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To cite this article: Axel von Stillfried & Raphael Pfaff (30 Sep 2026): Simulation-based sensitivity analysis of external influences on a locomotive-mounted weigh-in-motion system for freight wagons, Vehicle System Dynamics, DOI: [10.1080/00423114.2026.2739343](https://doi.org/10.1080/00423114.2026.2739343)

To link to this article: <https://doi.org/10.1080/00423114.2026.2739343>



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Published online: 30 Sep 2026.



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Simulation-based sensitivity analysis of external influences on a locomotive-mounted weigh-in-motion system for freight wagons

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ABSTRACT

To simplify the weighing of individual freight wagons on sidings, a measurement system is being designed that will determine the weight by measuring the traction force and acceleration of the shunting locomotive. To determine the influence of various resistances on the measurement result, a sensitivity analysis was carried out using a multi-body simulation model featuring an *Eanos* freight wagon and a road-rail vehicle. The qualitative influence of various parameters is shown individually. The results of the sensitivity analysis show that the condition of the freight wagon and driving behaviour have less of an impact, whereas the influence of the tracks and the wind must be taken into account when determining the weight. Based on these results, a measurement system can be set up and tested in real-world conditions in further studies.

ARTICLE HISTORY

Received 23 April 2026

Revised 23 June 2026

Accepted 27 August 2026

KEYWORDS

Weighing; freight wagon;
multi body simulation;
sensitivity analysis

1. Introduction

In addition to combined transport, industrial companies have the option of operating a private siding and using it to handle the receipt and dispatch of goods directly by rail. However, due to high operating costs, complex regulations and operational procedures, 80% of the sidings in Germany were decommissioned between 1994 and 2024 [1,2]. Knowledge of the weight of incoming and outgoing freight wagons at sidings is necessary both to ensure safe railway operations and to monitor the flow of goods and the associated cost accounting. Various weighting methods are currently mainly in use:

- Track scales are a piece of the siding infrastructure. Using load cells or strain gauges, the weight of wagons or axles is determined whilst they are in standstill or moving slowly [3,4]. Installation and maintenance represent a significant investment, particularly for small sidings.
- On-board systems are permanently installed on freight wagons to determine, among other things, the current load weight [5]. As the siding operator is not usually the wagon owner, they have only limited control over the equipment fitted to the wagons.

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- When weighing during or before loading (e.g. general cargo), the vehicle weight is calculated based on the vehicle's own weight and the payload. This method is also infrastructure-dependent and depends on the type of load.

The current situation calls for the development of a weighing system for freight wagons on sidings that is independent of infrastructure and wagon owners. Based on the connection $F = m \cdot a$ the mass of a wagon can be determined when measuring the longitudinal force and acceleration by installing the relevant sensors on a shunting locomotive. Various resistances acting on the wagon can influence the measurements, which is why they need to be examined more closely. This paper investigates with a multibody simulation the influence of the wagon resistances acting on freight wagons while determining its weight in order to determine the requirements for a measurement system and evaluation algorithm to be developed.

2. Modelling

The model implemented in this thesis consists of a road-rail vehicle with a freight wagon attached. The modelling of the vehicles and the track is described in the following. The multibody simulation software SIMPACK is used as the simulation environment.

2.1. Freight wagon

The model of the freight wagon is based on a four-axle open wagon of the *Eanos* type, in accordance with UIC Leaflet 571-2 for European standardised freight wagons [6]. The wagon has a length of $l_{\text{Eanos}} = 14.5$ m excluding buffers and a distance between bogie pivots of $d_{\text{bp}} = 10.7$ m. The freight wagon is designed for wheel set loads of up to $m_{\text{load,ws,max}} = 22.5$ t, the average tare weight is $m_{\text{tare}} = 23.5$ t [7]. In the model, the underframe and the load is modelled as two rigid bodies connected in the centre of the wagon by a stiffener. This connection reflects the wagon's torsional behaviour. It is modelled as a simplified cuboid, with the payload represented by varying its mass. Figure 1 shows the *Eanos* freight wagon model implemented in SIMPACK.

The bogies of the carriage are of the Y25 type with a wheelbase of $d_{\text{ws}} = 1.80$ m. In the simulation procedure, the wagon is unbraked, which is why the braking system has not been modelled. This model simulates both the behaviour of the suspension in the form of flexicoil springs and the damping behaviour of the Lenoir damping characteristic of Y25 bogies. The mass of the bogie, including wheel sets, is typically around $m_{\text{Y25}} = 4500$ kg [8]. The wheel sets are based on the BA004 design and weigh approximately $m_{\text{ws}} = 998$ kg when new [9]. The moments of inertia in rolling direction is set to $I_{yy} = 89.9$ kgm² [10]. The wheel radius can vary from $r_{\text{wheel,new}} = 460$ mm in the new condition to $r_{\text{wheel,old}} = 420$ mm in the worn condition. Mass and moment of inertia are adjusted in the model to the respective state of wear. It is assumed that all wheels have the same diameter. The wheel profile is a standard wear profile UIC S1002 according to DIN EN 13715 [11].

2.1.1. Flat spots

In operation flat spots occur relatively frequently in freight transport. This is usually caused by wheels locking up due to excessive braking or seised brakes, which is why both wheels

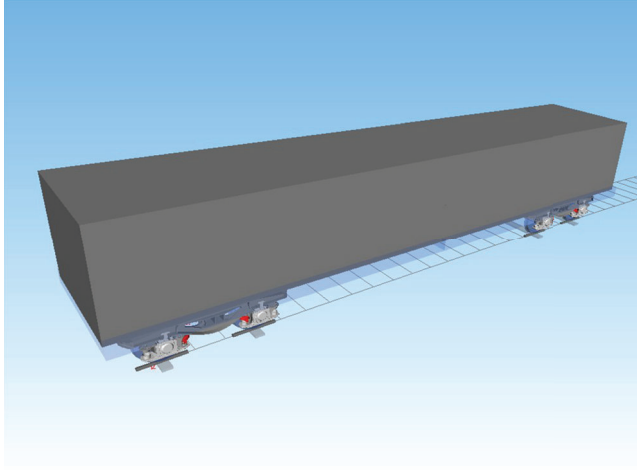


Figure 1. Simulation model of *Eanos* wagon.

of the wheelset are usually affected equally [12]. DIN EN 15313, which defines rules for the maintenance of wheel sets, specifies the length of an existing flat spot for an *Eanos* wagon as $l_{\max} = 80$ mm [13]. One method for modelling a flat spot is presented by MÜ in [14]. The deviation of the radius within the flat spot is given by

$$\Delta r = \frac{L_f^2}{8r_{\text{wheel}}} \left[1 - \cos \left(\frac{2\pi x}{L_f} \right) \right] \quad (1)$$

where Δr is the deviation from the nominal radius at the position $x = [0, L_f]$ and L_f is the length of the flat spot. For the purposes of the simulation, it is assumed that a flat spot occurs at most on one wheelset.

2.1.2. Wheel resistance

In the SIMPACK model, the wheel-rail contact is modelled using the *FASTSIM* model according to Kalker [15]. At low speeds of $v < 0.1 \frac{\text{m}}{\text{s}}$ and at standstill, the contact model switches to a classical spring-damper model, as the slip required for the calculations can no longer be defined. In general rail transport, a summary coefficient, which represents the bearing and rolling resistance, $f_R = [0.0007, 0.001]$ should be assumed for the wheel resistance [16]. In shunting operations, where individual freight wagons are moved or rolled down a hump, values ranging from $f_{R,\min} = 0.0005$ to $f_{R,\max} = 0.0062$ may occur [17]. Within the model, an additional force is applied to each wheel-rail contact, reflecting the variation of the wheel resistance coefficient. The wheel resistance coefficient is multiplied by the current respective wheel contact force, assuming that the same coefficient applies to all wheels:

$$F_{r,\text{wheel}} = f_r \cdot F_{z,\text{wheel}} \quad (2)$$

2.1.3. Starting resistance

Starting resistance is a resistance that occurs in addition to the wheel resistance during the initial movement of a wagon. It results from the uneven distribution of oil in the wheel-set

bearing [16]. When considering the running resistances of freight trains, it plays a minor role, as the high number of couplings and the resulting elongation of the train mean that the starting resistance acts on only a few carriages at any one time [18]. However, it can have a significant influence on short trains with fewer inter-carriage connections. COLE defines an additional factor T_s for velocities $v < v_s$ [19]:

$$T_s = \begin{cases} \frac{(M-1)(v-v_s)^2}{v_s^2} & \text{if } v < v_s \\ 1 & \text{if } v \geq v_s \end{cases} \quad (3)$$

The parameters M and transition speed v_s are given as $M = 5$ and $v_s = 3.6 \frac{\text{km}}{\text{h}}$. In this work, the factor T_s is applied as an additional coefficient to the wheel resistance.

2.1.4. Aerodynamic resistance

At low speeds, such as those encountered when shunting within sidings, the aerodynamic drag resulting from the vehicle's own motion is generally negligible. However, winds of sufficient strength can exert a considerable force on the vehicle, which must be taken into account. The resulting force F_{ae} can be determined using the drag coefficient c_d , the air density at standard pressure of $\rho_{air} = 1.225 \frac{\text{kg}}{\text{m}^3}$, the area A exposed to the wind, and the sum of the vehicle and wind speeds $\vec{v}$ [16]:

$$F_{ae} = \frac{1}{2} \cdot \rho_{air} \cdot c_d \cdot A \cdot v^2 \quad (4)$$

VOLLMER describes a detailed model for estimating the aerodynamic drag of freight wagons [20]. Based on tests with vehicle models, coefficients were determined which allow wagon-specific drag coefficients to be calculated. Based on the coefficients of an *Eaos* wagon with similar geometry, the longitudinal drag coefficient c_d can be determined as a function of load to be $c_{d,x,\text{unloaded}} = 1.42$ and $c_{d,x,\text{loaded}} = 0.95$. CHEN gives a drag coefficient of $c_w = 2.05$ for a long rectangular prism subjected to a side flow [21]. Based on this, the coefficients are estimated as $c_{d,y,\text{unloaded}} = 3.06$ and $c_{d,y,\text{loaded}} = 2.05$. In the model the wind vector is decomposed into its x and y components.

A model for describing wind gusts is provided by the 1-cosine-gust model in accordance with §25.341 of the Federal Aviation Regulations [22]. For the purposes of this study, the analysis is transformed into the time domain. The frequency and intensity of the wind gusts follow an intermittent frequency distribution, which can only be modelled using complex weather models. In this work, the strength of the gust is estimated using the factor $i = [0, 0.5]$ and the current wind speed v_w , whereby the gust can increase to 0.5 times the wind speed, and the duration $t_B = [2, 5]$ s. This leads to the definition:

$$v_B(t) = \begin{cases} iv_w \left(1 - \cos \frac{2\pi t}{t_B} \right) & \text{if } 0 < t < t_B \\ 0 & \text{if } t < 0 \text{ or } t > t_B \end{cases} \quad (5)$$

The prevailing wind direction is determined at random and remains constant.¹ The direction of the gusts may deviate by up to $\Delta\alpha = \frac{\pi}{6}$ from the prevailing wind direction.

2.1.5. Wagon connection

As wagon connection a UIC standard coupling in accordance with UIC Leaflet 520 is implemented [23]. Traction forces are transmitted via the draw hook, whilst pushing forces are absorbed by the buffers mounted on the sides [16]. The draw gear and buffer gear are supported on the carriage body by spring and damper elements. There is usually slack between buff and draft forces. DIN EN 15566 defines minimum and recommended parameters for the spring assemblies of the traction system [24]. Damping due to hysteresis is defined during loading and unloading. When the load changes from loading to unloading, there is a discontinuity in the spring characteristic curve, which poses a challenge in numerical programmes. WITT and JOBSTFINKE use a model which, at low speeds, interpolates between the upper and lower characteristic curves depending on the speed [25,26]. This model is also included in SIMPACK, which determines the spring force as following:

$$F = \begin{cases} F_{\text{spring}} \pm f_{\text{hys}} & \text{if } |v| \geq \epsilon_v \\ F_{\text{spring}} + \frac{v}{\epsilon_v} f_{\text{hys}} & \text{if } |v| < \epsilon_v \end{cases} \quad (6)$$

F_{spring} is the mean force between the characteristic curves, f_{hys} denotes the corresponding hysteresis value, and ϵ_v denotes the limiting speed at which the interpolation is performed. Figure 2(a) shows the spring characteristics of the implemented spring.

A distinctive feature of this work is that the compressive forces between the road-rail vehicle and the freight wagon are not transmitted via the side buffers but are also transmitted into the wagon via the draw hook (see Section 2.2.2). In the model, this behaviour is implemented according to the circuit diagram shown in Figure 2(b). Under draft force, the left branch behaves as described; under buff force, the right branch with a hard stop is active.

The coupling shaft of the UIC screw coupling is prevented from lateral deflection by a draw hook guide. The maximum lateral deflection is defined in accordance with DIN EN

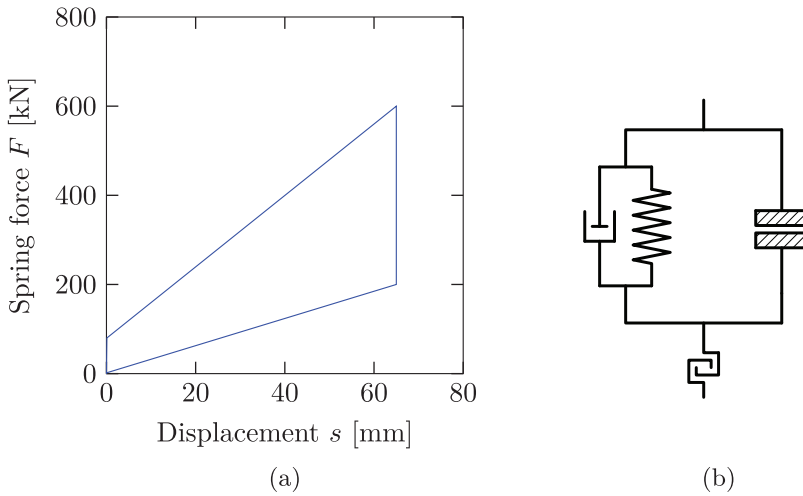


Figure 2. (a) Spring characteristics of the draw hook according to [25] and (b) Diagram of the force transmission implemented in the model.

16839 by the minimum clearance $b_{BR} \geq 400$ mm between the buffer and the draw hook for shunting personnel, the so-called *Berne space* [27]. This results in a lateral deflection of the modelled draw hook of $\alpha_{dh} = 12^\circ$. In the model, the rotational movement of the draw hook is counteracted by a dry friction damper, which damps the lateral movements of the coupling shaft.

2.2. Road-rail vehicle

The model of the rail-road vehicle is based on the battery-powered shunting robot *Rotrac E2* manufactured by G. Zwiehoff GmbH (see Figure 3). It has a weight of $m_{rrv} = 3800$ kg, is designed for trailing loads of up to $m_{trail,max} = 250$ t and can deliver a maximum power of $P_{rrv} = 22$ kW [28]. The motors are also used for braking.

2.2.1. Running gear

The traction force is transmitted to the rail via the rubber wheels. Each rubber wheel is individually driven and can rotate independently of the others. As the rubber wheels have a cylindrical profile and therefore cannot ensure track guidance, the vehicle is additionally fitted with non-driven steel rollers. According to DIN EN 15746, these must be subjected to a minimum contact force to prevent derailment [29]. Whilst the vehicle regulates the contact force via a hydraulic circuit, in the model a constant force of $F_{z,sw} = 2$ kN is applied to each steel roller. The profile is also cylindrical on the running surface and features a flange. The vehicle has no suspension, vertical impacts are damped by the elasticity of the rubber and the vertical guidance of the steel rollers.

2.2.2. Wagon connection

The road-rail vehicle is equipped with an automatic shunting coupler. The draw hook of the wagon to be moved is clamped between two locking bars, which can be released via a remote control to uncouple the wagon [5]. Shunting couplings do not have an energy-absorbing device in the thrust direction and are therefore designed so that the pushing

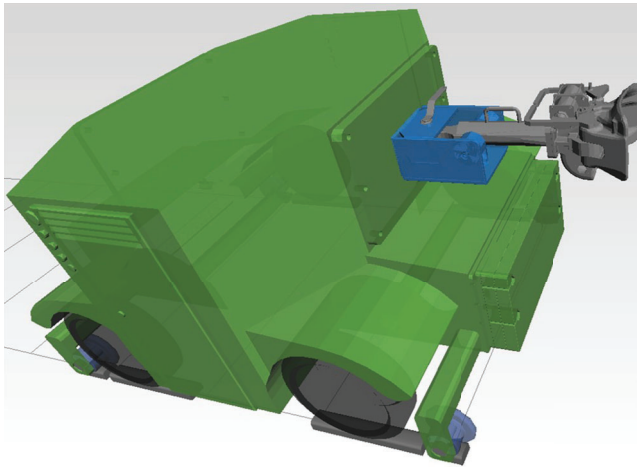


Figure 3. Simulation model of Rotrac E2.

forces continue to be transmitted via the buffers [16]. The road-rail vehicle used does not have side buffers, so the pushing forces are transmitted via the draw hook. A vertical pin connects the coupling shaft to the vehicle and allows rotation about the z -axis and, to a limited extent, about the y -axis. The movements are restricted by two lateral rubber buffers and one located below the coupling bar.

2.3. Track

The track on which the simulation is carried out is implemented using the corresponding *Track* module in SIMPACK. By arranging various track elements, lines with any geometry can be replicated. In this work, track elements are dimensioned in accordance with the regulation on the construction and operation of sidings (BOA) in North Rhine-Westphalia [30]. This sets the minimum track curve radius at $r_{\text{track,min}} = 140$ m and the vertical radius at gradient changes at $r_{z,\text{min}} = 300$ m. A maximum longitudinal gradient is not defined; however, tables for brake calculations are provided up to a gradient of $p_{\text{max}} = 25\text{‰}$. In addition to the geometry, track irregularities occur. It can be described by the quantities $h(x)$ for vertical deviation, $u(x)$ for lateral deviation, $\delta(x)$ for the cant angle and $s(x)$ for the deviation in track gauge [16]. The method according to FREDERICH for modelling track irregularities, presented in [31], is widely used. Using the function

$$\Theta = \frac{a}{\left(b + \frac{\Omega}{2\pi}\right)^3} \quad (7)$$

a power spectral density is defined, which can be used to generate track alignment errors that vary along the track based on the parameters a and b . FREDERICH defines the parameters for the freight classes *worst*, *average* and *best* track alignment; these are listed in Table 1. In SIMPACK, the track alignment errors generated previously can be applied to the track within the *Track* module. Since the values of a and b are not linear with respect to track condition, the function

$$a_h = a_{h,\text{best}} \cdot e^{\left[\ln\left(\frac{a_{h,\text{worst}}}{a_{h,\text{best}}}\right) \cdot \xi\right]} \quad (8)$$

with the track quality index $\xi = [0, 1]$ is introduced, where 0 is the best and 1 the worst condition. All other parameters can then be mapped from a_h .

Table 1. Unevenness and waviness for different track conditions.

Irregularity	Condition	Unevenness a [m ²]	Waviness b [1/m]
vertical alignment h	worst	$9.39 \cdot 10^{-7}$	$6.89 \cdot 10^{-2}$
	average	$1.31 \cdot 10^{-8}$	$2.94 \cdot 10^{-2}$
	best	$1.90 \cdot 10^{-10}$	$9.71 \cdot 10^{-3}$
lateral alignment u	worst	$2.74 \cdot 10^{-7}$	$3.13 \cdot 10^{-2}$
	average	$1.33 \cdot 10^{-8}$	$2.33 \cdot 10^{-2}$
	best	$6.33 \cdot 10^{-10}$	$1.30 \cdot 10^{-8}$
track gauge s	worst	$1.64 \cdot 10^{-7}$	$5.35 \cdot 10^{-2}$
	average	$4.68 \cdot 10^{-8}$	$7.96 \cdot 10^{-2}$
	best	$1.23 \cdot 10^{-8}$	$1.12 \cdot 10^{-1}$
cross level angle δ	worst	$4.07 \cdot 10^{-16}$	$5.57 \cdot 10^{-3}$
	average	$2.87 \cdot 10^{-15}$	$5.57 \cdot 10^{-3}$
	best	$4.07 \cdot 10^{-16}$	$5.57 \cdot 10^{-3}$

3. Test procedure

For the sensitivity analysis, the procedure shown in Figure 4 was carried out. The simulations were executed by using a SIMPACK-SIMULINK co-simulation, in which SIMULINK controls the road-rail vehicle. In a simulation run, the train is accelerated until it reaches a speed of $v_{\text{target}} = 1 \frac{\text{m}}{\text{s}}$. This is followed by a coasting phase of $t_{\text{roll}} = 1 \text{ s}$, after which the train is braked to a standstill. Figure 5 shows an example of the speed profile of the rail-road vehicle. Once the coupling slack has been passed through (point 1–2), the road-rail vehicle accelerates constantly. During the coasting phase, the motion becomes uneven due to the release of the draft gear and the resulting difference in velocity between the freight wagon and the rail-road vehicle (point 3–5). After passing the coupling slack once again (point 5–6), the road-rail vehicle brakes to standstill constantly (point 6–7). By breaking down the resistance of the freight wagon into an acceleration-dependent and an acceleration-independent term, the linear function

$$F_{\text{coup}} = (m_{\text{fw}} + m_{\text{rot}}) \cdot a_{\text{rrv}} + F_{\text{rest}} \quad (9)$$

can be derived. In this case, the acceleration of the road-rail vehicle a_{rrv} is the variable of the function. The sum of the mass and the rotational mass ($m_{\text{fw}} + m_{\text{rot}}$) corresponds to the slope of the line, whilst F_{rest} corresponds to the y-axis offset. Within the simulation, the coupling force F_{coup} and a_{rrv} are measured from this equation, resulting in a data pair for each time step. The resulting linear system can be solved by performing a linear regression. For the linear regression, only data points from the acceleration and braking phases are used. Figure 6 shows all the data points recorded during one simulation execution and the data points from the acceleration and braking phases used for the linear regression, as shown in Equation (9). The result of the subsequent linear regression is shown as a straight line. To minimise the error in the sensitivity analysis, the measured mass is subtracted by an average value of the rotational mass for Eanos freight wagons, $\bar{m}_{\text{rot}} = 1554.6 \text{ kg}$. To ensure that the results of all measurements are comparable, the simulated relative measurement error

$$\chi = \frac{m_{\text{meas}} - m_{\text{real}}}{m_{\text{real}}} \quad (10)$$

is defined as the difference between the measured mass m_{meas} and the actual mass m_{real} relative to the actual mass. The sensitivity indices can then be calculated on the basis of the relative measurement error. Table 2 summarises all the parameters that have been varied.

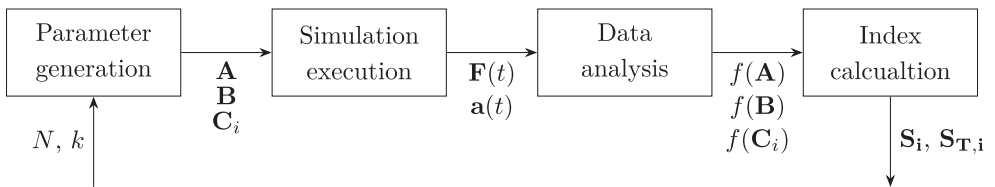


Figure 4. Procedure of the sensitivity analysis.

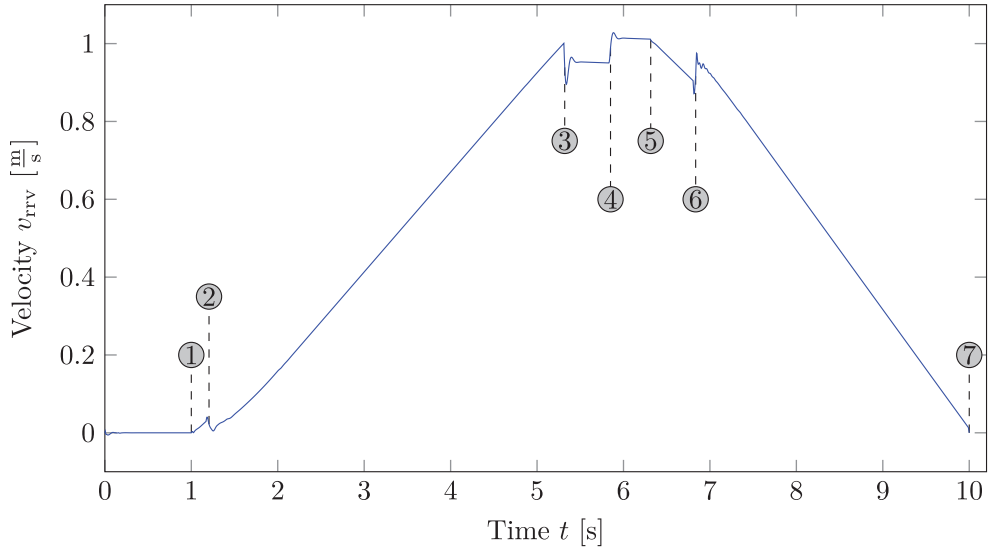


Figure 5. Velocity of the road-rail vehicle in a simulation run.

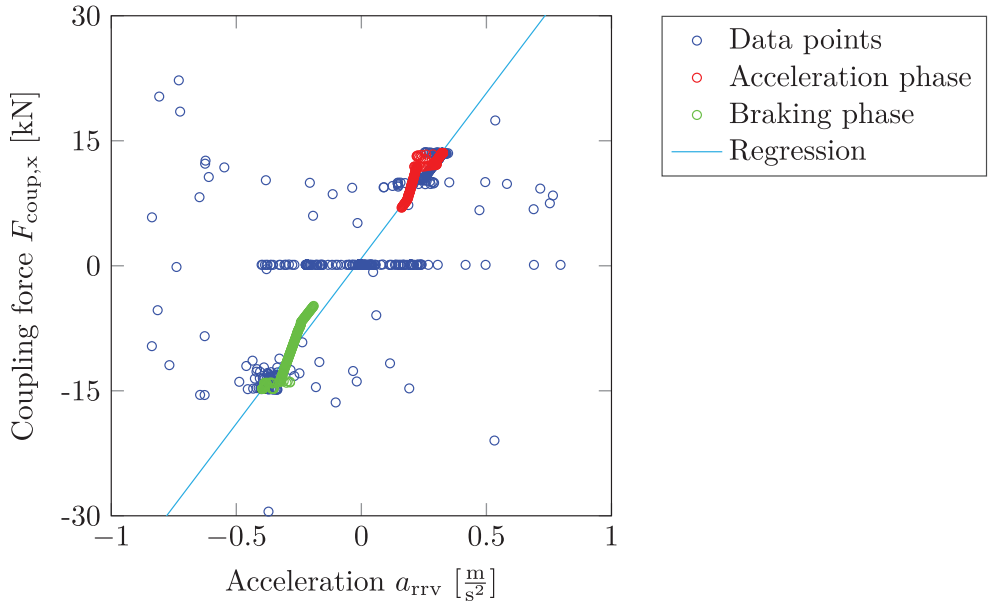


Figure 6. Linear regression with used data points of a simulation run.

3.1. Sensitivity analysis

Since, in the case of multidimensional, non-linear models, it is not possible to make straightforward statements about the influence of a parameter, sensitivity analyses are used to examine the variances of parameters in relation to the variances of the overall system [32]. A widely used method for variance-based sensitivity analysis is presented by SOBOL

Table 2. Variation parameters.

	Parameter	Minimum	Maximum
Freight Wagon	Mass load	0 t	66.5 t
	Wheel resistance coefficient	0.0005	0.0062
	Wheel radius	0.42 m	0.46 m
	Length flat spot	0 m	0.08 m
Track	Track gradient	−12.5‰	12.5‰
	Track curve radius	100 m	∞
	Track quality	Best	Worst
Rrv.	Maximum power	11 kW	22 kW
	Maximum torque	3.1 kNm	6.2 kNm
Wind	Wind speed	0 $\frac{\text{m}}{\text{s}}$	30 $\frac{\text{m}}{\text{s}}$
	Wind direction	0	π

in [33]: For each input parameter, a first-order sensitivity index

$$S_i = \frac{D_i}{D} \quad (11)$$

is defined. D is the variance of all output values of the model, D_i is the variance of the output values based on a fixed parameter x_i . In addition to that, the total effect index

$$S_{Ti} = S_i + S_{ci} \quad (12)$$

described by HOMMA in [34], describes the extent to which a parameter influences the result value indirectly via other parameters. S_{ci} is the sum of all sensitivity indices with higher order, in which i is included. In [35], building on [33,34], SALTELLI presents a practical algorithm for calculating sensitivity indices. Based on two independently generated $N \cdot k$ matrices **A** and **B**, where N is the sample size and k is the number of parameters, a matrix

$$\mathbf{C}_i = (b_1 \cdots a_i \cdots b_k) \quad (13)$$

is generated for each parameter. The model can then be run for each set of parameters consisting of **A**, **B** and $\mathbf{C}_i$. This yields the number of simulation runs required:

$$n_{\text{sim}} = N \cdot (k + 2) \quad (14)$$

In this work, a sample size of $N = 1024$ is used for all simulations. The sensitivity indices can then be calculated as follows:

$$S_i \approx \frac{\left(\frac{1}{N} \sum_{j=1}^N f(a_j) f(c_j) - \left(\frac{1}{N} \sum_{j=1}^N f(a_j) \right)^2 \right)}{\frac{1}{N} \sum_{j=1}^N f^2(a_j) - \left(\frac{1}{N} \sum_{j=1}^N f(a_j) \right)^2} \quad (15)$$

$$S_{Ti} \approx 1 - \frac{\left(\frac{1}{N} \sum_{j=1}^N f(b_j) f(c_j) - \left(\frac{1}{N} \sum_{j=1}^N f(a_j) \right)^2 \right)}{\frac{1}{N} \sum_{j=1}^N f^2(a_j) - \left(\frac{1}{N} \sum_{j=1}^N f(a_j) \right)^2} \quad (16)$$

To generate the parameter sets, quasi-random numbers are generated using the Monte Carlo method based on sequences according to SOBOL, as described in [36,37]. In [38],

ARCHER introduces the technique known as bootstrapping to determine the variance of the results. From the existing data set consisting of $f(\mathbf{A})$, $f(\mathbf{B})$ and $f(\mathbf{C}_i)$, $B = [1, N]$ new samples are drawn with replacement. For each set of parameters, the sensitivity indices are then calculated, which are now subject to variation. Subsequently, an interval covering 95% of the values is established, the width of which provides an indication of the robustness of the result. The implementation and presentation of results are based on the *Safe* toolbox in MATLAB [39].

4. Results

In order to examine the influence of individual parameters in isolation, simulations were first carried out using only some of the parameters. This was followed by a sensitivity analysis covering the entire parameter space.

4.1. Individual parameters

Initially, only the parameters of the freight wagon and the load are varied. In practice it represents a measurement without external influences, e.g. on a flat straight track in a factory hall. The varying parameters reflect the condition of the freight wagon, which can generally hardly be influenced during operation. In Figure 7, the simulated relative measurement error χ is plotted against the wheel resistance coefficient and the wheel radius. The colour of the data points represents the total mass of the freight wagon m_{fw} . Whilst the wheel resistance coefficient f_R has no discernible influence on χ , the wheel radius r_{wheel} clearly has. This influence arises from the relationship between the moment of inertia – and thus the rotational mass – and the wheel radius. However, the dominant influence lies in the mass of the goods wagon. On the one hand, this can be explained by the definition of χ (see Equation (10)); on the other hand, as the weight increases, the mass inertia is dominant against the vehicle resistances. Similar to the wheel resistance coefficient, the occurrence and length of a flat spot has no significant influence on the simulation error.

In further simulation runs, the effects of the track topology was also investigated. Figure 8 shows the effect of the track gradient and of the curve radius to the simulated

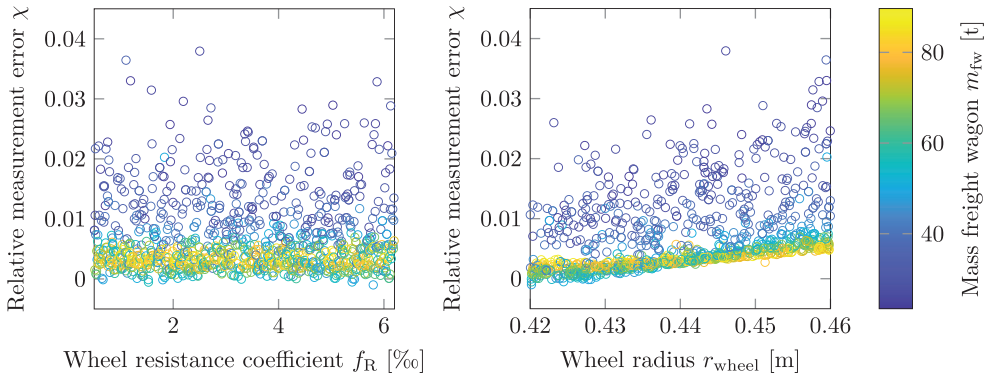


Figure 7. Results of only varying the freight wagon.

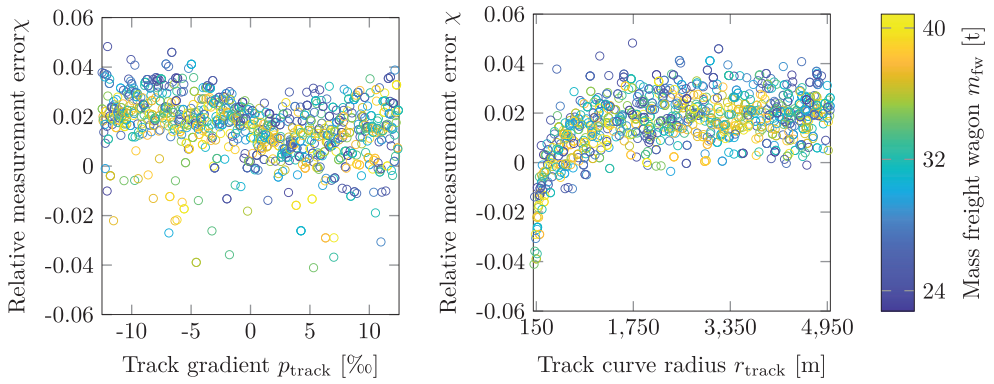


Figure 8. Results of varying the track topography.

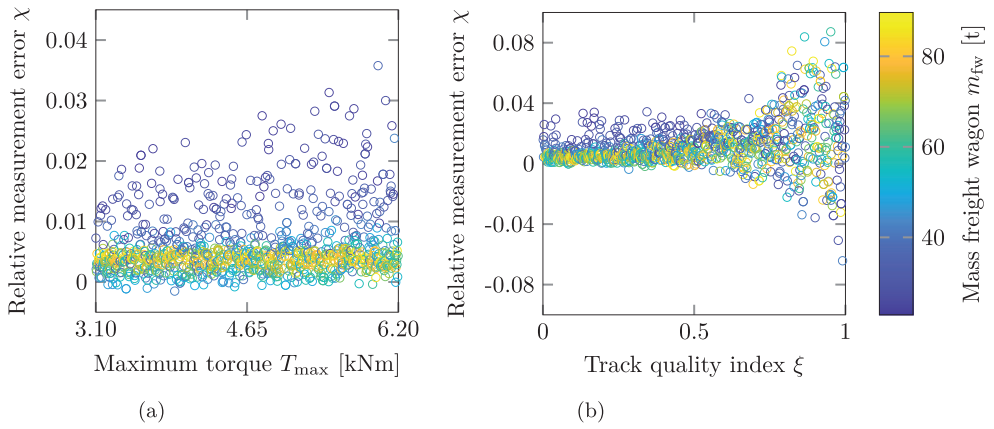


Figure 9. Results of varying (a) the behaviour of the road-rail vehicle and (b) the track quality.

relative measurement error. Due to the limit on the maximum trailing load of the road-rail vehicle on gradients, the weight of the freight wagon is limited to $m_{\text{fw,max}} = 40.8 \text{ t}$ in this test. It can be seen that the track gradient has little influence on the simulation error, whereas small radii in particular influence the simulation error. This is caused by the coupling being deflected laterally due to the lateral forces acting on it, meaning that a significant part of the forces is not transmitted in the x-direction of the coupling. The effect of the track gradient is, as long as it is constant, a x- and y-shift of the regression line (see Figure 6), but no change in its gradient.

Figure 9(a) was generated by simulation executions in which, in addition to the freight wagon, the running behaviour of the road-rail vehicle was also varied. No significant influence of the maximum drive torque could be detected, similar to the influence of varying the maximum drive power. In contrast, the simulated relative measurement error is strongly influenced by the track quality, as shown in Figure 9(b). Simulation errors increase significantly, particularly when track alignment is very poor. This is due, among other things, to track irregularities with short wavelengths and high amplitudes, which generate additional resistive forces that vary over distance and, consequently, temporally.

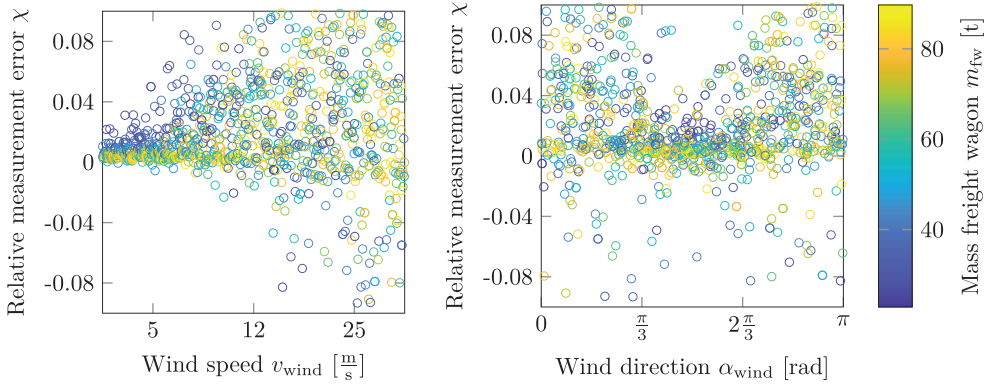


Figure 10. Results of varying the wind.

Eventually the external influence of wind speed and direction was investigated. The wind behaves as described in Section 2.1.4, meaning it does not blow constantly from a single direction but varies in speed and direction over time due to gusts. Figure 10 shows the influence of wind speed v_{wind} on the relative measurement error χ . From around $v_{\text{wind}} = 5 \frac{\text{m}}{\text{s}}$, the simulated relative measurement error is significantly affected and deteriorates as the wind speed increases. Due to gusts in strong winds, in some cases there are no longer any steady acceleration or braking phases, both of which affect the measurement error. Figure 10 also shows that the wind direction has a significant influence on the result. The inaccuracy is particularly apparent in the case of a tailwind ($\alpha_{\text{Wind}} = 0$) and a headwind ($\alpha_{\text{Wind}} = \pi$), whereas with a crosswind ($\alpha_{\text{wind}} = \frac{\pi}{2}$) its influence on the simulation error is much smaller.

4.2. Sensitivity analysis

Figure 11 shows the results of the simulation using all available parameters. The boxes represent the 95% interval of the sensitivity indices after bootstrapping, the mid line represents the mean value. The parameters were grouped as shown in Table 2. The values of the individual parameters were generated individually and the simulation runs were then combined within the groups, as described in Section 3.1. The first-order sensitivity indices are highest for the track with $S_{\text{track}} = 0.186$ and wind with $S_{\text{wind}} = 0.149$, whilst they are lower for the payload ($S_{m_{\text{load}}} = 0.050$) and the road-rail vehicle ($S_{\text{rrv}} = 0.186$), and are with $S_{f_w} = 0.007$ close to zero for the freight wagon. The sum of the indices $\sum_i S_i = 0.578 < 1$ shows that the system exhibits many non-additive features. The total effect indices reflect the same qualitative picture, but with higher values that also take non-additive features into account. Here too, the distance has the greatest influence on the variance of the relative measurement error, with $S_{T,\text{track}} = 0.855$. Next comes the influence of the wind ($S_{T,\text{wind}} = 0.718$) and the payload ($S_{m_{\text{load}}} = 0.557$). The relative to the first order index high value of the total effect index of the road-rail vehicle $S_{T,\text{rrv}} = 0.549$ is mainly caused by the fact that in adverse conditions (strong winds, poor track conditions) the vehicle is no longer able to accelerate and decelerate the freight wagon adequately, when the maximum power and torque is low. The freight wagon has with $S_{T,f_w} = 0.182$ once more the least influence of all parameter groups.

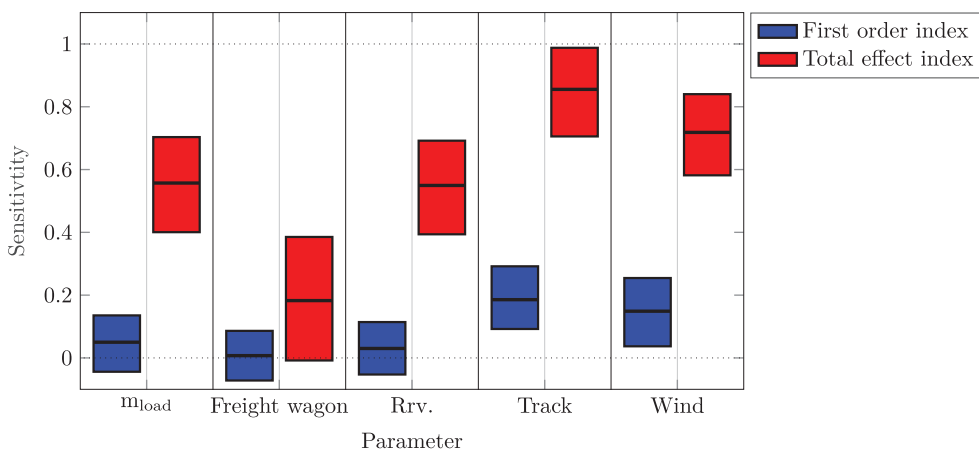


Figure 11. Results of the sensitivity analysis.

The global sensitivity analysis shows, that the main influence on the simulated relative measurement error is caused by the external parameters track and wind. It should be noted, that both are varied up to extreme conditions. Since the track layout can be measured and doesn't change quickly the influence can be subtracted in future algorithms. The biggest issue with wind is that it is time-dependent and cannot be predicted with certainty. A wind gauge may be necessary to reliably eliminate the interference. While the behaviour of the road-rail vehicle can be measured easily and the interference of the freight wagon is relative small, the interference of the payload itself has to be taken into account. It is helpful in this regard that the measurement error can be clearly attributed to a specific measured value. A correction coefficient dependent, for example, on the measurement result can reduce the measurement error.

5. Conclusion and outlook

For the development of a locomotive-mounted weigh-in-motion system various resistances acting on a freight wagon during shunting operations were modelled, and their effect on coupling-based weight determination was simulated. Each simulation run consists of an acceleration to $v_{\text{target}} = 1 \frac{\text{m}}{\text{s}}$, a rolling phase of $t_{\text{roll}} = 1 \text{ s}$ and subsequent braking to standstill. Under normal conditions, the proposed analysis method, which uses linear regression of the measurement data, achieves in the simulation an accuracy of 4%. It has become apparent that, of the external forces at play, it is mainly the wind and the track condition that can influence the measurements. Under poor conditions, the accuracy of the analysis method drops to 10%. The influence of condition of the freight wagon is relatively small. Next steps of the development process are the design and building a prototype of the system and measuring real-world data in order to validate the simulation.

Note

1. The wind direction relative to the vehicle changes during a turn.

Author contributions

CRedit: **Axel von Stillfried**: Conceptualization, Data curation, Formal analysis, Investigation, Methodology, Writing – original draft; **Raphael Pfaff**: Resources, Supervision, Writing – review & editing

Disclosure statement

No potential conflict of interest was reported by the author(s).

Funding

This research received no external funding.

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References

- [1] Deutscher Bundestag, Ed., Bundestagsdrucksache 19/9305, Berlin; 2019.
- [2] Bundesministerium für Verkehr, Ed., Verkehr in Zahlen 2025/2026, Flensburg; 2025.
- [3] Kochsiek M Ed. Handbuch des Wagens. Braunschweig: Vieweg; 1989.
- [4] Ehmke F. Vor- & Nachteile dynamischer Gleiswagen: Ein umfassender Guide, Essmann Waagen-Magazin. [accessed 2025 Oct 2]. [Online]. Available at <https://essmann.com/dynamische-gleiswagen-bauformen/>.
- [5] Janicki J, Reinhard H, Rüffer M. Schienenfahrzeugtechnik (DB-Fachbuch). 4th ed. Berlin: Bahn Fachverlag; 2020.
- [6] UIC Leaflet 571-2: standard wagons – Ordinary bogie wagons – Characteristics; 2001 Feb.
- [7] Eanos-x 056, DB Cargo Güterwagenkatalog. [accessed 2025 Oct 13] [Online]. Available at <https://gueterwagenkatalog.dbcargo.com/katalog/nach-gattung/e-offene-wagen/Eanos-x-056-5831764>.
- [8] Baur KG. Drehgestelle – bogies. Freiburg: EK-Verlag; 2006.
- [9] Neubau Radsatz BA004/BA182 – 22,5 t, Railmate.de. [accessed 2025 Oct 14] [Online]. Available at <https://railmate.de/Neubau-Radsatz-BA004-BA182-225-t-TSI-konform>.
- [10] Hoffmann M. Dynamics of European two-axle freight wagons [Ph.D. dissertation] Technical University of Denmark; 2006.
- [11] DIN EN 13715 Railway applications – Wheelsets and bogies – Wheels – Tread profile; 2020 Oct. doi: [10.31030/3147484](https://doi.org/10.31030/3147484)
- [12] Bernal E, Spiryagin M, Cole C. Wheel flat detectability for y25 railway freight wagon using vehicle component acceleration signals. Veh Syst Dyn. 2020;58(12):1893–1913. doi: [10.1080/00423114.2019.1657155](https://doi.org/10.1080/00423114.2019.1657155)
- [13] DIN EN 15313 Railway applications – In-service wheelset operation requirements – In-service and off-vehicle wheelset maintenance; 2024 Dec. doi: [10.31030/3572013](https://doi.org/10.31030/3572013)
- [14] Mu J, Zeng J, Wang Q, et al. Determination of mapping relation between wheel flat and wheel/rail contact force for railway freight wagon using dynamic simulation. Proc Inst Mech Eng FJ Rail Rapid Transit. 2022 May 1;236(5):545–556. doi: [10.1177/09544097211030373](https://doi.org/10.1177/09544097211030373)
- [15] Kalker JJ. A fast algorithm for the simplified theory of rolling contact. Veh Syst Dyn. 1982 Feb 1;11(1):1–13. doi: [10.1080/00423118208968684](https://doi.org/10.1080/00423118208968684)
- [16] Schindler C Ed. Handbuch schienenfahrzeuge. Hamburg: Eurailpress; 2014.
- [17] Scheuch M. Zum Stand der Laufzielbremsung in Ablaufanlagen und ihr Beitrag zur Industrie 4.0. Signal+Draht. Vol. 03/2020. 2020 Mar. p. 17–22.

- [18] Iwnicki S, Spiryagin M, Cole C, et al. Handbook of railway vehicle dynamics. 2nd ed. Boca Raton: CRC Press; 2019 Nov 14. p. 913. doi: [10.1201/9780429469398](https://doi.org/10.1201/9780429469398)
- [19] Cole C, Spiryagin M, Bosomworth C. Examining longitudinal train dynamics in ore car tippers. *Veh Syst Dyn*. 2017 Apr 3;55(4):534–551. doi: [10.1080/00423114.2016.1263393](https://doi.org/10.1080/00423114.2016.1263393)
- [20] Vollmer G. Luftwiderstand von Güterwagen [Ph.D. dissertation] TU Darmstadt, Darmstadt; 1989.
- [21] Chen Y, Lin H. The application of aerodynamics in the exterior design of racing cars. *Highlights Sci Eng Technol*. 2023 Apr 14;43:237–252. doi: [10.54097/hset.v43i.7425](https://doi.org/10.54097/hset.v43i.7425)
- [22] 14 CFR 25.341 – Gust and turbulence loads [accessed 2025 Oct 16] [Online]. Available at <https://www.ecfr.gov/current/title-14/part-25/section-25.341>.
- [23] UIC Leaflet 520: Wagon, choaches and vans – Draw gear – Standardisation; 2003 Dec.
- [24] DIN EN 15566 Railway applications – Railway Rolling stock – Draw gear and screw coupling; 2022 Oct. doi: [10.31030/3299346](https://doi.org/10.31030/3299346)
- [25] Witt T. Integrierte Zugdynamiksimulation für den modernen Güterzug [Ph.D. dissertation] Universität Hannover, Hannover; 2005.
- [26] Jobstfinke D. Güterzuglängsdynamik [Ph.D. dissertation] TU Berlin; 2020.
- [27] DIN EN 16839 Railway applications – Rolling stock – Head stock layout; 2025 Dec. doi: [10.31030/3298386](https://doi.org/10.31030/3298386)
- [28] G. Zwiehoff GmbH, Ed., Brochure Rotrac E2; 2025.
- [29] DIN EN 15746-1 Railway applications – Track – Road rail machines and associated equipment – Part 1: Technical requirements for travelling and working; 2021 May. doi: [10.31030/3223040](https://doi.org/10.31030/3223040)
- [30] Verordnung über den Bau und Betrieb von Anschlußbahnen (BOA); 1966.
- [31] Frederich F. Die gleislage – aus fahrzeugtechnischer sicht. *ZEV Glasers Ann*. 1984;108(12): 355–361.
- [32] Siebertz K, Van Bebber D, Hochkirchen T. Statistische versuchsplanung. Berlin, Heidelberg: Springer; 2017. doi: [10.1007/978-3-662-55743-3](https://doi.org/10.1007/978-3-662-55743-3)
- [33] Sobol IM. On sensitivity estimation for nonlinear mathematical models. *Mat Model*. 1990;2(1):112–118.
- [34] Homma T, Saltelli A. Importance measures in global sensitivity analysis of nonlinear models. *Reliab Eng Syst Saf*. 1996;52(1):1–17. doi: [10.1016/0951-8320\(96\)00002-6](https://doi.org/10.1016/0951-8320(96)00002-6)
- [35] Saltelli A, Ratto M, Andres T, et al. Global sensitivity analysis. The primer. Chichester, UK: John Wiley & Sons, Ltd; 2008. doi: [10.1002/9780470725184.fmatter](https://doi.org/10.1002/9780470725184.fmatter).
- [36] Sobol IM. On the distribution of points in a cube and the approximate evaluation of integrals. *USSR Comput Math Math Phys*. 1967;7(4):86–112. doi: [10.1016/0041-5553\(67\)90144-9](https://doi.org/10.1016/0041-5553(67)90144-9)
- [37] Sobol IM. Uniformly distributed sequences with an additional uniform property. *USSR Comput Math Math Phys*. 1976;16(5):236–242. doi: [10.1016/0041-5553\(76\)90154-3](https://doi.org/10.1016/0041-5553(76)90154-3)
- [38] Archer GEB, Saltelli A, Sobol IM. Sensitivity measures, anova-like techniques and the use of bootstrap. *J Stat Comput Simul*. 1997 May 1;58(2):99–120. doi: [10.1080/00949659708811825](https://doi.org/10.1080/00949659708811825)
- [39] Pianosi F, Sarrazin F, Wagener T. A matlab toolbox for global sensitivity analysis. *Environ Model Softw*. 2015 Aug 1;70:80–85. doi: [10.1016/j.envsoft.2015.04.009](https://doi.org/10.1016/j.envsoft.2015.04.009)