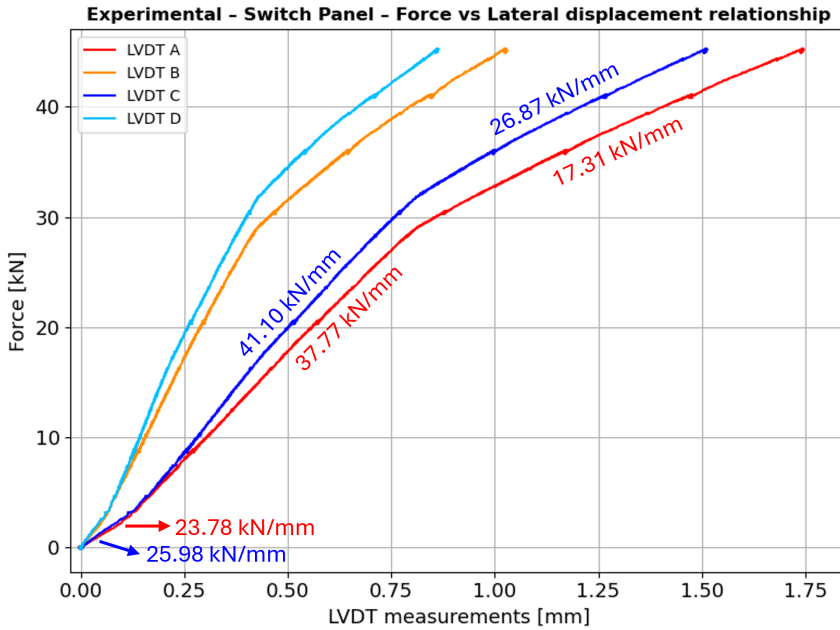


Fig. 6.35 Curves of applied force for the rails before the Switch Panel rails versus lateral rail displacement measured by the LVDT sensors. Blue and cyan correspond to the head and web of the curved closure rail (outer rail), respectively; red and orange to the head and web of the inner rail.

The lateral stiffness of the model is therefore calibrated mainly on the basis of the results obtained during the dynamic test with the measuring train, with the aim of obtaining lateral displacement amplitudes, considering a vertical charge, and of the same order of magnitude as those observed experimentally. The objective is not to reproduce the exact peak values measured, but to capture trends and their evolution.

Additionally, it should be noted that the stiffness values proposed in the literature are considered, which generally vary between 20 and 50 kN/mm (see Chapter 2). Finally, the value adopted for the model of the present project is 35 kN/mm.

To further investigate the influence of different stiffness regimes observed experimentally, additional cases with varying lateral stiffness values of the rail were simulated. In each simulation, both main rails were assigned

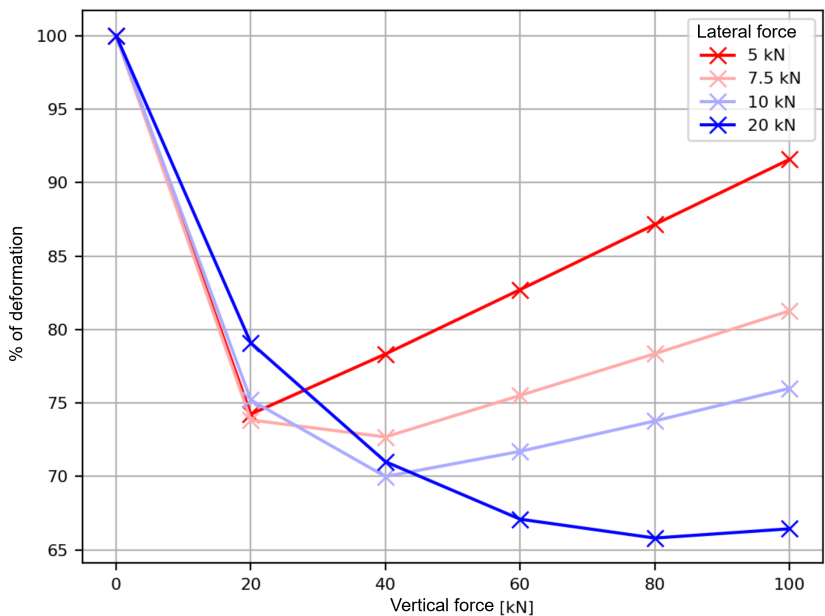


Fig. 6.36 Influence of vertical load on lateral displacement of the track. Result extracted from Quentin Quoidbach’s thesis (2025). It illustrates the interaction between the vertical and lateral components of the load, used in the AIGUIDYN project.

the same constant stiffness value along their entire length. The train velocity was set to 30 km/h for all cases, and both facing and trailing moves were considered. The average gauge variations were extracted for wheelsets A and C in the facing move, and wheelsets D and B in the trailing move.

The results in Fig. 6.37 show that the gauge displacement follows a decreasing exponential curve as a function of lateral stiffness, which is consistent with Hooke’s law, where displacement is inversely proportional to stiffness for a given force.

6.5.3 Dynamic vs static: discussion & analysis

The experimental results show that the track lateral stiffness depends on the value of the applied lateral force. This effect is observed in the given structure of the track used in this study — consisting of the rail, tie plate

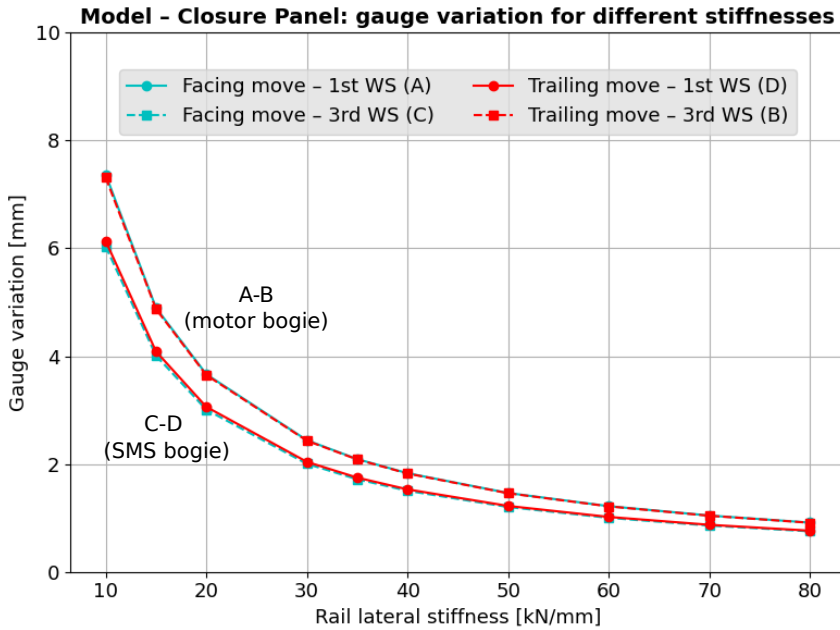


Fig. 6.37 Gauge increase for both facing and trailing moves, corresponding to the passage of the 1st and 3rd wheelsets (WS) when they reach the longitudinal position equivalent to the sensor location in the experimental setup. For higher rail lateral stiffness values, gauge increases are lower, following a decreasing exponential curve. A and B correspond to the motor bogie (heavier half), while C and D correspond to the SMS bogie (lighter half).

and sleeper — where a reduction in stiffness occurs when lateral forces exceeding approximately 30 kN.

A vertical force, when combined with a lateral one, can have an important influence on the lateral stiffness of the track. As seen in the literature, and in the study by Quoidbach [97], vertical forces tend to increase the lateral stiffness of the rail. This interaction between vertical and lateral loads is important for understanding the dynamic response of the track under real operating conditions. For this reason, a higher value than that measured by the static experimental campaign is considered to model the lateral equivalent stiffness of the wheel flange/rail contact. The lateral stiffness

used then in the base model (35 kN/mm) was defined by calibrating the displacements of the dynamic experimental campaign to obtain a similar order of magnitude of the displacements, taking the stiffness measured by the hydraulic jack as a reference.

The combination of static measurements with a hydraulic jack, which applies lateral forces only, with dynamic ones obtained during the passage of trains, allowed the development of a criterion for the calibration of stiffness in the numerical model. As a summary, while the static experimental campaign on track stiffness provides relevant insight into the mechanical behavior of the rail under static lateral forces, dynamic measurements better reflect the overall response of the infrastructure under train loading conditions.

7

Results in the Crossing Panel

THIS chapter presents the main findings from the experimental and numerical model data in the Crossing Panel, adjacent to the Closure Panel, which includes critical components such as the frog and the check rail. The analysis is structured into four complementary sections.

They focus on the analysis of the effects of key variables on the dynamic behavior of the track in this specific area. The analyzed variables are: train trajectory (**Section 7.1**), velocity (**Section 7.2**), center of mass location (**Section 7.3**), lateral track stiffness (**Section 7.4**) and worn check rail (**Section 7.5**).

Some global analyses already detailed in the previous chapter — particularly those concerning data processing and interpretation — are not repeated here, but referenced when necessary, to facilitate reading and avoid redundancy.

As done in the previous chapter, the experimental data analyzed in this chapter is grouped into two main categories: on-track systems — LVDTs and digital ruler — and on-board systems — Switch Measurement System (SMS) and on-board camera recordings. The results from these are combined with the numerical model outcomes to identify behavioral patterns, validate hypotheses, and propose guidelines for improving the understanding of switch & crossing dynamics in the Crossing Panel.

As an introductory reminder for reading the results, as illustrated in Fig. 8.1, in the facing move the 1st, 2nd, 3rd, and 4th wheelsets (WS) corre-

spond to A, B, C, and D, respectively. In the trailing move, they correspond to D, C, B, and A, respectively. For this case, when the graphs are plotted as a function of the wheelset position, they should be read from right to left. In both moves, the 1st and 3rd WS are the leading ones from each bogie.

7.1 Trajectory influence

This first section from this chapter analyzes the influence of the train trajectory — facing or trailing move — on variations in the check rail, and consequently on the flangeway gap, within the Crossing Panel. It focuses on the effect of track curvature prior to this area. The study combines experimental measurements with numerical simulations that allow for analysis beyond the theoretical configuration.

7.1.1 Experimental results

Sensors on the track

The DS installed in the crossing panel (see Fig. 7.1-2nd and Sect. 3.2.1) measures the lateral displacement of the check rail and the nearest running rail at the same longitudinal position. The sum of these displacements corresponds to the variation of the flangeway gap — the distance between the check rail and the nearest running rail — as shown in Fig. 7.1 for a train velocity of 30 km/h in the facing move. It should be noted that, for each move — facing and trailing —, the shape of the curve remains consistent across all velocity regimes, although the specific shape differs between the two moves.

When the leading wheelset **A** reaches the crossing gap, the check rail undergoes a lateral displacement of 1.0 mm outward from the gap, likely due to contact with the inner surface of the wheel flange (see Fig. 7.1-1st). Simultaneously, the nearest running rail also experiences a displacement outward from the gap, which contributes to widening the flangeway gap to nearly 2.5 mm.

When the second wheelset **B** reaches the crossing gap zone, the flangeway gap narrows. In this case, the check rail is slightly displaced inward, around 0.1 mm, and it contributes to the narrowing. Moreover, the nearest running rail is displaced inward, so it also contributes to the reduction the flangeway gap.

During the passage of the third wheelset **C**, that corresponds to the leading wheelset of the second bogie, both the check rail and the nearest

running rail present a similar behavior to that observed during the passage of the first wheelset **A**. Therefore, the flangeway gap widens.

Finally, during the passage of the fourth trailing wheelset **D**, the check rail suffers a slight displacement inward, toward the flangeway gap. However, the nearest running rail experiences a more significant displacement outward, eventually causing the flangeway gap to widen again.

For the trailing move, the train approaches from the right side of the figure, so the graph in Fig. 7.2 should be read from right to left, as in the previous sections.

Similarly, when the leading wheelset **D** reaches the crossing gap, the check rail undergoes a lateral displacement. In this case, the displacement is lower than in the facing move, measuring 0.2 mm, and occurs outward from the gap, probably due to wheel-flange contact. At the same time, the nearest running rail is also displaced outwards, contributing to a widening of the flangeway gap to almost 0.8 mm.

When the second wheelset **C** reaches the crossing gap, the lateral displacement of the check rail is approximately zero, while the running rail continues to move outwards, further widening the flangeway gap.

For the third (**B**) and fourth (**A**) wheelsets, the flangeway gap narrows when both wheelset pass through the crossing gap zone. This narrowing is mainly caused by the inward displacement of the running rail towards the check rail, which reduces the flangeway gap. In addition, between the passage of the third (**B**) and fourth (**A**) wheelsets, there is a noticeable peak in the widening of the flangeway gap, mainly due to the outward displacement of the running rail.

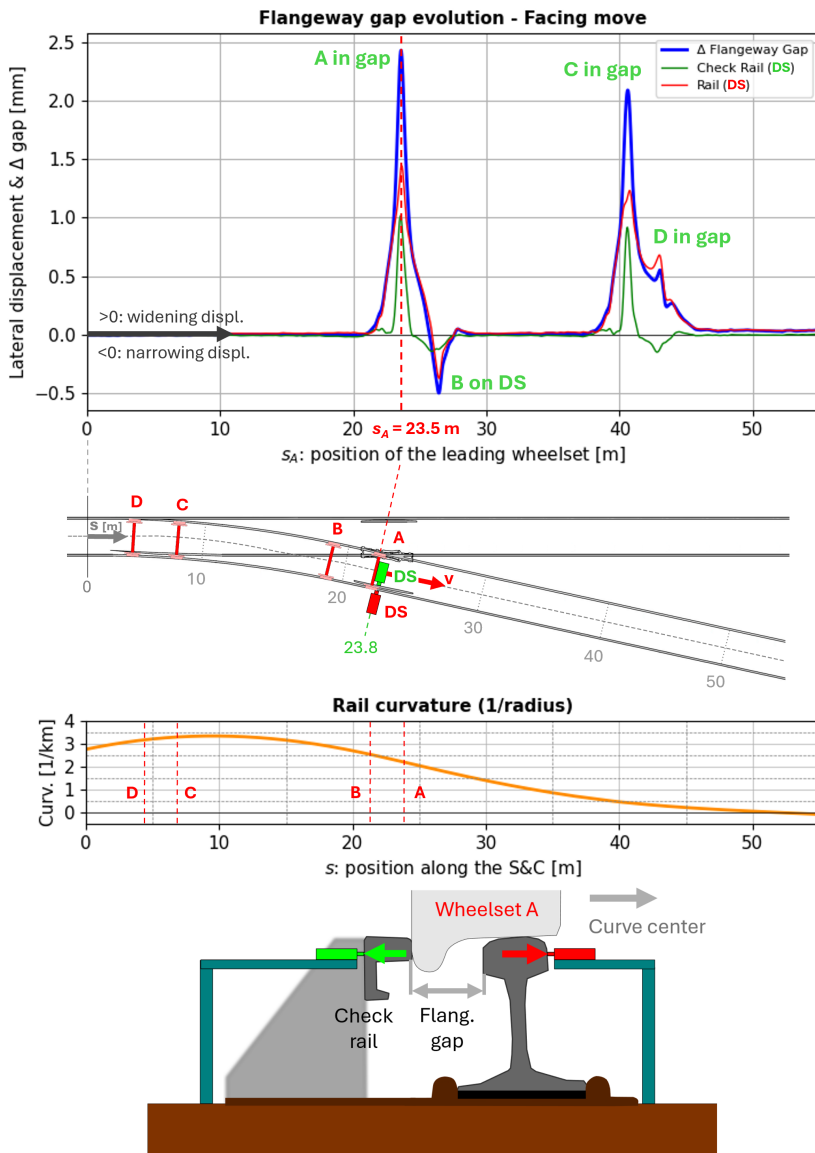


Fig. 7.1 Facing move at 30 km/h: (1st) Evolution of the flangeway gap and lateral displacements of the check rail (DS, green) and the nearest running rail (DS, red). (2nd) Wheelsets position when the leading wheelset is in the crossing gap. (3rd) Rail curvature with the corresponding wheelsets position marked on the graph. (4th) Cross-sectional view of the rail displacements.

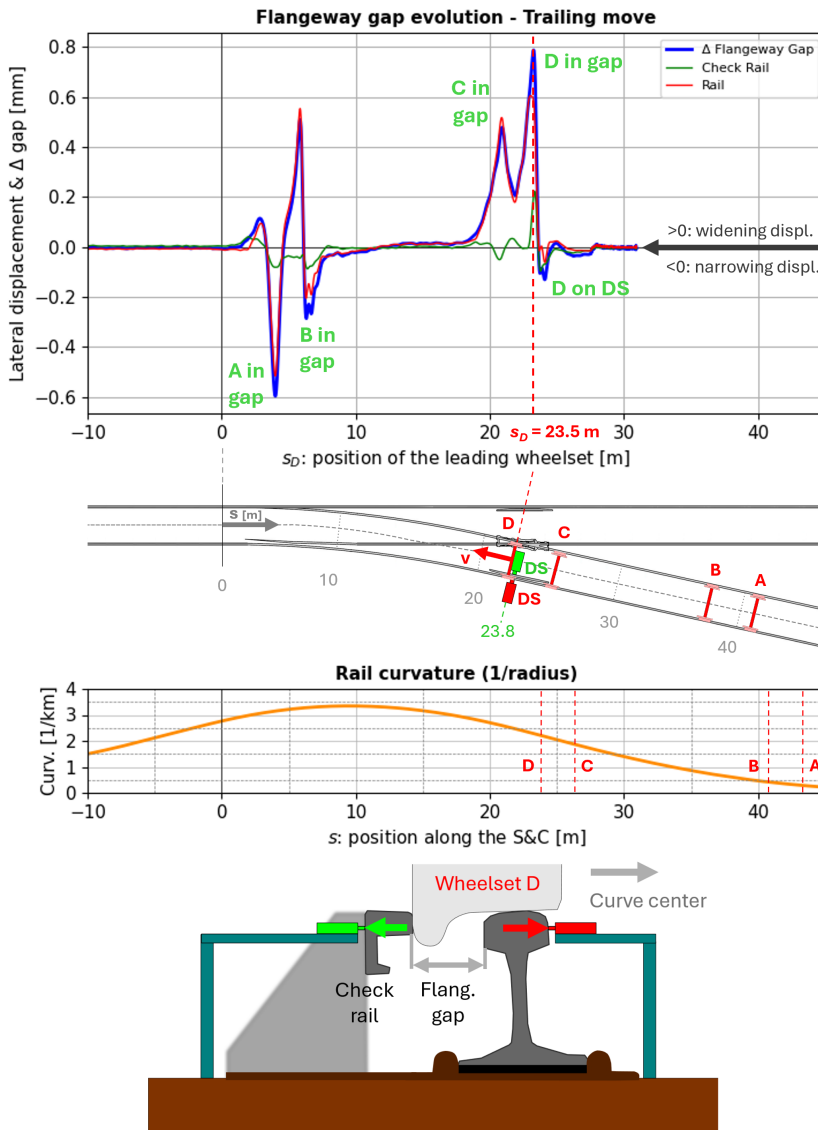


Fig. 7.2 Trailing move at 30 km/h: (1st) Evolution of the flangeway gap and lateral displacements of the check rail (DS, green) and the nearest running rail (DS, red). (2nd) Wheelsets position when the leading wheelset is in the crossing gap. (3rd) Rail curvature with the corresponding wheelsets position marked on the graph. (4th) Cross-sectional view of the rail displacements.

Comparison of the displacement peaks in the diverging route

The lateral displacement peaks of the check rail and the variation peaks of the flangeway gap were analyzed when the leading wheelset reached the crossing gap. Measurements were collected at different velocities and for both facing and trailing moves, resulting in two distinct curves (Fig. 7.3 and Fig. 7.4).

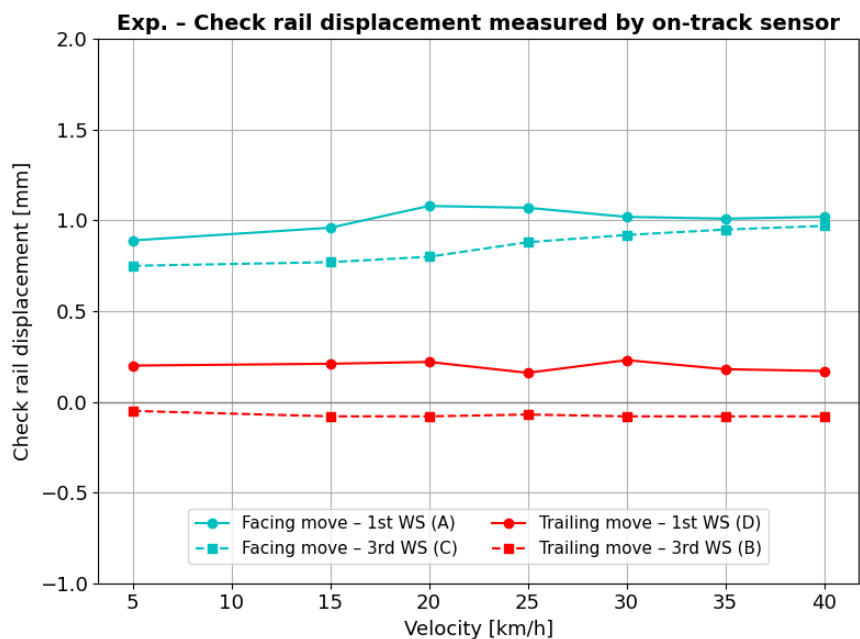


Fig. 7.3 Check rail displacement measured by the DS when the leading wheelsets of each bogie — the 1st and 3rd WS — are aligned with the sensors in the frog area, comparing facing and trailing moves at different speeds.

A clear influence of the train's direction on the measured values was observed: the peaks recorded during the facing move were consistently higher than those during the trailing move. Moreover, the displacements on the check rail and the flangeway gap variation values are always greater for the leading wheelset, the 1st WS, than for the 3rd WS.

As discussed in the closure panel analysis (Chap. 6), one hypothesis to explain this directional influence is related to the trajectory preceding the sensor area. This could be explained by the curvature values of the trajectory

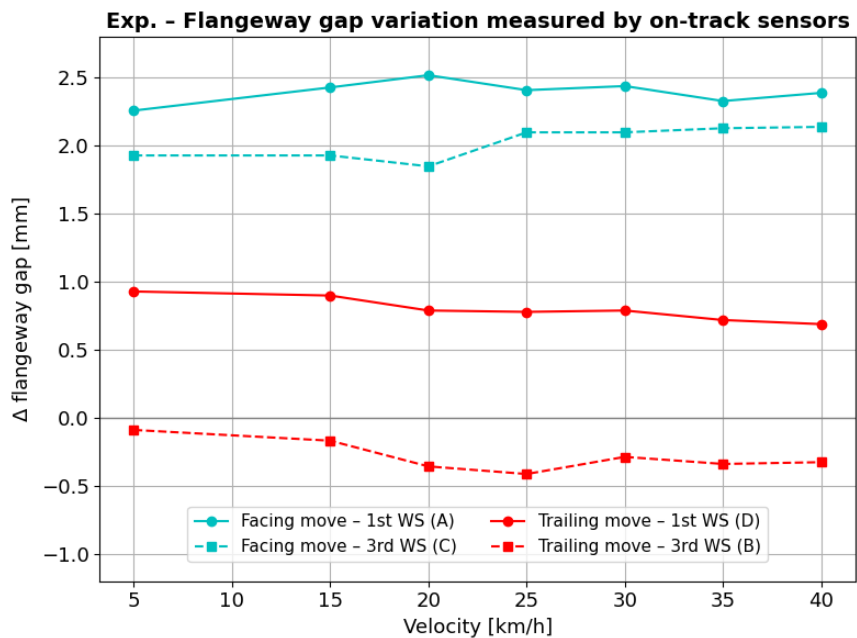


Fig. 7.4 Flangeway gap variation peaks measured by the DS when the leading wheelsets of each bogie — the 1st and 3rd WS — are in the crossing gap of the frog, comparing facing and trailing moves at different speeds.

that the 1st wheelset follows before reaching the crossing gap are higher for the facing move compared to the trailing move. Specifically, when 1st WS is in front of DS, the curvature of the rail in contact with the wheelsets ranges from 2.2 km^{-1} to 3.4 km^{-1} (Fig. 7.1) in the facing move, whereas in the trailing move, it ranges from 0 to 2.2 km^{-1} (Fig. 7.2). The same tendency is observed for the 3rd wheelsets in each move. This difference in curvature may partially explain the observed variations in the flangeway gap values.

Observations from camera recordings

The cameras positioned in front of wheelset D also capture relevant information from the closure panel. Fig. 7.5 presents two frames of the recordings as wheelset D is in the crossing gap zone: the frame on the top corresponds to the facing move and the frame on the bottom to the trailing move. In the trailing move, the wheels are slightly displaced to the left (look at the lateral flange displacement indicated by the arrows in the), suggesting that the flange of the right wheel might be in contact with the check rail.

The left frame corresponds to the local peak labeled "D in gap" in Fig. 7.1. In this case, there is no positive displacement of the check rail, as indicated by the graph, which suggests that there is no contact between the flange and the check rail when wheelset D is in the crossing gap zone. This observation is consistent with the video image, where the wheelset appears more displaced to the right compared to the trailing move.

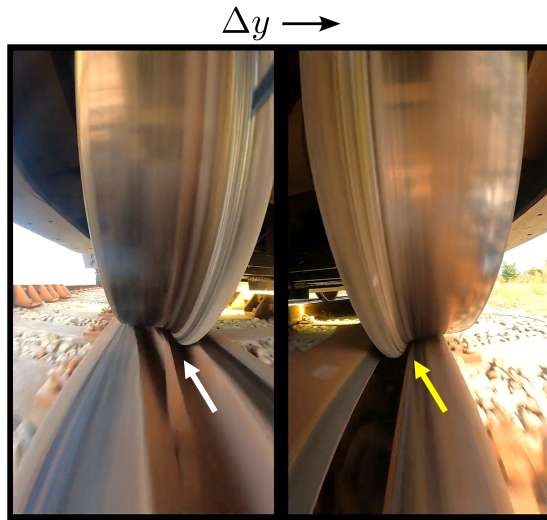
On the other hand, the right frame corresponds to the peak labeled "D in gap" in Fig 7.2, confirming that wheelset D is indeed in the crossing gap zone. Here, a significant peak in the displacement of the check rail is observed, as shown in the graph, indicating contact between the flange and the check rail. This aligns with the image, where the wheelset appears more displaced to the left compared to the facing move.

Through route

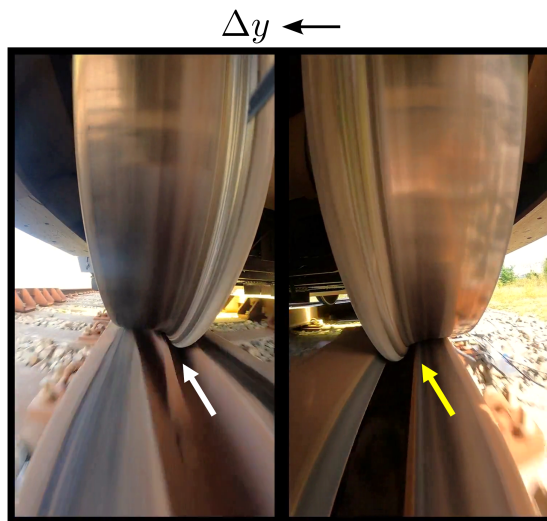
The equivalent LVDT sensor placed on the check rail for the through route is used to analyze the check rail behavior for the straight route (Fig. 7.6). This configuration allows the study of check rail displacement in the absence of flange contact, serving as a reference for comparison with curved route scenarios where some wheels enter the crossing zone without producing contact. Camera footage confirmed that no flange/check rail contact occurred during these passages.

At velocities of 10 and 50 km/h, displacements of the check rail towards the adjacent running rail were measured in both facing (Fig. 7.7) and trailing (Fig. 7.8) moves. For velocities equal to or greater than 70 km/h, a residual displacement was observed after the passage of the leading wheelset in both moves. Two main hypotheses are proposed to explain this phenomenon: (i) residual deformation of the sleeper, and (ii) permanent displacement of the sensor itself.

It is important to note that at the beginning of each passage, the displacement values were zeroed.



(a) Facing move



(b) Trailing move

Fig. 7.5 Camera recordings of wheelset D at a velocity of 30 km/h, as the train passes over the crossing nose. The top images correspond to the facing move, where the wheels are those of a trailing wheelset D. The bottom images correspond to the trailing move, where the wheels are those of a leading wheelset D. In the trailing move, the wheels are slightly more displaced to the left, toward the check rail (right wheel) and the crossing nose (left wheel).

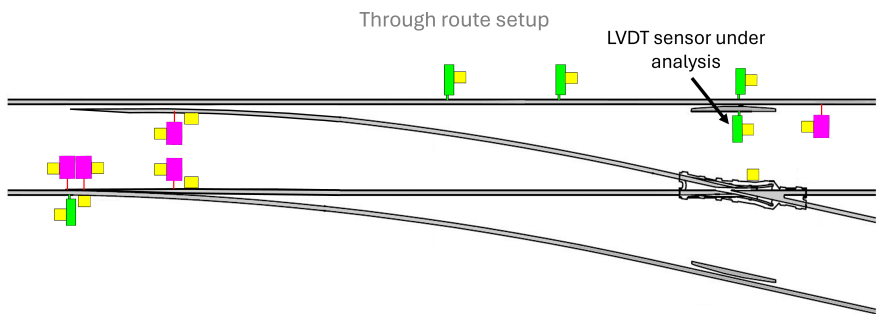


Fig. 7.6 Sensor layout for the through route configuration. The sensor under analysis in this section is indicated in the figure.

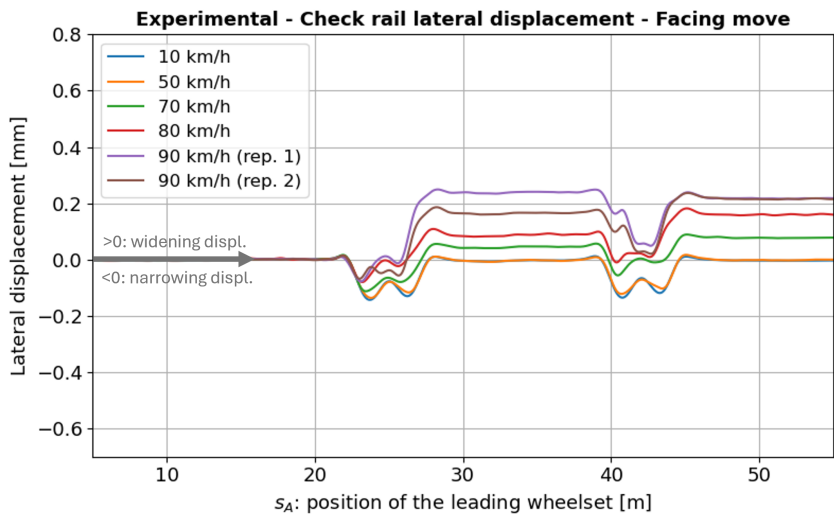


Fig. 7.7 Facing move: displacement of the check rail measured by the LVDT sensor during train passage on the through route. Negative values correspond to a movement towards the flangeway, indicating a tendency to reduce the flangeway gap.

Comparison between Track Sensors, Train Sensors, and Static Measurement

As previously discussed in the analysis of the closure panel, the SMS, positioned on the bogie 870 mm away from wheelset D, does not measure the displacements of the rail at the exact position where the wheel and

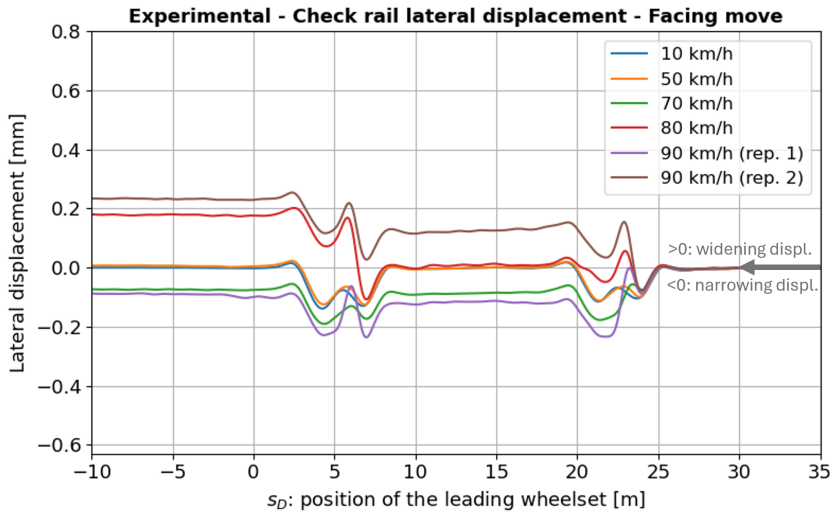


Fig. 7.8 Trailing move: displacement of the check rail measured by the LVDT sensor during train passage on the through route. Negative values correspond to a movement towards the flangeway, indicating a tendency to reduce the flangeway gap.

the rail are in contact, as the displacement sensors (DS) does. To enable a comparison at equivalent positions, the SMS readings have been aligned with the DS measurements, in Fig. 7.9 for the facing move and Fig. 7.10 for the trailing move. The SMS measurements of the flangeway gap between the peaks recorded by the DS for wheelsets C and D, are indicated by the dotted red line in the figures.

The measuring train provides a flangeway gap measurement alongside other parameters described in Section 3.2.2. To evaluate its variation relative to the static value, it is compared with the readings obtained using the digital ruler, considered as a *static reference*, and with the value obtained at a low velocity of 5 km/h, considered as a *quasi-static reference*, which represents a loading condition with the centrifugal effect minimized, as explained in Section 6.2. Fig. 7.11 illustrates the gauge values measured by the SMS and the digital ruler at the exact location where the DS are installed.

The two DS sensors in the Crossing Panel measure the lateral displacements of the check rail and the closest running rail at the same longitudinal position. The equivalent lateral variation of the flangeway gap value can be calculated by adding these displacements.

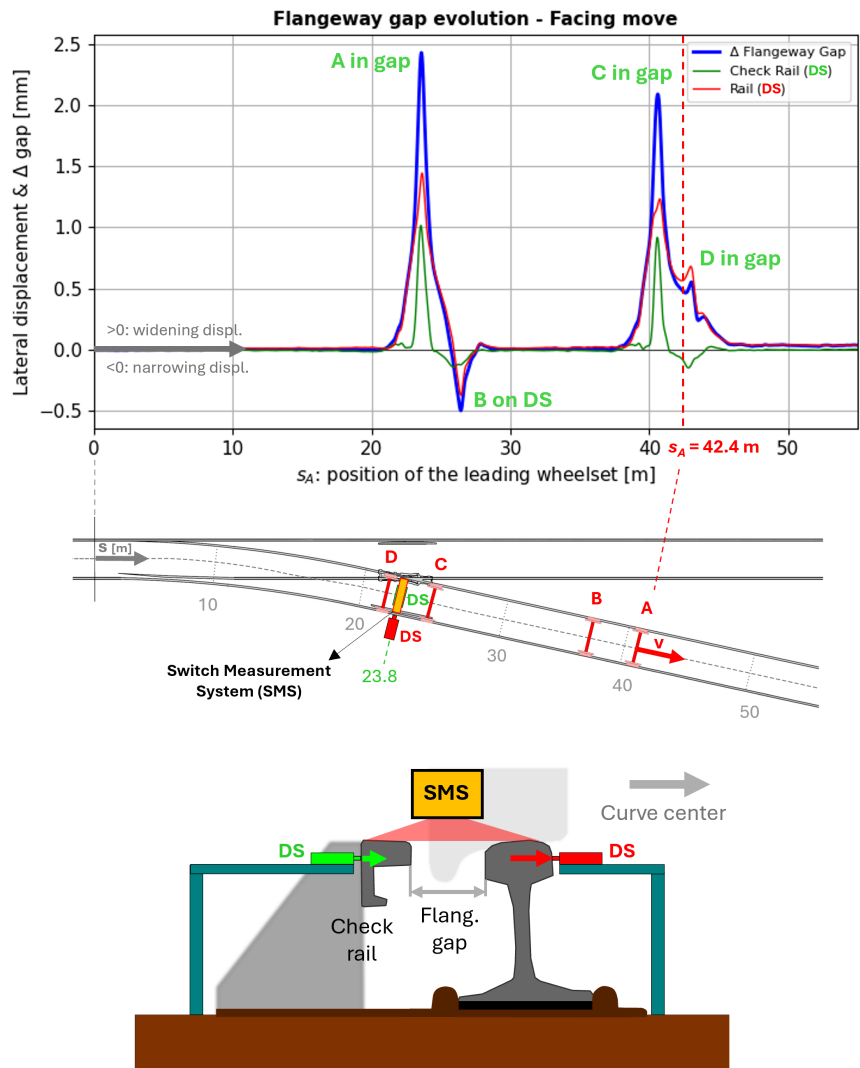


Fig. 7.9 Facing move at 30 km/h: (Top) Evolution of the flangeway gap and lateral displacements of the check rail (DS, green) and the nearest running rail (DS, red). (Middle) Wheelset positions when the SMS aligns with the DS location. (Bottom) Cross-sectional view of the running rail and check rail displacements when the SMS is at the same longitudinal position.

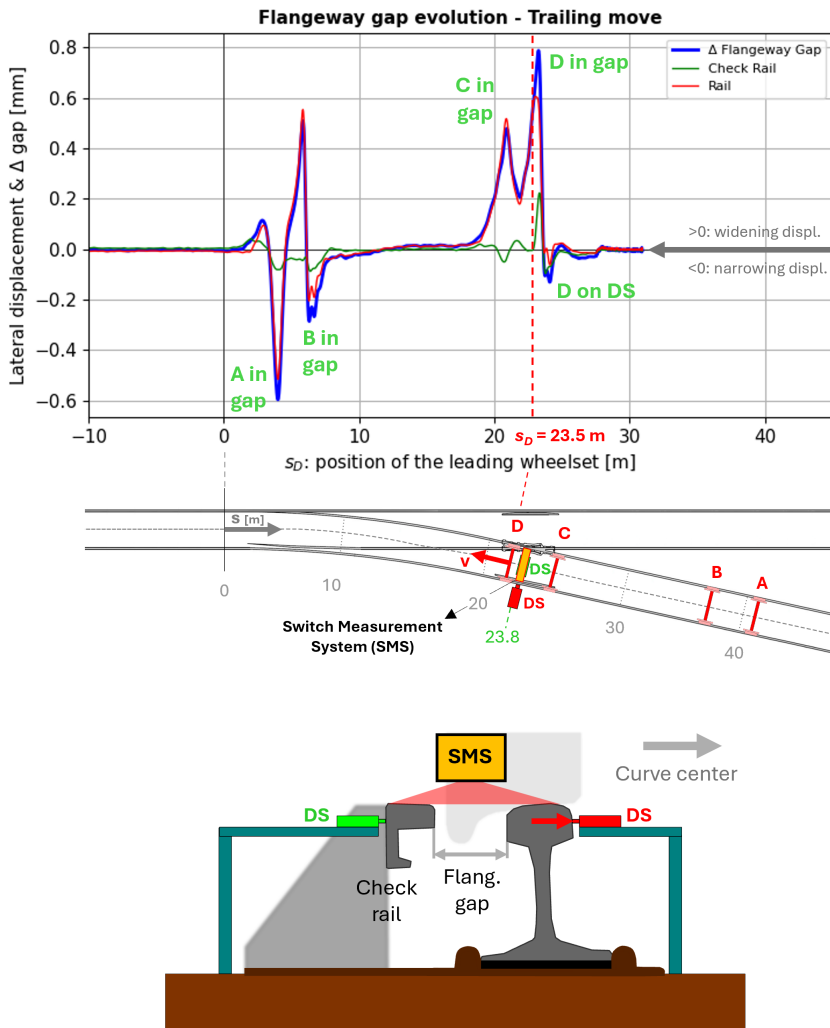


Fig. 7.10 Trailing move at 30 km/h: (Top) Evolution of the flangeway gap and lateral displacements of the check rail (DS, green) and the nearest running rail (DS, red). (Middle) Wheelset positions when the SMS aligns with the DS location. (Bottom) Cross-sectional view of the running rail and check rail displacements when the SMS is at the same longitudinal position.

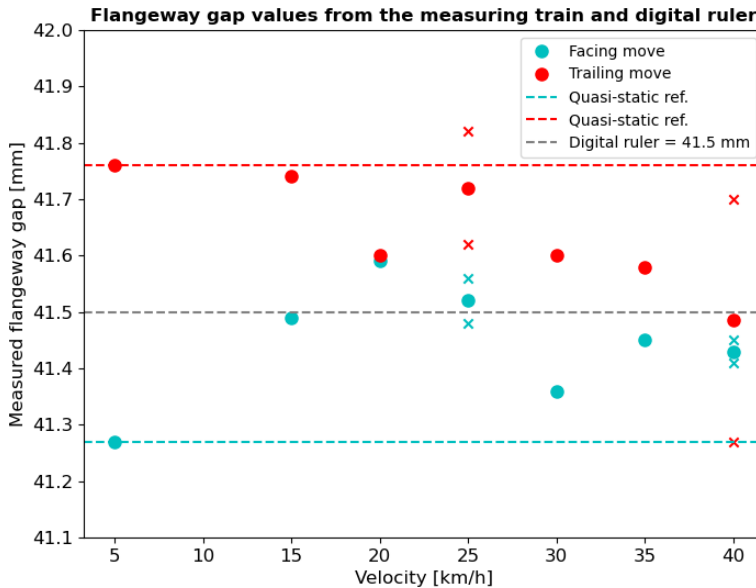


Fig. 7.11 Flangeway gap values measured by the train’s Switch Measurement System at different train velocities, compared with the digital ruler, when the SMS is located at the same longitudinal position as the DS in the Crossing Panel. For 25 km/h and 40 km/h, ‘x’ marks individual measurements, and the point on the same abscissa represents their average.

When comparing dynamic passages (5–40 km/h) measured by the SMS with the static reference (digital ruler), the following results are observed. For the facing move, Fig. 7.11 shows that the measuring train records a narrowing of the flangeway gap, about 0.14 mm at 30 km/h, while the DS register a widening of about 0.5 mm (Fig. 7.9). Even though the graph for the DS readings shows a local minimum at this point, it still indicates a widening instead of the narrowing recorded by the measuring train.

For the trailing move, both the SMS and the two DS detect a widening of the flangeway gap. However, the measuring train records a value of 0.10 mm, whereas the DS detect a variation closer to 0.6 mm (Fig. 7.10).

The comparisons between the dynamic passages (5–40 km/h) and the quasi-static reference are analyzed in Section 7.2.

7.1.2 Model results

The numerical model developed in Chapter 5 for the train on the turnout (Fig. 7.12) is able to capture the displacements of the check rail with the compliant flange–rail contact approach when the inner surface of the wheel flange is in contact, as well as the displacements of the running rail where the wheel tread is running. By computing the difference between these two displacements, the model can estimate the evolution of the flangeway gap.

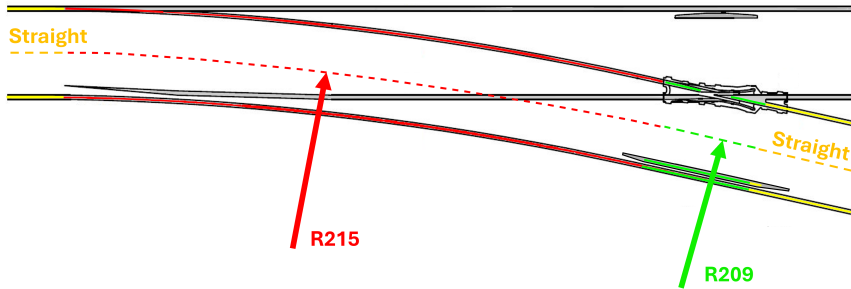


Fig. 7.12 Turnout geometry with two curved sections of 215 m and 209 m radius, defined as the base configuration in the numerical model.

Experimental data were collected at the location of the displacement sensor (DS), where contact occurs between the check rail and the 1st and 3rd wheelsets (WS) during facing moves. The model reproduces these contact conditions and the associated check rail displacements (see Fig. 7.13). Additionally, lower inward displacements into the flangeway gap were observed experimentally for the 2nd and 4th WS, when there is no contact with the wheel flange, probably due to check rail or sleeper flexion. These effects are not captured by the model, which does study the structural body deformation.

In trailing moves, experimental data show contact only with the 1st wheelset at the DS location. However, the model obtains a contact with both the 1st and 3rd wheelsets (Fig. 7.14).

This difference between both moves can be explained by the geometry of the turnout. When approaching from the facing move, the track follows a curve that extends beyond half the length of the check rail (Fig. 7.12) and then it transitions to the straight section. This curvature induces centripetal acceleration, causing earlier contact with the check rail. In contrast, the trailing move comes from a straight track when entering the check rail zone, resulting in delayed contact.

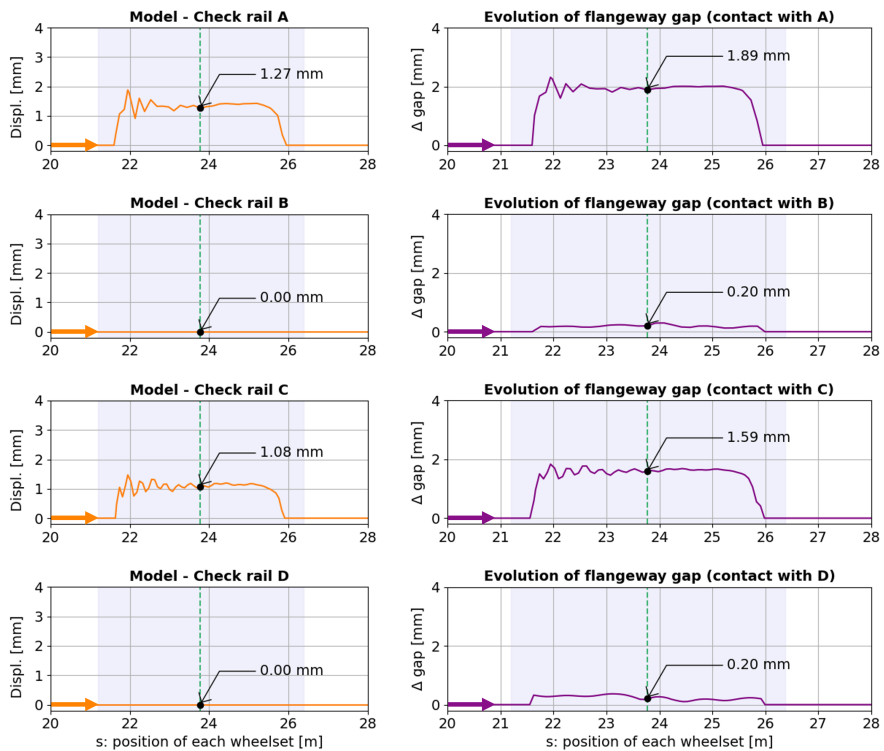


Fig. 7.13 Facing move: numerical simulation results validating the contact between the check rail and the 1st and 3rd wheelsets (left) and representing the flangeway gap variation (right) obtained at a train velocity of 30 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign. The purple zone highlights the check rail area.

In the base model this delayed contact can be observed in Fig. 7.14. The check rail zone is the one indicated with a purple background. Reading these trailing move **graphs from right to left**, the check rail starts at the position 26,3 m, and the flange contact for the 1st and 3rd WS starts is produced 2 m further (at the longitudinal coordinate of 24,5 m).

While in the facing move (Fig. 7.13), **graphs from left to right**, the contact with the check rail starts at the position 21,6 m, almost at the beginning of the check rail (21,2 m).

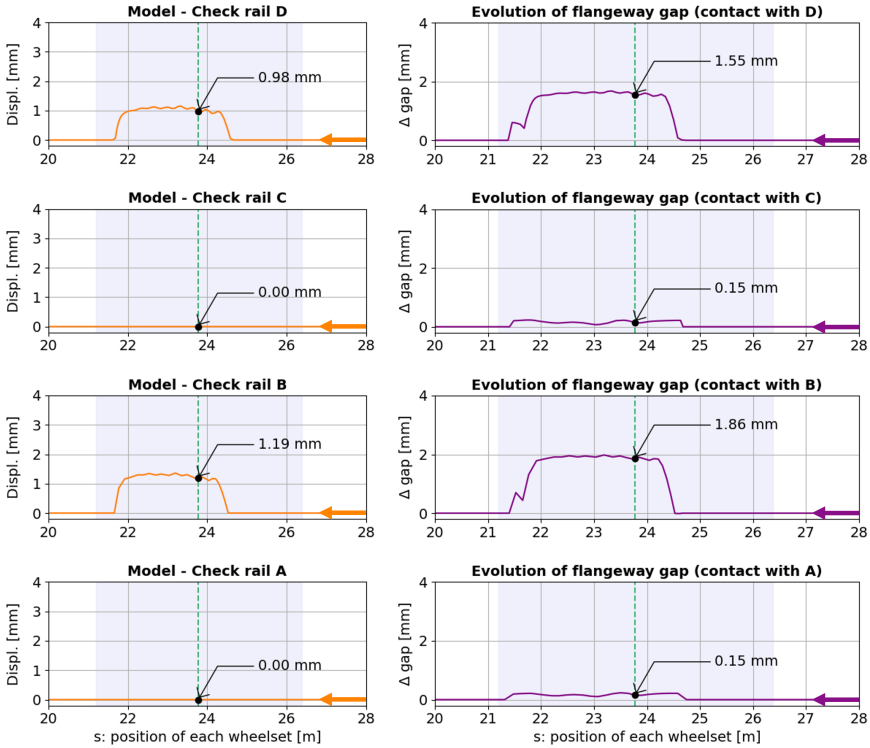


Fig. 7.14 Trailing move: numerical simulation results obtaining a contact between the check rail and the 1st and 3rd wheelsets (left) and representing the flangeway gap variation (right) at a train velocity of 30 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign. The purple zone highlights the check rail area. As it is the trailing move, note that the graphs must be read from right to left.

However, in the base model, whose values of curvature are taken from the theoretical plan, even if the contact is delayed, contact has already occurred at the sensor location (vertical dashed green line).

However, with a small modification of the origin of the curvature, or depending on the longitudinal position of the displacement sensor (DS) along the check rail, the sensor may record no contact, the beginning of contact, or contact occurring within a region of sustained maximum contact force.

Values taken for all velocities and represented in Fig. 7.15 and Fig. 7.16 show good agreement with the experimental data, particularly for the facing move. The displacements during the facing move are generally greater than those observed during the trailing move, except for the 3rd wheelset in the trailing configuration.

This discrepancy between the model and the experimental data for the 3rd wheelset in trailing move may be attributed to the timing of contact detection. While the model records the displacement once contact is fully or almost fully established, the experimental sensor might capture the beginning of contact or a pre-contact phase, depending on its exact longitudinal placement.

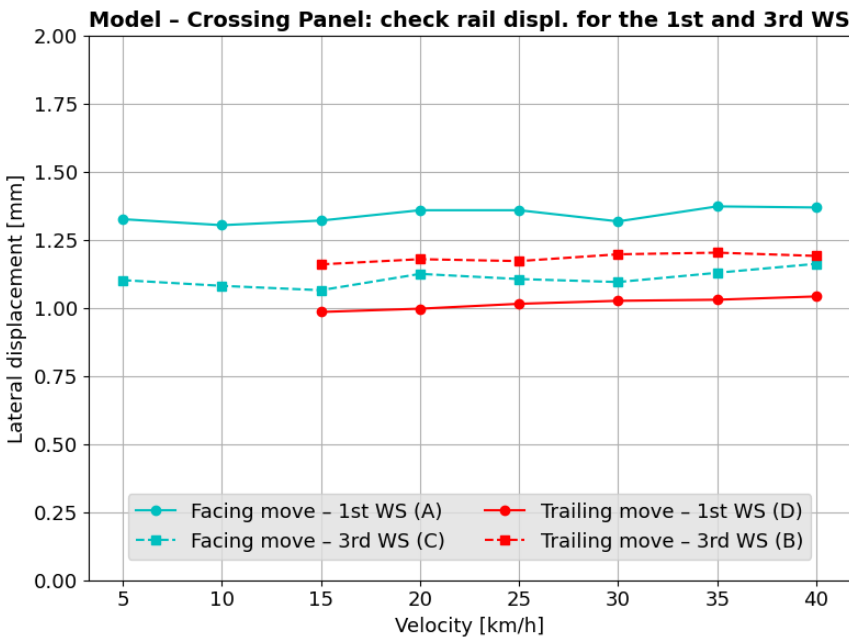


Fig. 7.15 Comparison of the lateral displacement of the check at the longitudinal position equivalent to the experimental sensor location, during wheel flange/check rail contact for the 1st and 3rd wheelsets, for both move directions: facing and trailing.

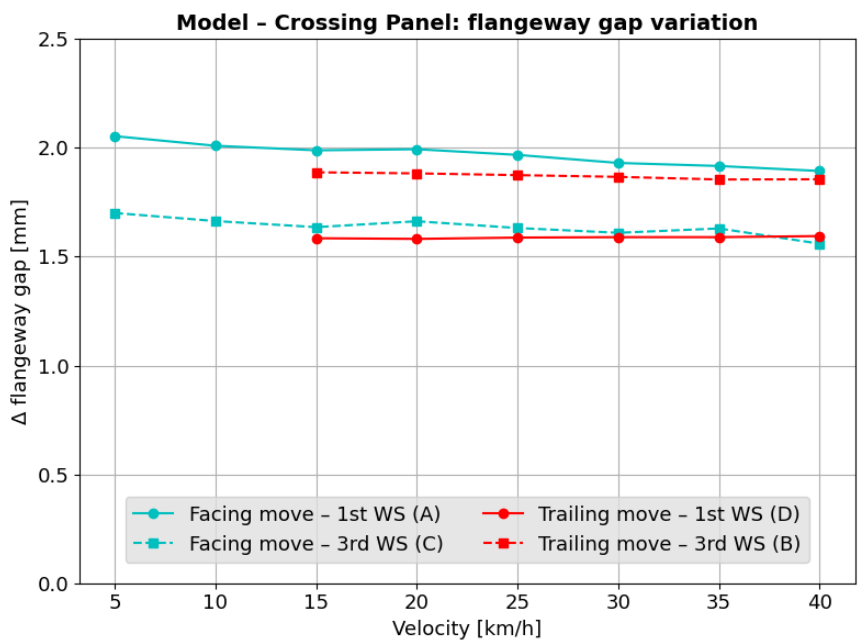


Fig. 7.16 Comparison of the flangeway gap variation at the longitudinal position equivalent to the experimental sensor location, during wheel flange/check rail contact for the 1st and 3rd wheelsets, for both move directions: facing and trailing.

7.1.3 Numerical exploration

The turnout geometry used in the MBS model is an approximate representation based on the official plan provided by Infrabel. It consists of a straight main route, followed by two consecutive curves with radii of 215 m and 209 m, and then a straight diverging route (Fig. 7.12).

However, the geometry measured by the EM130 (Fig. 7.17) train differs considerably from the theoretical layout, likely due to internal filtering performed by the device. The curvature, defined as $1/R$, reaches a peak value below 3.5 km^{-1} , which is lower than the expected value of 4.65 km^{-1} (corresponding to a radius of 215 m).

Moreover, in the zone where a radius of 209 m is expected — corresponding to the tighter radius in the Crossing Panel — the peak is not observed. Instead, the curvature is close to the inflection point of the curve, approaching zero.

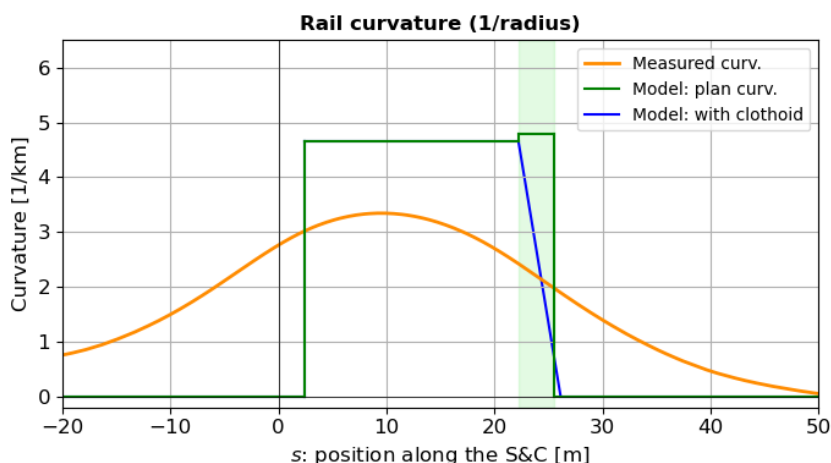


Fig. 7.17 Turnout curvature measured by the EM130 measuring train, curvature from the plan used for the model and curvature with clothoid used for the alternative model. The area in green marks the zone where a radius of 209 m (4.78 km^{-1} of curvature) is expected in the Crossing Panel.

To simulate a case similar to the one measured by the EM130, the second radius of 209 m was replaced by a clothoid (Fig. 7.18), with a length close to that of the arc corresponding to the original 209 m curve.

This alternative configuration, using a clothoid instead of a second

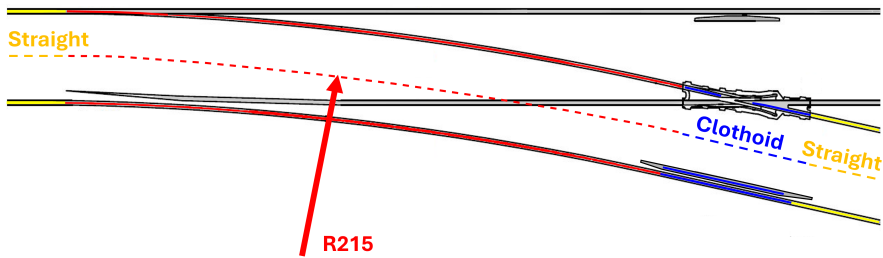


Fig. 7.18 Turnout geometry with a curved section of 215 m radius, followed by a clothoid of 3,90 m length, was chosen as an alternative configuration in the numerical model.

constant-radius curve, further delays the contact between the wheel flange and the check rail for the trailing move (Fig. 7.19). At the longitudinal position equivalent to the experimental sensor location, the model shows no contact between the 3rd wheelset and the check rail during trailing moves. These results (Fig. 7.20) are closer to the experimental observations and they support the hypothesis that, for the trailing move, the contact with the check rail has not yet occurred when passing in front of the sensor. Therefore, the sensor may be capturing the beginning or absence of contact rather than a full interaction.

This makes sense considering that the critical region of the crossing is the frog area, where the risk of derailment is highest. In trailing moves, the train is still aligned with the diverging route, reducing the possibility of deviation toward the through route, and therefore reducing the risk of derailment.

7.1.4 Dynamic vs static: discussion & analysis

Influence of the trajectory

The track dynamic behavior in the crossing panel was assessed by the analysis of flangeway gap variations, on the advice of our industrial partner, given the critical nature of this turnout zone. The static measurements with a digital ruler were taken as a reference for the geometric parameters of the crossing gap under the condition of no external forces. The dynamic measurements by displacement sensors (DS) and train's Switch Measurement System (SMS) showed considerable variations – in relation to the safety geometric criteria, which allow only 2–3 mm variation in certain dimensions

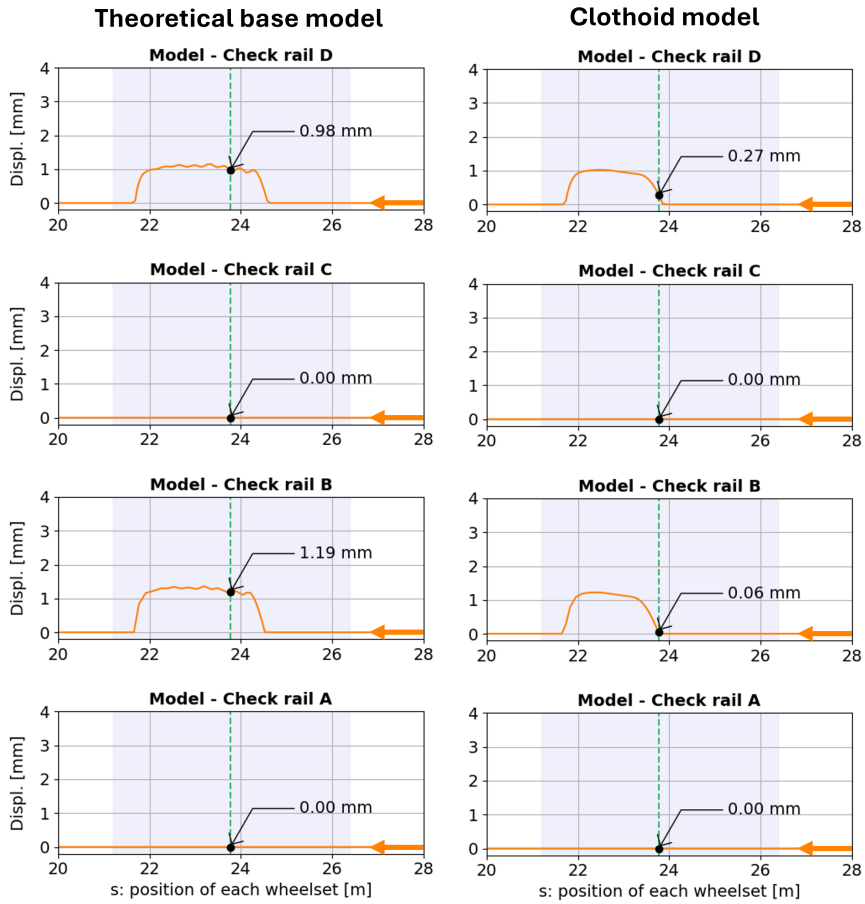


Fig. 7.19 Trailing move: numerical simulation results obtaining a contact between the check rail and the 1st and 3rd wheelsets for the theoretical base model (left) and the model with clothoid (right) at a train velocity of 30 km/h. The vertical dashed green line marks the equivalent location of the LVDT sensor from the experimental campaign. The purple zone highlights the check rail area. As it is the trailing move, note that the graphs must be read from right to left.

– caused by the train passage in this critical area.

The two DS captured the variations in the flangeway gap width by recording the lateral displacements of both the check rail and the nearest

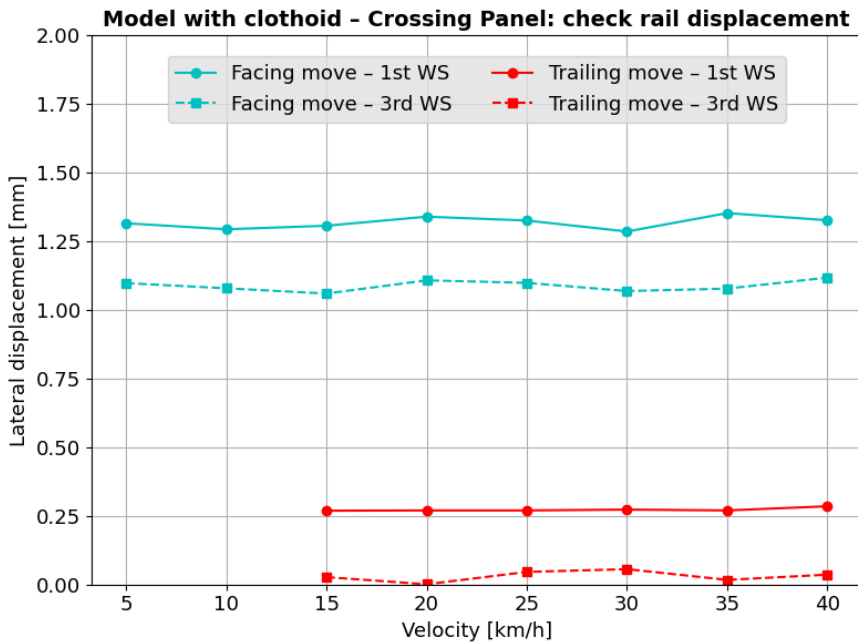


Fig. 7.20 Model of turnout with clothoid: comparison of the lateral displacement of the check rail at the longitudinal position equivalent to the experimental sensor location, during wheel flange/check rail contact for the 1st and 3rd wheelsets, for both move directions: facing and trailing.

running rail. The experimental results showed that the gap varied, either widening or narrowing, as the train wheelsets passed over the crossing panel. The widening of the flangeway gap was larger for facing moves than for trailing moves. This behavior is attributed to the curvature of the trajectory preceding the crossing: in facing moves, the train approaches from a curved track, inducing centripetal acceleration and earlier contact with the check rail. In trailing moves, the train enters from a straight track, resulting in delayed contact and reduced dynamic interaction.

This interpretation was also obtained by modifying the geometry in the numerical model, replacing the second curve in the base model with a clothoid that better approximated the curvature measured by the measuring train. This modification further delayed contact in trailing moves and aligned better with experimental observations.

A schematic summary of the displacements during the train passage for both facing and trailing moves in the base model is presented in Fig. 7.21 and Fig. 7.22, respectively.

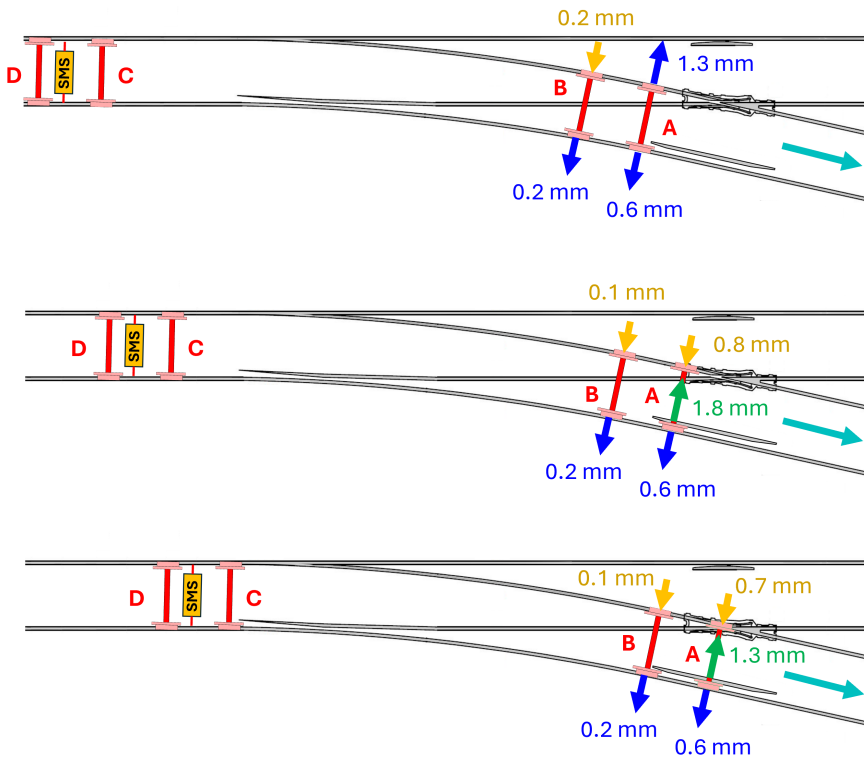


Fig. 7.21 Schematic representation of the rail and check rail displacements during some phases of train passage for the facing move.

Position of the SMS

As with the Closure Panel, the SMS, being positioned between two wheelsets, often produced results that differed from those of the DS fixed on the track. In facing moves, the SMS recorded a narrowing of the flangeway gap, while the DS measured a widening. However, both systems consistently demonstrated different dynamic behaviors depending on the train direction. For example, the SMS measured a narrowing in the facing move, with respect to the static reference, but a widening in the trailing move.

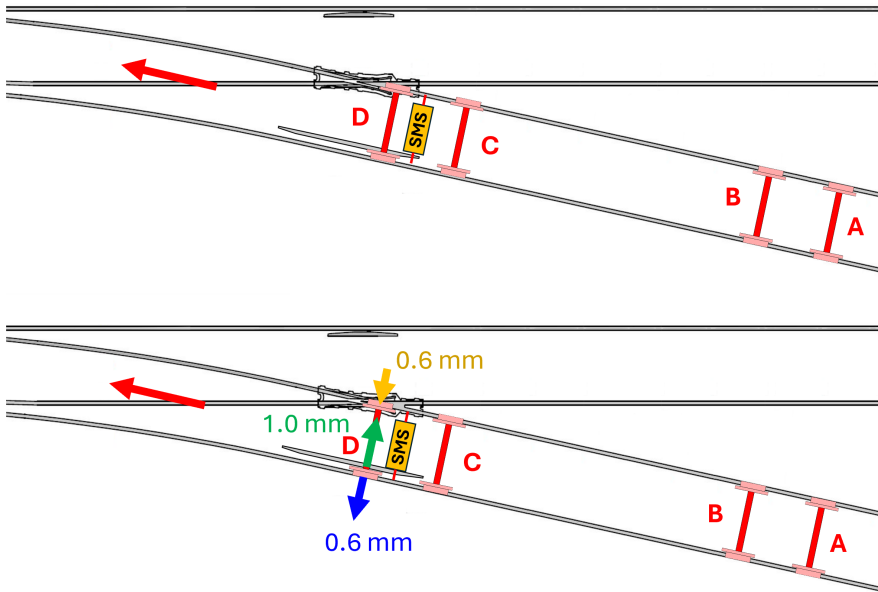


Fig. 7.22 Schematic representation of the rail and check rail displacements during some phases of train passage for the trailing move.

Comparison between dynamic and static data

- **Static vs quasi-static reference without flange contact:** in the absence of contact between the flange and the rail (e.g., on the 2nd and 4th wheelsets — the trailing wheelsets of each bogie), sensor data indicates that the check rail tends to move slightly inward toward the flangeway gap, both in facing and trailing movements. This can be verified by the measurement sensors in the through route, where the check rail also move inward. This behavior may be explained by the hypothesis of check rail or sleeper flexion, which is beyond the scope of the numerical model, as it only captures rail displacements under direct contact conditions and may neglect body deflection. These observed displacements correspond directly to the difference between the quasi-static load condition and the static geometry reference.
- **Static vs quasi-static reference with flange contact:** in the presence of contact of the check rail and the wheel flange inner surface, both the check rail and the running rail move outward, contributing to a widen-

ing of the flangeway gap (up to 2,0-2,5). This behavior is observed in DS measurements and reproduced by the numerical model.

Summary and implications

The Crossing Panel was analyzed using static, quasi-static (5 km/h), and dynamic measurements across various velocities and model configurations. The numerical model matches quite well with experimental data, particularly in facing move, and effectively captures the contact dynamics and flangeway gap evolution. The differences observed in the trailing move highlight that the measured displacements depend on both the location of the wheel–rail contact and the position of the sensor relative to that contact. These findings should be confirmed by future experimental setups which should consider multiple sensor placements along the check rail to better capture the full contact profile.

7.2 Velocity influence

As in the Closure Panel, to study the influence of velocity on the dynamic behavior of the Crossing Panel, all measurements from the experimental campaign are analyzed and compared with the model simulations, which allow extending the analysis to velocity conditions beyond those tested experimentally.

7.2.1 Experimental results

Concerning the experimental data, as already described no clear influence of velocity was observed on the check rail displacement (Fig. 7.3), nor on the flangeway gap variation (Fig. 7.4). For all tested velocities, the values remain relatively constant for the compared wheelsets — the 1st and the 3rd.

There is a slight tendency for the 3rd wheelset in the facing move to show increased displacement at higher velocities. In contrast, in the trailing move, a slight decrease is observed with increasing velocity.

The SMS system, which does not measure at the exact moment of contact between the flange and the check rail, also captured this trend: an increase in displacement for the facing move and a decrease in flangeway gap for the trailing move, when compared to the quasi-static case at 5 km/h, where centrifugal effects are minimal (Fig. 7.11).

These cases should be further explored using the numerical model, particularly at velocities beyond those tested in the experimental campaign.

7.2.2 Model results and numerical exploration

For velocities below 50 km/h, the model shows relatively constant behavior for both movement directions (Fig. 7.23), as seen in the experimental results. For example, the leading wheelset in the facing move (1st WS) shows an increase of less than 0.05 mm between 5 and 50 km/h.

At higher velocities, the check rail displacement tends to increase exponentially, particularly in the facing move, which is due to the curvature of the track before the crossing. This results in higher centripetal forces on the wheels. For instance, an increase of approximately 0.5 mm is observed between 50 and 70 km/h, equivalent to a 39% rise.

In the trailing move, which approaches the check rail from a straight track, the influence of velocity is lower due to reduced centripetal forces.

The wheelset with the highest increase (3rd WS) shows a change of less than 0.3 mm between 15 and 70 km/h, corresponding to a 25% increase.

Regarding the flangeway gap (Fig. 7.24), which depends on both the check rail and the running rail, the increase from 50 km/h onward computed by the model is less pronounced than in the facing move, and almost negligible in the trailing move.

Some local minima are observed instead of a consistent increase of the 1st and 3rd wheelsets. This is due to the nature of the check rail impact: the average displacement around the reference measurement point (equivalent to the LVDT sensor) is taken from the Fig 7.13 and Fig 7.14, but the range is narrow and contains many peaks. This makes it difficult to obtain a stable value, especially at velocities above 40 km/h. Nevertheless, rather than detailed oscillations, the focus here is on analyzing general trends.

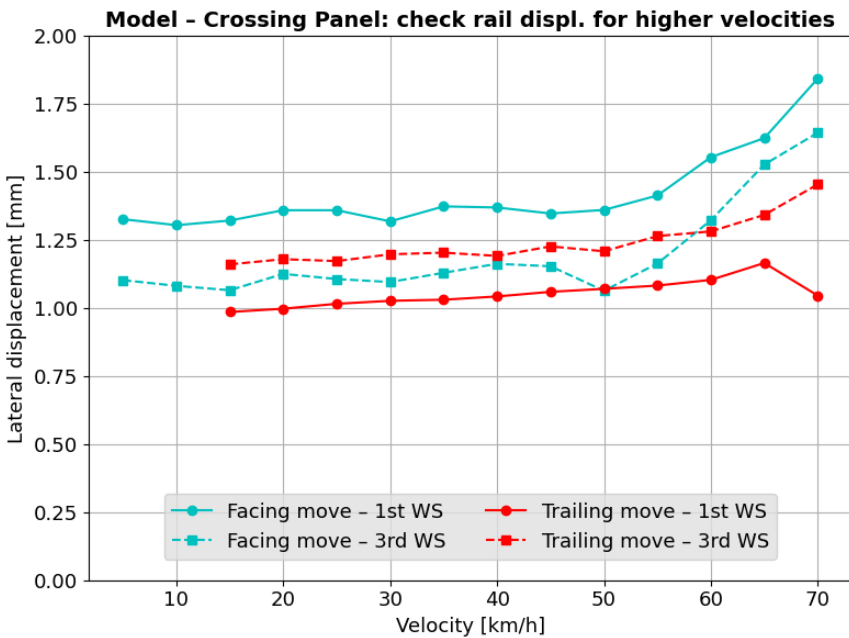


Fig. 7.23 Model comparison of the lateral displacement of the check rail — averaged around the longitudinal position corresponding to the experimental sensor location — during wheel flange/check rail contact for the 1st and 3rd WS, for both facing and trailing moves.

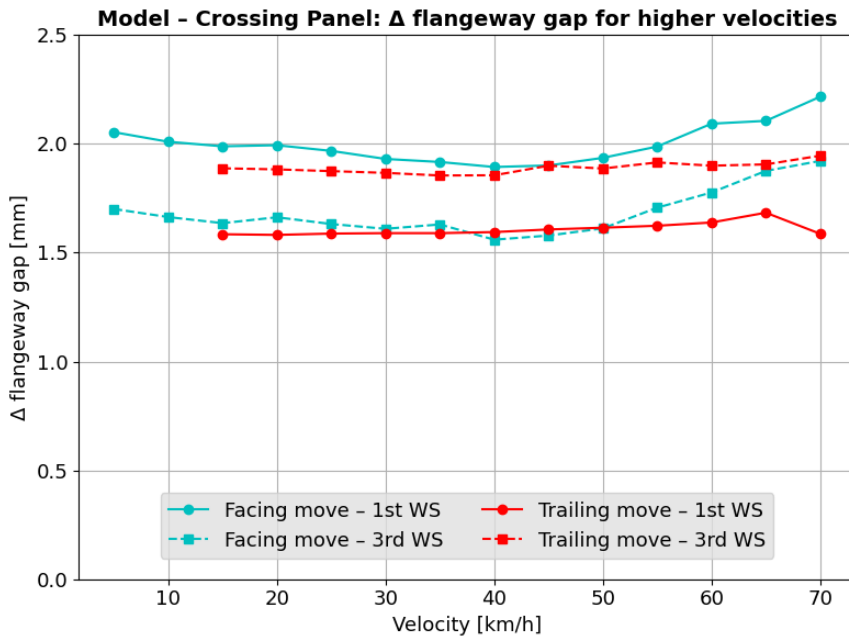


Fig. 7.24 Model comparison of the flangeway gap variation at the longitudinal position corresponding to the experimental sensor location, during wheel flange/check rail contact for the 1st and 3rd wheelsets, for velocities up to 70 km/h and for both move directions: facing and trailing.

7.2.3 Dynamic vs static: discussion & analysis

Similar to the closure panel, measurements in the crossing panel revealed no significant impact of train velocity on the variations observed. The multibody model, for velocities up to 40 km/h, within the same range as the experimentation, confirmed this negligible impact for low velocities.

However, for higher velocities, the numerical model reveals a quadratic increase in check rail displacement, particularly in facing moves. This behavior is attributed to the centripetal forces generated on the wheelsets by the curved trajectory preceding the crossing, which are absent in trailing moves that approach from a straight track.

In a way, all the results presented clearly show differences between measurements without load (i.e. no train on the track and measurement with a geometric ruler) and with load (i.e. a moving train and measurements

with sensors). For this loaded situation (the real one, in fact), the observed displacements are mainly a consequence of the bogie configuration (curvature of the track, the direction of travel, wheelset position as leading or trailing) rather than of actual dynamic effects such as train speed, centripetal acceleration, or carbody roll velocity, at least for the experimental velocities (≤ 40 km/h). We could therefore conclude that the differences in this range of velocities — in relation to geometric measurements — are ultimately more quasi-static, considering quasi-static as load cases with low influence of velocity - than dynamic in nature, and related to the way in which the wheels exert lateral efforts on the rails, via their tread creep forces and/or their flange pushing forces.

We can therefore conclude that the differences observed in this velocity range — relative to the geometric measurements — are primarily quasi-static rather than dynamic, with quasi-static referring to load cases where the effects come mainly from wheel-rail engagement, friction, and support conditions, rather than from velocities and accelerations. These differences are related to how the wheels exert lateral forces on the rails, through their tread creep and/or flange pushing forces.

7.3 Center of mass impact

As illustrated in Fig. 6.29, the heavier half of the train is positioned over the area corresponding to wheelsets A and B. This mass distribution already showed a significant influence in the Closure Panel analysis. In the following section, its impact is further analyzed for the crossing panel, specifically on the lateral displacement of the check rail.

7.3.1 Experimental results

In the experimental campaign, wheelset A was the leading wheelset only in the facing move, while wheelset D was the leading wheelset only in the trailing move.

Since the entry into the crossing panel is not symmetrical—being more curved in the facing move than in the trailing move—it is difficult to isolate the influence of the center of mass.

Moreover, the tests were not performed with both orientations of the train, which further limits the ability to assess the center of mass effect experimentally.

7.3.2 Model results and numerical exploration

As discussed in the previous chapter, numerical simulations enable predictions that were not feasible during experimentation. In particular, it allows reversing the train orientation to study its influence on the lateral displacement of the check rail.

From Fig. 7.25, a mean difference of 0.23 mm is observed between the heavier and lighter sides of the train in the facing move. In the trailing move (Fig. 7.26), the difference is slightly lower: 0.17 mm. It can be attributed to lower dynamic effects due to the straight trajectory before entering the crossing panel.

When the center of mass is centered (black curves), the difference between the 1st and 3rd wheelsets becomes negligible, especially in the trailing move.

7.3.3 Dynamic vs static: discussion & analysis

As previously seen for the Closure Panel, the results highlight a strong influence of mass asymmetry, in this case for the check rail lateral response.

The influence of which wheelset pushes the check rail — 1st or 3rd — is negligible and it has been verified for a case where the center of mass

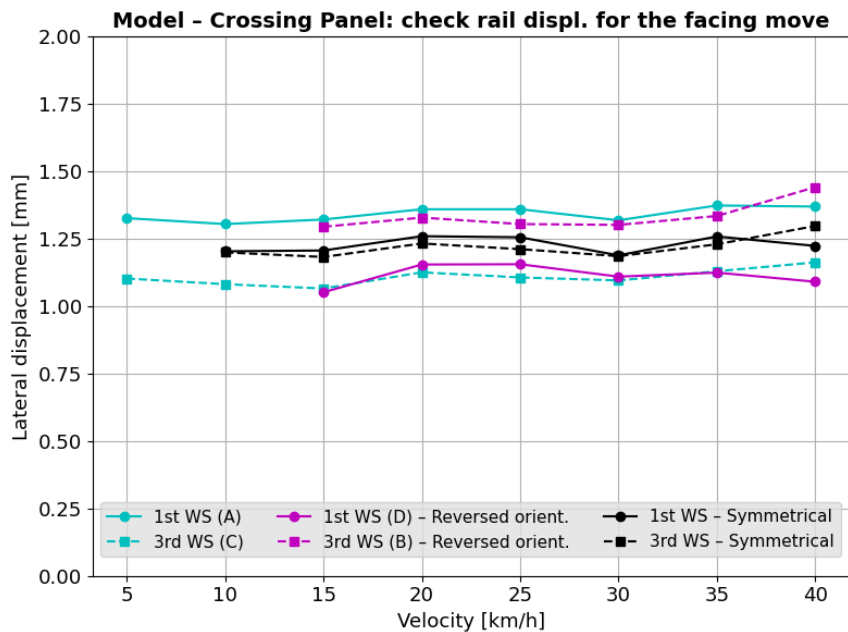


Fig. 7.25 Facing move: comparison of the lateral displacement of the check rail at the longitudinal position equivalent to the experimental sensor location, for the 1st and 3rd wheelsets, at speeds between 5 km/h and 40 km/h, for both orientations of the train.

is geometrically centered. As expected, the location of the heavier half of the train mass has a significant impact: the wheelset positioned under the heavier half shows higher lateral displacements than those under the lighter half.

The higher lateral displacement caused by the wheelset located under the heavier half of the mass can be explained by Newton's second law, although it must be noted that the carbody rotates together with the two bogies, influencing the overall lateral movement. Both halves of the train experience the same lateral acceleration, but the wheelsets under the heavier half experience a larger centripetal force. This force is transmitted through the wheel-rail contact and results in stronger flange contact with the check rail, which explains the larger lateral displacements.

The differences are slightly more pronounced in the facing move than in

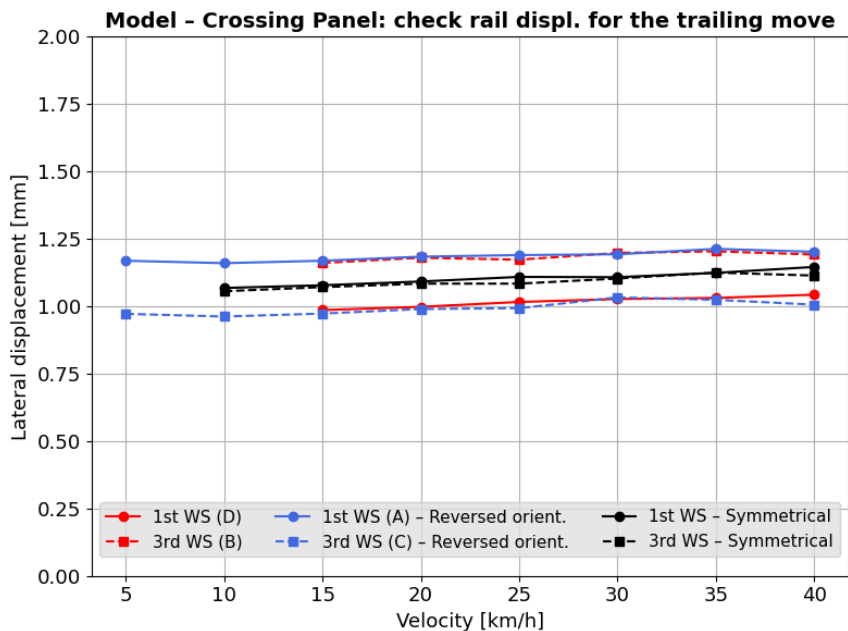


Fig. 7.26 Trailing move: comparison of the lateral displacement of the check rail at the longitudinal position equivalent to the experimental sensor location, for the 1st and 3rd wheelsets, at speeds between 5 km/h and 40 km/h, for both orientations of the train.

the trailing move, as previously discussed, probably due to inertial effects — specifically, the centripetal ones induced by the curvature of the track before entering the crossing panel.

Regarding the train's SMS and the orientation of the measuring train, it is important to keep in mind that changing the train orientation can significantly impact the measurements done by the SMS. Depending on the orientation and the move, the SMS may be positioned close to either a leading or trailing wheelset, which can result in two different interpretations of the measurement taken at the same turnout, for the same move, but with the opposite orientation.

7.4 Lateral rail and check rail stiffness impact

The objective of this section is to study the influence of the lateral stiffness of the track in the Crossing Panel. Specifically, it aims to characterize the equivalent lateral stiffness at the wheel/rail contact by considering the global response of the railway infrastructure in this zone. The stiffness of the check rail is varied to analyze its impact on rail displacement and energy dissipation. Both experimental and numerical approaches are compared to analyze stiffness configurations and to understand their consequences on the dynamic behavior of the turnout.

7.4.1 Experimental results

The hydraulic jack used to apply lateral forces to both rails was positioned, for this study, near the check rail area (Fig. 7.27), on the diverging route.

In this case, the configuration differs from those observed in the Switch and Closure Panels. Between the outer rail and the tie plate — a metal plate between rail and sleeper that holds the rail —, there was an elastic rail pad, while there was not for the inner rail.

The measured lateral displacements of each rail are shown in Fig. 7.28. As in the equivalent tests performed on the Switch and Closure Panels, the graphs indicate that the rail heads — red and blue — experience greater lateral displacement than their webs — orange and cyan, respectively — suggesting a rotational behavior or apparent flexion of the rails under lateral loading.

Different slope regimes are observed. For the inner rail, stiffness decreases with increasing force and lateral displacement. For the outer rail, three distinct regimes are identified:

- Up to approximately 5 kN: stiffness is 23.36 kN/mm
- Between 5 kN and 28 kN: stiffness is 19.03 kN/mm
- Beyond 28 kN: stiffness increases to 26.37 kN/mm.

As discussed in Section 2.1.3, these multi-stage behaviors are also reported in the literature. For example, Wang et al. (2024) [36] observed similar regimes in his experiments, attributed to shearing between the, sliding and collision between the plate and the anchor bolts.

The first regime change could be explained by a change from shearing (between the rail and the rail pad) to sliding (between rail and tie plate, or

between tie plate and sleeper). Concerning the third stage, a first possibility is that the pad comes into contact with another component of the tie plate, generating a localised increase in stiffness. Second explanation could be the proximity to the crossing nose, which may result in a larger support area and thus a local stiffening effect. Finally, a mechanical stop effect, acting as a constraint to the rail's rotational movement, could also contribute to the sudden changes in stiffness observed during the experiment.

The first regime change can be explained by the transition from shearing (between the rail and the rail pad, or between the rail and the tie plate) to sliding (either between the rail and the tie plate or between the tie plate and the sleeper). Concerning the third stage, two possibilities exist: first, the pad may come into contact with another component of the tie plate, such as the bolts, generating a localized increase in stiffness; second, the proximity to the crossing nose could result in a larger support area and thus a local stiffening effect.

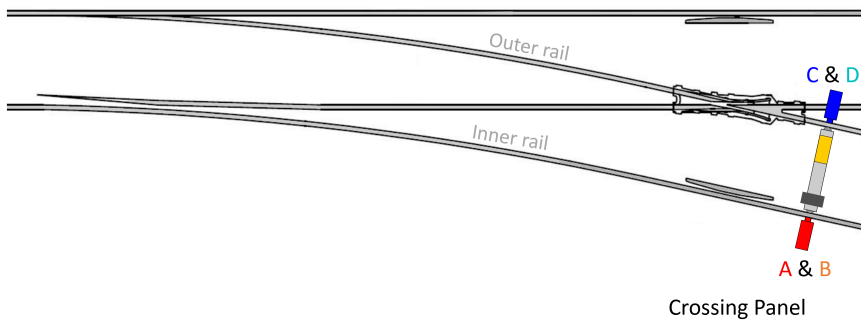


Fig. 7.27 Location of the sensors and the hydraulic jack on the Crossing Panel, next to the frog and the check rail. C and D indicate the head and web of the outer rail, respectively. A and B indicate the head and web of the inner rail, respectively.

7.4.2 Model results and numerical exploration

A direct comparison with the results of the static experimental campaign on track stiffness is not possible, as the stiffness of the check rail was not measured. Instead, the check rail stiffness value was calibrated using the displacements obtained from the dynamic experimental campaign, in order to achieve a comparable order of magnitude in the numerical model. A stiff-

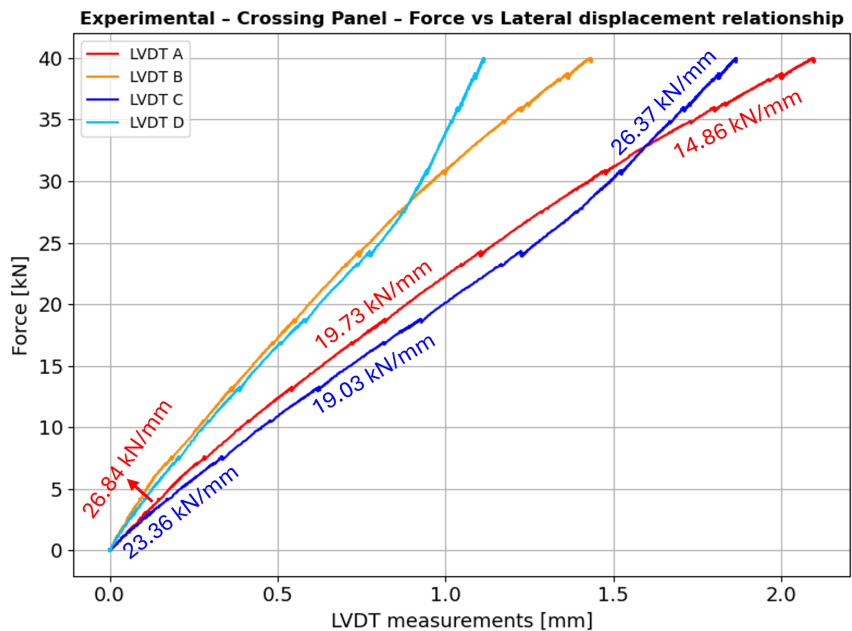


Fig. 7.28 Curves of applied force for the Crossing Panel rails versus lateral rail displacement measured by the LVDT sensors. Blue and cyan correspond to the head and web of the curved closure rail (outer rail), respectively; red and orange to the head and web of the straight closure rail (inner rail).

ness value of 50 kN/mm was selected, which is within the range reported in the literature for multibody models (between 15 and 120 kN/mm, see Table 2.1).

In the model, a sensitivity analysis was performed to evaluate the influence of check rail stiffness on both the dissipated energy and the variation of the flangeway gap, by simulating train passages at 30 km/h in both directions.

Rail wear is a concern for Infrabel’s railway structures manager, as it can increase the flangeway gap and, consequently, raise the risk of derailment. Therefore, this study investigates the influence of the check rail’s lateral stiffness on the energy dissipated at the contact between the wheel flange’s inner surface and the check rail, as well as its effect on the variation of the flangeway gap.

The dissipated power in the contact is calculated in the multibody model as the product of contact force and tangent velocity ($P = \vec{F} \cdot \vec{v}$). By integrating this power over time, the total dissipated energy is obtained. For each passage, the total energy dissipated corresponds to the sum of the energies due to contacts with the check rail, which occur with the inner flange of the 1st and 3rd wheelsets.

On one hand, the curve of dissipated energy (Fig. 7.29) shows a logarithmic increase that stabilizes beyond a lateral stiffness value of approximately 40–50 kN/mm.

In the facing move case, more energy is dissipated due to the curvature of the track before reaching the check rail zone: the contact occurs earlier and is more prolonged than in the trailing move case, where the train approaches through a straight track.

Both curves reach a plateau with increasing check rail stiffness, suggesting that beyond a certain threshold, further increases in stiffness do not significantly affect the amount of energy dissipated.

On the other hand, the curve in Fig. 7.30 that represents the flangeway gap variation shows a logarithmic decrease, which also stabilizes at higher stiffness values. In this regime, the direction of movement no longer has a significant influence, as the displacement of the check rail becomes minimal.

7.4.3 Dynamic vs static: discussion & analysis

Static and dynamic approaches have also proven to be complementary for the study of the lateral stiffness behavior in the Crossing Panel. Static tests provide measurements of rail displacement under controlled lateral loading, while dynamic modeling allows for the evaluation of contact dynamics and energy dissipation during realistic train passages.

In the static experimental campaign, multiple stiffness regimes were observed, consistent with findings in the literature. The first two regimes may correspond to a transition from shearing (between the rail and the rail pad, or between the rail and the tie plate) to sliding (either between the rail and the tie plate or between the tie plate and the sleeper). For the third regime, two hypotheses were discussed: the tie plate may come into contact with another component, such as the bolts, generating a localized increase in stiffness; or the proximity to the crossing nose may result in a larger support area and thus a local stiffening effect.

However, it was observed that these regimes differed depending on the presence of a rail pad: a stiffening was detected either due to the rail pad or

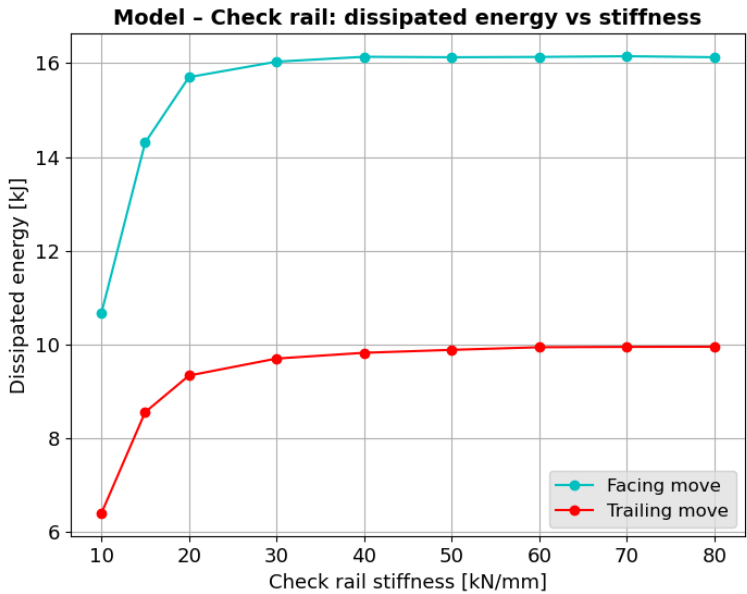


Fig. 7.29 Dissipated energy as a function of check rail stiffness for facing and trailing moves, for a train velocity of 30 km/h.

due to the proximity to the crossing, whereas no stiffening was observed for the inner rail, where no rail pad is present.

Regarding the numerical model, it shows that increasing the check rail stiffness reduces the flangeway gap opening, but increases the dissipated energy through tangential forces. However, beyond a certain stiffness value — approximately 40–50 kN/mm —, the dissipated energy does not increase significantly, indicating a plateau behavior.

In summary, both static and dynamic analyses are essential to evaluate the stiffness behavior in the Crossing Panel. Static experiments help to understand the relationship between lateral rail displacements and applied forces, and values obtained served as reference magnitudes for characterization of lateral stiffness in the model. Dynamic experiments helped calibrate this reference stiffness to obtain similar displacements, especially for the check rail, while dynamic simulations provide broader understanding of energy transfer and contact behavior under operational conditions.

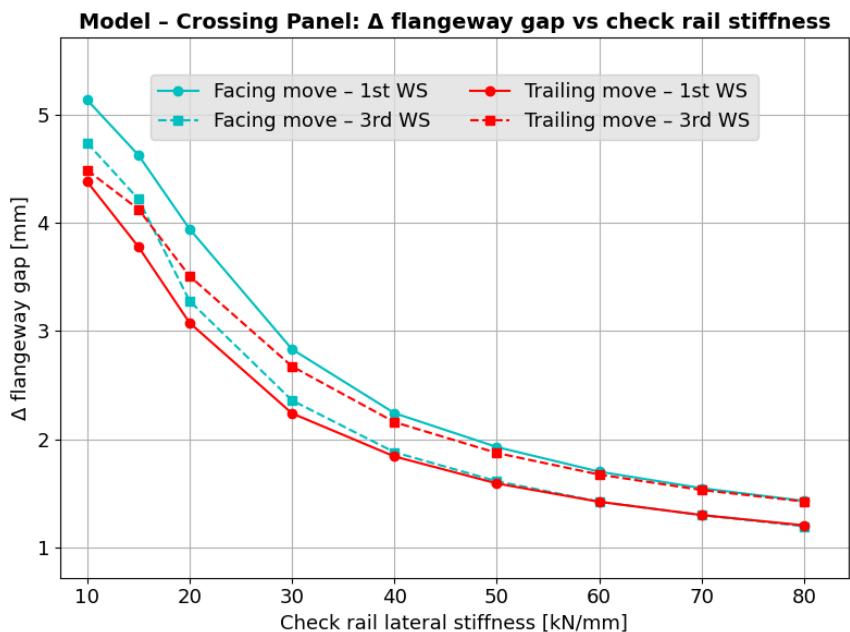


Fig. 7.30 Flangeway gap variation as a function of check rail stiffness for facing and trailing moves, for a train velocity of 30 km/h.

7.5 Impact of a worn check rail

In real-world applications, check rails are often installed with a slightly reduced flangeway gap for safety reasons. This tight configuration ensures that, if there is the presence of wear, the gap does not exceed the maximum tolerance limits. However, the mechanical consequences of check rail wear on wheel-rail interaction remain to be fully understood.

This section simulates three types of check rail configurations to study how worn check rails affect the dynamic behavior of a turnout, specifically the dissipated energy at the wheel flange/check rail friction contact. The dissipated energy, calculated as in the previous section, corresponds to that due to tangential (friction) forces between the check rail and the inner surface of the wheel flange. This energy is analyzed to compare which configurations may lead to greater wear and why.

The three configurations considered are: a tight check rail (38 mm flangeway gap), a nominal check rail (40 mm), and a worn check rail (42 mm).

7.5.1 Model results and numerical exploration

In the facing move (Fig. 7.31), the curve shows a logarithmic increase. There is only a small difference between the tight and nominal check rails, as the contact occurs almost simultaneously. In the case of the worn check rail, the contact happens later, resulting in less interaction with the check rail and therefore lower dissipated energy. Overall, the differences are small, especially between the tight and nominal check rails, indicating that a tight check rail does not seem to provoke more wear.

In the trailing move (Fig. 7.32), the differences between the tight–nominal and nominal–worn configurations are similar. The tight check rail results in higher energy dissipation due to its more active guiding function. Velocity also plays a role, as both the dissipated power and energy increase with speed. Particularly, energy increases logarithmically, which amplifies the differences in dissipated energy at higher velocities, for this range of velocities.

When comparing both moves (Fig. 7.33), the facing move clearly dissipates more energy. During the facing move, the check rail must ensure that the train, coming from a curved track, is guided to continue along the diverging route. In contrast, the trailing move starts from a straight track, so contact between the inner wheel flange and the check rail occurs later, leading to a shorter contact duration, and with similar contact forces. The

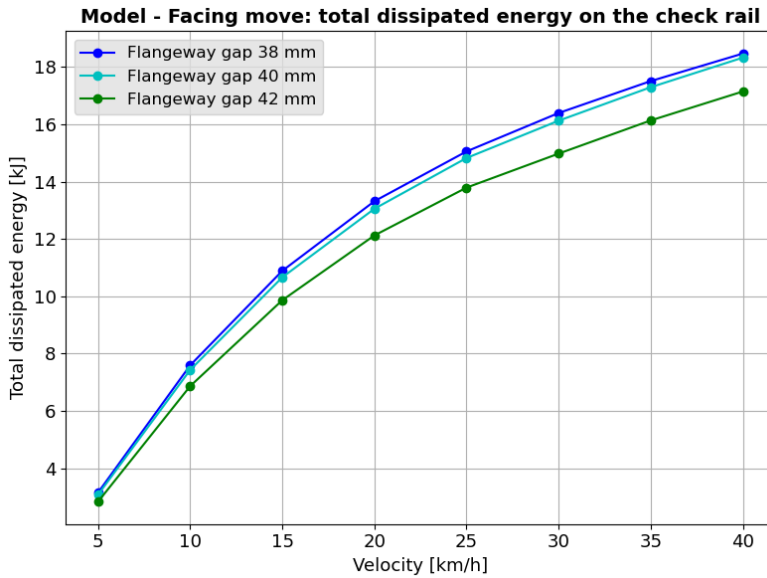


Fig. 7.31 Facing move: total dissipated energy for one train passage as a function of velocity, for a tight check rail (38 mm flangeway gap), nominal check rail (40 mm), and worn check rail (42 mm). The energy corresponds to the sum of all energy dissipated by the wheelsets that come into contact with the check rail during the passage.

difference in dissipated energy between the two moves also increases with velocity.

7.5.2 Dynamic vs static: discussion & analysis

The dynamic analysis performed using the multibody model allows studying aspects that cannot be captured by static or experimental dynamic measurements, even though one must be aware of its own limitations regarding the definition of the compliant contact of the flange.

In particular, this analysis shows that timing and duration of contact between the inner wheel flange and the check rail depend on the move direction (facing or trailing), the condition of the check rail (tight, nominal, worn) and the train velocity. These variables significantly influence the dissipated energy, which increases logarithmically with speed.

Concerning the contact forces, those during facing moves were found to have a peak due to an initial shock, but remained close in intensity to the

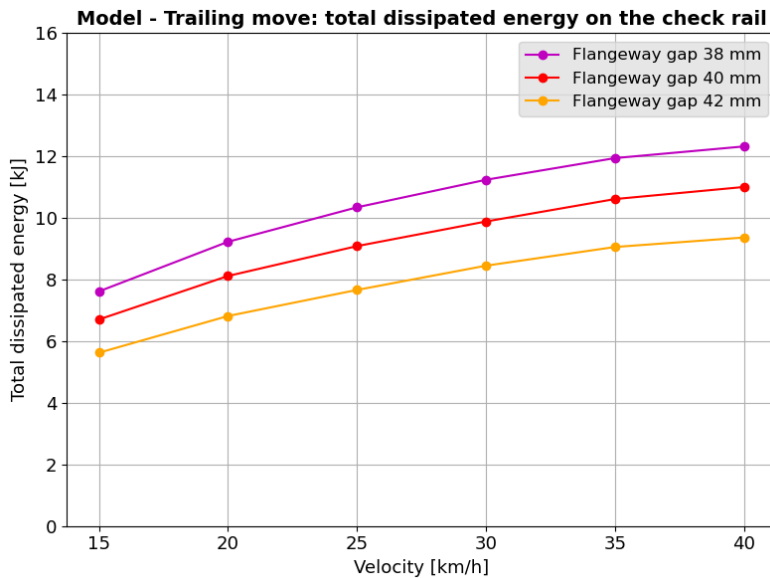


Fig. 7.32 Trailing move: total dissipated energy for one train passage as a function of velocity, for three check rail conditions.

trailing move forces during the stable contact phase, so their difference in intensity has a lower influence on the energy dissipated than the previously discussed parameters.

The direction of the move — facing or trailing — also has significant influence. The model shows that contact may be delayed or shortened, especially during trailing move or when the check rail is worn. In cases with a worn check rail, the lower energy dissipation could be misinterpreted as positive due to reduced wear. However, it could in fact indicate poor guidance, potentially increasing the risk of safety issues. As observed in the model simulation, for flangeway gaps $\geq 49 \text{ mm}$, a train coming from the facing move could be deviated onto the through route, potentially leading to derailment.

Velocity also has a direct influence on the interaction. Higher speeds amplify the dissipated energy and the differences between configurations, for the velocity range analyzed here. This suggests that maintenance strategies should consider operational conditions apart from geometric tolerances, such as train speed and direction of travel. Static evaluations may ignore

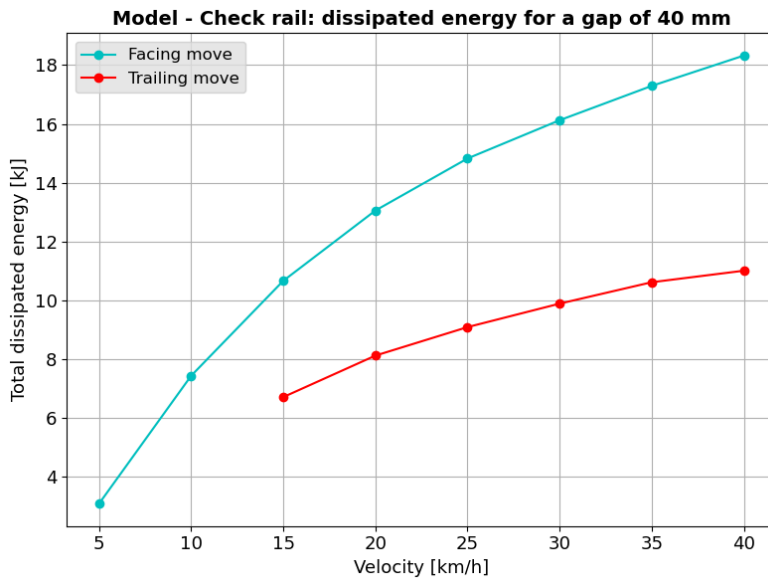


Fig. 7.33 Facing vs trailing move: total dissipated energy for one train passage as a function of velocity, for a nominal check rail (40 mm). The energy corresponds to the sum of all energy dissipated by the wheelsets that come into contact with the check rail during the passage.

these dynamic effects and underestimate the real dynamic effects on the check rail.

8

Discussion and synthesis

IN the preceding chapters, specific discussions have been provided for each parameter influencing the turnout dynamics. This chapter aims to summarize the main ideas and group them consistently, with the objective of addressing the research questions presented at the beginning of the thesis. The discussion is therefore structured around the three research questions that have guided the development of the project.

Q1. Influence of the location of the on-board measuring system on the recorded measurements

The first research question focuses on how the longitudinal position of the on-board measuring system on the train influences the recorded track geometry, particularly in relation to the actual local deformations at the wheel–rail contact, which are often the most critical for safety and maintenance.

The on-board Switch Measurement System (SMS), installed on the EM130 measuring train, is located between wheelsets C and D, closer to wheelset D, at approximately 870 mm (Fig. 8.1). This location has a direct impact on the collection and interpretation of dynamic measurements, as the system captures geometry between wheelsets rather than at the direct wheel/rail contact deformation.

To investigate this impact, experimental runs were conducted with the

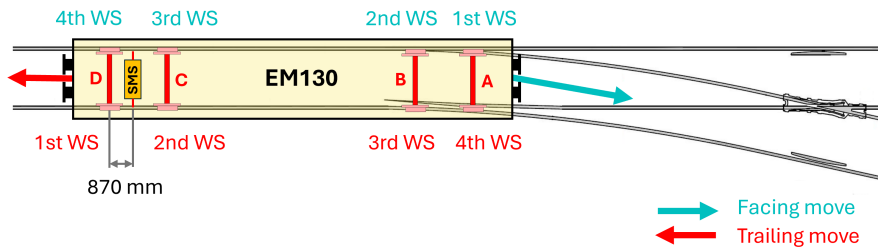


Fig. 8.1 Wheelset (WS) order and letter assignment for facing (in blue) and trailing (in red) moves. The Switch Measurement System (SMS) is located at a distance of 870 mm from wheelset D.

train passing over both the diverging and the through routes in several passages, always maintaining the same orientation and varying the speed in each run. In this configuration, the "rear" side of the train was oriented toward the main route, while the "front" side was oriented toward the diverging or straight route. Consequently, the SMS was positioned on the side facing the main route.

The results show that, in the Closure Panel of the diverging route, rail widening is observed when 1st and 3rd wheelsets (the leading wheelsets of each bogie) are in contact with the rail — a phenomenon confirmed by both the multibody model and experimental measurements. This deformation is mainly due to the contact between the wheel flange and the rail, as verified by the cameras focused on wheelset D. In contrast, the through route — although not a strict reference due to the absence of curvature — provides hints about the rail's behavior without flange contact: the rail tends to shift inward, a behavior also observed on the diverging route for wheelsets that do not have wheel flange/rail contact (2nd and 4th wheelsets, the trailing wheelsets of each bogie).

In summary, the fixed LVDT sensors systematically capture the local peaks of rail displacement at the exact wheel–rail contact locations, while the SMS measures the rail response between wheelsets C and D. This difference in measurement location explains the change in coherence between both systems depending on the direction of travel.

- **Facing move:** the peak widening is produced by the leading wheelset (A), but the SMS is located close to wheelset D, which is trailing in this configuration. As the LVDTs show, the trailing wheelsets generate an

inward shift rather than a widening. Consequently, the SMS would record a narrowing, not the widening peak captured by the LVDTs.

- **Trailing move:** in this case, the SMS is located close to the wheelset (D) that produces the largest widening peak. Since the LVDTs confirm that this wheelset generates a significant outward displacement, the SMS also records higher values than in the facing move, consistent with the LVDT peak.

Likewise, in the Crossing Panel, the SMS does not detect the peak widening of the flangeway gap — the distance between the check rail and the adjacent rail — which is actually captured by the LVDT sensors during experimentation and by the multibody model when the 1st and 3rd wheelsets are in contact in that zone.

Nevertheless, although the SMS does not detect these critical deformations, its measurements should not be considered incorrect. As discussed in the state of the art, the EN13848-2 (2020) [7] standard only specifies that “the sensors placed under the vehicle’s frame shall be as close as possible to one of the vehicle’s loaded axles to respect measurement conditions indicated in EN 13848-1 (2019) [8]”, without defining an exact location for the system. The discrepancies observed with respect to on-track sensors and the multibody model are therefore explained by the nature of the measurement: the SMS records the rail geometry between wheelsets, rather than the rail stiffness at the direct wheel/rail contact.

Based on the flangeway gap variation values measured by the on-track sensors at 30 km/h, several hypothetical SMS positions were evaluated (Fig. 8.2) to determine what the SMS would measure if installed elsewhere along the train. These positions are the actual location (870 mm from wheelset D), the geometric center of the train, locations close to each wheelset, and midpoints between wheelsets (see Table 8.1).

The results indicate that, for wheelsets A and C during the facing move, the SMS recorded values that could exceed the Immediate Action Limit (IAL) — a threshold of -2 and 3 mm¹ as defined by Infrabel [99] close to the limit of track intervention. In contrast, in all other cases — particularly during the trailing move — the measured values remained within acceptable tolerances.

¹As there is no upper threshold defined, the 3 mm taken refers to the maximum dimensional variation from the check rail to the crossing nose — also called the protection gauge — i.e., it includes both the check rail and crossing nose displacements. For the comparison presented here, the 3 mm value is taken only as an approximate reference. It is also known that the flangeway gap also depends on the running rail adjacent to it.

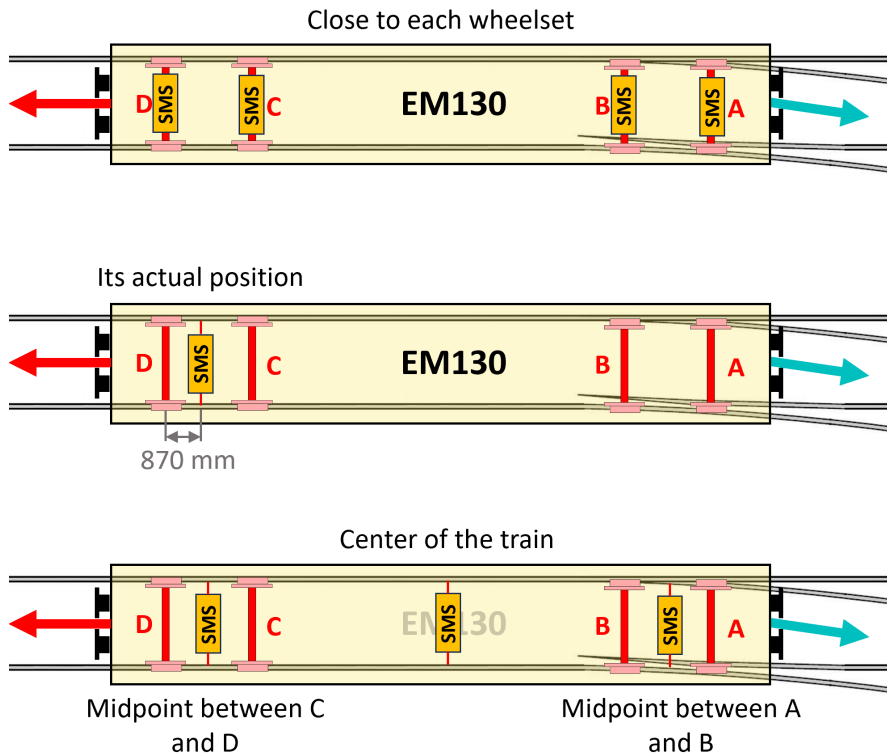


Fig. 8.2 Hypothetical SMS positions evaluated along the train. (Top) SMS installed close to each wheelset. (Middle) SMS at its actual position. (Bottom) SMS installed at the midpoints between C and D, between A and B, and at the center of the train.

This raises a fundamental question: should the SMS be positioned where dynamic effects are most pronounced, i.e., where the gap between static and dynamic measurements is maximal? Or should it be placed where dynamic influence is minimal, allowing for the evaluation of quasi-static geometric conditions such as wear or deformation without load?

If the goal is to capture the maximum dynamic effect, a strategic repositioning of the SMS may be necessary, as the current installation appears to underestimate rail deformation in the most sensitive regions. However, placing the SMS closer to the wheelsets introduces an operational limitation: by moving closer to the axles, the measurements become much more sen-

Table 8.1 Crossing Panel: based on the flangeway gap variation values measured by the on-track sensors at 30 km/h, this table presents the values that would have been detected by the Switch Measurement System (SMS) depending on its position along the train. The Immediate Action Limit (IAL) threshold is also included for reference. The value of 2 mm refers to the maximum narrowing, while the 3 mm corresponds to the limit of the widening dimension^a. For reference, the table cells are categorized by value ranges: **dark red** for values exceeding the IAL, **light red** for high values (above 65% of the IAL), **yellow** for moderate values (above 15%), **light green** for low values (above 5%), and **dark green** for values close to zero.

SMS possible positions	Facing [mm]	Trailing [mm]
Close to A	2.425	-0.597
Midpoint between A and B	0.404	0.295
Close to B	-0.502	-0.286
Center of the train	0.000	0.015
Close to C	2.092	0.476
Midpoint between C and D	0.525	0.209
At its actual position (870 mm from D)	0.479	0.596
Close to D	0.551	0.788
IAL	-2.00 & 3.00^a	

^a The 3.00 mm refers to the maximum dimensional variation from the check rail to the crossing nose, also called the protection gauge, i.e., it includes both the check rail and crossing nose displacements. For the comparison presented here, the 3.00 mm value is taken as a reference.

sitive to the running direction, which makes the interpretation of the data considerably more complex in practice. In this case, it is also worth questioning whether the same IAL threshold used for static measurements should be applied, given that the dynamic context is not directly comparable.

However, if the objective is to obtain measurements under static or quasi-static conditions, the optimal location would be near the geometric center of the train, as the dynamic deformation is minimal, as shown in Table 8.1. To comply with EN13848-2 [7], which requires the system to be placed close to loaded axles, positioning the SMS between wheelsets within each bogie would be more appropriate. In these configurations — specifically at the midpoints between wheelsets A and B, and between C and D — the measured values are approximately 0.5 mm or lower. These values indicate the presence of dynamic effects, but still far from the IAL threshold of -2 and 3 mm.

Q2. Influence of train direction, velocity, orientation and wear on rail dynamics

The experimental and numerical results allowed us to study the influence of several operational and geometric parameters. Their combined analysis provides insights into how train move direction (facing vs trailing), velocity, mass distribution and wear on rail affect the dynamic behavior of the track for the studied configuration — a right-hand turnout mounted on wooden sleepers. Some effects have a high influence on the track dynamics, others lower or negligible, and some remain inconclusive.

- **Move direction:** facing moves generally result in higher lateral displacements, especially in the Crossing Panel. On-track sensors detected a difference of approximately 0.8 mm in the check rail displacement between facing and trailing moves across all velocities, which contributed to a 1.5 mm difference in the flangeway gap. In line with this, the numerical model with a curvature defined as a clothoid in the Crossing Panel obtained a difference of about 1 mm in check rail displacement. This behavior is mainly attributed to the curvature of the track preceding the measured zone, which induces earlier and stronger flange contact forces due to centripetal acceleration.

In the Closure Panel, where measurements are taken in the middle of the curve, the difference in rail displacement peaks between facing and trailing moves is not significant. However, a slightly higher influence of the facing move was observed experimentally on the outer rail, although the results remain inconclusive due to the influence of the measurement order which may bias the results.

In contrast, in the Crossing Panel — located at the transition between the end of the curve and the straight track of the diverging route — the differences are more pronounced. In facing moves, the train approaches from a curved section, causing the wheels to shift toward the outer rail. As a result, the inner face of the flanges of the 1st and 3rd wheelsets come into contact with the check rail from the very beginning of the rail. The contact is present at the sensor location and persists along almost the entire length of the check rail.

On the other hand, in trailing moves, the train arrives from a nearly straight section. At the beginning of the Crossing Panel, the wheels do not require guidance from the check rail until the curve begins.

Therefore, contact occurs later than in facing moves, resulting in a smaller zone of check rail deformation and reduced energy dissipation — which may translate into lower wear. For example, at a velocity of 40 km/h, the total energy dissipated in the facing move is approximately 60% higher than in the trailing move.

- **Train orientation and mass distribution:** the experimental analysis of this parameter was limited due to the fixed train orientation and the limited repeatability of the tests, so no conclusions can be obtained from the experimental. However, the multibody model allowed for a more flexible investigation, where the orientation could be reversed. In the simulations, the heavier half of the train (wheelsets A and B) generated higher lateral forces and rail displacements, with around 0,3 mm of difference between flangeway gap peaks generated by each side — an order of magnitude comparable to the experimental results, which showed differences between 0.05 and 0.25 mm between wheelsets — and more than 0,3 mm in Closure Panel gauges. Simulations with nearly symmetric mass distribution showed that displacement peaks on both sides were almost identical, which confirms that mass asymmetry influences the rail response.

In fact, the track exhibits very different behaviors depending on the direction of travel and whether it is the bogie containing axles A and B or the one containing C and D. Thus, turning the train around changes the deformations observed. These are all the more pronounced when the SMS is positioned close to an axle. If the SMS is close to an axle, it will observe very different behaviors depending on whether this axle is the first or last of the bogie (1st or 3rd vs. 2nd or 4th), and also depending on whether the other bogie is still in the curve, has already exited it, or is still entering it.

- **Velocity:** for speeds up to 40 km/h and for this turnout configuration, the influence of velocity in the lateral displacements is limited and mostly quasi-static, both in the Closure and Crossing Panels. This is to say, the observed displacements are mainly a consequence of the bogie configuration — including track curvature, direction of travel, and the leading or trailing position of the wheelsets — rather than by dynamic parameters such as train speed or acceleration. Therefore, experimental data show stable displacement patterns across velocity regimes.

However, numerical simulations reveal that at higher speeds ($\geq 70\text{km/h}$), lateral displacements increase quadratically, mainly due to centripetal forces — which is consistent with the fact that this force depends on the square of the velocity.

Regarding energy dissipation, simulations obtained a logarithmic increase with velocity in the range of 5–40 km/h. For example, in the facing move, the total energy dissipated increases from approximately 3 kJ at 5 km/h (quasi-static reference) to 18kJ at 40 km/h, representing a 500% increase. This dynamic response is consistent, as power dissipation at the contact is proportional to both tangential forces and relative velocity between the inner surface of the wheel flange and the check rail.

- **Curve radius:** in the multibody model, the turnout configuration was modified to include a curve radius of 500 m. This modification results in a reduction of the dynamic effect of velocity for speeds up to 40 km/h, confirming that the additional lateral force observed at higher speeds for the R215 case is mainly centripetal. Since centripetal force is inversely proportional to the curve radius, the centripetal force for R500 is approximately twice lower compared to the nominal configuration of a 215 m radius. Consequently, the forces for a curve of R500 the forces mainly come from the quasi-static dynamics.

Even if not specifically analysed for the Crossing Panel, a reduction of the curvature radius in the Closure Panel could indirectly affect wheelset guidance in the Crossing Panel. For the trailing move, the changes would be negligible, since the vehicle approaches the turnout from the diverging route, which is straight. However, some differences could be expected in the facing move. A lower radius leads to higher centripetal forces and therefore higher lateral forces at the check rail contact. Increasing the radius (e.g., from R215 to R500) reduces these centripetal effects, resulting in lower lateral forces, smaller flangeway gap variations, and reduced wear on the check rail. This improvement comes from the reduced contact forces rather than from changes in the geometry of the Crossing Panel itself.

- **Worn check rail:** the model also shows that the worn check rail for this turnout configuration (simulated with a check rail displaced to widen the flangeway gap) results in delayed contact and reduced energy dissipation compared to tightened or nominal check rails (simulated

with a check rail displaced to narrow the flangeway gap). This lower energy dissipation could be misinterpreted as positive due to reduced wear, but it actually also indicates poor guidance and increased derailment risk. Tight or nominal check rails ensure earlier and more sustained contact, improving safety but increasing wear.

The results clearly reveal consistent differences between unloaded geometric measurements and those taken under load. In real operating conditions, the observed rail displacements are mainly driven by the train's configuration — such as track curvature, direction of travel, orientation of the train, and whether a wheelset is leading or trailing — rather than by dynamic effects like velocity or acceleration, at least within the tested velocity range (≤ 40 km/h). These variations appear to be quasi-static in nature, reflecting how the wheels apply lateral forces on the rails through tread creep and flange contact. Therefore, dynamic measurements should be interpreted considering the parameters that may influence the results depending on the train's configuration and trajectory. A summary table classifying the influence of each parameter on the different track responses is shown in Table 8.2.

Parameter	Closure Panel gauge	Flangeway gap	Check rail wear
Move direction (facing vs trailing)	Negligible	High ($\Delta \approx 1.5$ mm)	High ($\Delta > 60\%$)
Velocity (up to 40 km/h)	Low ($\Delta \approx 0.1$ mm)	Negligible	Very high ($\Delta > 500\%$)
Train orientation (rear vs front)	Moderate ($\Delta \approx 0.35$ mm) / Unclear (exp.)	Moderate ($\Delta \approx 0.30$ mm)	Not studied
Curve radius (R215 vs R500, for 40 km/h)	Moderate ($\Delta \approx 0.25$ mm)	Not studied	Not studied
Check rail condition (worn vs tight)	(N/A)	Not studied	Moderate $\Delta_{\text{fac.}} \approx 7.8\%$ $\Delta_{\text{trail.}} \approx 31.5\%$

Table 8.2 Summary of the influence of key parameters on track behavior, based on peak comparisons from both experimental and multibody model analyses.

Q3. Impact of lateral stiffness on railway dynamics

Lateral stiffness plays a critical role in the dynamic behavior of S&C, influencing rail displacements, gauge variation and energy dissipation.

The static experimental campaign revealed a nonlinear stiffness behavior when a force separating the rails was applied, with distinct displacement regimes depending on whether a rail pad was present. According to the literature, these regimes may correspond to shearing, sliding, and potential collision effects between the rail, the rail pad and the tie plate. In the Closure Panel, where the rail is directly placed on the tie plate without rail pad between them, stiffness decreased beyond 30 kN of lateral force, indicating a possible transition from static friction to sliding. In the Crossing Panel, three stiffness regimes were observed, possibly due to pad/plate interaction, proximity to the frog, or mechanical stops (collision).

Numerical simulations, calibrated with results from both experimental campaigns — dynamic and static — show that:

- Up to 40–50 kN/mm, higher check rail stiffness increases energy dissi-

pation due to greater contact forces on the inner surface of the wheel flange. Beyond this threshold, dissipation remains approximately constant.

- Increased check rail stiffness reduces lateral displacements and flange-way gap widening, particularly within the range up to 40–50 kN/mm.

To sum up, for stiffness values above 40–50 kN/mm, the effect of further increasing stiffness saturates in terms of flangeway gap variation and energy dissipation, indicating a limit to the benefits of increased rigidity.

General synthesis

The contribution to bridging the gap between static and dynamic measurements was analyzed through the perspective of the three research questions: the location of the measuring system (Q1), the operational and geometric parameters (Q2) and the lateral stiffness of the track components (Q3).

Both experimental measurements and multibody model simulations provide consistent evidence about the lateral loading applied to the track during train passage. This dynamic response was found to be primarily influenced by quasi-static effects related to train configuration and local geometry, rather than by actual dynamic effects such as train speed within the operational range. These influencing factors include track curvature, move direction, train orientation, rail stiffness, and whether a wheelset is leading or trailing. As illustrated in the examples for both facing (Fig. 8.3) and trailing (Fig. 8.4) moves, significant lateral deformation appear specifically in the curve, and these rails widen or narrow depending on the configuration of the wheelset — leading or trailing one. However, the difference of magnitude of the observed deformations in the analyzed velocity range remains limited.

Concerning the dynamic measurements by the measuring train, the SMS system records smaller deformations than on-track sensors because it is not located directly at the wheel/rail contact point, which reduces the discrepancy between static and dynamic measurements.

This raises questions about the recommendation in the standard to place the sensor closer to the wheelset, capturing the maximal dynamic effects or if the goal is to measure the “pure” geometry (i.e., unloaded), closer to static measurement with the digital ruler.

On one hand, if the objective is to evaluate static or quasi-static geometric conditions, such as wear or deformation minimizing the load effects, but

also complying with EN13848-2 (i.e., staying as close as possible to the loaded axles), placing the sensor at the midpoint of the wheelsets may be more appropriate. In this configuration, the dynamic influence on the SMS measurement is reduced, and the resulting data reflect a loaded condition with lower dynamic effects than if the sensor were located directly at the wheel-rail contact point.

On the other hand, if the goal is to capture the maximum dynamic effect, then the sensor should be placed where the gap between static and dynamic measurements is higher. By positioning the sensor closer to the wheelset, the SMS becomes more sensitive to various parameters (facing/trailing, reversed train configuration, etc.), which in some cases leads to significant variations compared to the IAL thresholds (as summarized in Table 8.2).

This reflection highlights the need to clarify whether the SMS system should prioritize dynamic sensitivity or geometric accuracy, particularly when comparing its outputs to reference thresholds such as those established by the IAL. In other words, should the system aim to detect the maximum gap between static and dynamic measurements in order to study and control dynamic effects? Or, on the contrary, should it focus on minimum-gap cases, highlighting quasi-static conditions that primarily characterize the geometry of the track?

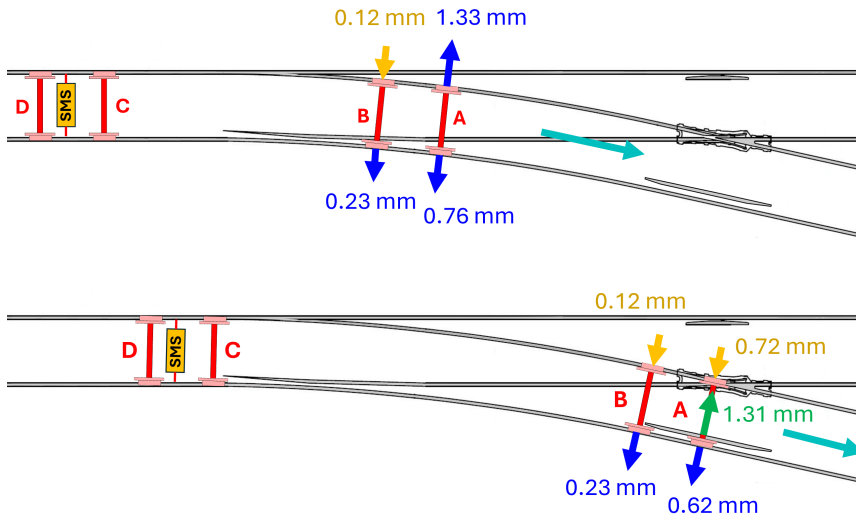


Fig. 8.3 Facing move: rail displacement estimated from the lateral forces computed in the multibody model (MBS). The leading wheelset A first reaches the Closure Panel (top figure), followed by the Crossing Panel (bottom figure), at a velocity of 30 km/h.

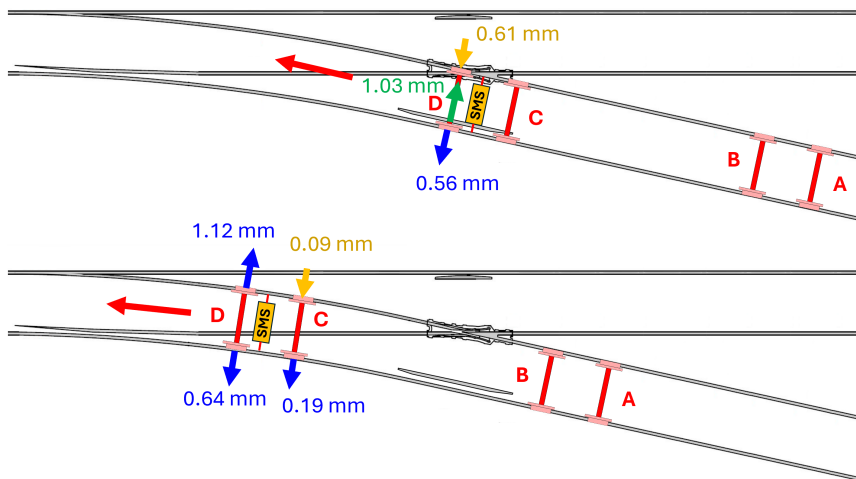


Fig. 8.4 Trailing move: rail displacement estimated from the lateral forces computed in the multibody model (MBS). The leading wheelset D first reaches the Crossing Panel (top figure), followed by the Closure Panel (bottom figure), at a velocity of 30 km/h.

PART V

Conclusion and perspectives

9

General conclusions

THE central challenge of this thesis was to investigate and understand the gap between static measurements — obtained with the digital ruler — and dynamic measurements — performed with the EM130 measuring train.

To address this, the dynamic effect of a train passing over a switch & crossing (S&C), specifically a right-hand turnout, was analyzed through a combination of experimental campaigns and multibody dynamics modeling. Two large-scale experimental campaigns were conducted: one dynamic, using the EM130 measuring train equipped with both on-board and on-track systems, and one static, applying controlled lateral loads with a hydraulic jack. These were complemented by the use and adaptation of a multibody model of the measuring train running through a turnout. The combination of these approaches provided valuable input for an in-depth analysis of the research questions formulated in this project.

First, regarding the influence of the location of the on-board measurement system (SMS), which is positioned between two wheelsets, its measurements do not capture the peak displacements when the wheel/rail contacts are produced. This system configuration provides robust and repeatable measurements, but the geometry measured between wheelsets results from the complex deformation of the rail and substructure caused by localized peaks. Therefore, its measured value depends on its relative position on the

train. This is an important aspect to consider when interpreting SMS data and aligning it with reference thresholds.

Second, operational parameters such as train direction, orientation, and mass distribution were found to have a greater influence on rail deformation than speed within the studied velocity range (≤ 40 km/h). Facing and trailing moves generated different rail responses, particularly in the Crossing Panel, where check rail contact and associated energy dissipation varied significantly. Moreover, the asymmetric distribution of train mass was shown by the model to produce differences in gauge widening and flangeway gap. These findings indicate that the track's response is predominantly governed by quasi-static factors that result from the complex guiding mechanism of the train in the turnout, rather than by purely dynamic effects, both within the operational velocity range of the present turnout and for an alternative turnout with a higher radius of 500 m.

Third, the influence of lateral stiffness was confirmed as a key factor in turnout dynamics design. Static tests revealed nonlinear stiffness regimes, which depend on the track configuration - presence or not of a rail pad - while dynamic measurements helped calibrate the lateral stiffness of rail contacts in the model, accounting for vertical load effects. Numerical simulations calibrated against both campaigns showed that stiffness influences the magnitude of rail displacements and energy dissipation, but its effect reaches a plateau beyond a threshold of stiffness value (40–50 kN/mm). This reinforces the need to consider configuration of track and stiffness for S&C evaluations.

The conclusions drawn from the three research questions demonstrate that the distinction between static and dynamic measurements depends on the context in which the measurements are taken. Depending on the train's direction and orientation, the position of the Switch Measurement System (SMS) may be closer to either the leading or trailing wheelsets, which affects the interpretation of the results. Putting the SMS closer to the wheelsets may make it more sensitive to maximum dynamic effects, as it would be nearer to the wheel/rail contacts. In contrast, when it is closer to the trailing wheelsets, it tends to reflect quasi-static behavior. As a result, the measurement configuration must be carefully considered. Operationally, a wheelset will always be either leading (1st and 3rd) or trailing (2nd and 4th), depending on the train orientation or move direction.

In practice, this work provides guidance for the design and interpretation of inspection strategies: whether the priority is to capture maximum dynamic effects for safety analysis — where the gap between static and

dynamic measurements is largest — or to characterize quasi-static geometry for long-term maintenance — where the gap is smaller. Analyzing the measurement configuration, identifying which parameters are influenced by it, and clarifying the objective of the measuring system are key steps to bridge the gap between static and dynamic measurements.

10

Perspectives

The work conducted in this thesis provides hints for future research and applications. These perspectives concern both possible (improvements and) additional work directly related to AIGUIDYN project and suggestions of how the results can be used for practical railway maintenance. It concerns new experimental campaigns, the extension of the multibody or FEM model, and considerations for maintenance data analysis and normative frameworks.

10.1 Future experimental campaigns

For hypothetical future campaigns similar to those carried out in this work — involving turnout instrumentation and repeated train passages — a series of considerations are listed that could help improve both the campaign itself and its subsequent analysis.

Repeatability Additional repetitions with both train orientations would allow a deeper analysis of the influence of velocity and orientation. These parameters were mainly explored through the multibody model, but experimental validation remains limited.

Sleeper, rail and check rail instrumentation Future experimental campaigns could include the instrumentation of rails, check rails and sleepers

using optical fibers. While sleepers could not be studied in this work, rails and check rails were partially instrumented. The current model only considers these elements when wheel/rail contact occurs. Complementary experiments could consider instrumenting the check rail and rail with optical fibers to capture its deformation distribution even when it is not in direct contact.

Static loaded reference case A static-loaded case could be introduced by stopping a train within the turnout, positioning a wheelset close to a sensor. This would provide a reference load case at 0 km/h, against which dynamic passages at different velocities could be compared.

Other turnout configurations Future projects could extend the analysis to turnouts with larger radius on concrete sleepers or to other layouts, investigating how sensor installation strategies should adapt to different support configurations.

10.2 Model development

Future improvements could be introduced to further develop the numerical models, creating a closer link between numerical predictions and experimental data.

The different track gauges observed in the Closure Panel, as seen in the plan, could be incorporated into the improved model. This update would primarily influence the timing of contact events within the Closure Panel and may slightly affect the resulting variations in lateral displacements.

The Switch Panel, not studied in this work, could also be developed using experimental measurements as input for the model.

Regarding the work of Quoidbach [97], the inclusion of the check rail and the crossing gap in the FEM model remains an important step toward a more realistic representation and understanding of wheel–rail (and check rail) interactions.

10.3 Maintenance and norm

This research raises important considerations regarding maintenance strategies and normative thresholds: should the Switch Measurement System (SMS) be positioned where dynamic effects are most pronounced (maximizing the gap between static and dynamic measurements), or where they

are minimal (reflecting quasi-static geometric conditions such as wear or deformation under load)?

The measuring train captures two distinct aspects of the track: its geometry and its stiffness, whereas a digital ruler only measures the geometry. Its relative position with respect to the wheelsets can also have an impact on the measured data. This raises the question of whether a single IAL threshold should be applied to all measurement contexts — static with a digital ruler, dynamic with the SMS near a trailing wheelset, or dynamic with the SMS near a leading wheelset — given that these situations are not directly comparable.

The methodology developed in this work provides a basis for reconsidering current tolerance thresholds [99], which are now applied equally regardless of whether the measurement is static or dynamic. Considering the measurement configuration during track inspections would help to improve both reliability and safety, by distinguishing between safety-critical dynamic effects and long-term geometric degradation.

PART V

Appendices



Lateral centripetal force on a single wheel

We consider a train with a total mass of 66 100 kg, traveling on a curved track segment. Only half of the train is inside the curve. The mass distribution is not symmetrical: the half where the motor is placed accounts for 55% of the total mass and the other one for 45%.

For the purpose of this calculation, we assume that the lateral centripetal force generated by the curved motion is **entirely distributed by the first 4 wheels** of the half that is inside the curve.

Assumptions:

- Total mass: $m = 66\,100\text{ kg}$
- Radius of curvature: $R = 215\text{ m}$ and $R = 500\text{ m}$
- Centripetal force formula:

$$F_c = \frac{mv^2}{R}$$

- Velocities considered:
 - $v = 5\text{ km/h} = 1.39\text{ m/s}$

A | Lateral centripetal force on a single wheel

- $v = 40 \text{ km/h} = 11.11 \text{ m/s}$
- $v = 70 \text{ km/h} = 19.44 \text{ m/s}$

Case 1: heavier half in the curve ($m_1 = 0.55 \cdot 66\,100 = 36\,355 \text{ kg}$)

Radius [m]	Speed [km/h]	Velocity [m/s]	Lat. force wheel/rail [N]
215	5	1.39	326
215	40	11.11	20 876
215	70	19.44	63 932
500	5	1.39	140
500	40	11.11	8 977
500	70	19.44	27 491

Table A.1 Theoretical centripetal force acting on a single wheel for the heavier half of the train ($m_1 = 36\,355 \text{ kg}$).

Case 2: lighter half in the curve ($m_2 = 0.45 \cdot 66\,100 = 29\,745 \text{ kg}$)

Radius [m]	Speed [km/h]	Velocity [m/s]	Lat. force wheel/rail [N]
215	5	1.39	267
215	40	11.11	17 080
215	70	19.44	52 308
500	5	1.39	115
500	40	11.11	7 344
500	70	19.44	22 492

Table A.2 Theoretical centripetal force acting on a single wheel for the lighter half of the train ($m_2 = 29\,745 \text{ kg}$).



Rail gauge analysis across curve radii

Model R215: gauge at different velocities

Velocity [km/h]	Outer rail displ. [mm]	Inner rail displ. [mm]	Δ gauge [mm]	Outer/Gauge [%]
5	1.244	0.768	2.012	61.83
10	1.255	0.767	2.022	62.07
15	1.266	0.766	2.032	62.30
20	1.298	0.751	2.049	63.35
25	1.307	0.761	2.068	63.20
30	1.334	0.758	2.092	63.77
35	1.405	0.726	2.131	65.93
40	1.437	0.688	2.125	67.62
45	1.484	0.645	2.129	69.72
50	1.572	0.595	2.167	72.53
55	1.672	0.546	2.218	75.37
60	1.743	0.505	2.248	77.56
65	1.782	0.477	2.259	78.91
70	1.804	0.449	2.253	80.07

Table B.1 Facing move. Outer and inner rail highest peak displacement as a percentage of gauge at different speeds obtained from the model simulations. These values correspond to the contact of wheelset A (the 1st one) with the rail and are averaged around the sensor location used in the experimental tests.

Velocity [km/h]	Outer rail displ. [mm]	Inner rail displ. [mm]	Δ gauge [mm]	Outer/Gauge [%]
5	1.021	0.637	1.658	61.58
10	1.025	0.638	1.663	61.64
15	1.034	0.642	1.676	61.69
20	1.055	0.624	1.679	62.84
25	1.072	0.645	1.717	62.43
30	1.102	0.616	1.718	64.14
35	1.120	0.641	1.761	63.60
40	1.160	0.616	1.776	65.32
45	1.239	0.517	1.756	70.56
50	1.236	0.442	1.678	73.66
55	1.273	0.416	1.689	75.37
60	1.336	0.427	1.763	75.78
65	1.384	0.454	1.838	75.30
70	1.482	0.459	1.941	76.35

Table B.2 Facing move. Outer and inner rail second highest peak displacement as a percentage of gauge at different speeds obtained from the model simulations. These values correspond to the contact of wheelset C (the 3rd one) with the rail and are averaged around the sensor location used in the experimental tests.

Velocity [km/h]	Outer rail displ. [mm]	Inner rail displ. [mm]	Δ gauge [mm]	Outer/Gauge [%]
15	1.258	0.761	2.019	62.30
20	1.279	0.755	2.034	62.87
25	1.294	0.761	2.055	62.98
30	1.328	0.757	2.085	63.69
35	1.330	0.766	2.096	63.45
40	1.392	0.750	2.142	65.01
45	1.463	0.667	2.130	68.68
50	1.524	0.579	2.103	72.47
55	1.590	0.519	2.109	75.39
60	1.696	0.491	2.187	77.53
65	1.749	0.489	2.238	78.15
70	1.817	0.485	2.302	78.91
75	1.899	0.464	2.363	80.37
80	1.935	0.430	2.365	81.84

Table B.3 Trailing move. Outer and inner rail highest peak displacement as a percentage of gauge at different speeds obtained from the model simulations. These values correspond to the contact of wheelset B (the 3rd one) with the rail and are averaged around the sensor location used in the experimental tests.

Velocity [km/h]	Outer rail displ. [mm]	Inner rail displ. [mm]	Δ gauge [mm]	Outer/Gauge [%]
15	1.050	0.631	1.681	62.46
20	1.073	0.624	1.697	63.23
25	1.097	0.622	1.719	63.82
30	1.124	0.624	1.748	64.30
35	1.148	0.613	1.761	65.19
40	1.196	0.586	1.782	67.12
45	1.235	0.552	1.787	69.11
50	1.279	0.517	1.796	71.21
55	1.328	0.479	1.807	73.49
60	1.360	0.444	1.804	75.39
65	1.400	0.412	1.812	77.26
70	1.456	0.379	1.835	79.35
75	1.541	0.350	1.891	81.49
80	1.540	0.333	1.873	82.22

Table B.4 Trailing move. Outer and inner rail second highest peak displacement as a percentage of gauge at different speeds obtained from the model simulations. These values correspond to the contact of wheelset D (the 1st one) with the rail and are averaged around the sensor location used in the experimental tests.

Model R500: gauge at different velocities

Velocity [km/h]	Outer rail displ. [mm]	Inner rail displ. [mm]	Δ gauge [mm]	Outer/Gauge [%]
5	1.094	0.784	1.878	58.27
10	1.096	0.783	1.879	58.33
15	1.101	0.781	1.882	58.49
20	1.106	0.779	1.885	58.65
25	1.106	0.784	1.890	58.52
30	1.121	0.778	1.899	59.02
35	1.137	0.762	1.899	59.87
40	1.151	0.745	1.896	60.71
Mean				58.86

Table B.5 Facing move. Outer and inner rail displacement as a percentage of gauge at different speeds for R500. These values correspond to the contact of wheelset A (the 1st one) with the rail and are averaged around the sensor location used in the experimental tests.

Velocity [km/h]	Outer rail displ. [mm]	Inner rail displ. [mm]	Δ gauge [mm]	Outer/Gauge [%]
15	1.091	0.779	1.870	58.34
20	1.091	0.779	1.870	58.34
25	1.098	0.785	1.883	58.33
30	1.113	0.777	1.890	58.89
35	1.110	0.782	1.892	58.67
40	1.136	0.766	1.902	59.72
Mean				58.72

Table B.6 Trailing move. Outer and inner rail displacement as a percentage of gauge at different speeds for R500. These values correspond to the contact of wheelset B (the 3rd one) with the rail and are averaged around the sensor location used in the experimental tests.



Statistical analysis of experimental track measurements

Two different trends can be observed in the graph shown in Figure 6.25. Up to the 4th run (included), the red curve — representing the trailing move — increases noticeably, while from the 6th train passage to the last one, the curves become more stable. This trend suggests that the rail system is still adapting or settling under repeated loading. From the 6th passage to the last one, all curves tend to stabilize, indicating that the system reaches a stable state. In this phase, the influence of the passage order becomes less pronounced.

Several ANOVA tests are performed to analyze the influence of the following variables:

- Order: order in which the measurements were taken
- Location: motor side vs. non-motor side
- Trajectory: facing vs. trailing move
- Speed: 5, 15, 20, 25, 30, 35, 40 km/h

The first ANOVA is performed separately for the first four runs without the variable Speed (Table C.1), and then a second one including this variable (Table C.2).

In both cases, the variables Order and Trajectory are found to be statistically significant.

Table C.1 ANOVA - First 4 Runs

Source	Nparm	DF	Sum of Sq.	F Ratio	p-value
Order	1	1	0.09674405	16.9428	0.0026*
Trajectory	1	1	0.27196225	47.6289	<0.0001*
Location	1	1	0.02755600	4.8259	0.0556
Order*Trajectory	1	1	0.00307520	0.5386	0.4817
Order*Location	1	1	0.00000405	0.0007	0.9793
Trajectory*Location	1	1	0.00378225	0.6624	0.4367

Table C.2 ANOVA - First 4 Runs with Speed

Source	Nparm	DF	Sum of Sq.	F Ratio	p-value
Order	1	1	0.09185460	15.1486	0.0046*
Trajectory	1	1	0.27196225	44.8519	0.0002*
Location	1	1	0.02755600	4.5445	0.0656
Order*Trajectory	1	1	0.00307520	0.5072	0.4966
Order*Location	1	1	0.00000405	0.0007	0.9800
Trajectory*Location	1	1	0.00378225	0.6238	0.4524
Speed	1	1	0.00288171	0.4753	0.5101

A third ANOVA is also performed for all runs (Table C.3). In this case, the variables Order, Trajectory, the interaction Order*Trajectory and the interaction Speed*Trajectory are statistically significant.

An ANOVA is then performed for the last four runs (6th to 9th, both included) and the results are presented in C.4. In this more stable phase, the variable Order is no longer significant so it confirms the stabilization observed in the graph.

Finally, an ANOVA is performed on runs 6 to 9, excluding the second measurement of the facing move at 40 km/h (Table C.5). This analysis obtained the significant influence of Trajectory, Location and several interaction terms.

Table C.3 ANOVA - All Data

Source	Nparm	DF	Sum of Sq.	F Ratio	p-value
Order	1	1	0.25486462	36.9068	<0.0001*
Trajectory	1	1	0.22467600	32.5352	<0.0001*
Location	1	1	0.02777778	4.0225	0.0550
Order*Trajectory	1	1	0.04476185	6.4819	0.0169*
Order*Location	1	1	0.00218407	0.3163	0.5785
Trajectory*Location	1	1	0.02689600	3.8948	0.0588
Speed	1	1	0.00197120	0.2854	0.5975
Speed*Trajectory	1	1	0.06773564	9.8088	0.0041*

Table C.4 ANOVA - Runs 6 to 9 with the second measurement of the facing move at 40 km/h

Source	Nparm	DF	Sum of Sq.	F Ratio	p-value
Order	1	1	0.00888040	2.9976	0.1216
Trajectory	1	1	0.00514806	1.7378	0.2239
Location	1	1	0.00566256	1.9114	0.2042
Trajectory*Location	1	1	0.03676806	12.4113	0.0078*
Speed	1	1	0.00175781	0.5934	0.4633
Speed*Trajectory	1	1	0.07335923	24.7629	0.0011*
Speed*Location	1	1	0.02366823	7.9894	0.0223*

Table C.5 ANOVA - Runs 6 to 9 without the second measurement of the facing move at 40 km/h

Source	Nparm	DF	Sum of Sq.	F Ratio	p-value
Order	1	1	0.00388294	3.2823	0.1129
Trajectory	1	1	0.01672272	14.1357	0.0071*
Location	1	1	0.01520595	12.8536	0.0089*
Trajectory*Location	1	1	0.01811078	15.3091	0.0058*
Speed	1	1	0.00885280	7.4833	0.0291*
Speed*Trajectory	1	1	0.03591009	30.3548	0.0009*
Speed*Location	1	1	0.00709413	5.9967	0.0442*

In conclusion, the measurement order is statistically significant in the early phase of the experiment. In the later phase, however, the system stabilizes, and other variables, such as trajectory, location, and speed, become more relevant. Due to the limited repetability in the experimental measurements, these variables are interpreted based on the model results.



Publications and conference contributions

- R. Acosta Suñé, P. Fisette, N. Docquier. Oral presentation and proceedings paper: *Analysis of the Track Irregularities in Railway Switch & Crossing via the Multibody Approach*, ECCOMAS Thematic Conference on Multibody Dynamics, Lisbon, Portugal, 2023. [100]
- R. Acosta Suñé, P. Fisette, N. Docquier. Oral presentation and proceedings paper: *Dynamic Measurement of Switch & Crossing Geometry*, Proceedings of the Sixth International Conference on Railway Technology, Civil-Comp Press, 2024. [101]
- R. Acosta Suñé, P. Fisette, N. Docquier. Journal paper (submitted, 2026): *Field Measurement of Turnout Geometry: Influence of the On-Board Sensor System Location*, submitted to *Vehicle System Dynamics*.

Bibliography

- [1] SNCB. *Voyageurs montés dans les trains un jour ouvrable moyen – octobre 2024*. Accessed: July 2025. 2025. URL: <https://www.belgiantrain.be/-/media/corporate/2025/voyageurs-montes-2024-def.ashx>.
- [2] Infrabel. *Annual Report 2019*. Accessed: July 2025. 2019. URL: https://infrabel.be/sites/default/files/generated/files/report/AnnualReport2019_0.pdf.
- [3] Région de Bruxelles-Capitale. *Plan régional de mobilité Good Move 2020–2030*. Accessed: July 2025. 2020. URL: https://environnement.brussels/sites/default/files/user_files/1_pres_bmd_goodmovephbkdm_0.pdf.
- [4] SPF Mobilité et Transports and Georges Gilkinet. *Vision Rail 2040: Le rail, colonne vertébrale de la mobilité en Belgique*. Online. Approved by the Council of Ministers on May 6, 2022. 2022. URL: <https://gilkinet.belgium.be/sites/default/files/articles/Vision%20Rail%202040.pdf>.
- [5] Infrabel. *Track length and number of track devices per region*. <https://opendata.infrabel.be/explore/dataset/kilometers-sporen-en-aantal-spoortoestellen-per-gewest/>. Accessed: July 2025. 2025.
- [6] Plasser & Theurer Export von Bahnbaumaschinen GmbH. *Typenplan EM130 – Track geometry measuring vehicle*. Technical Drawing E.EM130.05D_01_KO3255. Internal technical documentation for Belgian EM130 measurement train. Plasser & Theurer Export von Bahnbaumaschinen GmbH, 2008.

- [7] *Railway applications – Track – Track geometry quality – Part 2: Measuring systems – Track recording vehicles*. Supersedes EN 13848-2:2006. Available through national standardization bodies such as BSI (UK), NBN (Belgium), or DIN (Germany). Brussels: European Committee for Standardization (CEN), 2020.
- [8] *Railway applications – Track – Track geometry quality – Part 1: Characterization of track geometry*. English version, identical to DIN EN 13848-1:2019-06. Brussels: European Committee for Standardization (CEN), 2019.
- [9] Raúl Acosta Suñé. *AIGUIDYN PhD Project*. Accessed: 2025-12-10. 2021. URL: <https://www.robotran.be/2021/12/21/aiguidyn-phd-project/>.
- [10] Niloofar Minbashi et al. “Turnout Degradation Modelling Using New Inspection Technologies: A Literature Review”. In: *Current Trends in Reliability, Availability, Maintainability and Safety*. Ed. by Uday Kumar et al. Lecture Notes in Mechanical Engineering. Springer International Publishing, 2016, pp. 49–63. DOI: 10.1007/978-3-319-23597-4_5.
- [11] Mihai Florin Rusu. “Automation of railway switch and crossing inspection”. Accessed: August 2025. PhD Thesis. University of Birmingham, 2015. URL: <https://etheses.bham.ac.uk/7348/1/Rusu17PhD.pdf>.
- [12] Allan M. Zarembski, Thomas L. Euston, and John W. Palese. “Development, Implementation, and Validation of an Automated Switch Inspection Vehicle (ASIV)”. In: *International Heavy Haul Association Conference*. Accessed: August 2025. Calgary, Canada, 2011. URL: <https://railknowledgebank.com/Presto/content/GetDoc.axd?ctID=MTk4MTRjNDUtdWQOMy00OTBmLT1lYWUtZWZjM2U20TE0ZDY3&rID=NDIxOQ==&pID=NzIx>.
- [13] K. Zschiesche and A. Reiterer. “Optical measurement system for monitoring railway infrastructure—A review”. en. In: *Appl. Sci.* 14.19 (Sept. 2024). DOI: 10.3390/app14198801, p. 8801.
- [14] Jana Izvoltova et al. “3D Laser Scanning of Railway Line”. In: *Communications - Scientific letters of the University of Zilina* 15.4 (2013). DOI: 10.26552/com.C.2013.4.80-84, pp. 80–84. ISSN: 13354205, 25857878. DOI: 10.26552/com.C.2013.4.80-84. (Visited on 01/16/2025).

- [15] Clemens Schoone. "Monitoring a turnout of a railway or tramway line". Patent EP2165915A2. European Patent Application, published on 24 March 2010. 2010. URL: <https://worldwide.espacenet.com/patent/search/family/042933013/publication/EP2165915A2>.
- [16] Jeremiasz Hauck and Piotr Gniado. "Laser Scan Compression for Rail Inspection". In: *Sensors* 24.20 (2024), p. 6722. DOI: 10.3390/s24206722. URL: <https://doi.org/10.3390/s24206722>.
- [17] András Farkas. "Measurement of Railway Track Geometry: A State-of-the-Art Review". In: *Periodica Polytechnica Transportation Engineering* 48.1 (2020). DOI: 10.3311/PPtr.14145, pp. 76–88. ISSN: 1587-3811, 0303-7800. DOI: 10.3311/PPtr.14145. (Visited on 01/29/2025).
- [18] Graeme James Yeo. "Monitoring Railway Track Condition Using Inertial Sensors on an In-Service Vehicle". PhD thesis. Birmingham, UK: University of Birmingham, 2017.
- [19] Network Rail. *New Measurement Train (NMT)*. <https://www.networkrail.co.uk/running-the-railway/looking-after-the-railway/our-fleet-machines-and-vehicles/new-measurement-train-nmt/>. Accessed: August 2025. n.d.
- [20] P. Weston et al. "Perspectives on railway track geometry condition monitoring from in-service railway vehicles". In: *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility* 53.7 (July 2015). DOI: 10.1080/00423114.2015.1034730, pp. 1063–1091. DOI: 10.1080/00423114.2015.1034730.
- [21] Aleš Hába et al. "S-CODE: Switches and Crossings Optimal Design and Evaluation". In: *Proceedings of the VII. International Scientific Conference of the Faculty of Transport Engineering*. Pardubice, Sept. 2018.
- [22] André Van Leuven. *New concepts for turnouts in urban rail transit infrastructures*. Tech. rep. Deliverable 7.2. Project acronym: TURNOUTS. Contract N° TST3-CT-2003-505592, FP-6-505592. Funded by the European Community under the Sixth Framework Programme (Priority 6: Sustainable Development, Global Change & Ecosystems). Belgium: Dynamics, Structures & Systems International (D2S), Jan. 2007.

- [23] Shift2Rail Joint Undertaking. *IN2TRACK Final Brochure*. https://projects.shift2rail.org/s2r_ip3_n.aspx?p=IN2TRACK. Research into Enhanced Track, Switches and Structures. Final results of the IN2TRACK project under Horizon 2020, Grant Agreement No. 730841. 2019.
- [24] Union Internationale des Chemins de Fer (UIC). *INNOTRACK – Innovative Track Systems*. <https://cordis.europa.eu/project/id/31415>. Funded by the EU Sixth Framework Programme (FP6), Grant Agreement No. TIP5-CT-2006-031415. Project duration: September 2006 – December 2009. 2009.
- [25] Elias Kassa et al. “Optimisation of track gauge and track stiffness along a switch using a multibody simulation tool”. In: *Proceedings of the 21st International Symposium on Dynamics of Vehicles on Roads and Tracks (IAVSD)*. Also included in INNOTRACK Deliverable D3.1.5. IAVSD. Stockholm, Sweden, 2009. URL: https://www.charmec.chalmers.se/innotrack/deliverables/sp3/d315-f4-recommendation_of_and_scientific_basis_for_optimisation_of_switches_and_crossings_part1.pdf.
- [26] Wanming Zhai, Kaiyun Wang, and Chengbiao Cai. “Fundamentals of vehicle–track coupled dynamics”. In: *Vehicle System Dynamics* 47.11 (2009), pp. 1349–1376. DOI: 10.1080/00423110802621561. URL: <http://dx.doi.org/10.1080/00423110802621561>.
- [27] Wanming Zhai and Kaiyun Wang. “Lateral interactions of trains and tracks on small-radius curves: simulation and experiment”. In: *Vehicle System Dynamics* 44.Suppl. (2006), pp. 520–530.
- [28] Elias Kassa and Jens C.O. Nielsen. “Dynamic interaction between train and railway turnout: full-scale field test and validation of simulation models”. In: *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility* 46.S1 (2008). DOI: 10.1080/00423110801993144, pp. 521–534. DOI: 10.1080/00423110801993144.
- [29] J. Drozdziel, B. Sowiński, and W. Groll. “The Effect of Railway Vehicle-Track System Geometric Deviations on its Dynamics in the Turnout Zone”. In: *Vehicle System Dynamics* 33.sup1 (1999), pp. 641–652. DOI: 10.1080/00423114.1999.12063118.

- [30] J. Shih et al. "Understanding the impact of train run-throughs on railway switches using finite element analysis". In: *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit* 233.4 (2019). DOI: 10.1177/0954409718795759, pp. 359–369. DOI: 10.1177/0954409718795759.
- [31] Jou-Yi Shih et al. "Dynamic characteristics of a switch and crossing on the West Coast main line in the UK". In: *Railway Engineering Science* 30.2 (2022), pp. 183–203. DOI: 10.1007/s40534-021-00269-4.
- [32] Xiangming Liu et al. "Experimental tools for railway crossing condition monitoring". In: *Measurement* 129 (2018), pp. 424–435. DOI: 10.1016/j.measurement.2018.07.062.
- [33] DB Systemtechnik GmbH. *ESAH – Diagnostic technology: electronic systems analysis at frogs*. Technical Report. Accessed: August 2025. DB Systemtechnik GmbH, 2018. URL: https://www.db-systemtechnik.de/resource/blob/3271262/9d6b94f31fb96f4c83c6c44de193ffc5/Aktuell_weichenherzstueckdiagnose_esah_englisch-data.pdf.
- [34] Can Liu et al. "Experimental Study on the Transverse Stiffness of WJ-8 Rail Fastening". In: *Applied Mechanics and Materials* 596 (2014), pp. 3–6. DOI: 10.4028/www.scientific.net/AMM.596.3. URL: <https://doi.org/10.4028/www.scientific.net/AMM.596.3>.
- [35] Xiang Liu et al. "Experimental study on mechanical and deformation properties of rail WJ-8B type fastener system in longitudinal push-out". In: *Steel and Composite Structures* 54.4 (2025), pp. 359–370. DOI: 10.12989/scs.2025.54.4.359. URL: <https://doi.org/10.12989/scs.2025.54.4.359>.
- [36] Xian Wang et al. "Research on fastener lateral nonlinear stiffness characteristics and its impact on wheel-rail stick-slip vibration". In: *Nonlinear Dynamics* 112.19 (Oct. 2024), pp. 16917–16939. DOI: 10.1007/s11071-024-09990-1.
- [37] Z. Chen, M. Berg, and M. Dhanasekar. "Dynamic response of railway turnouts to wheel impacts and its implications to maintenance". In: *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit* 230.2 (2016), pp. 390–402. DOI: 10.1177/0954409716642485.

- [38] A. M. Zarembski. "Rail Rollover - The State of the Art". In: *Bulletin of the American Railway Engineering Association* 79.664 (Sept. 1977), pp. 12–20.
- [39] J. Choros, B. Marquis, and M. N. Coltman. "Prevention of Derailments Due to Concrete Tie Rail Seat Deterioration". In: *ASME/IEEE Joint Rail Conference and Internal Combustion Engine Division Spring Technical Conference*. American Society of Mechanical Engineers. 2007, pp. 173–181.
- [40] B. P. Marquis, M. Muhlanger, and D. Y. Jeong. "Effect of Wheel/Rail Loads on Concrete Tie Stresses and Rail Rollover". In: *ASME Rail Transportation Division Fall Technical Conference*. American Society of Mechanical Engineers. 2011, pp. 143–150. DOI: 10.1115/RTDF2011-67025.
- [41] H. Wu and B. Kerchof. "Management of Wheel/Rail Interface to Prevent Rail Rollover Derailments". In: *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit* 228.6 (2014), pp. 673–686. DOI: 10.1177/0954409714522222.
- [42] Jabbar Ali Zakeri and Meraj Barati. "Utilizing the track panel displacement method for estimating vertical load effects on the lateral resistance of continuously welded railway track". In: *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit* 0.0 (2015), pp. 1–6. DOI: 10.1177/0954409713508937.
- [43] E.G. Berggren. "Railway Track Stiffness: Dynamic Measurements and Evaluation for Efficient Maintenance". PhD thesis. Stockholm, Sweden: KTH Royal Institute of Technology, 2009.
- [44] E.G. Berggren, A. Nissen, and B.S. Paulsson. "Track deflection and stiffness measurements from a track recording car". In: *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit* 228.5 (2014), pp. 570–580. DOI: 10.1177/0954409713506205.
- [45] M. Sadri and Michaël Steenbergen. "Effects of railway track design on the expected degradation: Parametric study on energy dissipation". In: *Journal of Sound and Vibration* 419 (2018). TU Delft - Railway Engineering, pp. 321–338. DOI: 10.1016/j.jsv.2018.01.029. URL: <https://doi.org/10.1016/j.jsv.2018.01.029>.

- [46] Abdollah Malekjafarian et al. "Railway Track Loss-of-Stiffness Detection Using Bogie Filtered Displacement Data Measured on a Passing Train". In: *Infrastructures* 6.6 (2021), p. 93. DOI: 10.3390/infrastructures6060093. URL: <https://www.mdpi.com/2412-3811/6/6/93>.
- [47] Yann Bezin and Björn A. Pålsson. "Multibody simulation benchmark for dynamic vehicle-track interaction in switches and crossings: modelling description and simulation tasks". In: *Vehicle System Dynamics* 61.3 (2021), pp. 644–659. DOI: 10.1080/00423114.2021.1942079. eprint: <https://doi.org/10.1080/00423114.2021.1942079>. URL: <https://doi.org/10.1080/00423114.2021.1942079>.
- [48] Simon Iwnick. "Manchester Benchmarks for Rail Vehicle Simulation". In: *Vehicle System Dynamics* 30.3-4 (1998), pp. 295–313. DOI: 10.1080/00423119808969454.
- [49] Yan Bezin et al. "Multibody simulation benchmark for dynamic vehicle-track interaction in switches and crossings: results and method statements". In: *Vehicle System Dynamics* 61.3 (2023). DOI: 10.1080/00423114.2021.1959038, pp. 660–697. DOI: 10.1080/00423114.2021.1959038.
- [50] Björn A. Pålsson et al. "A comparison of track model formulations for simulation of dynamic vehicle-track interaction in switches and crossings". In: *Vehicle System Dynamics* (2021). DOI: 10.1080/00423114.2021.1983183. DOI: 10.1080/00423114.2021.1983183.
- [51] Björn A. Pålsson. "A Parameterized Turnout Model for Simulation of Dynamic Vehicle-Turnout Interaction with an Application to Crossing Geometry Assessment". In: *Lecture Notes in Mechanical Engineering*. Springer, 2020, pp. 351–358. DOI: 10.1007/978-3-030-38077-9_41. URL: https://doi.org/10.1007/978-3-030-38077-9_41.
- [52] Wanming Zhai. *Vehicle-Track Coupled Dynamics: Theory and Applications*. Singapore: Springer Nature Singapore Pte Ltd and Science Press, 2020. ISBN: 978-981-32-9282-6. DOI: 10.1007/978-981-32-9283-3. URL: <https://doi.org/10.1007/978-981-32-9283-3>.
- [53] K. Knothe and S.L. Grassie. "Modeling of railway track and vehicle-track interaction at high frequencies". In: *Vehicle System Dynamics* 22.3/4 (1993), pp. 209–262.

- [54] G. Diana et al. "Interaction between railroad superstructure and railway vehicles". In: *Vehicle System Dynamics* 23.Suppl. (1994), pp. 75–86.
- [55] J.C.O. Nielsen and A. Igeland. "Vertical dynamic interaction between train and track – influence of wheel and track imperfections". In: *Journal of Sound and Vibration* 187.5 (1995), pp. 825–839.
- [56] B. Ripke and K. Knothe. "Simulation of high frequency vehicle-track interactions". In: *Vehicle System Dynamics* 24.Suppl. (1995), pp. 72–85.
- [57] W.M. Zhai, C.B. Cai, and S.Z. Guo. "Coupling model of vertical and lateral vehicle/track interactions". In: *Vehicle System Dynamics* 26.1 (1996), pp. 61–79.
- [58] R.D. Frohling. "Low frequency dynamic vehicle–track interaction: Modelling and simulation". In: *Vehicle System Dynamics* 28.Suppl. (1998), pp. 30–46.
- [59] J. Oscarsson and T. Dahlberg. "Dynamic train/track/ballast interaction – Computer models and full-scale experiments". In: *Vehicle System Dynamics* 28.Suppl. (1998), pp. 73–84.
- [60] K. Popp, H. Kruse, and I. Kaiser. "Vehicle-track dynamics in the mid-frequency range". In: *Vehicle System Dynamics* 31.5–6 (1999), pp. 423–464.
- [61] Y.Q. Sun and M. Dhanasekar. "A dynamic model for the vertical interaction of the rail track and wagon system". In: *International Journal of Solids and Structures* 39 (2002), pp. 1337–1359.
- [62] Y.Q. Sun, M. Dhanasekar, and D. Roach. "A three-dimensional model for the lateral and vertical dynamics of wagon-track systems". In: *Journal of Rail and Rapid Transit* 217 (2003), pp. 31–45.
- [63] X.Y. Lei and L.J. Mao. "Dynamic response analyses of vehicle and track coupled system on track transition of conventional high speed railway". In: *Journal of Sound and Vibration* 271.3–5 (2004), pp. 1133–1146.
- [64] A. Lundqvist and T. Dahlberg. "Dynamic train/track interaction including model for track settlement evolution". In: *Vehicle System Dynamics* 41.Suppl. (2004), pp. 667–676.
- [65] Y.L. Xu and Q.S. Ding. "Interaction of railway vehicles with track in cross-winds". In: *Journal of Fluids and Structures* 22.3 (2006), pp. 295–314.

- [66] M. Pletz et al. "A Dynamical Wheel Set - Crossing Model Regarding Impact, Sliding and Deformation". In: *Proceedings of the 8th International Conference on Contact Mechanics and Wear of Rail/Wheel Systems*. Firenze, Italy: Ab Editore, 2009, pp. 801–808.
- [67] M. Pletz, W. Daves, and H. Ossberger. "Dynamic Finite Element Model of a Wheel Passing a Crossing Nose". In: *Proceedings of the 10th International Conference on Computational Structures Technology*. Ed. by B.H.V. Topping. Stirlingshire, UK: Civil-Comp Press, 2010.
- [68] Martin Pletz, Werner Daves, and Heinz Ossberger. "A wheel passing a crossing nose: Dynamic analysis under high axle loads using finite element modelling". In: *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit* 226.6 (2012), pp. 603–611. DOI: 10.1177/0954409712448038.
- [69] C. Andersson and T. Dahlberg. "Load impacts at railway turnout crossing". In: *Vehicle System Dynamics* 33 (1999), pp. 131–142.
- [70] S. Alfi and S. Bruni. "Mathematical modelling of train–turnout interaction". In: *Vehicle System Dynamics* 47 (2009), pp. 551–574.
- [71] J. Xu, P. Wang, L. Wang, et al. "Effects of profile wear on wheel–rail contact conditions and dynamic interaction of vehicle and turnout". In: *Advances in Mechanical Engineering* 8 (2016), pp. 1–14.
- [72] N. Bosso et al. "Effects of geometric track irregularities on vehicle dynamic behaviour when running through a turnout". In: *Vehicle System Dynamics* 61.3 (2023). DOI: 10.1080/00423114.2021.1957127, pp. 782–798. DOI: 10.1080/00423114.2021.1957127.
- [73] S. Bruni et al. "State-of-the-art and challenges of railway and road vehicle dynamics with multibody dynamics approaches". In: *Multibody System Dynamics* 49 (2020). DOI: 10.1007/s11044-020-09735-z, pp. 1–32. DOI: 10.1007/s11044-020-09735-z.
- [74] E. J. M. Hiensch and N. Burgelman. "Switch Panel wear loading – a parametric study regarding governing train operational factors". In: *Vehicle System Dynamics* 55.9 (2017). DOI: 10.1080/00423114.2017.1313435, pp. 1384–1404. DOI: 10.1080/00423114.2017.1313435.

- [75] Rodrigo Fernando Lagos et al. "Rail vehicle passing through a turnout: analysis of different turnout designs and wheel profiles". In: *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit* 0 (2012). DOI: 10.1177/0954409712445114, pp. 1–16. DOI: 10.1177/0954409712445114.
- [76] Dassault Systèmes. *Simpac – Multibody Simulation Software*. <https://www.3ds.com/products/simulia/simpac>. Accessed: August 2025. n.d.
- [77] Erik Kassa and Jens C. O. Nielsen. "Lateral dynamic interaction between train and railway turnout: field measurements and validation of simulation models". In: *Vehicle System Dynamics* 46.S1 (2008), pp. 521–534. DOI: 10.1080/00423110801993144.
- [78] Z. Ren, S. Sun, and W. Zhai. "Study on lateral dynamic characteristics of vehicle/turnout system". In: *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility* 43.4 (2005), pp. 285–303. DOI: 10.1080/00423110500083262. URL: <http://dx.doi.org/10.1080/00423110500083262>.
- [79] Geismar. *Règle digitale de mesure de la géométrie de la voie et des appareils de voie – GARNET*. Technical brochure. Accessed via [RD-garnet-pms-fr-expo.pdf](#). Geismar, Dec. 2021.
- [80] Micro-Epsilon France S.A.R.L. *induSENSOR DTA (LVDT) – Capteurs de déplacement inductifs linéaires*. Fiche technique, consultée en septembre 2025. Saint Germain en Laye, France, 2025. URL: <https://www.micro-epsilon.fr/distance-sensors/inductive-sensors-lvdt/indusensor-dta-gauge/>.
- [81] SICK AG. *Laser Sick O2*. Equipment specification. Retrieved from <https://www.sick.com>. 2025.
- [82] Micromega Dynamics. *Recovib Tiny: Miniature Vibration Recorder*. <https://micromega-dynamics.com/products/recovib/miniature-vibration-recorder/>. Accessed: September 24, 2025. 2025.
- [83] N. Docquier, A. Poncelet, and P. Fisette. "ROBOTRAN: A powerful symbolic generator of multibody models". In: *Mechanical Sciences* 4.1 (2013), pp. 199–219. ISSN: 2191-916X. DOI: 10.5194/ms-4-199-2013. URL: <https://ms.copernicus.org/articles/4/199/2013/>.

- [84] P. E. Nikravesh and E. J. Haug. "Generalized Coordinate Partitioning for Analysis of Mechanical Systems with Nonholonomic Constraints". In: *Journal of Mechanisms, Transmissions, and Automation in Design* 105.3 (1983), pp. 379–384. DOI: 10.1115/1.3258702.
- [85] Jean-Claude Samin and Paul Fiset. *Symbolic Modeling of Multibody Systems*. Dordrecht: Springer Science+Business Media, 2003. ISBN: 978-90-481-6425-7. DOI: 10.1007/978-94-017-0287-4.
- [86] N. Docquier et al. *Quandyga Engineering*. <https://www.quandyga.com>. Accessed: August 2025. 2023.
- [87] J.J. Kalker. "Wheel-rail rolling contact theory". In: *Wear* 144.1 (1991), pp. 243–261. ISSN: 0043-1648. DOI: [https://doi.org/10.1016/0043-1648\(91\)90018-P](https://doi.org/10.1016/0043-1648(91)90018-P). URL: <https://www.sciencedirect.com/science/article/pii/004316489190018P>.
- [88] Oldřich Polach. "Characteristic parameters of nonlinear wheel/rail contact geometry". In: *Vehicle System Dynamics* 48.1 (2010), pp. 19–36. DOI: 10.1080/00423111003668203. URL: <http://dx.doi.org/10.1080/00423111003668203>.
- [89] Edwin A.H. Vollebregt, Christoph Weidemann, and Andreas Kienberger. "Use of CONTACT in multi-body vehicle dynamics and profile wear simulation: Initial results". In: *Proceedings of the IAVSD Symposium*. 2011. URL: <https://kalkersoftware.org/downloads/vollebregt2011b-iavsd2011.pdf>.
- [90] Emanuele Crivellente. "An Extension of Kalker's Linear Theory for the Simulation of Limit Conditions". Master's Thesis. Politecnico di Milano, 2013. URL: <https://www.politesi.polimi.it/handle/10589/83182>.
- [91] Hugo Magalhães et al. "Implementation of a non-Hertzian contact model for railway dynamic application". In: *Multibody System Dynamics* 48 (2020), pp. 41–78. DOI: 10.1007/s11044-019-09688-y.
- [92] Luca Romano, Marco Maglio, and Stefano Bruni. "Transient wheel–rail rolling contact theories". In: *Tribology International* 186 (2023). DOI: 10.1016/j.triboint.2023.108611.
- [93] A. Jaschinski. "On the application of similarity laws to a scaled railway bogie model". Doctoral Thesis. Delft, The Netherlands: Delft University of Technology, 1990. URL: <https://resolver.tudelft.nl/uuid:b0bd4e61-8edf-41f2-a996-d3f3b5266687>.

- [94] P. J. Vermeulen and K. L. Johnson. “Contact of Nonspherical Elastic Bodies Transmitting Tangential Forces”. In: *Journal of Applied Mechanics* 31.2 (1964), pp. 338–340. DOI: 10.1115/1.3629626. URL: <https://asmedigitalcollection.asme.org/appliedmechanics/article/31/2/338/388614/Contact-of-Nonspherical-Elastic-Bodies>.
- [95] Ahmed A. Shabana. “Nadal’s Formula and High Speed Rail Derailments”. In: *Journal of Computational and Nonlinear Dynamics* 7.4 (2012), p. 041003. DOI: 10.1115/1.4006730. URL: <https://doi.org/10.1115/1.4006730>.
- [96] Marine Boxus. “Impact of railway dynamics on wheel-rail interaction: modeling, sensitivity analysis and experimental validation”. Supervisors: Nicolas Docquier, Paul Fisette. Master’s thesis. Louvain-la-Neuve, Belgium: École polytechnique de Louvain, Université catholique de Louvain, 2024. URL: <http://hdl.handle.net/2078.1/thesis:45856>.
- [97] Quentin Quoidbach. “Analyse de l’impact de la dynamique du train sur la précision des mesures géométriques ferroviaires: Modélisation et validation expérimentale”. École Polytechnique de Louvain. Promoteurs: Laurent Delannay, Nicolas Docquier. Lecteur: Hadrien Rattez. MA thesis. Louvain-la-Neuve, Belgique: Université catholique de Louvain, 2025.
- [98] P. Fisette, K. Lipinski, and J.C. Samin. “Dynamic Behaviour Comparison between Bogies : Rigid or Articulated Frame, Wheelset or Independent Wheels”. In: *Vehicle System Dynamics* 25.sup1 (1996), pp. 152–174. DOI: 10.1080/00423119608969193. eprint: <https://doi.org/10.1080/00423119608969193>. URL: <https://doi.org/10.1080/00423119608969193>.
- [99] Infrabel. *PTR V2.1 – Prescriptions Techniques et Réglementaires de la Voie*. Internal document. AVIS 20 I-AM/2019. Not publicly available. Bruxelles, 2019.
- [100] Raúl Acosta Suñé, Paul Fisette, and Nicolas Docquier. “Analysis of the Track Irregularities in Railway Switch & Crossing via the Multi-body Approach”. In: *ECCOMAS Thematic Conference on Multibody Dynamics*. Lisbon, Portugal, July 2023.

- [101] R. Acosta Suñé, P. Fisette, and N. Docquier. “Dynamic Measurement of Switch & Crossing Geometry”. In: *Proceedings of the Sixth International Conference on Railway Technology: Research, Development and Maintenance*. Ed. by J. Pombo. Vol. 7. Civil-Comp Conferences. Edinburgh, United Kingdom: Civil-Comp Press, 2024, Paper 6.3. DOI: 10.4203/coc.7.6.3.

