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# Asymmetric Two-Axle Rail Vehicle with Active Wheelset Steering

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## Abstract

A concept for a two-axle vehicle with a focus on simplicity and cost-effectiveness is proposed as a future generation regional rail vehicle. The proposed vehicle is equipped with a novel asymmetric running gear configuration with active suspension to alleviate the issue of poor curving performance and ride comfort commonly found in two-axle vehicles. This study describes the proposed running gear configuration and the modelling and simulation of the active wheelset steering designed to improve the vehicle's curving performance. The active wheelset steering is designed around a feedback control loop for the two actuators fitted on the vehicle with a focus on having a simple, cost-effective system. A control strategy based on radial wheelset position and actuator length was chosen. The simulation results demonstrate an improved curving performance and confirm the effectiveness of the active wheelset steering.

**Keywords:** two-axle vehicle, rail vehicle, running gear, active wheelset steering, radial steering, regional train

# 1 Introduction

Regional railways provide an environmentally friendly means of transportation within regions and serve as feeders for the main railway network. They have also been credited for promoting territorial integration in Europe and have brought great economic growth to the connected regions [1, 2]. However, many of these lines have been abandoned due to high operational costs.

One of the main contributors to the operational costs of rail transport is the vehicle. This reflects the trajectory of rail vehicle development. Over the past few decades, rail vehicles have grown to be faster, produce less wheel-rail wear, and have better ride comfort. However, these improvements have also led to increased weight and cost. These side effects, together with competition from road transport, become a barrier to continued regional rail transport operations. To address this issue, an innovative regional rail vehicle concept is proposed as part of the regional railway revitalisation strategy brought forward in the Europe's Rail Joint Undertaking Flagship Project 6 - FutuRe [3].

Regional railway lines serve areas of relatively low density , which means that the ideal vehicle size is relatively small compared to current-generation main line rail vehicles. The small size requirement, along with the simplicity- and cost-focused principles, suggests that a two-axle vehicle is a good platform for this project.

Two-axle rail vehicles are a long-known concept, dating back to early rail transport. It was preferred for its simplicity, low weight, and cost-effectiveness. All of these attributes align well with the objective of the FutuRe project. However, two-axle rail vehicles are known to suffer from poor curving performance and ride comfort due to the running gear setup. Different solutions have been proposed to address the dynamics issue of two-axle rail vehicles, one of which is the use of active suspension.

Active suspension for two-axle rail vehicles has been a topic of discussion in several different studies. Mei and Goodall described several wheelset control strategies for a two-axle vehicle [4], some of which have become the foundation for more advanced wheelset control studies and are still very relevant today. Qazizadeh et al. applied an $H_\infty$ control method to improve the curving performance of a two-axle vehicle, ensuring robustness of the control system [5]. Heckmann et al. introduced the Next Generation Train concept, a family of train concepts with actively controlled independently rotating wheels running gear [6]. Giossi et al. presented active wheelset steering solutions for a two-axle vehicle with u-shaped running gear frame [7], which also highlights the effect of the physical configuration. Although all of these studies two-axle rail vehicles, their exact running gear configurations differ, which in turn influence the active suspension control itself. This study aims to explore this aspect by investigating active wheelset steering in conjunction with a novel running gear configuration.

## 2 Vehicle concept

This study concerns a regional rail vehicle concept as proposed by Persson et al. in [8]. Some properties of the vehicle concept are listed in Table 1. The vehicle is studied as part of Europe's Rail Joint Undertaking project FutuRe, where the goal is to increase the viability of regional railways by decreasing the total cost of ownership. Thus, the design focus is to present a simple and inexpensive solution.

| Property | Value |
|---|---|
| Number of passenger | 100 |
| Vehicle length | 16 m |
| Maximum axle load | 16 t |
| Maximum speed | 120 km/h |

Table 1: Key properties of the regional rail vehicle concept

The concept involves a two-axle rail vehicle comprising a carbody and two sets of single-wheelset running gear with a unique configuration. The principle of the proposed running gear concept is to have a very simple and lightweight running gear while maintaining a safety and comfort level similar to current generation bogie-equipped rail vehicles. The design consideration is extended to also achieve good stability, curving performance, and wear characteristics, which have historically been a drawback for two-axle rail vehicles.

The starting point of this running gear design is to include only the essential components and select the simplest possible versions of each. This yields a setup that consists of a solid wheelset, a single motor-gearbox traction system, a tread brake system, and suspension elements for a single-step suspension system.

The main component of the suspension system is chosen to be a set of coil springs. The coil spring should have a fairly high spring rate to limit displacements due to load and vehicle dynamics. This leads to a suspension system that would ensure safe running but has deficiencies in ride comfort and curving performance.

To achieve satisfactory ride comfort and curving performance, the coil springs are paired with a set of active suspension elements. The active suspension elements include vertical actuators and wheelset steering actuators. This setup shares some similarities with the running gear concept studied by Giossi et al. [7]. However, the configuration has been further simplified by omitting the running gear frame and the lateral actuators found in the concept in the referenced study.

Another area of simplification is the active wheelset steering mechanism. Contrary to the usual practice, the proposition in this project is to use only one actuator for each wheelset. The actuator is set up in series with the bushings on one side of the wheelset, while the other side retains the full passive longitudinal guidance, as shown in Figure 1. This simplified setup results in an asymmetric running gear configuration which is simple and lightweight but causes some peculiarities in the vehicle behaviour. The implications of this design are discussed later in this study.

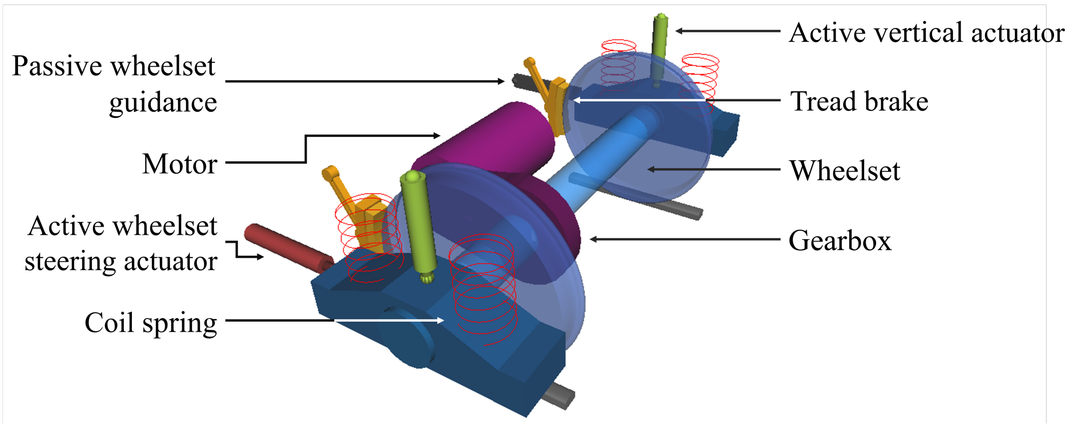



Figure 1: Running gear configuration with active suspension systems

# 3 Active wheelset steering mechanism

A free wheelset naturally assumes a perfect rolling position when negotiating a curve due to its conicity. However, suspended wheelsets may not always achieve this position, as the suspension limits the displacement of the wheelset relative to the carbody. This is especially true for vehicles with a long wheelbase, such as a two-axle vehicle. To alleviate this limitation, an active wheelset steering could be used.

The basic intention of active wheelset steering is to move the wheelset to the desired position during curving. In this study, active wheelset steering is carried out using one actuator per wheelset. The actuator is connected between the carbody and one of the axleboxes in longitudinal direction. Meanwhile, the other axlebox is connected to the carbody via passive wheelset guidance comprising a rod and bushings. This passive side of the running gear is designed to be the pivot point of the wheelset during curving; the entire wheelset positioning effort will be borne by the actuator on the other side [9].

Despite the novel asymmetric arrangement of the steering actuators, its working mode remains the same as a typical steering actuator, i.e., it is connected longitudinally to the axlebox to control the wheelset yaw movement. This conventional working mode opens up the possibility to apply many different steering control strategies, such as perfect steering control [10], yaw relaxation or compensation [11], and control of the relative angle between wheelsets [12]. However, the focus of this study is to evaluate the viability of the novel running gear configuration. Therefore, a simple control strategy based on radial steering has been chosen.

## 3.1 Active wheelset steering control strategy

The control target of the radial steering strategy is to have the wheelsets positioned radially according to the track curvature, as shown in Figure 2a. This wheelset position is achieved by controlling the actuator length.

The yaw angle required for the wheelset to achieve a radial position, $\psi$, can be calculated based on the vehicle wheelbase $L$ and the curvature $1/R$ as

$$\psi = arcsin\left(\frac{L}{2} \cdot \frac{1}{R}\right). \tag{1}$$

Because $L$ is normally very small compared to $R$, a small angle approximation can be used. With this approximation, Eq. 1 can be simplified to

$$\psi = \frac{L}{2R}. \tag{2}$$

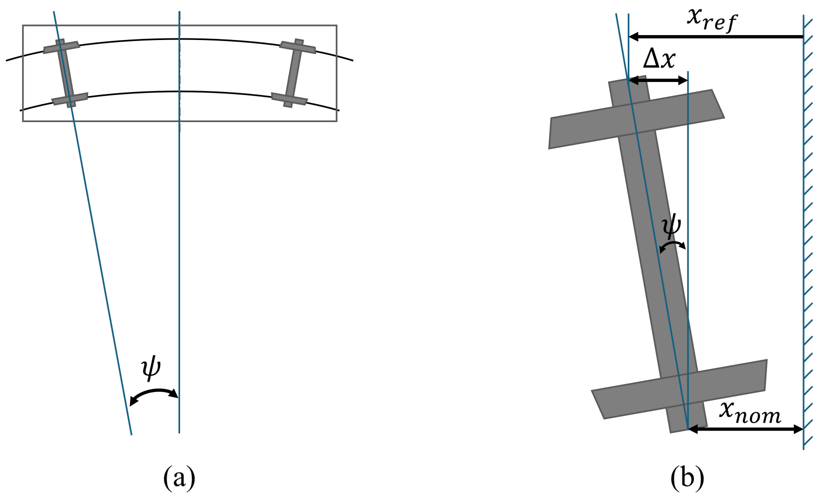



Figure 2: Radial control strategy (a) wheelset position and (b) reference length for the actuator

This wheelset yaw angle can subsequently be used to determine the reference length of the actuator. Since the wheelset pivots around its passive end, the reference length for the actuator can be calculated using the longitudinal position difference between the left and the right axleboxes at that yaw angle. Figure 2b shows a schematic of the calculation of the reference length for cases where the actuator is located on the side of the vehicle further away from the centre of the curve. This results in a reference length of

$$x_{ref} = x_{nom} + \Delta x, \tag{3}$$

with $\Delta x$ having a positive value. For cases where the actuator is located on the side of the vehicle closer to the centre of the curve, the sign should be flipped.

The value of the longitudinal position difference of the axleboxes $\Delta x$, as described in Figure 2b, can be calculated as

$$\Delta x = 2b_L \cdot tan(\psi), \tag{4}$$

where $2b_L$ is the lateral distance between the left and right suspension elements and $\psi$ is the wheelset yaw angle as described in Eq. 2. Substituting Eq. 2 into Eq. 4 and using small angle approximation for the wheelset yaw angle, $tan(\psi) \approx \psi$, $\Delta x$ becomes

$$\Delta x = b_L \frac{L}{R}. \tag{5}$$

The same $\Delta x$ value is used as the base value for the reference length of both actuators, with a difference only in the sign. By substituting Eq. 5 into Eq. 3, the reference length for the actuators can be calculated as

$$x_{ref} = x_{nom} \pm b_L \frac{L}{R}. \tag{6}$$

However, a modifying factor was added later; this will be discussed in the following subsection.

## 3.2 Active wheelset steering control system

The active wheelset steering control is built around a feedback control for the actuators with a PID-type controller. Figure 3 shows the schematic of the control loop. The control loop involves taking a measurement of the actuator lengths from the vehicle model and using them as a feedback variable. Meanwhile, the reference length values are calculated based on the vehicle wheelbase, the lateral distance between the actuator and the passive wheelset guidance, and the track curvature.

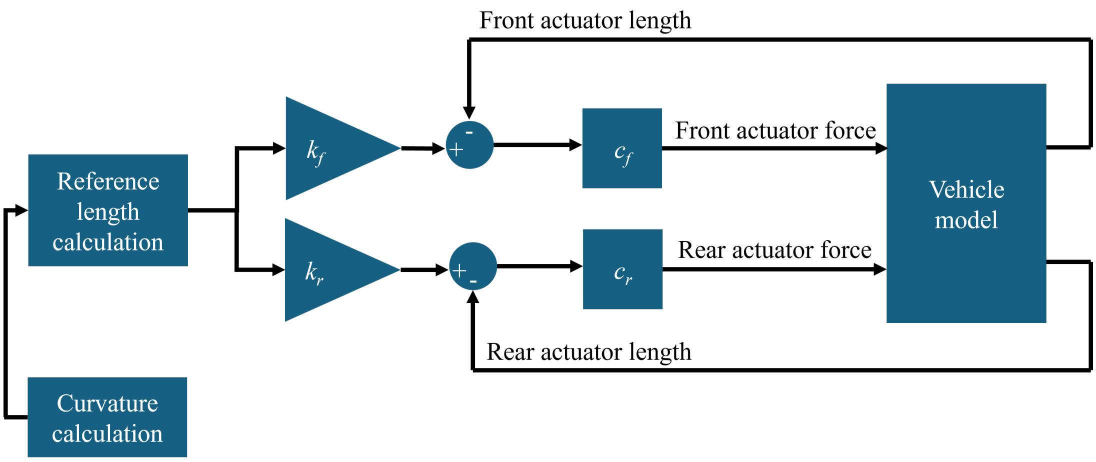



Figure 3: Schematic of the active wheelset steering control loop

Since the wheelbase and suspension lateral distance are constant, the only free variable in the reference length calculation is the track curvature. The track curvature could be estimated in real-time based on the carbody yaw rate and vehicle speed, which has been shown to be a feasible method in [7]. All three variables involved in the calculation are easy to measure with widely-available sensors. The result is a simple and inexpensive system, in line with the overarching principles of the project.

Electrodynamic actuators are expected to be used as steering actuators in this running gear. This type of actuator is not normally used for steering due to the relatively low force capability, but the expected force required for this active wheelset steering implementation is low, which leads to the possibility of using electrodynamic actuators. In addition, electrodynamic actuators have a high acceleration capability, which could support dynamic wheelset yaw control if necessary and avoid the need for a separate actuator for enhancing lateral comfort.

Electrodynamic actuators are based on the electrodynamic principle of a Lorentz force acting upon a conductor conducting a current. The Lorentz force, defined as

$$\mathbf{F}_{\mathbf{Lorentz}} = \mathbf{i} \cdot l \times \mathbf{B}, \tag{7}$$

depends on the current $\mathbf{i}$, the magnetic induction $\mathbf{B}$, and the length of the conductor $l$ [ref]. In this study, however, actuator modelling is less critical because the primary focus is on quasi-static performance. Therefore, the actuators in the control system are considered ideal and are not modelled in detail.

# 4 Simulation and Results

The vehicle model and proposed active wheelset steering control were evaluated in a series of co-simulations. These were performed using SIMPACK and Simulink coupled together.

A model of the vehicle was created in the multibody simulation software SIMPACK. The vehicle is modelled as a multibody system with seven bodies: A carbody, two wheelsets, and two pairs of axleboxes. These bodies are interconnected using force elements that represent the suspension components of the vehicle as described in Section 2.

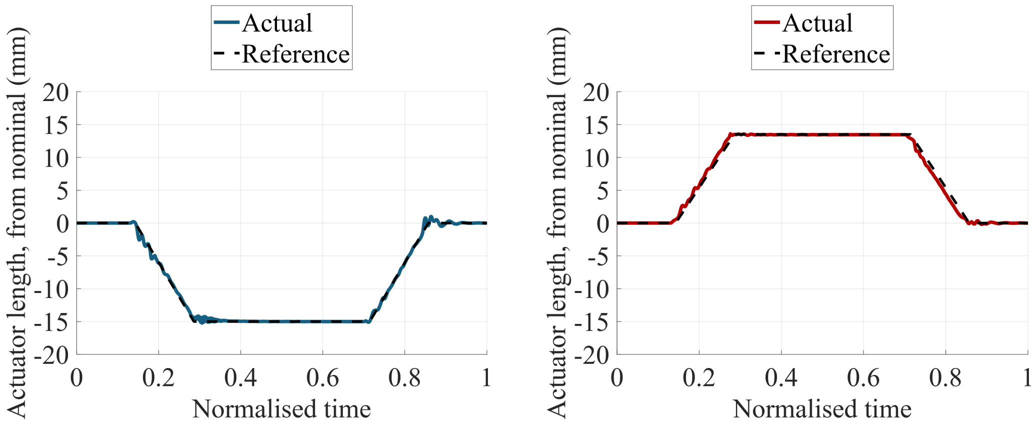
In this running gear, most of the longitudinal stiffness affecting the curving behaviour comes from the longitudinal links between the carbody and the axleboxes. The passive vehicle is equipped with the same passive wheelset guidance on both sides of the running gear, leaving the wheelset pivot point around its centre as normal. Meanwhile, in the vehicle with active suspension, one of these passive wheelset guidances is replaced by an actuator. This results in asymmetric longitudinal stiffness, which causes the wheelset pivot point to shift away from the middle.

Simulation results show that the wheelset pivot point moved depending on the actuator force. This movement compromises the effectiveness of the active wheelset steering control. To solve this issue, the passive bushings in the vehicle with active suspension are adjusted to have higher stiffness values. This limits the displacement of the passive-side axlebox, thereby fixing the wheelset pivot point around there. Table 2 contains the longitudinal stiffness parameters of the vehicle model.

| Parameter | Stiffness (N/m) |
|---|---|
| Coil spring longitudinal stiffness | $0.5 \times 10^6$ |
| Passive wheelset guidance bushing stiffness (passive vehicle) | $3.5 \times 10^6$ |
| Passive wheelset guidance bushing stiffness (active vehicle) | $25 \times 10^6$ |

Table 2: Stiffness parameters of the two-axle regional rail vehicle model

The actuators in the vehicle are modelled as a force element which takes external control input. During co-simulation, the control inputs for the vehicle model are determined by a control loop running in Simulink. The control loop was created according to the design in Subsection 3.2. The choice of controller types and parameters was a result of an iteration process during controller tuning, where the goal was to uphold simplicity and minimise the actuator forces while maintaining adequate reference tracking performance. A PI controller was used to control the front actuator, $c_f$, to eliminate steady state error. Meanwhile, a P controller was used for the rear actuator, $c_r$, as the simple controller was found to have adequate performance in controlling the rear wheelset. The tuning process for the controllers was performed at a reference running case of a 600 m radius curve with 100 mm track cant and zero non-compensated lateral acceleration (NLA).




(a) (b)

Figure 4: Controller performance at 600 m curve with 0 NLA. (a) Front actuator length, (b) Rear actuator length

The controller performance can be evaluated by the resulting actuator lengths and forces of the tuned active wheelset steering control. Figure 4 shows the actuator lengths recorded during a simulation of passing through a 600 m curve. Both actuators managed to closely track the reference length throughout the curve with only minor oscillations concentrated around the transition curve endings. This result demonstrates the effectiveness of the chosen controllers.

As shown in Figure 4b, the reference length for the rear actuator is slightly lower compared to the front actuator. In this case, it is 13.5 mm for the rear, compared to 15 mm for the front. This is a result of an adjustment of the control target for the rear actuator. It was based on the observation that when the front wheelset achieves radial position, the rear wheelset managed to position itself well and attain a low wear number even without any control input. This means that controlling the rear actuator using the same reference as the front one would unnecessarily counteract the wheelset's natural curving ability. Additionally, exploratory testing on the rear controller parameters showed that having a large control force acting on the rear wheelset was observed to induce vehicle instability. Therefore, multipliers $k_f$ and $k_r$ were introduced to individually set the control targets for the front and the rear actuator, respectively. In this study, the control target for the rear actuator was set to $90\%$ of the front actuator's target.

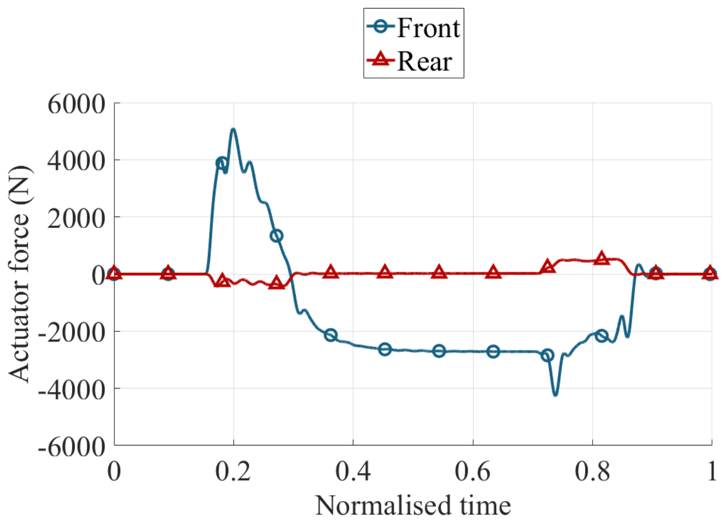



Figure 5: Actuator force at 600 m curve with 0 NLA

With the lower target value, the rear actuator was active only in the transition curves. During steady state curving, the actuator force dropped to just above zero (see Figure 5). This force pattern is expected, given the physical configuration of the running gear where the actuator is connected in series with the bushings. This serial connection relieves the longitudinal stiffness of the suspension system and allows wheelset conicity to move the wheelset toward the desired position during curving, as is the case with a free wheelset. This effect is also known as the yaw relaxation principle [10]. The same near-zero steady state force pattern is not observed in the front actuator data, likely due to the asymmetric configuration of the running gear. Although the actuator is connected in series with the bushings on one side of the running gear, the actuator still has to work against the passive wheelset guidance on the other side of the running gear and the coil springs. Nevertheless, the force value remains sufficiently low, peaking around 5 kN in the transition curve and levelling around 2.8 kN at steady state. This result supports the proposition that electrodynamic actuators are suitable for this application.

Despite only requiring little force, the active wheelset steering greatly improved the curving performance of the two-axle regional rail vehicle model. Figure 6 shows a comparison of wheelset displacements between the passive vehicle and the vehicle with active suspension. In contrast to the passive vehicle, the vehicle equipped with the asymmetric running gear with active suspension system managed to travel through the curve without flange contact on any of its wheelsets. The vehicle is still in an underradial position, with the front wheelset shifting outward and the rear wheelset inward. However, shifting the wheelsets away from flange contact is enough to reduce the wheel-rail wear and is in line with the aim of presenting a simple solution for the vehicle concept.

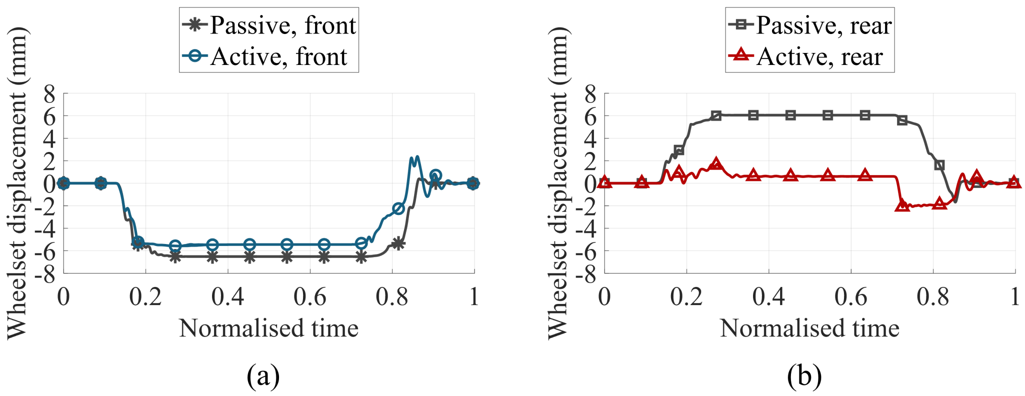



Figure 6: Wheelset lateral displacement at a 600 m curve with 0 NLA. (a) Front wheelset, (b) Rear wheelset

The wear numbers of wheelset 1 and 2 are plotted in Figure 7. Active wheelset steering managed to reduce the wear numbers by more than 80%. The wear number of the front wheelset dropped from around 420 to 80, while the wear number of the rear wheelset dropped from around 300 to 30 for the steady state curving part. Even lower wear numbers could be achieved, as demonstrated in other studies, but keeping the actuator force low is crucial in enabling the use of electrodynamic actuators. Furthermore, running with a very low wear increases the risk of rolling contact fatigue (RCF). Actuator force optimisation and wear-RCF balancing are thus better goals for future enhancement of this active wheelset steering scheme.

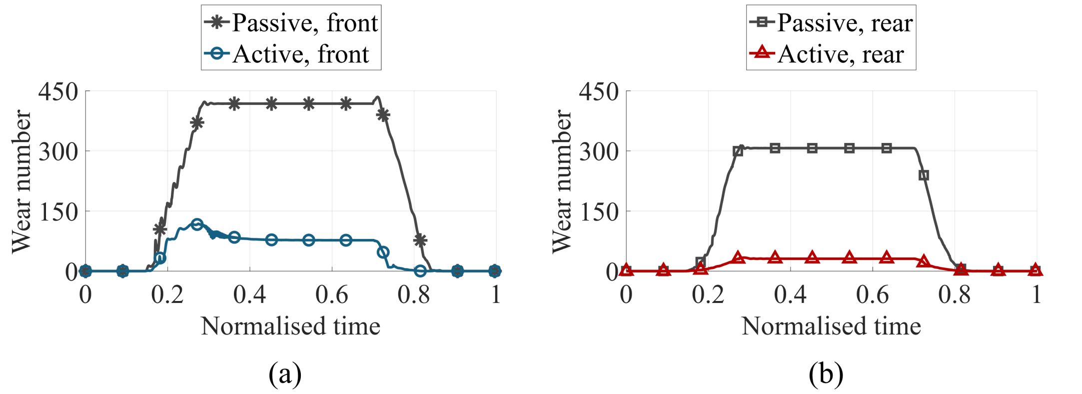



Figure 7: Wear number at a 600 m curve with 0 NLA. (a) Front wheelset, (b) Rear wheelset

# 5 Conclusions

A two-axle regional rail vehicle equipped with a novel asymmetric running gear configuration and active wheelset steering was modelled, simulated, and evaluated. Both the physical running gear configuration and the active wheelset steering system were designed with simplicity in mind. The active wheelset steering control strategy is based on the radial wheelset position modified to lower the rear wheelset control target. The results show that the control works well together with the asymmetric running gear configuration, resulting in a significant improvement in curving performance of the vehicle, with the vehicle model achieving no flange contact and a low wear number.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