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# Investigation of the dynamic parameters of the DAC

## System analysis with consideration of the EP brake

On behalf of Technical Innovation Circle for Rail Freight Transport (TIS)

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## List of abbreviations

| | |
|---|---|
| AAR | Association of American Railroads |
| AC | Automatic coupling |
| DAC | Digital automatic coupling |
| EP brake | Electro-pneumatic brake |
| ERRI | European Railway Research Institute |
| GSA | Global sensitivity analysis |
| CDFC | Cumulative difference of the force collectives |
| CT | Combined transport |
| MBS | Multibody simulation/multibody system |
| OAT | One-At-A-Time |
| OSShD | Organisation for Cooperation of Railways |
| RFT | Rail freight transport |
| UIC | International Union of Railways (Union internationale des chemins de fer) |

## 1. Summary

This study examines how the selection of the dynamic parameters for the digital automatic coupling (DAC) influences the tolerable longitudinal compressive forces of freight wagons, the longitudinal dynamics of freight trains, shunting impacts operations and the operational stability of the coupling. The aim is to determine the effects of each design, the underlying mechanisms of these effects, and the limiting conditions that require consideration. The study uses the following methods: multibody simulation and Global Sensitivity Analysis (GSA), the evaluation of journey data from operations and counting procedures for the determination of force collectives.

The results show a clear conflict of objectives between a design optimised for longitudinal dynamics and a design that is advantageous for tolerating shunting impacts in marshalling yards. In terms of longitudinal dynamics, the optimal solution would tend towards a rigid connection. This would include a high spring preload in both the tensile and compressive directions, ideally a degressive characteristic curve, short spring travel distances and the minimum degree of coupling slack. Buffers and draw gear optimised exclusively for powerful impacts between wagons in marshalling yards would provide soft suspension and a high degree of energy absorption. This would be achieved with a low spring preload, a highly progressive characteristic curve and long spring travel distances.

This conflict of objectives cannot be resolved conclusively within the existing framework. For this reason, the study investigates the influence of the coupling design on operational stability based on load assumptions and the resulting force collectives. The objectives are either the longest possible service life at a defined weight/cost of the coupling, or the lightest possible (and thus more cost-effective) coupling design, assuming a certain service life. The results show that different designs have advantages for different rail freight segments. In intermodal traffic, for example, a coupling that is more optimised for longitudinal dynamics proves advantageous, whereas a coupling that is optimised for shunting impacts has significant benefits in wagonload traffic. Thus, different classes of couplings should be available with dynamic parameters that are adapted to specific types of traffic or forms of production. The use of wagons fitted with couplings of different classes is not considered critical for short- and medium

length trains. Further investigation is required to assess the use of different couplings in long trains.

Compared to trains with conventional pneumatic brakes, the forces in the coupling caused by longitudinal dynamics are significantly reduced when the wagons are equipped with EP brakes. This reduction in longitudinal forces allows either the dynamic parameters to be optimised for shunting impacts, or the development of new train and operating concepts with longer, heavier trains and the same degree of running safety.

The dynamic parameters have a comparatively minor influence on the tolerable longitudinal compressive forces of freight wagons. The decisive factors here are the geometric parameters of the wagon and the coupling. A long coupling arm tends to have a positive effect on the tolerable longitudinal compressive forces. On average, the presence of a stabilisation joint permits higher tolerable longitudinal compressive forces. However, for four-axle vehicles with Y25 bogies, couplings without stabilisation joints can also provide adequate running safety. Therefore, no general recommendation for or against a stabilisation joint can be derived from the results.

## 2. Background to the study

The introduction of the digital automatic coupling (DAC) in Europe is the most important innovation project in rail freight transport for decades. In addition to automatic coupling and the faster operational processes that result from this, the DAC is seen as a key technology for the comprehensive digitisation of rail freight transport.

While there are open questions regarding operational and financial aspects, it is also necessary to determine the best possible design for the DAC. This concerns the design of the draft gear and its dynamic parameters, as well as the configuration of the DAC and the issue of whether a stabilisation joint is required. To ensure an optimal design, it is first necessary to determine the operational conditions under which DAC is used as precisely as possible, i.e. the longitudinal forces due to acceleration, braking and shunting impacts in marshalling yards. The challenge here is to take account of the wide variety of production forms and traffic that exist in rail freight transportation and lead to vastly different requirements regarding the operational strength of the coupling. Designing the coupling with the optimum characteristics to handle the stresses it will experience in operation offers two possibilities, depending on requirements: either the service life can be increased, or the costs and weight of the coupling can be reduced, generating efficiency gains for rail freight traffic.

The aim of this study is to show which parameters of the DAC affect the forces generated by longitudinal dynamics, shunting impacts and the tolerable longitudinal compressive forces to ensure running safety. It also seeks to determine how these parameters affect the forces. Based on the forces occurring, it is possible to make statements about the force collectives that provide pointers for creating an operationally stable DAC design. The aim is not to examine specific versions of the coupling, which is already available on the market, but to show, independently of this, what possibilities exist for optimising its design and how these possibilities affect the forces and force collectives.

The study is divided into four sections: the investigation of the tolerable longitudinal compressive forces, of the longitudinal forces due to freight train longitudinal dynamics and shunting impacts in marshalling yards and – based on this – the investigation of the design with regard to the operational stability of the coupling under different limiting conditions.

# 3. State of the art

This chapter first presents the state of the art with regard to the questions addressed by this study. The last fundamental studies into the dynamic parameters for an AC in rail freight transport were mostly carried out in the 1970s/80s by the UIC or the European Railway Research Institute (ERRI).

## 3.1 AC draft gear requirements according to UIC 524

Requirements for the AC draft gear based on actual operating conditions were last examined in detail in **UIC leaflet 524** [1]. It considered the longitudinal dynamics during train movement and shunting operations, where forces occur in the couplings due to shunting impacts in the sorting sidings. To this extent, it is similar to the present study. Unlike the present study, however, shunting impacts were not considered from the point of view of the coupling's operational stability, but only with regard to the protection of loads at "normal impact speeds" and wagons at "relatively high impact speeds".

The document defined various general requirements for the draft gear (selection):

- Simplest, lightest possible construction of the draft gear
- Maintenance-free operation for 6 to 10 years, if possible

This catalogue of requirements was always with reference to the UIC-AK or Intermat design principle, which implies the presence, e.g. of a stabilisation joint[1] and other fixed design parameters, such as a fixed coupling arm length. Draft gears designed in accordance with this specification should be compatible with freight wagons that meet the requirements of **UIC leaflets 530-1/2**.

**UIC 524** defines three groups of draft gears which differ in their design according to the types of wagons for which they are intended [1]. Group I is intended for two-axle wagons, Group II for four-axle wagons and Group III for six-axle wagons. The principal

[1] The stabilisation joint is also referred to in the literature as an "all-side stabilisation joint". It is an additional component, between the coupling arm and the draft gear, which deflects or attenuates transverse forces caused by an offset or angle between two adjacent wagons under longitudinal compressive forces. This increases running safety under longitudinal compressive forces [2, 3]. The function of the stabilisation joint is discussed in more detail in relation to modelling in Chapter 5.1.4.

design criterion is the maximum impact speed between two fully loaded wagons of the same design, at which no damage to the vehicles occurs. It also defines limit values for spring stiffness, preload force and minimum quasi-static energy absorption with regard to the longitudinal dynamics. All three groups share the same requirements for the maximum stroke length until the stop is reached, the minimum damping and the preload force. With regard to stability, the draft gears should be able to tolerate compressive forces of up to 2 000 kN and tensile forces of up to 1 000 kN without plastic deformation. There are large differences in the static, quasi-static and dynamic energy absorption during deflection as well as in spring stiffness. The information on minimum energy absorption is based on the following assumptions (selection):

- Distribution of two-axle and four-axle wagons: 50:50 %
- Distribution of wagon mileage empty/partially loaded/full: 33.33% respectively
- The load and wagon body absorb 25% of the impact energy, the rest is absorbed by the draft gear
- The force in the impacting coupling is equivalent at the same impact speed, regardless of how many wagons are involved in the impact
- Shunting processes every 100 km, on average
- Average annual mileage of wagons 25 000 km

The probability distribution shown in Table 1 is assumed for potential impact speeds. This distribution was determined by means of measurements and statistical surveys within the **ORE D 74** working group and reflects the conditions prevalent in shunting operations in the 1970s, e.g. with a high proportion of marshalling yards using stop blocks and many other conditions that no longer apply today.

**Table 1: Probability distribution of impact speeds according to UIC 524 [1]**

| Impact speed | Probability |
|---|---:|
| > 5 km/h | 85% |
| > 8 km/h | 51% |
| > 11 km/h | 17% |
| > 13 km/h | 9% |
| > 15 km/h | 3.5% |

The data sheet defines the test conditions under which the behaviour of the draft gear should be tested in operation. The tests should only be performed for emergency braking of a single typical freight train with a mass of 1 200 t and “unfavourable mass distribution” in brake position P, which can be regarded as a worst-case scenario in terms of the longitudinal compressive forces. However, evaluation of the results reveals that there are gaps in the specifications. For example, they do not specify the level of longitudinal force that should be used as the upper limit for braking, or the initial speed from which this braking should take place [1].

## 3.2 Freight train longitudinal dynamics

COLE describes the longitudinal dynamics of a train as the motion of the vehicles in the direction of the track. This includes not only the running dynamics, i.e. the motion of the train as a whole, but also the relative movements between the vehicles themselves [4]. In the case of trains with automatic central buffer couplings, driving and braking forces generate relative movements with forces in the elastic connections between the wagons via the draft gear of the couplings. Force-free relative movements result from design-related slack, e.g. in the draft gear or coupling head. In trains hauled by locomotives, the driving forces are typically applied by one or more locomotives at the front of the train, while the braking forces are decentralised and applied by the locomotive and the wagons, whereby the brake control mechanism plays a key role [5].

The longitudinal dynamics of freight trains play an important role in the design of the DAC for several reasons and are therefore a central area of investigation in this study [5]:

- Stress on the coupling due to vibrations in the train during operation with regard to the operational stability of the coupling
- Potential generation of high tensile forces that can lead to the coupling or other draw gear components tearing off
- Ensuring operational safety under the influence of longitudinal compressive forces

In recent decades, a great deal of research has been conducted into the longitudinal dynamics of freight trains, with increasing use of simulation technology. However, much of this work relates to rail freight traffic in the AAR area, which has only very limited comparability to freight transport in Europe in terms of the limiting conditions (e.g. very high degree of coupling slack and direct release brakes).

Particularly noteworthy are the studies conducted by UIC and ERRI in the 1980s and 1990s, which were performed with a view to increasing freight transport speeds, among other things. At the European level, the *TrainDy* simulation program, which was developed and extensively validated by the University of Tor Vergata in Rome together with the brake manufacturer Faiveley, is considered to be the state of the art in the simulation of the UIC air brake on freight trains and its effects on longitudinal dynamics [5, 6].

The existing approaches were analysed, combined and further developed by JOBSTFINKE in his dissertation [5]. The simulation models used in this study for the investigation of longitudinal dynamics are based on these investigations, further details in Chapter 0.

## 3.3 Tolerable longitudinal compressive forces for freight wagons with an automatic central buffer coupling

**UIC leaflet 530-1/2** deals with the requirements for the running safety of freight wagons and defines both the corresponding limit value for the tolerable longitudinal compressive forces and the test procedure to determine this limit value for a particular wagon type [7, 8]. For new freight wagons, it is recommended that the tolerable longitudinal compressive force is at least 600kN. This can be verified using the analytical calculation method described in the **ERRI report B125/RP5** or **RP6** [3]. As an alternative to the analytical calculation, Annex G presents a method for determining the tolerable longitudinal compressive forces by means of running tests. However, this method relates to vehicles with screw couplings and side buffers. A different procedure for freight wagons with automatic couplings is not specified separately. To determine the longitudinal compressive force that can be tolerated for a given type of freight wagon, Annex C provides various diagrams that define corresponding limit curves based on the geometric parameters of the wagon for bogie freight wagons. The procedures for determining these limit curves are described in **ERRI report B12/RP49** [9].

The ERRI report **B125/RP5** presents an analytical method for calculating the tolerable longitudinal compressive forces for freight wagons as a parameter for assessing running safety. This method was developed jointly by the UIC and OSJD. The calculation differentiates between two-axle and four-axle freight wagons, based on the effect of the stabilisation joint. The study is not based on measurement results, but on theoretical considerations of the geometry and equilibrium conditions of freight wagons running in a three-wagon train between two frame wagons. The critical value for the tolerable longitudinal compressive forces is determined by superimposing the most unfavourable conditions possible during operation. The influence of friction in force-rail contact is not considered. The input parameters for the calculation method are the geometric dimensions of the wagons, the masses and the torsional stiffness of the wagon body and bogie frame including attachments, and the characteristic curves of the primary springs. Most parameters for the coupling are stored as fixed parameters for the dimensions and the installation position of the UIC-AK [3].

The aim of the **ERRI report B12/RP49** is to investigate the influence of wagon parameters on the derailment safety of freight wagons on twisted tracks. For this purpose, two-axle and four-axle freight wagons of different types and geometric dimensions are considered with the aim of determining the maximum permissible torsional stiffness of the wagon body. The calculations were determined for a large number of parameter combinations in accordance with the method presented in **ERRI report B125/RP5** and the limit curves were derived from these [3]. The observations were based on a three-wagon train consisting of a wagon/wagon type being investigated and two so-called frame wagons, running in front of and behind this wagon. The following limiting conditions were defined as prerequisites [9]:

- All parameters for the automatic coupling correspond to UIC-AK or Intermat.
- Three types of frame wagon were permitted: Fc(s) wagons, Rs-680 wagons and wagons of the same type as the type of freight wagon under investigation
- For each wagon type, the 40 scenarios specified in B125/RP5 (2 track bends, 5 wagon combinations, 2 loading states of the frame wagons and 2 wheelset positions in the track for the wagon under investigation) are analytically calculated

and the tolerable longitudinal compressive force determined as the minimum of the longitudinal compressive force calculated in the respective scenarios to an accuracy of 50kN.

The report explicitly states that if the above essential parameters are changed, the limit curves defined in **UIC leaflet 530-1/2** become invalid [7, 8].

Report **B12/RP59** presents the procedure for determining the tolerable longitudinal compressive forces using multibody simulation. The procedure was used exclusively for freight wagons with screw couplings and side buffers. A coupled three-wagon train was modelled for the study, assuming a short two-axle Tdg 5930 freight wagon at the front and a long four-axle Rs 680 freight wagon as the frame wagons (see Fig. 1:). The frame wagons were simplified as being guided centrally in the track; the flange-to-rail clearance is not used. The admissibility of this simplification was tested by comparison with a reference model, in which the frame wagons were integrated as complete models and the deviations proved to be negligible [10].


**Fig. 1: Freight wagons of the types Tds (left) and Rs 680 (right) [11, 12]**

The tests were simulated in S-bends with radii of 150 m and 190 m respectively. To verify the procedure, the simulation results for the selected wagon types under investigation were compared with corresponding measurement results. The limit values defined in **UIC leaflet 530-2** were used as criteria for reaching the critical speed [8]. For the bogie freight wagons under investigation, there was good agreement between the simulation and tests. The lifting of a non-guiding wheel always proved to be the critical state for the tolerable longitudinal compressive force [10].

## 3.4 Shunting impacts between freight wagons with an automatic central buffer coupling

Investigations of forces and energy dissipation in the case of shunting impacts between wagons with automatic couplings during shunting were carried out within the framework of the **ERRI report B36/RP25** and **RP27** [13, 14]. Based on sample tests, the programs TULIP 1/2 were developed for the simulation of shunting impacts between wagons. For this purpose, the freight wagons and buffer units were converted into a simplified mathematical model that allowed the properties of various draft gears and load securing devices to be evaluated. In addition, it was able to predict the behaviour of the load during the impact. The results of the simulation include the forces and strokes in the draft gears as well as the accelerations of the wagon and load.

The simulation was based on a mechanical equivalent model of wagon, load and coupling, which consists of rigid bodies connected by force elements. The model of the draft gears can be adapted to the characteristics of different designs, such as friction cone, elastomer or hydro-gas spring units. Based on the measurement results, an “apparent wagon elasticity” was modelled, which takes into account the absorption of impact energy by the wagon design. The tests for this were carried out using the Es, Eas, Gbs and Rs wagon types. Energy absorption by the load and load securing equipment was also tested experimentally and incorporated into the model. Freight wagon designs and specific load types cannot be considered independently of each other, so not all wagon types were tested with all loads. The following cargos were considered: one or more concrete blocks (fastened or tied down), one or more steel blocks, and gravel as a bulk material. The testing programme was based on a total of 11 typical wagon design and load combinations.

The study produced the following key findings:

- the type of load and the load securing equipment have an influence on the level of impact forces
- under high impact forces, the wagon and load no longer behave like rigid bodies → deformation and energy dissipation occur in the wagon structure and possibly in the load
- the buffers absorb only approx. 75% of the theoretical impact energy (see UIC 524)

- the behaviour on impact is strongly dependent on the wagon type (e.g. chassis, wagon structure etc.)

## 3.5 Comparison with today's limiting conditions and requirements for the DAC

The UIC leaflets and the ERRI reports provide important basic data for this study. The various experimental results are particularly valuable. As the basic design of the wagons, e.g. the running gear, has not changed significantly since these investigations were conducted and no more recent studies of vehicles with automatic couplings (AC) are available, the test results provide valuable input data for the simulation calculations.

At the same time, a comparison of the conditions on which the reports are based reveals clear discrepancies with the situation in rail freight transport (RFT) today. Compared to the basic assumptions used for the design of the draft gears in accordance with **UIC 524**, the basic conditions have changed significantly, as shown in Table 2[1].

**Table 2: Comparison of assumptions according to UIC 524 and current conditions**

| | UIC 524 [1] | Current RFT |
|---|---|---|
| **Average annual mileage of freight wagons** | 25 000 km | Examples according to [15]:<br>Car transporter wagons 44 000 km<br>Tank wagons 50 000 km<br>Container wagons 150 000 km |
| **Share of two-axle/four-axle freight wagon fleet** | 50/50 | Data 2018 [16]<br>Germany 18/78<br>Austria 16/81<br>Switzerland 19/78 |
| **Wagon mileage empty/partially loaded/fully loaded** | 1/3 each | Very different depending on the form of production/mode of transport, e.g. intermodal transports are rarely fully and often partially loaded, bulk transports are often ½ fully loaded, ½ empty |

Within the framework of this study, data provided by clients on the impact speeds in marshalling yards with a modern hump yard control system were compared with the values according to **UIC 524** (see Fig. 2:). Both datasets refer to velocity ranges within certain intervals. In the absence of other information, it is assumed that the probabilities of specific speeds within an interval are distributed more or less equally. UIC 524 does not specify the highest anticipated shunting impact speed. In this study, the maximum speed is therefore assumed to be 18 km/h. The comparison shows significant differences: While fewer than 15% of the impacts occurring in a modern system take place at more than 5 km/h, precisely the opposite is the case with the distribution according to **UIC 524**, i.e. 85% of the shocks occur at impact speeds of more than 5 km/h [1]. This study assumes that some proportion of old marshalling yards in the European network are still in operation and that not all newer facilities are equipped with the most modern control systems. Thus, the distribution according to **UIC 524** must also be taken into account. These two distributions are regarded as the extreme poles of the velocity distributions occurring during shunting operations.

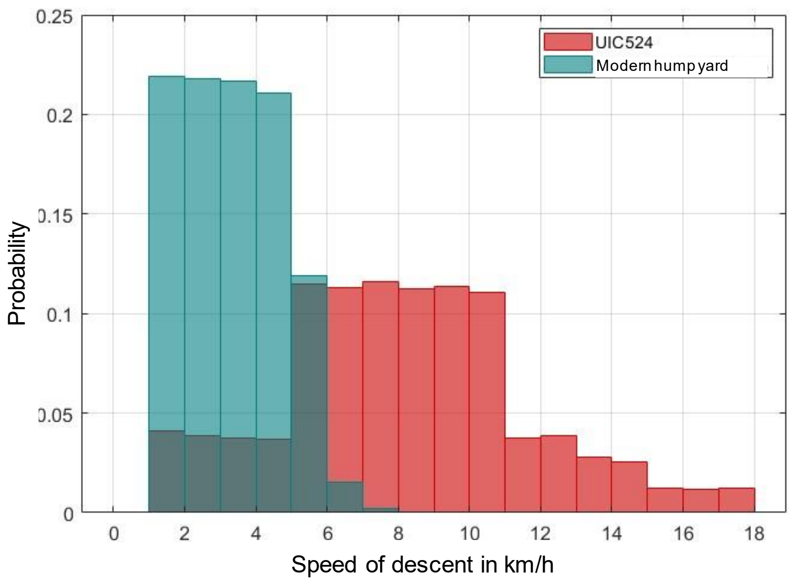



**Fig. 2: Comparison of impact speeds according to UIC leaflet 524 and for a modern hump yard, in-house presentation based on information from project partners and [1]**

When considering the impact forces themselves, the measurement campaign carried out within the framework of the **ERRI reports B 36 RP 25/27** provides a valuable fund of data for this study. However, two of the four types of freight wagons considered there are no longer in use today (Es and Gbs wagons). The loads considered by these reports no longer adequately reflect the reality of today's freight traffic, e.g. they do not consider palletised goods or intermodal loads at all. The methodology used to obtain the measurements makes it very difficult to separate the effects of wagon, load and spring apparatus effectively when creating simulation models. This is because two of the parameters were always varied simultaneously during the tests (e.g. Test 1 with wagon type A, load B and buffer C, Test 2 with wagon type X, load Y and buffer C etc.).
The studies presented in Chapter 3.3 on the tolerable longitudinal compressive forces of freight wagons with automatic couplings provide information on the methodology to be used in the tests, e.g. the choice of frame wagons or the critical track infrastructure. The studies also derive suitable criteria for derailment detection and derailment mechanisms. However, all investigations were conducted on an analytical basis and, for example, the influence of wheel-rail contact was not taken into account. The test procedure allowed numerous changes to the wagon parameters but practically none to the coupling parameters, i.e. the results always refer to the configuration of the UIC-AK/Intermat and cannot be easily transferred to other design principles or coupling dimensions.

## 3.6 Operational limiting conditions in RFT production forms

To determine the requirements for the coupling, it is necessary to consider the operational limiting conditions that exist in RFT production forms. The following section considers the three most important forms of production: block train transport, single wagon or wagonload traffic and combined transport (CT) or intermodal transport.
In block train transport, all the wagons in a freight train are transported together from the same dispatch station to the same destination station without passing through a train formation facility. These transports are used, e.g. for bulk goods, automobile transports or tank wagons. For smaller consignments, wagons are moved in single wagonload traffic. Here, in contrast to block train transport, individual wagons or groups of wagons from different dispatch stations are joined together to form a single train in

train formation facility. Later, at another train formation facility, this train is divided into wagons or groups of wagons which are then transported to their respective destinations. The transport thus passes through at least one train formation facility and a train typically consists of different wagons of different types. In intermodal transport, a consignment is transported using at least two different modes of transport in succession. In most cases, this means a combination of road and rail transport, but transport by water is also possible. Combined transports are mostly used for containers [17, 18].

There are major differences between the operational limiting conditions for these three forms of production, e.g. how often they pass through train formation facilities. The annual mileage of freight wagons also varies greatly. While container wagons in intermodal transport often cover 150,000 km or more per year, the corresponding figure for car transporter wagons (block train transport) or tank wagons (block train or single wagonload transport) is considerably lower, at 44,000 and 50,000 km per year respectively.

The aim of this study is to investigate the influence of the different conditions described above on the design of the DAC and whether segmentation in the production forms should also be reflected in segmentation for the coupling.

# 4. Research methodology

This chapter presents the methodology used in the study. To investigate the tolerable longitudinal compressive forces as well as the longitudinal tensile dynamics and shunting impacts (Chapter 0), multibody simulations are performed using the SIMPACK program, which is generally recognised in the field of rail vehicle dynamics.

Since the composition of freight trains can vary greatly in terms of length, load, wagon types, etc., and since there can be an almost infinite number of possible configurations, so-called “Monte Carlo” methods are used to create simulation scenarios. These statistical methods enable researchers to make reliable statements about the influence of individual parameters and the behaviour of the entire system within the framework of sensitivity analyses (Chapter 4.2), despite the large number of possible train configurations, the variety of influencing parameters and their variance.

Depending on the sub-investigation, different parameters – resulting from the technical and operational limiting conditions (Chapter 4.3) – play a role in these procedures.

Scenarios are generated for the respective investigations based on the relevant parameters. Depending on the sub-investigation, these scenarios are used to determine the tolerable longitudinal compressive forces, the longitudinal compressive and tensile forces occurring due to longitudinal dynamics, the compressive forces and wagon accelerations occurring during shunting impacts and the force collectives with regard to the operational stability (Chapter 4.4).

Using the results of the simulations, investigations are conducted into the operational stability of the coupling. Here, the corresponding force collectives are determined on the basis of their amplitude and frequency using counting methods. These investigations are based on the most realistic possible load assumptions for the braking and acceleration processes occurring during operation. These assumptions are derived from data from the project “Construction and Testing of Innovative Freight Wagons” (Chapters 4.6 and 4.5).

## 4.1 Multibody simulation

Multibody simulation programs (MBS programs), such as SIMPACK, Adams Rail or VAMPIRE, are standard investigative tools in rail vehicle dynamics. MBS programs can simulate oscillating systems, such as trains or wagons in general, using rigid or flexible bodies and force elements that transmit the forces and torques between them. The model is intended to represent the motion behaviour of the real system as closely as possible. However, simplifications are made at some appropriate points which have little or no influence on the results of the simulation [19]. Based on the physical simulation model, these programs determine mathematical models, which are generally solved by time-step integration. Other methods, e.g. an analytical solution, are not capable of achieving the required results due to the complexity and large number of non-linear relationships in rail vehicle dynamics. [20]

The more accurate the model, the more accurate the reproduction of the real system behaviour in the simulation. To achieve a good agreement between the model and reality, it is advisable to carry out a plausibility check or validation for the model. This compares the results of the simulation and measurements for individual cases. Based on these cases, other configurations can also be investigated using the models [21]. The main advantage of using multibody simulation programs is that it offers lower cost/faster investigation of new scenarios, e.g. different train compositions or loads, than measurement-based techniques. Nevertheless, measurement results – if available – are essential for confirming the accuracy of the simulation results and a sufficiently good correlation with reality [22].

## 4.2 Global sensitivity analysis

Global Sensitivity Analysis (GSA) methods are used to investigate and map the influence of the presented parameters on the respective results – in the case of longitudinal dynamics, e.g. the longitudinal compressive forces occurring. These methods can be used to study how the variance of one or more input parameters affects variances in the results of the respective investigation. A variety of mathematical methods with different advantages and disadvantages are available for this purpose, e.g. the *Elementary Effects Test* (EET), *Fourier Amplitude Sensitivity Test* (FAST) or *Dynamic*

*Identifiability Analysis* (DYNIA). PIANOSI, SARRAZIN and WAGENER have integrated the most important of these procedures into their so-called SAFE toolbox, making them easily accessible for various investigations [23].

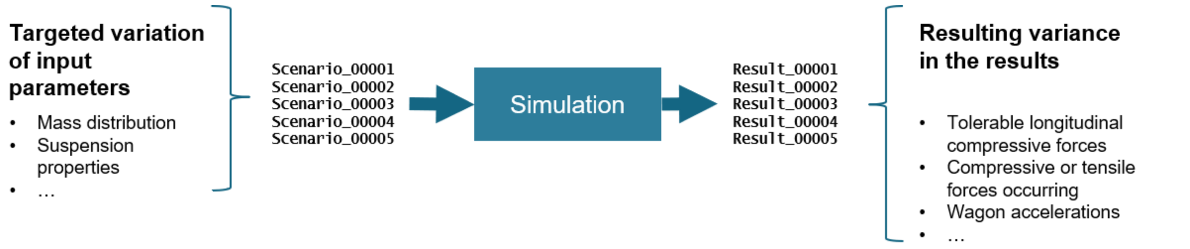



**Fig. 3: Application of GSA to problems in rail vehicle dynamics**

Depending on the method used, the number of scenarios may differ, i.e. the number of simulations required in principle to make valid statements about the influence of parameters [24]. This variable plays a key role for each of the sub-investigations, since even with state-of-the-art computing technology and *parallel computing*[2] the simulation time per scenario is significant. For example, if it is assumed for the simulation of an impact between two wagons that the two wagons in question each have a variable mass of 20 to 90 t, collide in 5 t increments and a variable impact speed of 1 to 10 km/h in 0.5 km/h increments, this would already result in $15 \cdot 15 \cdot 19 = 4275$ cases. Assuming a simulation time of one minute per scenario, the CPU computing time would already be almost 3 days, if all the scenarios are calculated consecutively. If the simulation also uses several variables for the energy absorption by the wagon body and the load, these cases, and thus the required total computing time, multiply even further. In addition to the required computing time, the choice of the most appropriate method for a certain goal depends on the number of input parameters and the general behaviour of the models (Does the model behave in a linear manner or are there non-linear influences?) [24]. With regard to the longitudinal dynamics of trains, studies by the ERRI, for example, found that the behaviour of freight trains and the influence of various

[2] *Parallel computing* is the parallel use of various computer resources simultaneously, e.g. several cores or computers, to solve a problem. In this case, different simulation runs are calculated in parallel on different computer cores. Compared to serial calculation, this allows resources to be used more efficiently, since scenarios often require different amounts of processing time [25].

parameters cannot simply be extrapolated from a few cases to the whole. Some parameters can be very influential under certain circumstances. However, in other combinations they can play quite a minor role in the longitudinal compressive and tensile forces that occur. For this reason, it is necessary to consider the various parameter specifications as comprehensively as possible in order to be able to make general statements [26].

In his dissertation, JOBSTFINKE investigates the applicability of GSA methods to questions of the longitudinal dynamics of freight trains and comes to the conclusion that the *variance-based sensitivity analysis* (VBSA) method – also known as the *Sobol method* – permits a very good separation of influential and non-influential parameters. Based on this work, it is also possible to make statements about generally advantageous parameter specifications for the parameters that can be influenced [5, 23, 24, 27]. The results of the investigation have been incorporated into this study at various points and the selected procedure is also applied to the other sub-investigations.

The method is well suited for use in deterministic studies, i.e. a certain combination of input parameters always leads to the same result. For this purpose, the range of the input parameters is divided according to the *Sobol method*, which ultimately results in the scenarios that are considered within the framework of the respective sub-investigation. In contrast to methods in which just one parameter is varied between different scenarios, so-called *One-At-A-Time*(OAT) procedures, the Sobol method permits several parameters to be changed at once. This allows researchers to make valid statements about the influence of individual parameters on the overall result while significantly reducing the number of scenarios [24].

However, in addition to the variance of the input parameters, the applied methodology is another factor that influences the results. The choice of the result variable to be considered provides an example of this: Since the output for the application of the method must be one-dimensional, the result of the sensitivity analysis can potentially depend, e.g. in the case of longitudinal dynamics, on whether the maximum values for longitudinal compressive or longitudinal tensile forces or an average value of these are used for the sensitivity analysis [5].

Furthermore, the variance of the input parameters plays an important role. The influence of a single parameter on the result can differ, depending on how large the variance of this parameter is and also on the size of the variance in the other influencing parameters [28].

## 4.3 Analysis parameters

The aim of this investigation is to obtain information on the influence of the dynamic parameters of the DAC on tolerable longitudinal compressive forces, longitudinal dynamics, i.e. longitudinal tensile and longitudinal compressive forces generated by shunting impacts, and on the operational stability of the entire coupling. First, it is necessary to identify those parameters which could potentially have an influence. These are based firstly on the investigations presented in Chapter 3 and, secondly, on considerations regarding the interactions between technologies and operations within the railway system. If the goal is to achieve a targeted optimisation of the parameters, it is vital to know which parameters can actually be influenced, e.g. by targeted design adjustments, and which parameters must be assumed to be variable in railway operations. Depending on the sub-investigation, different parameters play important roles.

Two groups of parameters that are important for several sub-investigations within the framework of this study are described in more detail below. These are the parameters that control the spring characteristics, i.e. ultimately the dynamic parameters of the DAC, as well as the mass distribution and mass arrangement of wagons within trains. Fig. 4: shows an overview of the parameters which can be used to vary the characteristic curve of the draft gear. The requirement when selecting these control parameters is that they can be used to generate practically any, technically conceivable, characteristic curves for draft gears. These may include the characteristic curves of technical designs that are already on the market, but also any other variations. Depending on the sub-investigation, these parameters are provided twice, so that the characteristic curves for the tensile and compressive directions can be varied independently of each other.

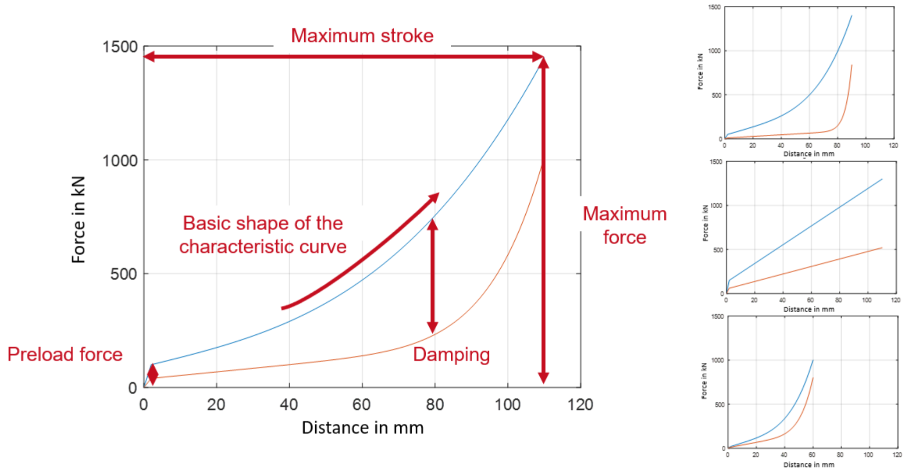



**Fig. 4: Parameters for controlling the spring characteristics**

The right-hand section of Fig. 4: shows an example of three different characteristic curves, which can be adjusted by varying the coupling parameters. The preload force, the maximum stroke and the maximum force – hereinafter referred to as end force – are parameters that can be read from the characteristic curve and are thus relatively easy to interpret. However, auxiliary variables must be introduced for the basic shape of the characteristic curve and for the damping. The basic shape of the characteristic curve – hereinafter also referred to as the spring characteristic – indicates how the coupling force between the preload force and the end force behaves as a function of the spring travel. This parameter is defined so that a value of 1 corresponds to a linear progression. Values below 1 indicate degressive characteristics, values above 1 indicate progressive characteristics. The characteristic curve of the return stroke is a measure of the dissipated energy within a load cycle. This behaviour is called the damping of the spring. Depending on the damping value, the characteristic curve of the return stroke is determined using the characteristic curve of the deflection. This discussion refers initially only to the static characteristic curve, which is sufficient for the consideration of the longitudinal dynamics [5]. For shunting impacts in marshalling yards, more extended studies of the dynamic behaviour of the characteristic curve are required. These will be explained in more detail in Chapter 5.3.2, because they are closely linked to the modelling process.

Fig. 5: shows an overview of the mass arrangements and distributions under investigation. Essentially, it distinguishes between two configurations – a block train in which

all the wagons have the same mass, and a mixed train which has the required total mass but consists of different wagons or groups of wagons with different masses. The mass distributions indicate the relative frequency with which trains exhibit a certain load state. In the 50/50 distribution, it is assumed that a train will run fully loaded in 50% of cases and empty in the other 50%. In the 30/60/10 distribution, the probability of an empty train is 30%, the probability of a half-loaded train is 60% and the probability of a fully loaded train is 10%.

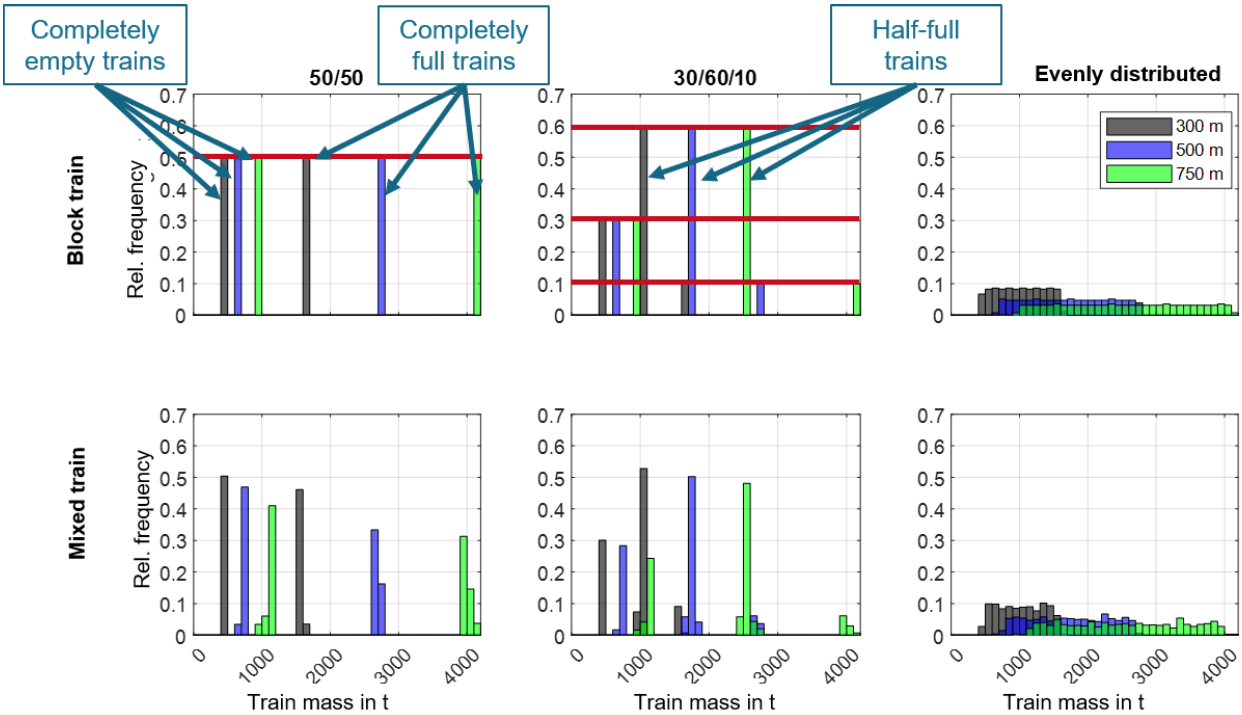



**Fig. 5: Matrix of mass distribution (horizontal) and mass arrangement (vertical)**

In the third variant with equal distribution of mass, all load states are equally likely. The extent to which the mass is dependent on the mass distribution for the entire train depends on the length of the respective train. Three different train lengths are considered here: 300m, 500m and 750m. The total masses of trains with different lengths differ due to the differing empty masses of the wagons, even for a completely empty train.

The detailed analyses for the individual sub-investigations – see Chapters 4.3.1 to 4.3.4 – show that individual parameters must be considered differently for different investigations. When looking at the respective value ranges, it should be noted that the aim is to identify trends. In individual cases, this may mean that the value ranges of some individual parameters do not cover all conceivable special cases or may go well beyond what is currently used in freight transport. When analysing the influence of parameters, it is important to try to consider all parameters as independently as possible and ensure that certain combinations are not excluded in advance. Retrospectively, however, the analysis can be narrowed down to certain areas.

### 4.3.1 Tolerable longitudinal compressive forces

The parameters that are included in the investigation of tolerable longitudinal compressive forces are divided into three groups:

a) Vehicle-related parameters (see Table 3)
b) Coupling related parameters (see Table 4)
c) Parameters related to the test procedure (see Table 5)

The value ranges considered in each case are based on information from the literature as well as assumptions made by the TU Berlin within the framework of this study.

**Table 3: Vehicle-related parameters for the investigation of tolerable longitudinal compressive forces**

| Parameter | Value range | Reference |
|---|---|---|
| Pivot pin spacing | 7 – 20 m | [29 bis 31] |
| Overhang (pivot pin to coupling pin) | 1.5 – 3.0 m | [7, 8] , [29 to 31] |
| Torsional stiffness of the wagon body | $0.5 – 3.0 \cdot 10^{10}$ kNmm²/rad | [3] |
| Centre of gravity of the wagon body in the z-direction above the upper edge of the rails | 1 – 1.5 m | Assumption by TU Berlin |
| Height of the coupling pin in the z-direction above the upper edge of the rails | 0.95 – 1.045 m | [32] |

An overview of the wagon parameters is shown in Fig. 6:. As already explained, this study only considers freight wagons with Y25 bogies. Thus, parameters such as the wheelset spacing in the bogie cannot be changed. Based on observations of real wagons, however, the pivot pin spacing does vary over a relatively wide range. In this study, the wagon overhang is defined as the distance between the pivot pin and the bolt of the DAC in the nominal state. The information on torsional stiffness is closely linked to the modelling process (see Chapter 5.1.2). The elasticity of the wagon body, which depends on the type of freight wagon, is considered in the form of a one-dimensional torsional stiffness.

These parameters are varied to show the principal influence of different wagons. Indirectly, the pivot pin spacing also depends on the wagon type, as certain types are characterised by long and others by short wagons. The height of the coupling pin is determined by the load state of the wagon. However, to ensure compatibility with other wagons it is strictly limited in accordance with **UIC leaflet 522** [32].

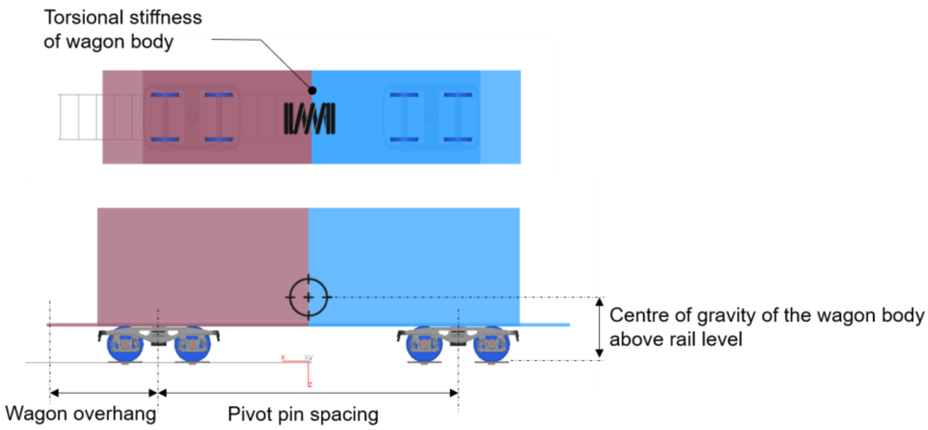



**Fig. 6: Overview of wagon parameters**

The dynamic parameters of the coupling, which were explained at the beginning of this chapter, are also varied. For the investigation of the tolerable longitudinal compressive forces, only the compressive and not the tensile direction is relevant. In addition, the relief of the coupling is not important. For these reasons, only that part of the characteristic curve which characterises the deflection in the compressive direction in the static case is important. In addition to the dynamic parameters, the study also considers two design parameters: the length of the coupling arm and the presence of a stabilisation joint.

**Table 4: Coupling-related parameters for the investigation of tolerable longitudinal compressive forces**

| Parameter | Value range | Reference |
|---|---|---|
| Coupling arm length | 1.025 – 1.4 m | Assumption by TU Berlin |
| Stabilisation joint | Yes/No | |
| Spring preload* | 0 – 300 kN | Assumption by TU Berlin |
| Spring end force* | 1,000 – 2,000 kN | [1], assumption by TU Berlin |
| Spring progressivity* | 0.5 (degressive) - 5 (strongly progressive) | [5] |

* Only for the compressive direction

Finally, the study investigates the extent to which the limiting conditions of the test procedure affect the tolerable longitudinal compressive forces. The main questions here are which frame wagons, i.e. the wagons that run before and after the wagon being tested, are used in the tests, and which geometric characteristics these wagons possess. No generally accepted, standard procedure for testing tolerable longitudinal compressive forces has yet been established. Relevant standards for running safety such as DIN EN 15839 only refer to wagons with side buffers [33]. The usual approach for determining the tolerable longitudinal compressive forces involves superimposing worst-case scenarios for the interactions between the wagon arrangement, the loads of the respective wagons and the track geometry [3]. Thus, it is by no means the case that there is an immediate risk of derailment every time the tolerable longitudinal compressive force is exceeded since this coincidence of unfavourable circumstances in operation is extremely unlikely. For this reason, the test conditions in this study are not considered fixed, but variable.

**Table 5: Parameters for the investigation of the tolerable longitudinal compressive forces in relation to the test method**

| Parameter | Value range | Reference |
|---|---|---|
| Height of the coupling pin in the frame wagon in the z-direction above the upper edge of the rails | 0.95 – 1.045 m | [32] |
| Pivot pin spacing of the frame wagons* | 7 m – 20 m | [29 to 31] |
| Wagon overhangs of the frame wagons* | 1.5 m – 3.0 m | [7, 8] , [29 to 31] |

* stated separately for the front and rear frame wagons

### 4.3.2 Longitudinal dynamics

The parameters that are included in the investigation of longitudinal dynamics are divided into three groups:

a) Parameters affecting the wagons or locomotive
b) Parameters affecting the train as a whole
c) Parameters affecting the coupling design

The value ranges considered in each case are based on information from the literature as well as assumptions made by the TU Berlin within the framework of this study. Due to the large number and complexity of the influencing factors – especially with regard to the brake system – only a selection of the parameters is listed and explained here. For a more detailed consideration, please refer to the dissertation by Mr JOBSTFINKE [5]. Not all parameters play a role in all operating situations that are investigated here. For example, the starting tractive effort of the locomotive is not relevant when considering braking operations. Conversely, the friction material used in the brake plays no role in terms of the starting process.

**Table 6: Parameters for the investigation of the longitudinal dynamics related to the wagons and locomotive (selection)**

| Parameter | Value range | Reference |
|---|---|---|
| Wagon mass | 20 – 90 t | [29 to 31] |
| Friction material | Composite brake pad, disc brake | Assumption by TU Berlin |
| Efficiency of the brake linkage in relation to the nominal value of 0.83 | 0.85 – 1.1 | [5] |
| Starting tractive effort of the locomotive | 150 – 450 kN | [5] |

Table 6 shows a selection of parameters relating to wagons and locomotives. In terms of the longitudinal dynamics, however, it is not so much the characteristics of a particular wagon that are important, but rather the distribution of the characteristics within a train. JOBSTFINKE describes these distributions of individual parameters within a train using the following three factors: M (mean parameter – the average value of the parameter over the train), G (group parameter – the groups in which a certain parameter occurs in the train) and D (difference parameter – the size of the maximum differences for a parameter between different wagons in the train) [5]. It is therefore not possible to distinguish clearly between the parameters for the wagons in Table 6 and the parameters that affect the entire train in Table 7. For example, although the mass arrangement influences the way that the wagon masses are distributed in the train, the masses themselves depend on the range in which the masses of the individual wagons can vary.

**Table 7: Parameters for the investigation of the longitudinal dynamics related to the entire train**

| Parameter | Value range | Reference |
|---|---|---|
| Total mass of the train | Depending on train length/number of wagons, this is dependent on the maximum weight of the wagons and the mass arrangement and distribution | [29 to 31] |
| Mass arrangement | Block train, mixed train | Assumption by TU Berlin |
| Mass distribution | 30/60/10, 50/50, evenly distributed | Assumption by TU Berlin |

With regard to the coupling, the parameters already described play a role in controlling the characteristic curve of the coupling. In its analyses of longitudinal dynamics, the study distinguishes between the tensile and compressive directions. In contrast to the investigation of the tolerable longitudinal compressive forces, the energy dissipation and coupling slack potentially play a major role here.

**Table 8: Parameters for the investigation of the longitudinal dynamics related to the coupling (selection)**

| Parameter | Value range | Reference |
|---|---|---|
| Coupling slack | 0 – 40 mm | [5] |
| Preload of the spring* | 1 – 300 kN | [5] |
| Progressivity of the spring characteristic curve* | 0.5 (degressive) - 5 (strongly progressive) | [5] |
| Energy dissipation during a load cycle* | 0.1 (almost no energy dissipation) – 1 (complete energy dissipation) | [5] |
| Spring end force* | 1000 – 2000 kN | [5] |
| Maximum spring travel until the end force is reached* | 20 -150 mm | [5] |

* each separated according to compressive and tensile direction

### 4.3.3 Shunting impacts

The parameters included in the investigation of shunting impacts are divided into three groups:

a) Parameters affecting the general conditions for shunting impacts
b) Parameters affecting the wagons
c) Parameters affecting the coupling design

The value ranges considered in each case are based on information from the literature, data provided by project partners and assumptions made by the TU Berlin within the framework of this study.

The general conditions for shunting impacts are essentially characterised by the distribution of the impact speeds. In reality, these are distributed within a certain range and therefore cannot be stated as absolute values and reduced to one figure. The distribution indicates the probability that the speed will be higher or lower than a certain value during an impact. This distribution depends on the specific conditions, i.e. principally the technical equipment used by marshalling yards. This equipment can vary considerably and little information is available on the subject.

**Table 9: Parameters for the investigation of shunting impacts in relation to the general conditions of the impacts**

| Parameter | Value range | Reference |
|---|---|---|
| Impact speed | Modern hump yard, UIC 524, UIC 524 mod | Data from project partners, [1], Assumption by TU Berlin |

As can be seen in Fig. 7:, three different distributions are included in the analysis: the distribution according to UIC 524 [1], the distribution for a modern gravity hump yard provided by the project partners within the scope of this study, and a modified speed distribution based on UIC 524. This third distribution is an assumption which takes into account the large discrepancy between the other two distributions, which correspond to particularly good conditions (modern hump yard) and very bad conditions (UIC 524). This distribution was generated by scaling data from UIC 524 downwards and is therefore referred to as "UIC 524 mod" in the following section.

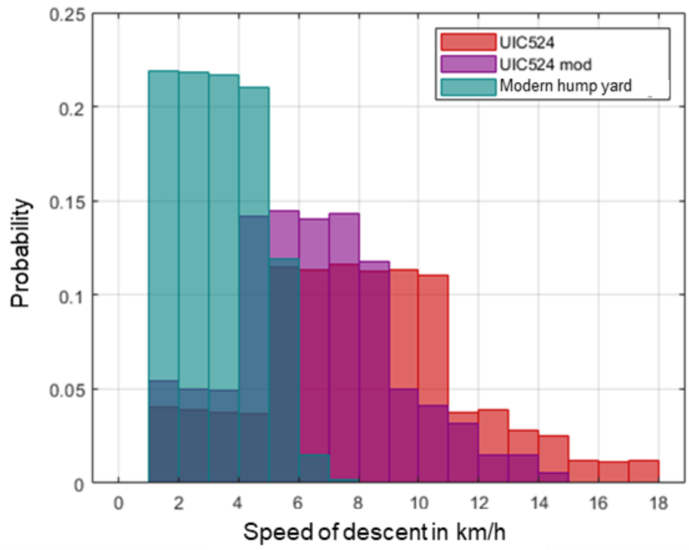



**Fig. 7: Velocity distributions considered for the investigation of parameter optimisation with respect to shunting impacts, in-house representation according to [1]**

The mass of the wagon and its load also plays a role when considering shunting impacts. In accordance with the basic assumptions used in this study, only four-axle freight wagons are considered here. Wagon elasticity also has a potential influence, as shown by the considerations in **ERRI report B36 RP27** [14]. However, the type of energy absorption differs depending on the type of wagon and load. Accordingly, this investigation distinguishes between the different types examined in **ERRI report B36 RP27**, using two different approaches, which are discussed in more detail in the chapter on modelling (see Chapter 5.3). The amount of energy absorbed also depends on the path over which the corresponding elastic deformation of the wagon is applied – referred to here as the wagon elasticity value.

**Table 10: Parameters for the investigation of shunting impacts in relation to the wagons**

| Parameter | Value range | Reference |
|---|---|---|
| Mass of the load of the impacting wagon | 0 – 70t | [29 to 31] |
| Mass of the load of the impacted wagon | 0 – 70t | [29 to 31] |
| Type of wagon elasticity | 1 - 7 | [14], assumption by TU Berlin |
| Wagon elasticity value | 5 – 40 mm | Assumption by TU Berlin |

The definition of the parameters affecting the coupling design is carried out in the same way as for the values in the investigation of longitudinal dynamics shown in Table 11.

### 4.3.4 Operational stability

The parameters of the operational stability test refer to the operational conditions to which a freight wagon, and thus its coupling, is subjected. These conditions are divided, according to their causes, into stresses resulting from longitudinal dynamics and stresses resulting from impact shocks. Some of the investigation parameters are relevant to both cases and therefore referred to here as superordinate parameters. The parameters are listed in Table 11 to Table 13. The value ranges considered in each case are based on information from the literature, evaluated data from the project “Construction and Testing of Innovative Freight Wagons" (see Chapter 4.6, hereinafter IFW data) and assumptions made by the Technical University of Berlin in the context of this study.

**Table 11: Parameters for the operational stability investigation related to longitudinal dynamics**

| Parameter | Value range | Reference |
|---|---|---|
| Number of braking operations per 100 km | ½ IFW data - 2 IFW data[3] | Evaluation of IFW data |
| Percentage of emergency braking operations per 100 km | | |
| Number of acceleration operations per 100 km | | |
| Mass distribution of trains | 0 (= 100% mixed trains) – 1 (= 100% block trains) | Assumption by TU Berlin |
| Average train length | 320 m – 500 m | Assumption by TU Berlin |
| Brake pad material | 0 (= 100 % composite shoe) – 1 (= 100 % disc brake) | Assumption by TU Berlin |
| Number of inclination changes per 100 km | 0 - 20 | Assumption by TU Berlin |
| Average inclination | 2 – 8 ‰ | Assumption by TU Berlin |

**Table 12: Parameters for the operational stability investigation relating to shunting impacts**

| Parameter | Value range | Reference |
|---|---|---|
| Number of marshalling yard stops per 100 km | 0.01 - 1 | [1], assumption by TU Berlin |
| Distribution of hump yard speeds | 0 (= 100% modern hump yards) - 1 (= 100% hump yards with speed distribution according to UIC 524) | See 3.4 |
| Average group size of the wagons running down the hump | 1 - 3 | Assumption by TU Berlin |

[3] ½ IFW data means that braking or accelerating takes place half as often as stated in the data evaluated in Chapter 4.6 for the IFW, 2 IFW data means it takes place twice as often.

Table 13: Superordinate parameters for the operational stability investigation

| Parameter | Value range | Reference |
|---|---|---|
| Design of the spring apparatus | Optimised for longitudinal dynamics / Optimised for shunting impacts | Based on sub-investigations of longitudinal dynamics and shunting impacts |
| Mass arrangement within trains | 50/50<br>30/60/10<br>equally distributed | Assumption by TU Berlin |
| Annual mileage | 40,000 km – 200,000 km | [15] |

## 4.4 Generation and simulation of investigated scenarios

The influence of the parameters for the different sub-investigations is investigated using deterministic methods. It is therefore based on deterministic combinations of parameters, known as “scenarios” in the context of this study. In the case of longitudinal dynamics, for example, one specific scenario corresponds to a freight train that is assembled and loaded in a certain way, has couplings with defined parameters and performs a certain driving manoeuvre, such as emergency braking to a standstill or starting up to a certain target speed [5].

For all sub-investigations, the scenarios are generated using similar principles, although the details may differ. For the individual sub-investigations, the following steps are used for the generation of scenarios, simulation and the subsequent data evaluation and further processing:

### a) Tolerable longitudinal compressive forces

To investigate the tolerable longitudinal compressive forces, the variable parameters and their respective value ranges are first defined (see 4.3.1). Based on these definitions, the individual scenarios are generated using the Sobol method, whereby the parameters are varied (see 4.2). *Parallel computing* is used to calculate the tolerable longitudinal compressive forces for the individual scenarios. Several simulations are required for each scenario. In each case, the acting longitudinal compressive force is changed and the relevant variables in wheel-rail contact are evaluated with regard to the derailment criteria. The tolerable longitudinal compressive force at which the vehicle will not derail is approached from above and below using an iterative process. The determination is accurate to 10 kN. The results of the parallel simulations are stored centrally and evaluated once all the calculations have been completed, e.g. identifying the influential parameters.

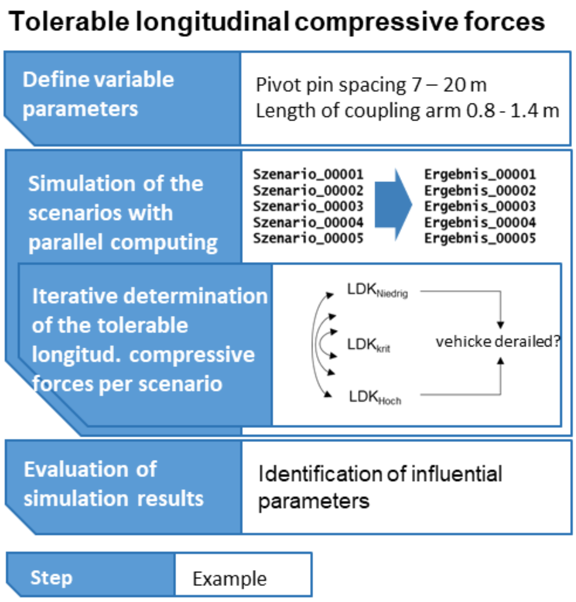



**Fig. 8: Generation of simulation scenarios for the investigation of tolerable longitudinal compressive forces**

## b) Longitudinal dynamics

When investigating longitudinal dynamic processes, it is important to distinguish between different operating situations which are considered separately. These include, for example, operational and emergency braking from various speeds to a standstill, or achieving a certain target speed, or acceleration processes with or without disruption to traction. For this investigation, it is also necessary to distinguish between the various basic train types which have specific characteristics. Three different train lengths are considered here: 300 m, 500 m and 750 m. Furthermore, the investigation only considers trains that use a uniform friction material. Within the scope of this study, block brakes with composite brake pads and disc brakes are considered. More detailed studies, e.g. for trains consisting of wagons with different friction materials, were carried out by JOBSTFINKE in his dissertation [5].

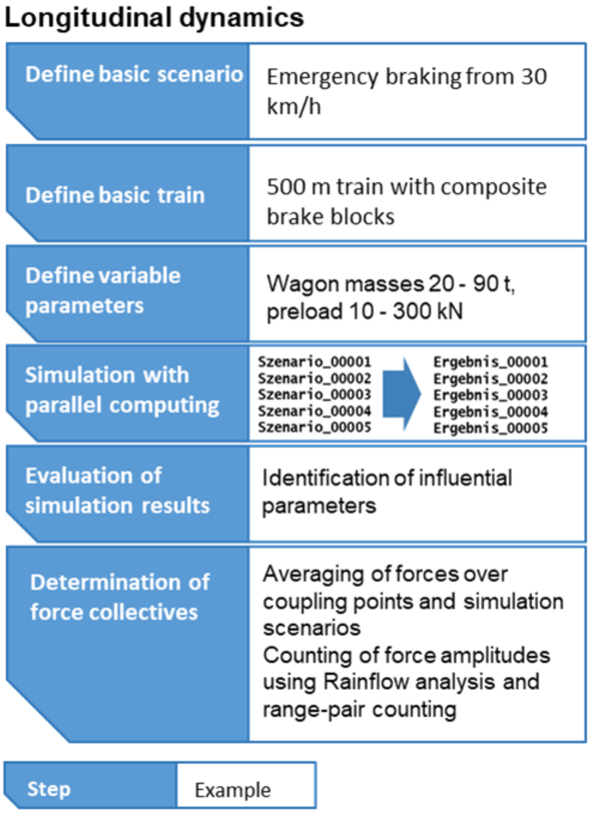



**Fig. 9: Generation of simulation scenarios for the investigation of longitudinal dynamics**

In the next step, the other variable parameters are determined in the same way as described for the tolerable longitudinal compressive forces with their respective value ranges (see 4.3.2). The scenarios are generated from these using the Sobol method. The simulations of the individual scenarios are run in parallel using *parallel computing*. The results are then processed further, e.g. including suitable filtering of the longitudinal forces. They are filtered in the same way as the results produced by JOBSTFINKE and exclusively in the time range [5]. To provide a basis for the investigation of the operational stability, counting methods are used to determine force collectives from the time records of the force curves (see 4.5).

### c) Shunting impacts

The procedure for the investigation of shunting impacts is largely analogous to the procedure for longitudinal dynamics. Once the basic scenario has been defined, the variable parameters are determined and the simulation is run using *parallel computing*.

The influential parameters are then identified and the force curves during the impact processes prepared for the operational stability tests.

**Shunting impacts**

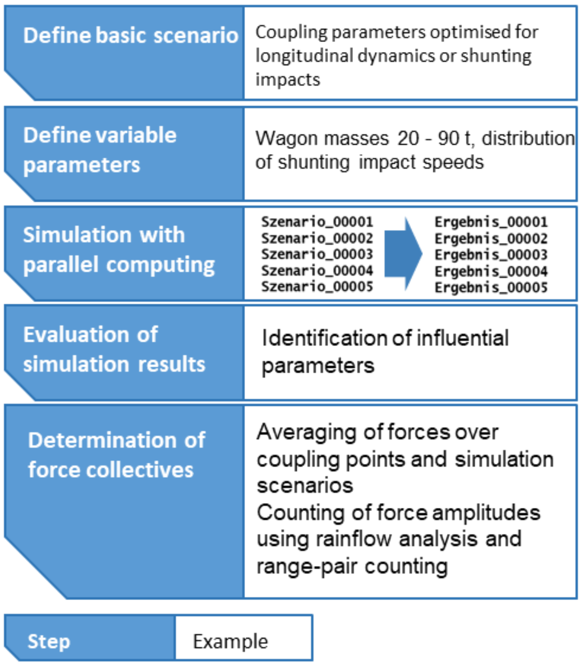



**Fig. 10: Generation of simulation scenarios for the investigation of shunting impacts**

## d) Operational stability

The methodology for the investigation of operational stability differs significantly from the other sub-investigations in that no multi-body simulation calculations are performed. Instead, the investigation is based on the results generated from the sub-investigations into the longitudinal dynamics and shunting impacts. The first step, however, is the definition of the variable parameters and their respective value ranges, whereby most of the parameters refer to the operational limiting conditions in RFT (see 4.3.4).

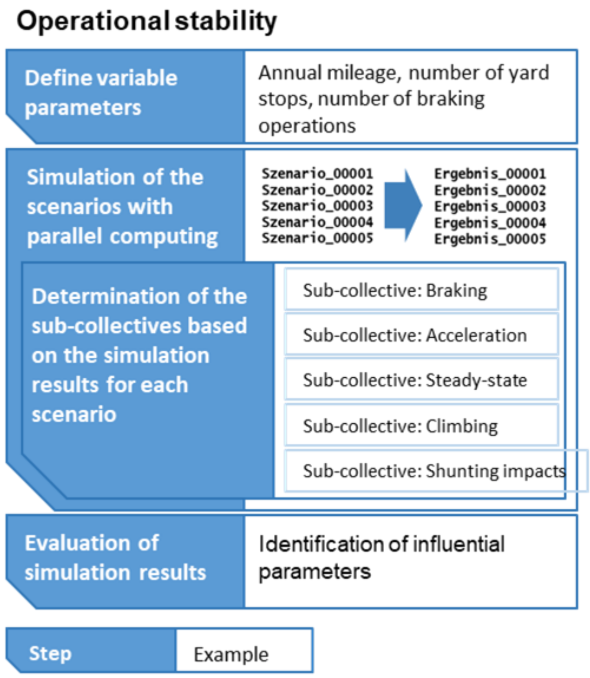



**Fig. 11: Generation of simulation scenarios for the investigation of operational stability**

From these, the operating scenarios to be investigated are now generated using the Sobol method and calculated using parallel computing. For each scenario, the sub-collectives are calculated for the individual operational situations to which a wagon or coupling is exposed in the course of its service life, e.g. braking and acceleration processes, steady-state and coasting phases, uphill and downhill gradients and processes in marshalling yards.

If the objective is to calculate force collectives for individual scenarios under specific load assumptions and limiting conditions, the previously stored sub-collectives for specific operating scenarios are used. The basic procedure is shown in Fig. 12:. The analysis first distinguishes between a variety of operational situations: acceleration, braking, steady-state running, change of inclination and shunting impacts. The relevant parameters for each of these operating situations are extracted from the scenario. The limiting conditions, e.g. the mass arrangement or the average train length, determine which collective must be loaded from the database. Based on the load assumptions that specify how often this operating situation occurs over the observation period of one year, the collective is superposed to a sub-collective for this operating situation. A similar approach is taken for the other operating situations. Finally, the individual sub-collectives are combined to form an overall collective for a specific scenario.

This information is only included here for the sake of completeness and serves as an overview. More details on the formation of force collectives are provided in the following chapter.

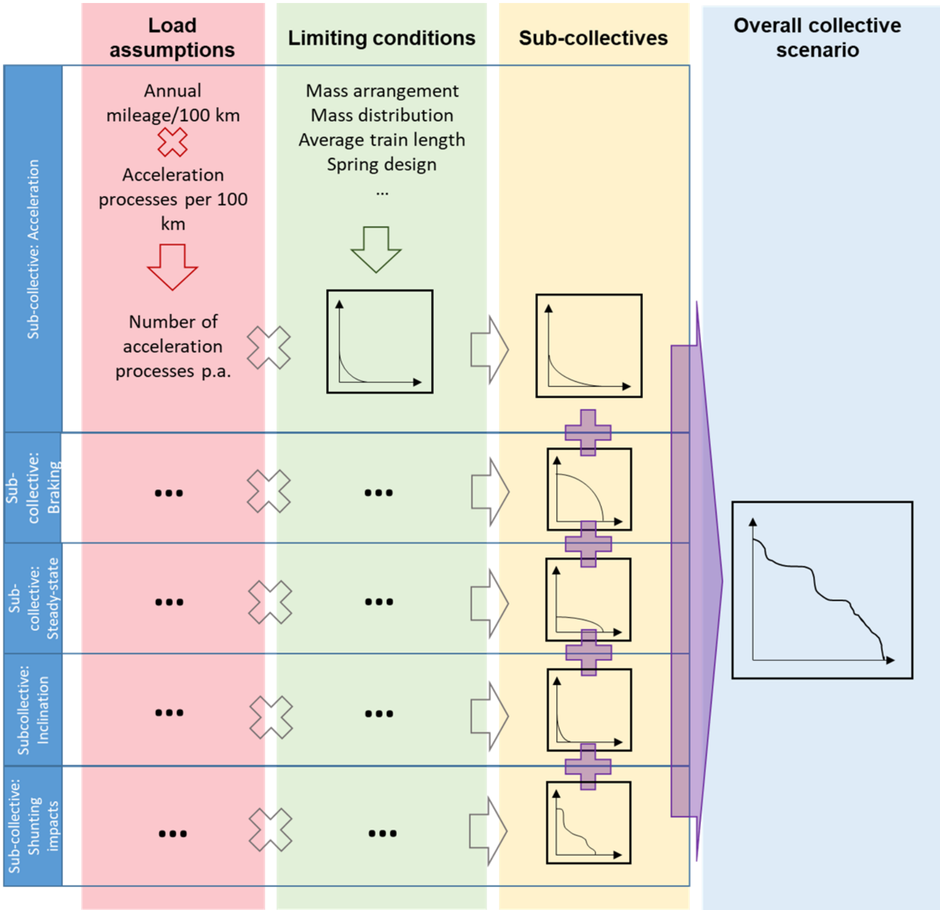



**Fig. 12: Superposition of sub-collectives to the overall collective**

## 4.5 Determination of force collectives in connection with the operational design of the DAC

The aim of the operational stability investigation is to determine to what extent the optimisation of the dynamic parameters, resulting from the investigation of the longitudinal dynamics and shunting impacts, affects operational stability. For this reason, this analysis only considers those dynamic parameter values that have proven to be optimal based on the selected value ranges in the respective sub-investigation.

### 4.5.1 Operational stability in connection with the DAC

During operation, the freight train couplings are subjected to oscillating stresses resulting from longitudinal dynamics while the train is in motion and shunting impacts in marshalling yards. In contrast to static loads, alternating loads require special consideration, since this form of stress can lead to fractures, even if the effective stress amplitudes are well below the yield or fracture limit [34].

During the design phase, a suitable design or dimensioning is required in order to ensure that the sub-assembly or component can withstand the required number of load cycles – essentially derived from the specified service life of the component – without cracking or fracturing due to the vibrations. The decisive factor here is not the forces that occur, but the stresses that result from the forces, i.e. the component stresses [35]. Given the same force amplitude, a component with a large cross-section is damaged much less by alternating stress than a component with a small cross-section. The same applies to the materials and their characteristic values.

This design is based on reasonable assumptions regarding the amplitudes and frequencies of the component stresses that can be expected during operation. Here, the material, design, production-related, operational and environmental conditions must also be taken into account [34 bis 36]. Thus, when applying the design to the DAC, knowledge of the stresses resulting from longitudinal dynamics and shunting impacts is required.

The design is differentiated according to the type of process selected. In a fatigue-resistant design, components are designed in such a way that they can theoretically withstand an infinite number of fatigue cycles and can therefore be used for any length of service life – as long as the load amplitudes on which the design is based are not exceeded during operation (see Fig. 13:). A fatigue-resistant design requires a correspondingly large and solid dimensioning of components [34].

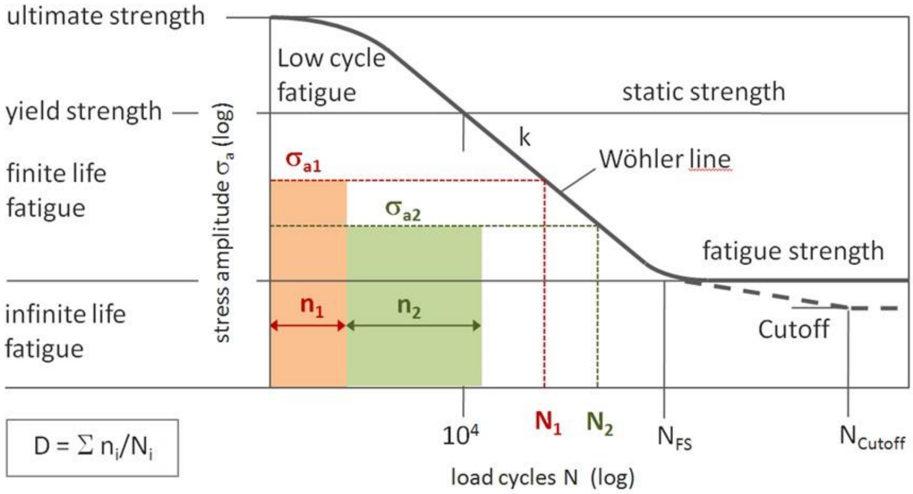



**Fig. 13: Fatigue strength and operational stability in the Wöhler diagram [37]**

In contrast, a limited useful life is assumed for an operationally stable design. Taking safety factors into account, these data are used to determine the design required for the specified service life. The component design is based on assumptions regarding the operating stresses for the minimum number of cycles to failure during the specified service life. Compared to a fatigue-resistant design, a certain number of higher stresses are also permitted. Thus, in contrast to the fatigue-resistant design, the component dimensions can be lower. The operationally stable design can therefore provide the basis for lightweight construction [35].

Many of the tests and procedures relating to operational stability were developed in the railway sector and go back to August Wöhler who designed wheel tyres and wheelset axles in the second half of the 19th century. The assumptions of the *Wöhler curve*, which shows the relationship between the number of cycles to failure and stress, are still valid in a modified form today [38].

*Counting methods* are used to determine the loads and frequencies of stresses occurring during the service life. The frequency with which certain stress amplitudes occur is generally recorded in *stress collectives*. The frequencies and amplitudes recorded in these stress collectives can be compared using the Wöhler curve. Using damage accumulation hypotheses, the collective can be transformed to a single equivalent damage, e.g. according to Miner's rule [34, 35].

In this study, the challenge is to achieve results that are as robust and meaningful as possible, while at the same time considering the DAC as generic, i.e. not a specific design. Based on the simulations, it is possible to make statements about the acting forces. However, to record stress collectives, the corresponding cross-sections in the coupling are also required as input conditions. These, in turn, are only available for certain designs.

Consequently, the approach used in this study is shifted from the level of the stress collectives to the level of the force collectives that can be generated from the simulation data. Force collectives do not allow any direct statements to be made about the operational stability because they cannot be directly compared with characteristic values, such as the Wöhler curve, and thus no conclusions can be drawn regarding component failure. However, consideration of the force collectives does make it possible to identify the most important parameters for an operationally stable design (longitudinal dynamics/shunting impacts) and, if a specific technical realisation of the coupling is known, these force collectives can be converted into stress collectives.

### 4.5.2 Rainflow analysis and range-pair counting

The force collectives described in the previous section are derived from the time records of the forces acting in the couplings – determined by the investigations of longitudinal dynamics or shunting impacts – using counting methods. *Rainflow analysis* and then *range-pair counting* are used for this purpose. These two procedures are presented in the following overviews.

Rainflow analysis is a very clear method that reproduces the material-mechanical processes of vibrational stress very accurately [35]. For this reason, it has established itself as one of the standard methods for the analysis of operational stability and is used in many different fields of engineering, e.g. by Sturm for investigating the load-dependent component behaviour for the weight optimisation of drive shafts for motor vehicles, or by Brunnhofer for investigating the operational stability of bogie attachments in regional trains [39, 40].

The rainflow analysis model is very visual (see Fig. 14:). It is based on the fact that the time plot of the force curve is rotated by 90° and that rain flows from top to bottom, hence the name. The force progression is divided into different classes with a defined width based the amount of force. The analysis records the classes between which the imaginary rain forms closed circuits. For each of these circuits, the amplitude is recorded based on the starting and target class and the frequencies of the amplitudes occurring are stored as the rainflow matrix.

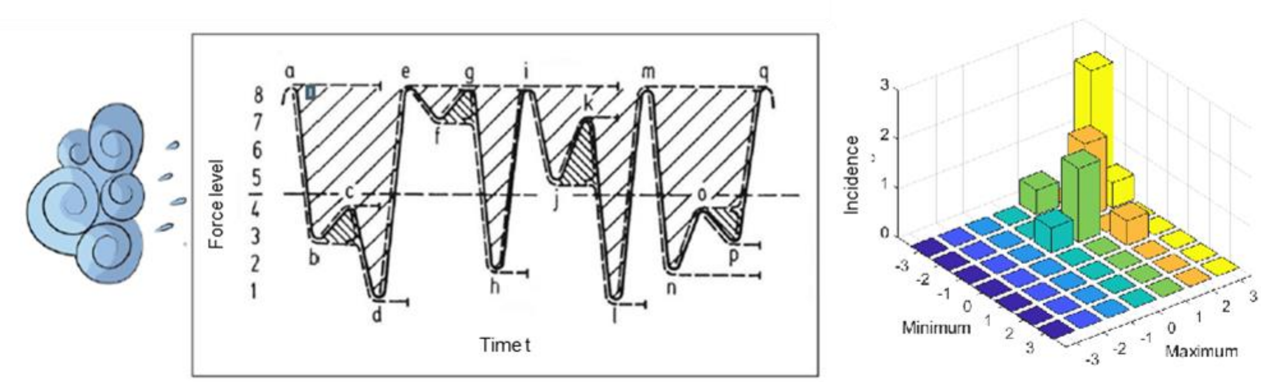



**Fig. 14: Rainflow analysis model (left) and example of a rainflow matrix (right), modified presentation in accordance with [35]**

Especially when evaluating shorter time records with a correspondingly limited number of fatigue cycles, it is necessary to include not only closed cycles but also *residuals*, i.e. non-closed cycles [35]. The rainflow method is a *two parameters' counting method*, i.e. the rainflow matrix provides information about the amplitude, based on the minima and maxima, as well as the mean values [34].

The rainflow matrix can now be used to determine the associated force collective with the aid of a one parameter counting procedure, in this case *range-pair counting*. First, the direction in which a certain force amplitude is passed through is equalised. The fatigue cycles with the same amplitude are then totalled. The result is the force collective for the evaluated time record of the force path, in which the force amplitudes are plotted over their respective frequency.

### 4.5.3 Determination of force collectives based on simulation results

To identify meaningful force collectives for the coupling based on the simulation results for longitudinal dynamics and shunting impacts, it is necessary to consider more than the different operating situations for the train as a whole. In the case of longitudinal dynamics, the position of the coupling within the train plays an important role. In terms of the forces occurring, it makes a considerable difference whether a coupling is in the first, middle, last or any other position within the train during a driving manoeuvre [5].

In the course of its life, a freight wagon and its coupling will typically run in a large number of freight trains with different compositions. As the length of the period during which the life of a freight wagon is studied is extended, the probability that the relative frequency of the position of this wagon within the trains in which it is placed will approach an equal distribution increases, in accordance with the law of large numbers [41]. For this reason, the force or stress collectives on which the design is based should not be determined on the basis of the superposition of extreme cases, but with the aid of statistically substantiated assumptions. One worst case scenario, for example, would be the assumption that a wagon always runs in the middle of freight trains, where the greatest longitudinal forces generally occur. In this case, only the simulation results for the middle wagon would be considered in the operational stability investigation [5, 42].

If this assumption were used, the loads or stresses resulting from the longitudinal dynamics would be estimated relatively reliably upwards but would result in the vast majority of couplings being over-dimensioned. The more often wagons are used in different trains, e.g. in wagonload traffic, the less realistic this assumption becomes. Even in intermodal transport, where wagons often run in fixed formations over long periods of time, it is unlikely that a wagon will always occupy the same position in the train throughout the service life of its coupling. The same applies to the load state. Here, the RFT production form in which a particular wagon is used plays a major role: in extreme cases, it could be assumed that a wagon always runs with a full load, while in fact during operation it only runs empty, partially loaded or fully loaded, depending on the production form (see 3.6).

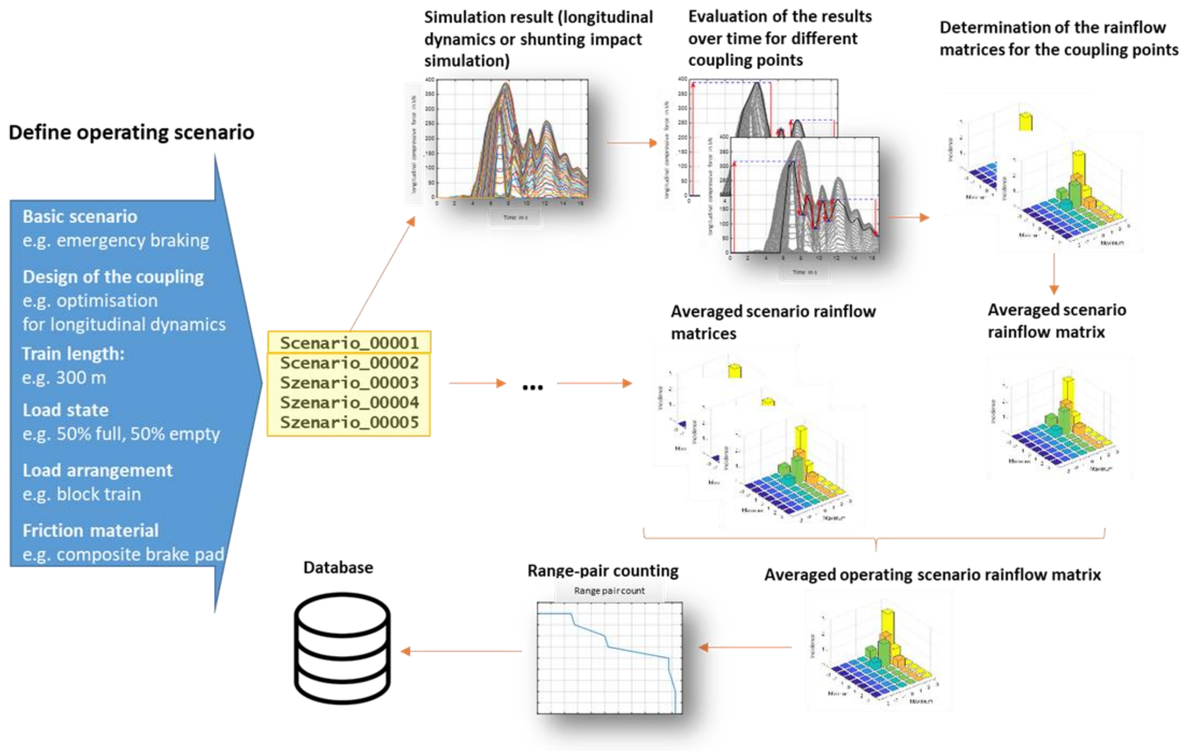



**Fig. 15: Determination of the force collective for an operating scenario (longitudinal dynamics)**

The forces occurring in the couplings as a result of the longitudinal dynamics simulation are therefore averaged over the individual coupling points, using the first, middle and last coupling points. The averaging does not take place at the level of forces, as this would result in inadequate consideration of the high forces that are particularly relevant for the operational stability analysis, but at the level of the rainflow matrices. Next, an averaged rainflow matrix is produced for each operating scenario – consisting of the basic scenario, the respectively optimised coupling parameters, the train length, the assumptions regarding load distribution and configuration and the friction material – and converted into the corresponding force collective using range-pair counting. An overview of the procedure described is shown Fig. 15: .

For shunting impacts, a similarly differentiated approach is also required. Here, there is only one basic scenario – the process in the marshalling yard. The operating scenario includes the respective optimised coupling design, the load distribution and the distribution of shunting impact speeds. The procedure for determining the force collectives is analogous to the procedure described in Fig. 15: for longitudinal dynamics.

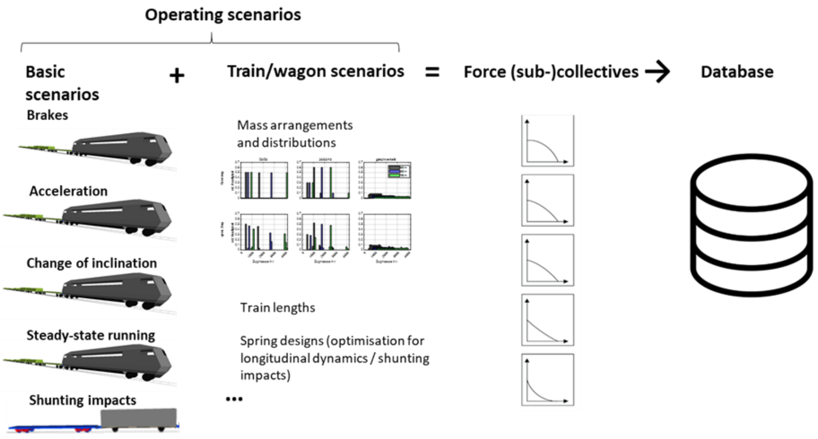



**Fig. 16: Determination of force collectives for various operating scenarios**

The force collectives for the individual operating scenarios are stored in a database as shown in Fig. 16:. The overall collective is assembled from this data during the operational stability analysis. It is made up of the various sub-collectives from the basic scenarios in accordance with the load assumptions (see 4.3.4).

### 4.5.4 Determination of a reference collective

The sensitivity analysis procedure for the identification of the most important parameters for a partial investigation was presented in Chapter 4.2 and requires the results for a scenario to be reduced to a single value. To quantify and compare the influence of operational and technical parameters on the force collectives, the force collectives are compared scenario-by-scenario with a reference collective. In contrast to stress collectives, where the graphs can be compared with the Wöhler curve, there is no such standard procedure for the comparison of force collectives [34]. In the context of this study, the procedure is intended to reveal the parameter influences for which the absolute values of the collective are initially of secondary importance. Instead, the relative differences play the decisive role.

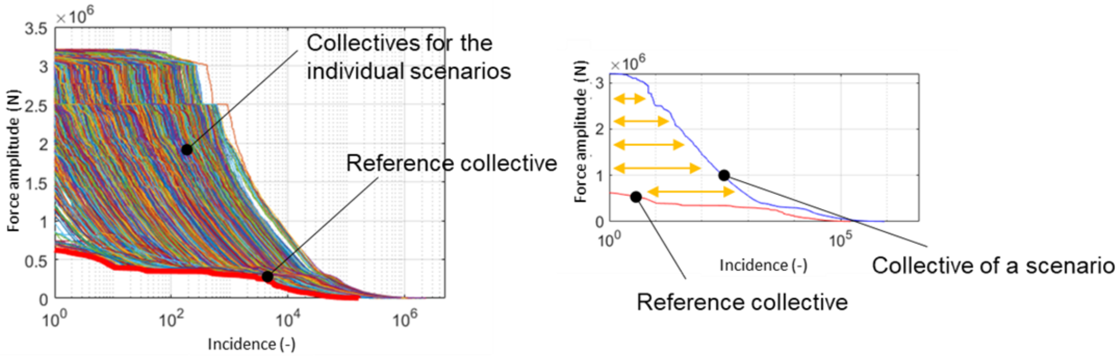



**Fig. 17: Determination of a reference collective**

The corresponding reference collective is formed as a minimum from the collectives of all considered scenarios as shown in Fig. 17:. To express the difference between the collective of a given scenario and the reference collective as a single value, the differences in frequency are calculated for each class of force amplitude. These differences are then added up – this total is referred to as the cumulative difference of the force collectives. At this point, it should again be pointed out that the formation of this value ultimately also influences the result of the sensitivity analysis. Since this is not a standard procedure, Chapter 6.5.1 examines various procedures for the formation of the differences and the weighting of particularly high forces using the results of the sensitivity analysis for the partial investigation of the operational stability.

## 4.6 Determination of load assumptions for railway operation based on data from the project “Innovative freight wagon”

Although the methodology presented in the previous chapter can be used to derive force curves and force collectives for various operational situations to which a freight wagon or coupling may be exposed in the course of its service life, the operational situations themselves cannot be simulated. Force collectives are only meaningful if they are based on realistic assumptions about stresses, loads and their frequencies. This would apply in the same way to the corresponding stress collectives. Specifically, the following questions arise:

a) How often does a wagon typically brake within a certain interval, e.g. 100 km?
b) How high is the proportion of emergency braking operations in relation to the total number of braking operations?

c) How often does the wagon brake?
d) How high is the proportion of steady-state and coasting phases relative to the total distance travelled?
e) On average, how often do wagons pass through marshalling yards and what are the shunting impact speeds?

These questions are difficult to answer without concrete operating data and can only be narrowed down by rough estimates. Moreover, it is not possible to make generalisations here because it can be reasonably presumed that operational conditions vary greatly between different forms of production, e.g. intermodal and wagonload transport.
Only very limited experience or measured values from operations were available within the framework of this project, e.g. number of stops in marshalling yards and shunting impact speeds at different marshalling yards. However, for the limiting conditions that primarily affect the longitudinal dynamics – essentially braking and acceleration processes – it was possible to draw on data from the project "Construction and Testing of Innovative Freight Wagons", which was carried out on behalf of the BMVI between 2016 and 2019 by a consortium led by DB Cargo and VTG (henceforth referred to as: IFW data). During this project, a demonstrator freight train completed test runs in various configurations in several European countries and on various routes within Germany, covering a total distance of approx. 150,000 km [43].
This study had access to measurement data for completed trips during the period March to October 2018 and evaluated GPS data on position and speed as well as the pressure curves for the main brake pipe. However, for reasons not specified, these data do not cover the entire operating test period in Germany and, moreover, could not be evaluated consistently.
For the evaluation, the recorded journeys were broken down into separate sections according to the respective operating situation. The speed profile and the MBP pressure curve were used for classification purposes (see Fig. 18:). The following running conditions were identified and the respective start and target speeds were recorded in classes with a range of 10 km/h each:

- Service braking
- Emergency braking
- Acceleration
- Steady-state running
- Coasting

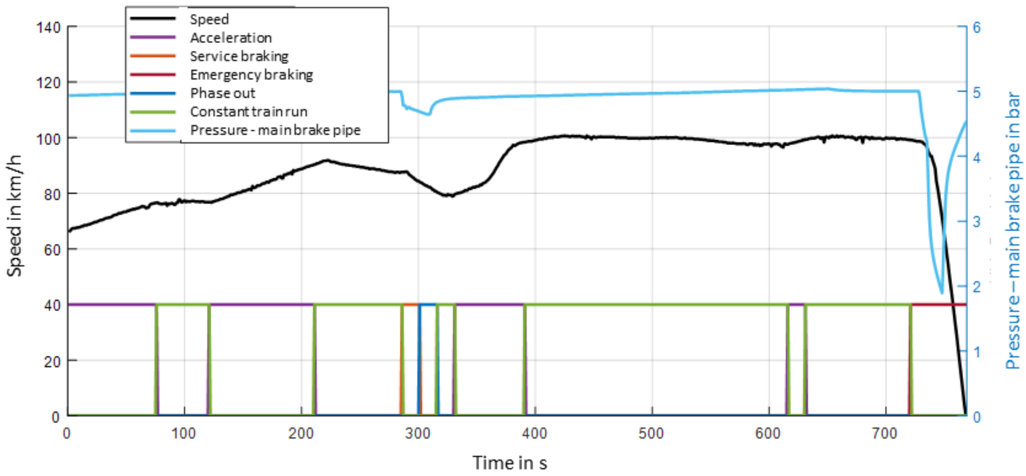



**Fig. 18: Sample classification of different operating situations based on the speed and MBP pressure curve**

The absolute frequency, the distance travelled and the time span were statistically recorded over the entire measurement period. Average values were determined from these measurements and scaled to a distance of 100 km for further calculations. Sample results for the braking processes are shown in Fig. 19:.

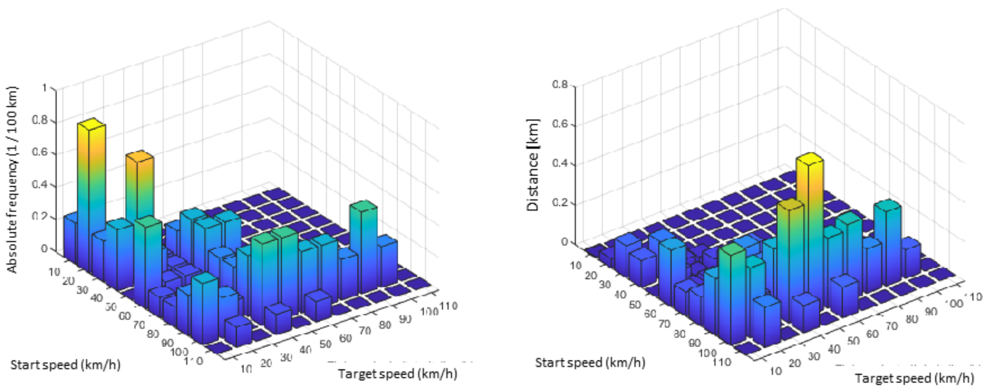



**Fig. 19: Evaluation of IFW data for to relative frequency and distance of braking processes**

The distinction between operational and emergency braking is based on the pressure drop measured in the main brake pipe. Selected results from the data evaluation are listed in Table 14. For the braking and acceleration processes, the absolute number was added up independently of the start and target speed.

**Table 14: Selected results from the evaluation of the IFW data**

| Evaluation variable | Value |
|---|---|
| Average number of braking operations per 100 km | 8.5 |
| Proportion of emergency braking operations relative to the total number of braking operations | 1.38 % |
| Average number of acceleration processes per 100 km | 19.6 |
| Average proportion of steady-state and coasting phases in the total distance travelled | 75.1 % |

The IFW data were collected during a field test in real-world railway operations, so it is assumed that the results are reasonably representative for the entire freight train fleet, even if there is a residual uncertainty for the reasons mentioned above.

# 5. Structure of the simulation models

This chapter presents overview of the various MBS models for investigating tolerable longitudinal compressive forces, longitudinal dynamics and shunting impacts. The models used are adapted to the purposes of the respective sub-investigations and therefore follow different modelling approaches. One key difference lies in the number of degrees of freedom: while one-dimensional models are sufficient for investigations of the longitudinal dynamics and shunting impacts within the scope of this study, a three-dimensional vehicle model is required for the investigation of tolerable longitudinal compressive forces.

## 5.1 Investigation of tolerable longitudinal compressive forces

### 5.1.1 Overview of the models and general approach

The test wagon model used in the push tests is embedded in a train consisting of two frame wagons and a locomotive. For the investigation of the tolerable longitudinal compressive forces, an MBS model of a freight wagon is constructed, consisting of submodels of the wagon body, two bogies and four wheelsets. The vehicle models are connected using models of an AC. The general structure of the model with the various model levels is shown in Fig. 20.

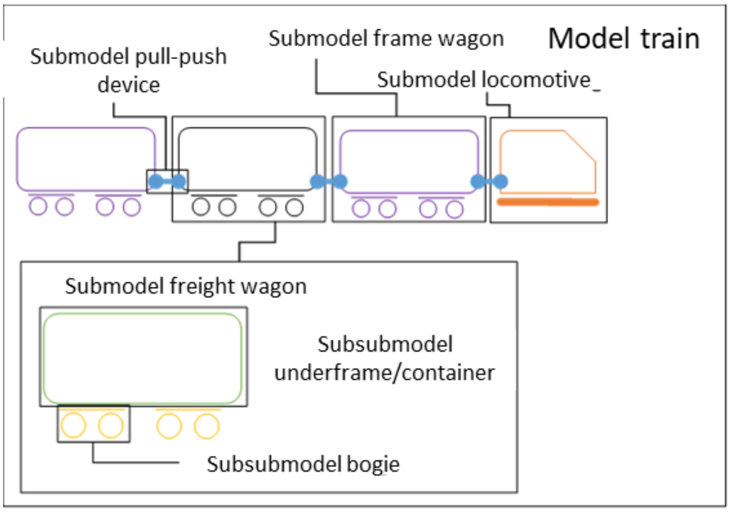



**Fig. 20 Structure of the MBS model for the investigation of tolerable longitudinal compressive forces**

Due to the large number of simulations required within the scope of this study, the focus is on producing the most efficient execution of the simulation and the most accurate reproduction of the real running behaviour of the vehicles. To find a suitable compromise, a highly simplified model of the frame wagons and the locomotive was used. This approach is in line with the methodology proposed in ERRI report B 12 FP 59 (see 3.3) [10].

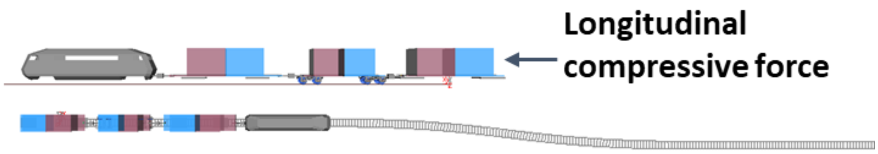



**Fig. 21: Simulation of push tests by determining the tolerable longitudinal compressive forces**

To determine the tolerable longitudinal compressive forces, the three-wagon train passes through an S-bend with a curve radius of 150 m and an intermediate straight, with the locomotive keeping the train at a constant speed of 5 km/h. The longitudinal compressive force is applied to the last wagon (see Fig. 21:). The test speed is based on the specifications in DIN EN 15839, where a speed of between 4 and 8 km/h is recommended for the performance of push tests [33]. The wheel-rail contact variables are evaluated for the middle wagon, i.e. the wagon under investigation. The criteria for the classification of a derailment according to DIN EN 15839 are applied [33]. The following sections present overviews of the structures of the individual sub-models.

### 5.1.2 Freight wagon model

The model of the freight wagon for which the tolerable longitudinal compressive forces are to be investigated consists of two sub-models: the model for the two bogies and the model of the wagon body. In the context of this study, only freight wagons with the Y25 bogies shown in Fig. 22: are considered. This is the most frequently used bogie in RFT in Europe [44].

Fig. 22: Y25 freight wagon bogie [45]

In addition to the two wheelsets, the bogie consists of four axle bearing housings, a bogie frame, a primary restraint and the connection to the wagon body via a pivot pin. The primary restraint consists of two-stage coil springs and load-dependent friction damping. The bogie is assigned to the rigid axle bogies [46, 47]. The modelling approach is based on various studies by KEUDEL, HECHT und SCHELLE at the Faculty of Rail Vehicles at the TU Berlin [48 bis 50]. The structure of the bogie model is shown schematically in Fig. 23:.

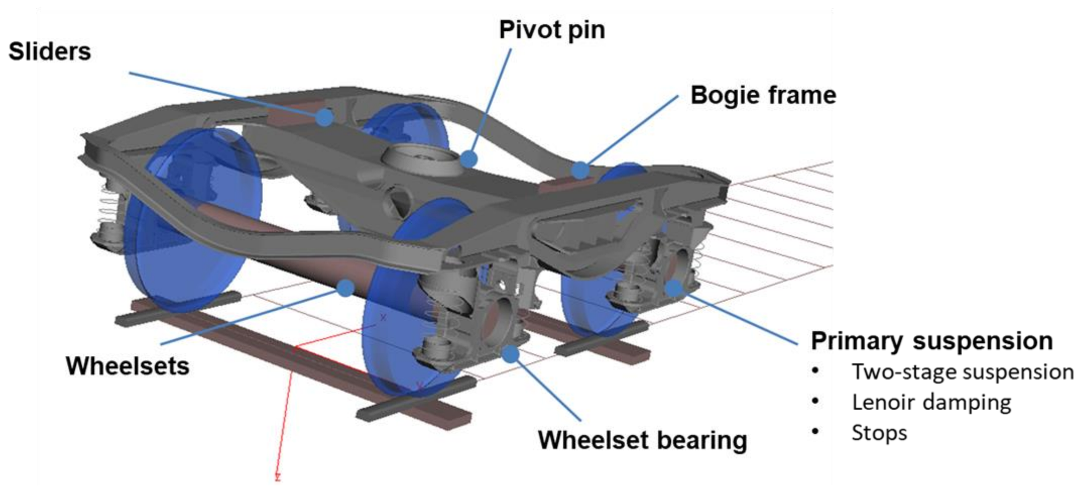



Fig. 23: Schematic structure of the MBS model for the Y25 bogie

The structure of the primary restraint is shown schematically in Fig. 24:. It acts between the bogie frame (1) and axle bearing housing (2). The bogie frame is supported on two spring assemblies. To achieve load-dependent stiffness, each spring assembly consists of an inner and an outer spring. The outer spring is always under load, the inner spring is shorter and only engages when a certain load is reached.

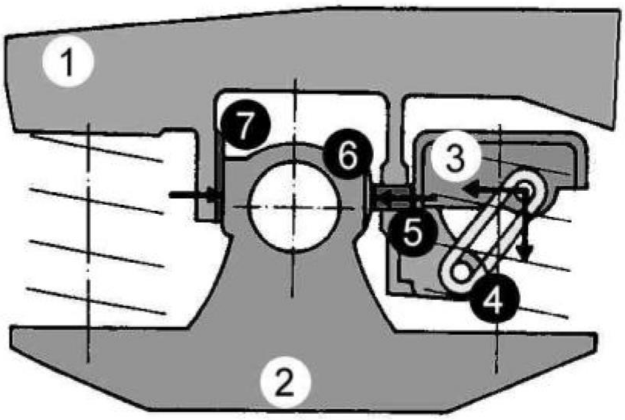



**Fig. 24: Structure of the primary restraint for the Y25 bogie [51]**

The primary restraint incorporates load-dependent friction damping. This is provided by a Lenoir damper. One of the two spring assemblies is supported on a spring plate (3), which is connected to the bogie frame via a link. As a result, part of the spring force is transmitted via the rod in a longitudinal direction to the axle bearing housing and generates a normal force on both the friction plate (6) and the sliding surface (7). If a relative movement occurs between the axle bearing housings and the bogie frame, this generates a friction force that dampens the movement in the vertical direction.

The bogies and the wagon body are connected by a pivot pin and two lateral sliders. These support the wagon body [48]. The pivot pin permits rotational movement, while simultaneously blocking translational movements, and transmits driving and braking forces between the bogie and the wagon body.

The model of the wagon body is connected to the model of the two bogies via the pivot pin. The structure is shown schematically in Fig. 25:. The model consists of two separate bodies on which the mass of the wagon body – including the load if necessary – is evenly distributed. A one-dimensional rotational stiffness acts between the two bodies around the longitudinal axis of the wagon and is used to create a simplified model of the torsional stiffness of the wagon body. This depends on the type of wagon, for example, intermodal wagons or flat wagons are torsionally quite soft, while covered freight wagons or tank wagons have greater torsional stiffness. This modelling approach for the consideration of torsional stiffness is also used in ERRI report B 12 RP 59 and can generally be considered adequate for this problem [10].

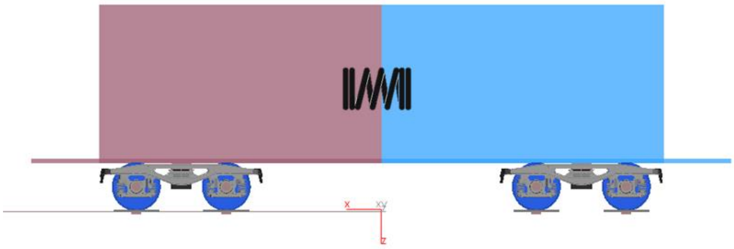


**Fig. 25: Wagon body model with one-dimensional torsional stiffness**

The parameters of the wagon that are potentially important for the investigation of tolerable longitudinal compressive forces, such as the pivot pin spacing or the torsional stiffness of the wagon body, can be varied in the model using automatic simulation controls based on the scenarios to be simulated.

### 5.1.3 Model of the frame wagons/locomotive

As mentioned at the beginning of this section, simplified models of the frame wagons should be used to reduce the computing time – simplified modelling of the bogies and the connection between the bogies and the wagon body. The bogies are modelled as a rigid bodies, which are guided in the middle of the track at a constant height. Consequently, the lateral movement of the wheelsets in the track channel is not modelled. In contrast to the modelling approach used in ERRI report B 12 RP 59, however, the model does consider the stiffnesses of the primary and secondary stages in the longitudinal and lateral directions as well as the rotation of the wagon body around the vertical axis. The interaction between the wagon body and the bogie in the x/y plane is simulated by a simplified substitute model. The model structure is shown schematically in Fig. 26:.

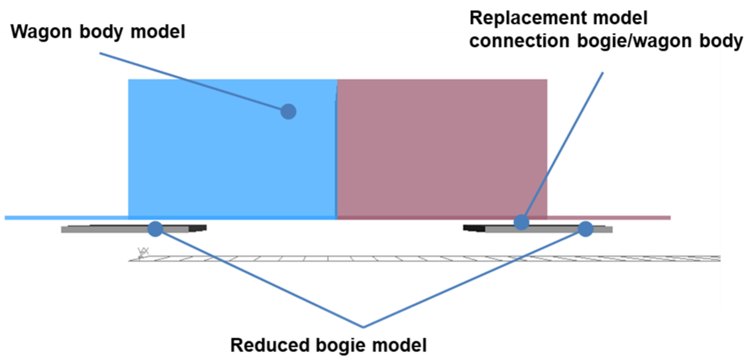



**Fig. 26: Schematic structure of the MBS model for the frame wagons**

The model of the locomotive is only responsible for moving the three-wagon train through the bend at a constant speed. Thus, the model consists of a single body that can move only along the axis of the track with one degree of freedom while maintaining a constant speed.

### 5.1.4 Model of the DAC with stabilisation joint

The model of the automatic centre buffer coupling with stabilisation joint comprises a total of seven bodies, which are shown in the overview in Fig. 27:. The model consists of the two coupling arms incl. coupling heads, the coupling pins, the pressure plates of the stabilisation joint and – on each side – a connection to the bodies of the two vehicles connected to the coupling. The main focus of the modelling process is the implementation of the transmission of tensile and compressive forces via the draft gear and the mode of action of the stabilisation joint. This is shown schematically in Fig. 28:.

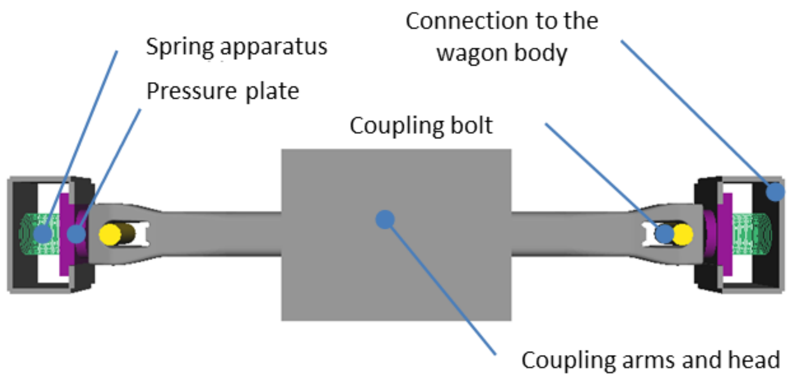



**Fig. 27: Overview of the MBS model for the AC with stabilisation joint**

In case of tensile forces, the contact between coupling arm and coupling pin is tightened and the coupling pin transmits the tensile force to the coupling housing. This causes the draft gear to deflect and, according to its physical properties, a force to build up which is transmitted to the wagon body via the hard stop of the pressure plate. Thus, in the tensile direction, there is no degree of freedom between the coupling housing and coupling pin, so they can both be considered as one body in the simulation.

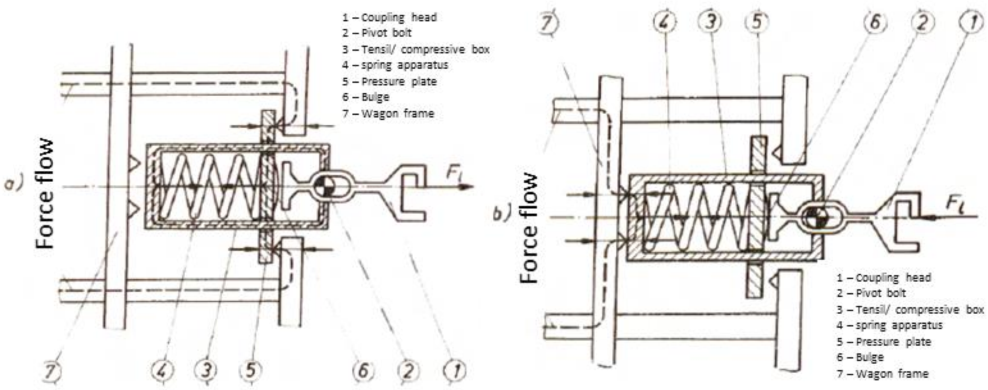



**Fig. 28: Flow of forces during a) tensile or b) compressive loading of an AC with stabilisation joint, modified presentation in accordance with [2]**

Under compressive loads, the contact between the coupling arm and coupling pin is slack. Instead, the pressure force is transmitted directly to the pressure plate via the contact surface on the front side of the coupling arm. The draft gear is compressed by the force acting on it. The resulting force acts first on the coupling housing, which stops in the x-direction and thus transmits the force to the wagon body. The draft gear can be installed in a pre-tensioned state. From the system topology, it can be seen that the draft gear itself is only ever loaded when compressed, both when transmitting tensile and compressive forces. Thus, the characteristic curve of the draft gear displays *hysteresis*, i.e. the force-travel characteristics of loading and unloading are not congruent. Since a quasi-static approach is used to determine the tolerable longitudinal compressive forces, the static characteristic curve of the draft gear can be used and changes in the characteristic curve due to dynamic effects can be neglected.

The stabilisation joint of the AC acts only in case of compressive loads. The stabilising effect comes into play in the event of lateral, vertical or angular displacement of the two wagons connected by the coupling. In particular, it serves to deflect or reduce the transverse forces – resulting from longitudinal compressive forces – transmitted through the coupling. The deflection of the transverse force is based on the principle that when compressive forces are transmitted, the force is transferred between the flat surface of the coupling arm and the convex contact surface on the pressure plate. The displacement of the contact point changes the load application line and thus the amount and direction of the transverse forces that occur.

The mode of action of the stabilisation joint can be divided into three or four phases, depending on the offset between the adjacent wagons. This is illustrated in Fig. 29: using the example of an effective longitudinal compressive force of 750 kN.

a) In the neutral position (1) there is no lateral deflection of the coupling, the force is transmitted centrally between the two adjacent wagons via the draft gear and coupling arm and there is no transverse force acting between the wagons.
b) If a lateral offset occurs between the wagons, the joint is in the stable position (2). By shifting the contact point, the joint deflects the transverse force so that it counteracts the misalignment. This can be seen in Fig. 29: from the fact that a transverse force with a negative sign occurs in the event of a positive deflection in the transverse direction.
c) In case of a larger transverse offset, the joint reaches the indifferent position (3). The resulting line of action of the longitudinal compressive force between the two contact points is parallel to the longitudinal axes of the vehicle, so that the transverse force disappears to 0.
d) If the offset in the transverse direction goes beyond the indifferent position, the joint can no longer stabilise the transverse displacement between the wagons.

The offset and transverse force now act in the same direction (unstable range (4)). In this range, however, the stabilisation joint ensures that while the amount of lateral force is no longer deflected, compared to a design without a stabilisation joint, it is at least significantly reduced.

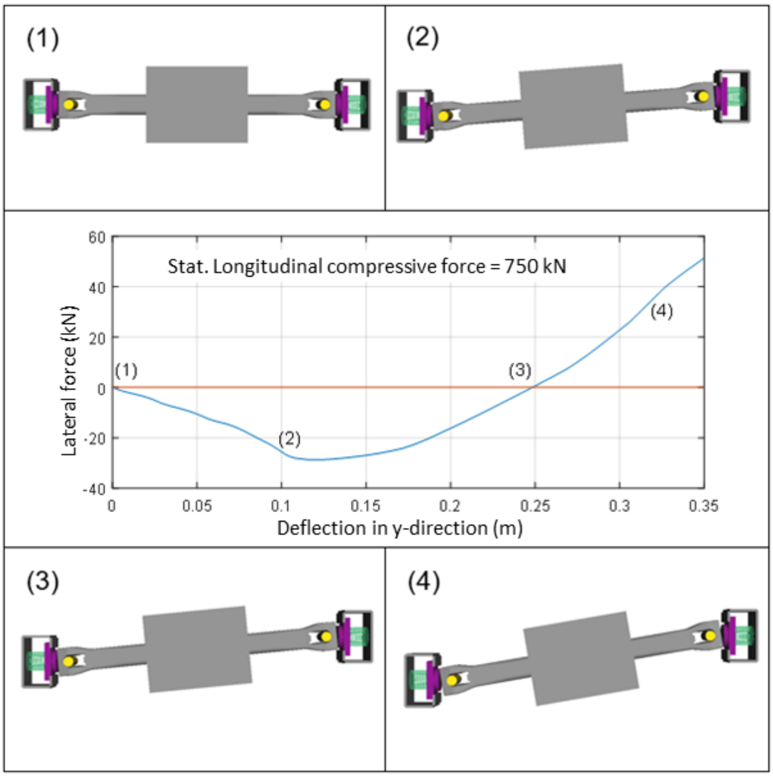



**Fig. 29: Mode of action of the stabilisation joint depending on the transverse offset between two wagons**

### 5.1.5 Model of the DAC without a stabilisation joint

The structure of an AC without a stabilisation joint (see Fig. 30) differs very clearly from the structure of an AC with a stabilisation joint. The design consists of a coupling rod that is attached to the wagon body using a hinge joint, enabling it to move along its longitudinal axis relative to the wagon body. In this design, the springs which act on the coupling when the coupling is loaded with tensile or compressive forces are separated from each other. The springs themselves are only loaded with compressive forces, as in the AC with a stabilisation joint. In the same way, the springs have a characteristic hysteresis curve and are therefore modelled separately according to the tensile and compressive directions, similar to the procedure described above. The corresponding force elements act between the coupling rod and the connection to the wagon body. The springs can be installed in a pre-tensioned state, as in the AC with a stabilisation joint.

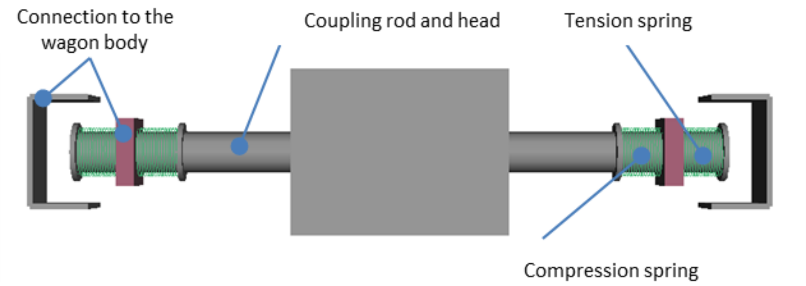



**Fig. 30 Design of the coupling model without a stabilisation joint**

In contrast to the coupling with a stabilisation joint, the centre self-align function, i.e. the generation of a torque that counteracts the misalignment between two wagons, is not a pure design parameter in the AC without a stabilisation joint, but a system characteristic. Due to the design of the draft gear, a restoring torque between the coupling arm and the wagon body is generated when the coupling arm is deflected in the rotational direction. This is based on the bending stress of the spring assemblies. The draft gear and the connection to the wagon body are rotationally symmetrical, thus the centre self-align function has the same effect on deflections in all spatial directions. The self-aligning torque depends on the acting longitudinal compressive force or tensile force. Due to the mirror-symmetry of the springs only the amount of force and the angle of deflection has an influence.

## 5.2 Longitudinal dynamics

### 5.2.1 Overview

The simulation model for the longitudinal dynamics consists of two parts, which run together in a *co-simulation*. The first part is the model of the train, which includes the various wagons, locomotive and couplings. This model also reproduces the routing and the running resistances of the individual vehicles. The model was created using the MBS program SIMPACK. The second model, which is linked to the train model, is the pneumatic brake model for freight trains. This behaviour was modelled in MATLAB using mathematical relationships. The overall model was validated using *TrainDy* software, which is considered to be the state of the art in the field of longitudinal dynamics for freight trains [5].

### 5.2.2 Model of the pneumatic brake

The model of the pneumatic brake takes into account the controlling function of the driver's brake valve from the locomotive, the air flow in the main brake pipe, the behaviour of the control valve and the brake cylinder as well as the brake mechanism on each individual wagon. The modelling approach is fundamentally based on the considerations of Witt [20]. The model can reproduce the function of both the block brakes and disc brakes.

The behaviour of the air flow within the main brake pipe (MBP) is modelled as a one-dimensional, isothermal, frictional flow of a compressible fluid, which is characterised by its continuity and momentum equation [20]:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial x} = F_1(\dot{m_f}) \qquad 1$$

$$\frac{\partial \rho u}{\partial t} + \frac{\partial (\rho u^2 + c_{air}^2 \cdot \rho)}{\partial x} = F_2(\rho, u, \dot{m_f}) \qquad 2$$

Here, ρ is the density, *u* is the flow velocity, $m_f$ is the mass flow of the fluid and $c_{air}$ the speed of sound. The variable $F_1$ describes density changes within the fluid due to inflows and outflows, e.g. at the driver's brake valve, the service brake accelerators and the control valves. The term $F_2$ represents momentum changes in the flow, caused in part by the above-mentioned sources and sinks as well as by the friction of the air on the surface of the MBP [20].

The function of the control valve, which converts the pressure in the MBP into the corresponding pressure in the brake cylinder via several valves, is implemented in the model using an equivalent circuit with the same function. The approach is essentially based on work by Witt and Cantone et al [6, 20, 52]. The build-up of pressure in the brake cylinder takes place in three phases, which are illustrated by the diagram shown in Fig. 31:. Phase I during filling, also known in the literature as the *application stroke*, takes place until the brake pads are applied to the wheel or the brake shoes to the brake discs, at which point the pressure $p_{AS}$ is achieved. In Phase II, known as the *in-shot,* the brake cylinder is further filled as quickly as possible within a specified time $t_{MD}$ until a minimum pressure $p_{MD}$ is reached. In Phase III, the brake cylinder pressure increases until the maximum pressure is achieved in accordance with the brake demand by the MBP. Depending on the inherent time lag of the control valve and the pressure drop in the MBP, the brake cylinder pressure build-up in Phase III is limited either by a limit curve or a transfer function. This subject is dealt with in greater detail in the dissertation by JOBSTFINKE [5]. The described behaviour is reproduced in the model using a function, which is presented in Fig. 31: based on the variables defined for this purpose.

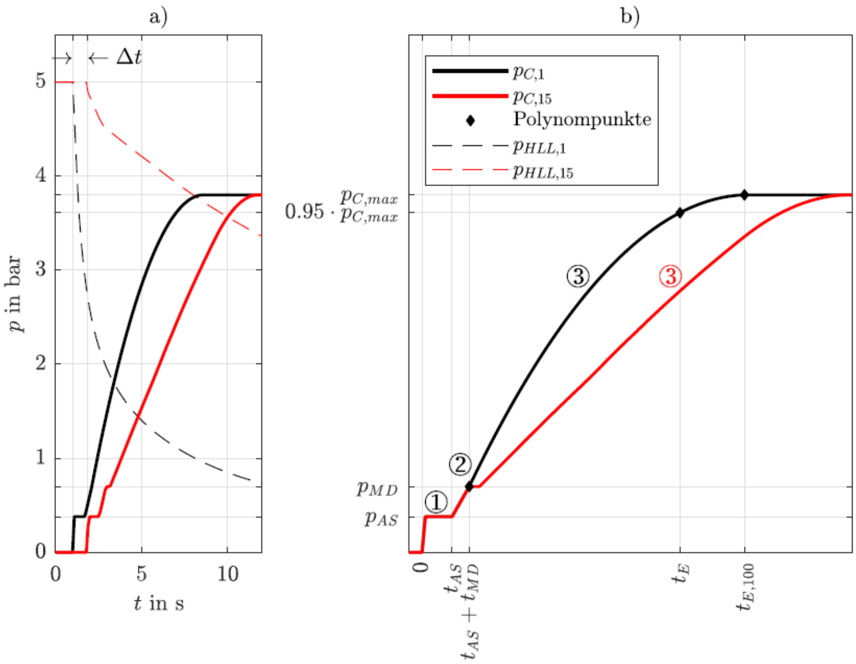



**Fig. 31: Example of pressure build-up in the brake cylinders at the first and 15th wagons of a 750 m train over the absolute time (a) and the modelling parameters (b) [5]**

The model also includes the function for the automatic or manual load change. This valve changes the maximum pressure that can be achieved in the brake cylinder depending on the load state. In case of a manual load change, it is possible to switch between two different values – 1.3 bar or 3.8 bar. In reality, the operator switches manually between the two levels when the total mass of a wagon rises above or falls below a specified level. The corresponding adjustment is also made in the model. During an automatic load change, the maximum brake cylinder pressure is controlled proportionally up to a certain total mass and then remains constant at the maximum brake cylinder pressure level [5, 6, 20, 52].

The final step models the transmission of the braking force from the brake cylinder via the brake mechanism to the wheelset. The model takes account of the lever ratios within the brake linkage as well as the corresponding efficiency in the transmission ratio and the behaviour of the brake linkage adjuster. This enables the contact pressure of the block against the wheel or the brake shoes against the brake disc to be calculated. The resulting braking force at the wheel depends on both the contact pressure $F_A$ itself and the friction speed $v$. According to Karwazki, this relationship can be modelled using the following formula (3) [53]. The value $\mu$ corresponds to the effective friction force and thus indicates the relationship between contact pressure $F_A$ and effective braking force. The coefficients $k_1$ to $k_5$ are dependent on the friction material used (grey cast iron, composite brake pad, LL-pad, disc brake etc.). The calculated braking force acting on the wheelset is the interface to the MBS model of the train.

### 5.2.3 MBS model

The MBS model for investigating the longitudinal dynamics of the train comprises the sub-models for the locomotive, the wagons and the couplings and has one degree of freedom along the track coordinate. The structure is shown schematically in Fig. 32:.

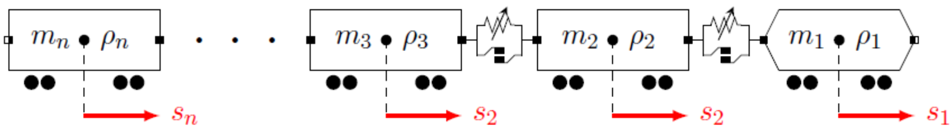



**Fig. 32: Schematic representation of the overall model of the train [5]**

The wagon model consists of a body with a certain total mass $m_{ges}$, whereby the rotational mass ratios resulting from the movement of the wheelsets are taken into account via the *mass factor* [54]. Various external forces act on the wagon: the braking force $F_B$, calculated using the pneumatic brake model, the running resistance $F_R$ and the air resistance $F_L$ (see Fig. 33: a). The two resistance forces differ depending on the type of freight wagon and the load state.

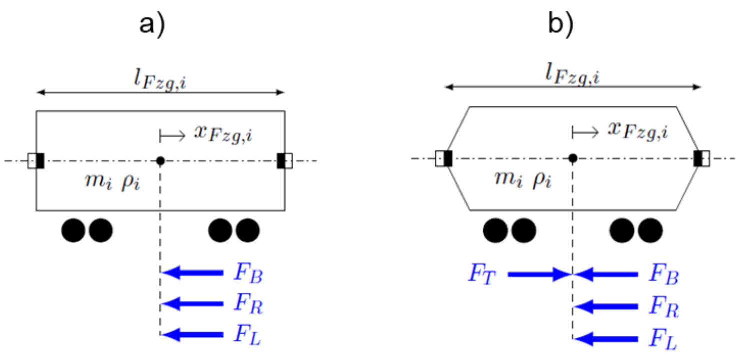



**Fig. 33: External forces acting on the models for a) wagons and b) locomotive, modified presentation in accordance with [5]**

The model of the locomotive (see Fig. 33: b) is basically similar in design, although the two models differ in terms of the mass factor and the resistance forces. However, the tractive force $F_Z$ and the electrodynamic braking force $F_{ED}$ of the locomotive are also taken into account. These variables are determined based on the total power, the available wheel-rail adhesion and other conditions, such as the limitation of the electrodynamic braking force in locomotives.

The model of the couplings connects the locomotive and the first wagon as well as the other wagons with each other. The structure is shown schematically in Fig. 34: , the value $l_{ZSE}$ represents the nominal length of the coupling. The model consists of two force elements (FE), each acting in the direction of compression or tension in which the behaviour of the spring apparatus is modelled. These are hysteresis models with different characteristic curves that depend on the direction of the stroke. These curves take account of the energy absorbed by the draft gear.

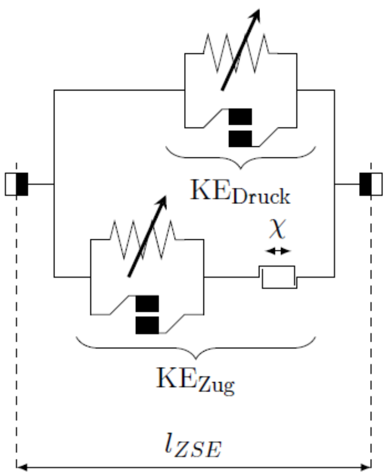



**Fig. 34: Schematic representation of the DAC model for the investigation of longitudinal dynamics [5]**

Due to the comparatively low-frequency processes in the longitudinal dynamics of trains, it is not necessary to model the dynamic characteristic curve. Only the respective static characteristic curve can be implemented [6, 20, 55]. The progression of the characteristic curve is read in from an external source and can correspond both to real characteristic curves for specific couplings and realistic but synthetically-generated progressions. At the transition between tensile and compressive stress or vice versa, the model takes account of the design-related coupling slack $\chi$ .

Many of the parameters, both for the pneumatic brake model and the MBS train model, are fundamentally variable and can be automatically varied and adapted to a particular scenario by a simulation control system. This is the basis for automatic simulation control (see Chapter 4.5).

## 5.3 Shunting impacts

The MBS model for the investigation of shunting impacts is primarily based on the analyses of shunting impacts with different types of freight wagons and loads presented in Chapter 3.4. As in the investigation of the longitudinal dynamics, the investigation of shunting impacts uses a one-dimensional MBS model with only one degree of freedom along the track axis.

In accordance with the statement in UIC leaflet 524 that the force in the couplings involved in the impact is independent of the number of wagons involved in the impact, the tests are carried out with a model consisting of two wagons – one impacting and one impacted – and the corresponding couplings between them, with one exception [1]. In contrast to the investigation of the longitudinal dynamics, various dynamic effects must be considered due to the significantly higher speeds involved in the processes. Specifically, these include the speed dependence of the characteristic curve for the draft gear, energy absorption by the wagon, attachments and the load itself, the elasticity of the load securing equipment and the friction of the load on the floor or walls of the loading space.

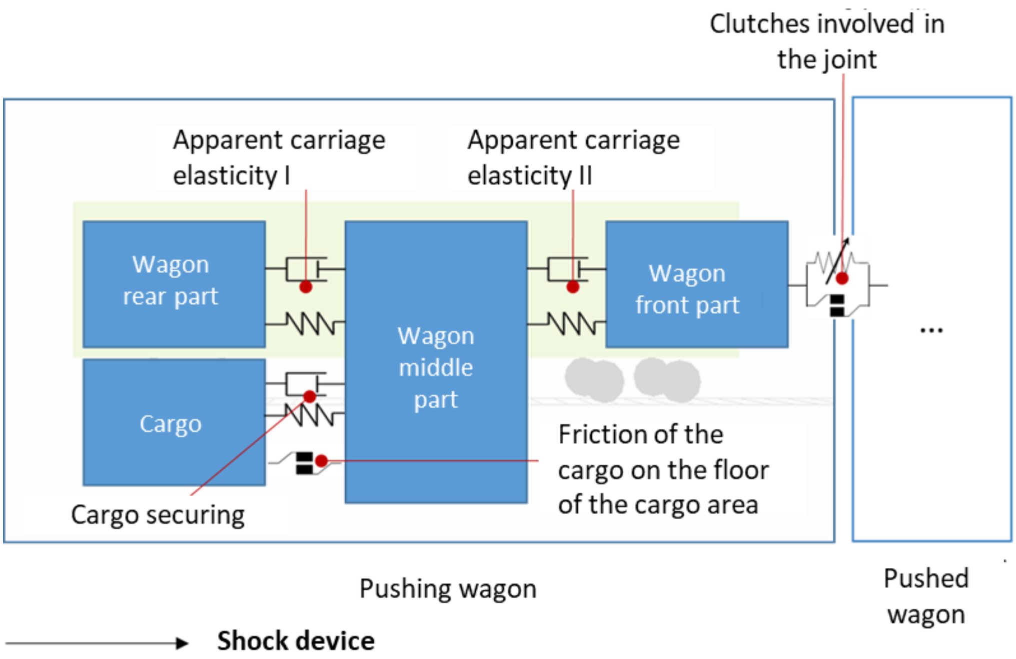



**Fig. 35: Overview of the MBS model for the investigation of shunting impacts**

### 5.3.1 MBS model of the wagons

The elasticity of the wagon structure and its attachments has a significant influence on the energy absorbed by the wagon during the impact. This behaviour is referred to as the apparent wagon elasticity, in accordance with ERRI report B 36/RP 27. It takes into account both the elastic behaviour and the energy absorption by the wagon construction during the impact [14]. For this purpose, the wagon model for the investigation of the shunting impacts is divided into three parts, with force elements representing the behaviour of the apparent wagon elasticity positioned between each of them.

To determine this apparent wagon elasticity for different cases (wagon type A runs into wagon type B or C with load D or E etc.), two approaches are investigated. First, this behaviour is reproduced in the MBS model using a physical substitute model in which the spring and damping elements are implemented with suitably adjusted parameters. The structure of this model is quite simple since the wagon elasticity can be modelled without adjustments for different wagon masses and impact speeds. This modelling approach is used to simulate typical cases from the ERRI report B 36/RP 25. The results are broken down in Fig. 36:. The three columns show different cases that were investigated. First, impacts between two Rs wagons were simulated, with the masses being constant in each case and the speeds varied in five steps. The second step considered impacts with a different type of wagon (Eas instead of Rs) and variable vehicle mass, but with the same buffer system. In the third step, impacts between two Eas wagons with different masses and impact speeds were again investigated, but this time with a different buffer system. The results of the simulations are compared with the corresponding measurement results from the ERRI report B 36/RP25. The comparison shows that there is good agreement between the measurement and simulation for the first two cases with buffer system A. In the third case, however, there are clear outliers. Here, the simulation supplies forces that are clearly too high in some cases and too low in others, so that it is no longer possible to speak of a systematic error. Consequently, this physical substitute model cannot easily be adapted to the corresponding behaviour.

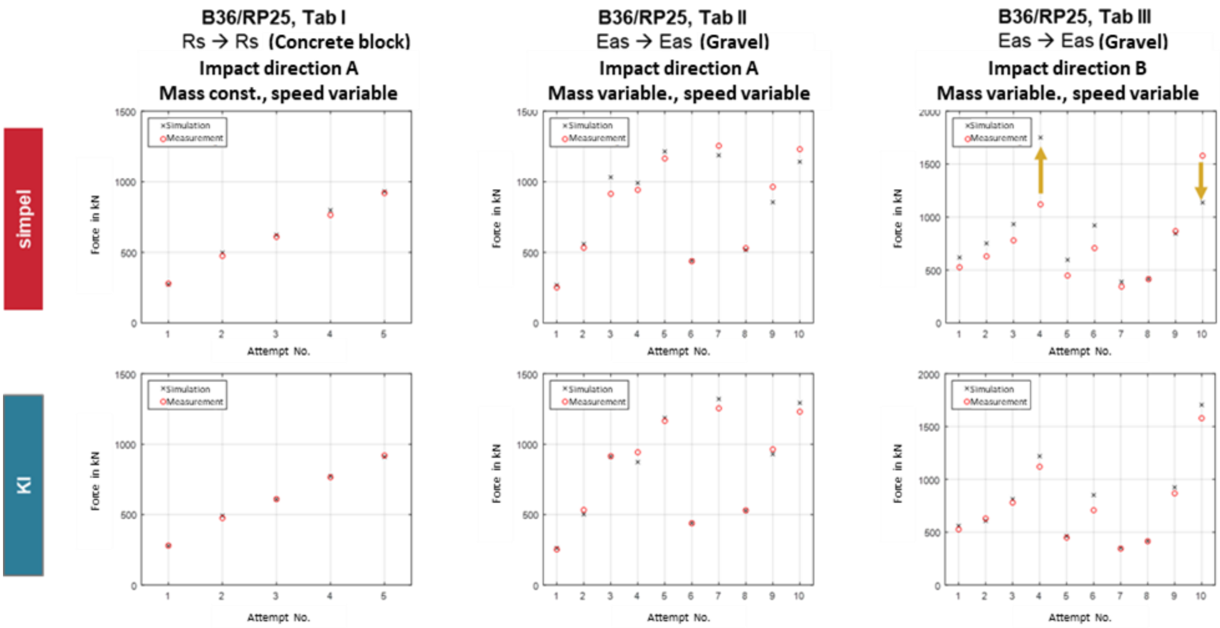



**Fig. 36: Comparison of methods for determining the apparent wagon elasticity using the simple approach (above) and the AI-based approach (below)**

In the second modelling approach, the behaviour of the apparent wagon elasticity is approximated using artificial intelligence (AI). The input variables are the corresponding wagon masses, impact speeds, wagon types etc. in accordance with the specifications of the tests conducted in the ERRI reports B36/RP25 and B36/RP27 [13, 14]. These results are also presented in Fig. 36:. It can be clearly seen that the model shows quite good agreement between the test and simulation results for all cases. It is

therefore used for further investigations. The physical reasons for behaviour of this model are not fully understood, so it cannot be transferred to other loads and wagon types with total confidence. In the available literature, however, test results are only available for very few cases, as explained in Chapter 3.4. These are used, e.g. to make extrapolations to other wagon types. The more results that are known from shunting impact tests for other types of goods, cargoes, and combinations of these and other factors, the greater the accuracy of the results that can be achieved with this method.
In addition to the apparent wagon elasticity, the model also uses force elements to take account of the stiffness of the load securing equipment and the friction of the load on the wagon floor or walls (see Fig. 35:).

### 5.3.2 MBS model of the coupling

Due to the dynamic effects described above, the model of the DAC must also be extended in comparison with the investigation of the longitudinal compressive forces, in order to map the dynamic behaviour of the draft gear. For the investigation of shunting impacts, it is not sufficient to know only the static characteristic curve of the coupling and implement this in the model. The challenge here is to be able to depict various possible characteristics of the DAC using the model. This must include differentiating between draft gear with elastomer and annular spring elements, which differ fundamentally in their dynamic behaviour. In addition, there are manufacturer-specific differences in the details due to the different materials, geometries, etc. used. As a result, a highly parameterised model must be used to permit the mapping of a wide range of different possible behaviours. The typical examples of characteristic curves shown below are based on data provided by the partners in this project. The respective behaviour was implemented in the model according to these specifications.
The dynamic behaviour for two different examples of draft gears with annular spring elements is shown in Fig. 37:. With this type of design, the static and dynamic characteristics differ from each other only slightly and are even congruent in some cases (annular spring 1). In the other case shown, the characteristic curves initially run almost congruently in the range of small deflections. For larger spring deflections, the dynamic characteristic curve is below the static characteristic curve but rises sharply shortly before reaching the stop. To reproduce this behaviour, general models from the relevant literature on modelling can be used.

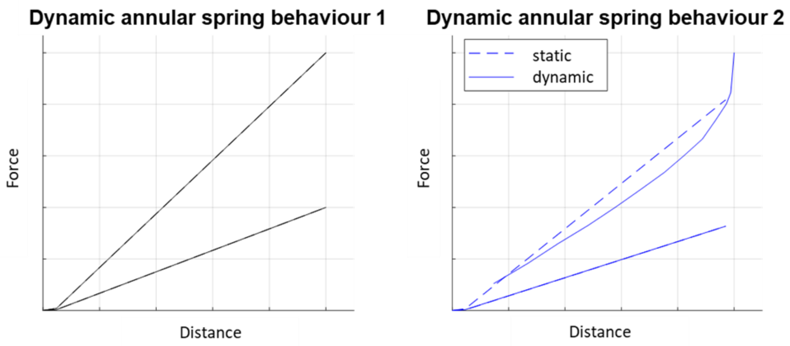



**Fig. 37: Dynamic behaviour of draft gears with annular spring elements**

In draft gears with elastomer elements, there are significantly greater differences between the static and dynamic behaviours due to material properties. Again, manufacturer-specific differences become apparent here. The characteristic curves for two typical examples of draft gears with elastomer springs are shown in Fig. 38:. In the range of very small deflections, the dynamic characteristic curve is below the static characteristic curve in Case 1, while in Case 2 it lies significantly above it. For larger deflections, both draft gears initially exhibit essentially similar behaviour, with the dynamic characteristic curve lying above the respective static curve. However, there are again clear differences in the range of high deflections near the stop. In Case 1, the dynamic characteristic curve achieves a lower end force than the static characteristic curve, while in Case 2 it is exactly the other way round with the dynamic characteristic curve continuously above the static characteristic curve.

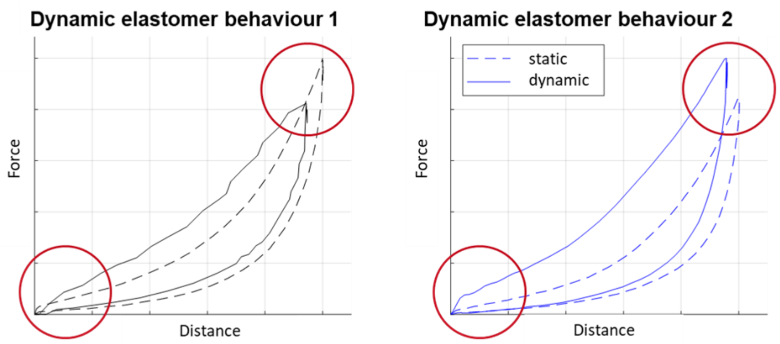



**Fig. 38: Dynamic behaviour of draft gears with elastomer elements**

In addition to the dynamic effects, the model of the coupling must take account of the coupling process itself. Initially, the impacting and impacted wagons are not connected to each other – coupling only occurs at the moment when the two coupling planes are one above the other.

# 6. Results

This chapter presents the results of the investigation based on the methodology and models presented. The analysis is based on the various sub-investigations. The objective is to find out how the parameters can be optimally selected for specific cases and in which direction an optimisation will have an effect.

## 6.1 Tolerable longitudinal compressive forces

The investigation of the tolerable longitudinal compressive forces focuses on several objectives: In relation to the core topic of this study, the aim is to examine to what extent dynamic parameters have an influence on the tolerable longitudinal compressive forces. In addition, the influence of two general design parameters of the DAC – the coupling arm length and the presence of a stabilisation joint – are examined more closely. A third important objective is to investigate the extent to which parameters of the test procedure influence the tolerable longitudinal compressive forces.

### 6.1.1 Sensitivity analysis

Within the framework of the sensitivity analysis, the most influential parameters should be identified first and their respective characteristics then analysed in a second step. A distinction is made between the coupling with and the coupling without a stabilisation joint. The results are presented for a typical pivot pin spacing of 8 m. The other wagon parameters are still considered variable.

The results of the sensitivity analysis for the DAC with a stabilisation joint are presented in Fig. 39:. The upper section of the chart first shows all parameters with their respective parameter influence – the dotted line marks the boundary between influential and non-influential parameters. The parameters are arranged in the order of their influence. The diagram also shows whether these parameters are characteristics of the wagon, the coupling or the test procedure for determining the tolerable longitudinal compressive forces. The lower section of the chart classifies the parameters – the parameters are first clustered into influential and non-influential parameters. In this case, the length of the coupling arm has the greatest influence on the tolerable longitudinal compressive forces. The coupling arm length is primarily a geometric parameter which influences the position of the vehicles and the angles between them. Thus, it also has a major influence on the tolerable longitudinal compressive forces. This value is regarded as variable in the context of this project, as it is essentially a variable design parameter of the DAC. In contrast to earlier observations (see Chapter 3.3), no specific value is defined here. The other parameters which have been found to have an influence on the tolerable longitudinal compressive forces relate to the applied test method. These are the underlying height difference between the frame wagons and the wagon being investigated as well as the respective pivot pin spacings of the frame wagons. Since these parameters have a major influence and the test procedure for the tolerable longitudinal compressive forces of freight wagons with a DAC is not yet covered by DIN EN 15839, these parameters must be precisely defined in a standardised test set-up that must still be developed [33]. The remaining parameters, which include the dynamic parameters that are the focus of this study, prove to have little influence here.

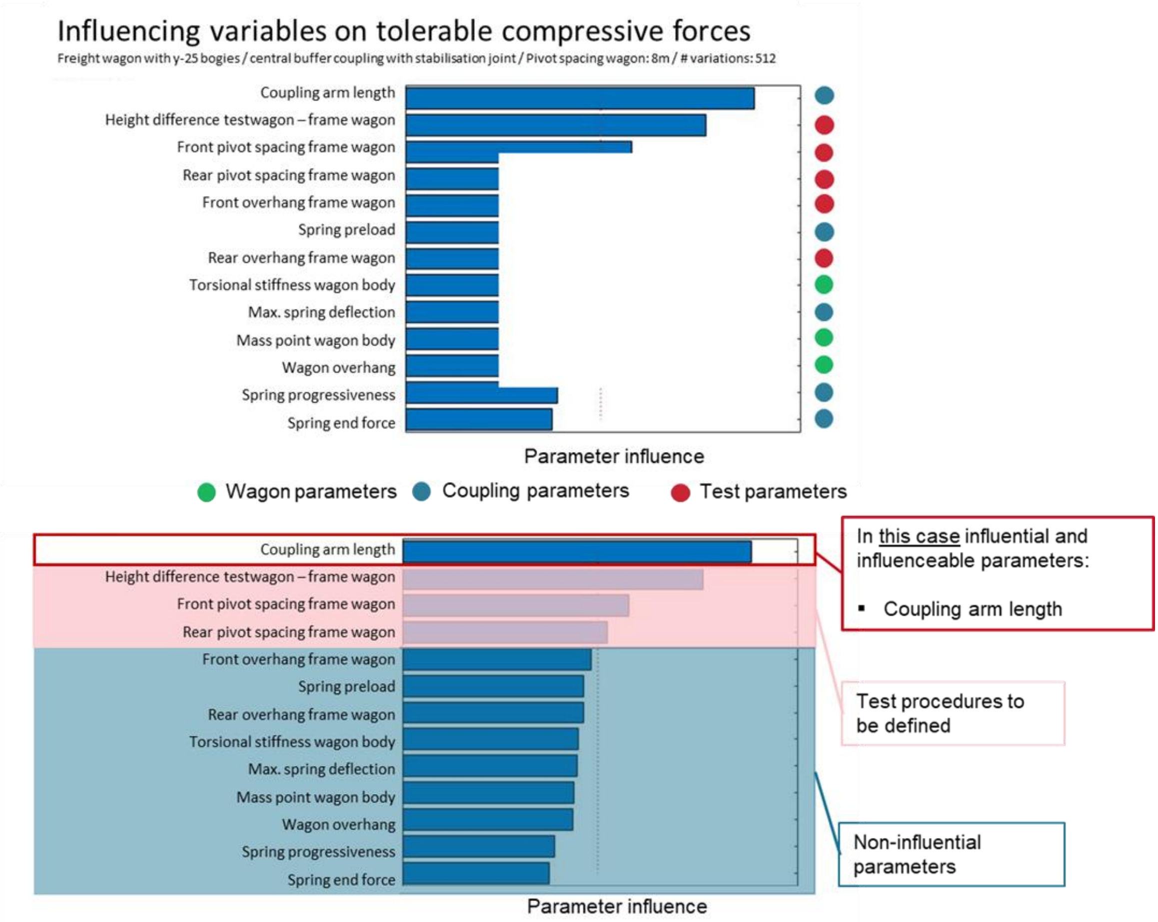



**Fig. 39: Identification of influencing variables on tolerable longitudinal compressive forces for the DAC with stabilisation joint**

The second step is to analyse which parameter values have an effect on the tolerable longitudinal compressive forces in a DAC with a stabilisation joint as well as the mode of action of these parameter specifications. Fig. 40: shows the sum frequencies[4] for various parameter values. The presentation is separated according to the three groups of parameters already stated above – both influential and less influential parameters are listed here. For each parameter, it is shown how the value of this parameter affects the tolerable longitudinal compressive forces. For clarity, the parameters are divided into five graded groups from low to high values. This means, for example, that the group with the lowest value contains all those cases where the value of the respective parameter falls within the bottom quintile of the value range. The opposite is true for the group with the highest value, i.e. this includes all cases where the value is in the top 20% of the value range. The groups in between are graded accordingly. The influential and less influential parameters are also easy to identify in the presentation of the sum frequencies: for the influential parameters, the graphs of the individual groups are well-spaced and highly fanned out, whereas for the less influential parameters the graphs lie close together and sometimes intersect several times.

[4] For more information on the presentation and interpretation of sum frequencies, see Annex A.

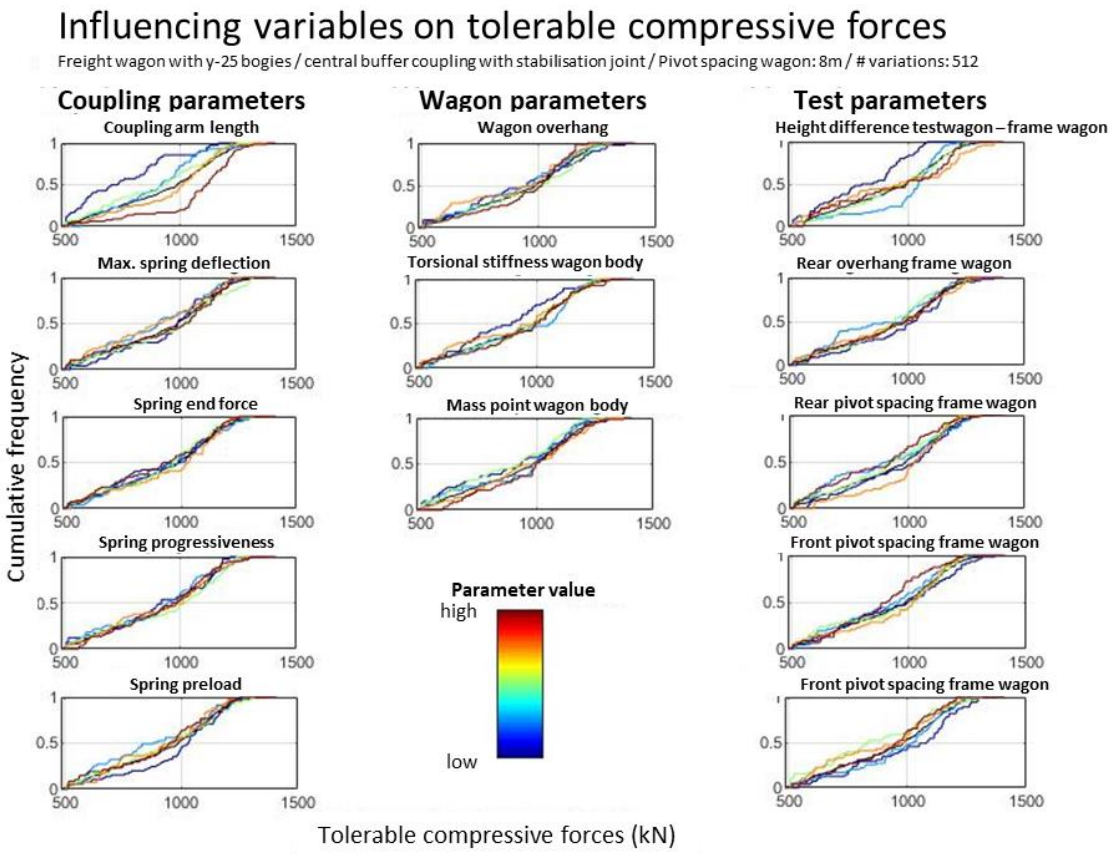



**Fig. 40: Parameter values of the tolerable longitudinal compressive forces for couplings with stabilisation joints**

Before considering specific parameters, it should be noted that the range of tolerable longitudinal compressive forces is very wide. At the lower end of the scale, there are several scenarios in which parameters have been unfavourably superimposed, resulting in tolerable longitudinal compressive forces that are below the 600 kN required by UIC 530 1/2. At the upper end of the scale, however, there are also cases that lead to tolerable longitudinal compressive forces of up to 1,400 kN.

Only the influential parameters will be discussed in this analysis of the parameter values. The results show that the longest possible coupling arm leads on average to significantly higher tolerable longitudinal compressive forces. In the above-mentioned test procedure parameters, the influence cannot be seen so clearly. This is probably due to the fact that the geometric parameters of the frame wagons cannot be viewed completely independently of each other and there are interactions between them. The influence of the height difference is not entirely clear either – this could be related to the non-linear behaviour of the stabilisation joint in the case of angular displacement.

The analysis is carried out in the same way for the coupling without a stabilisation joint. Selected results from the sensitivity analysis are listed in Fig. 41:. There are various similarities and differences between this and the previous analysis. In both cases, the length of the coupling arm plays a key role, but here the wagon overhang of the wagon under investigation is added as an influential wagon-specific parameter. As the tolerable longitudinal compressive force is a characteristic of the wagon itself, this parameter is considered variable. This only applies to freight wagons that are to be newly constructed and not to existing freight wagons, which of course cannot be changed in terms of their geometry. In addition, two parameters relating to the test procedure play an important role: the height difference between the tested wagon and the wagon conducting the test as well as the overhang of the rear frame wagon.

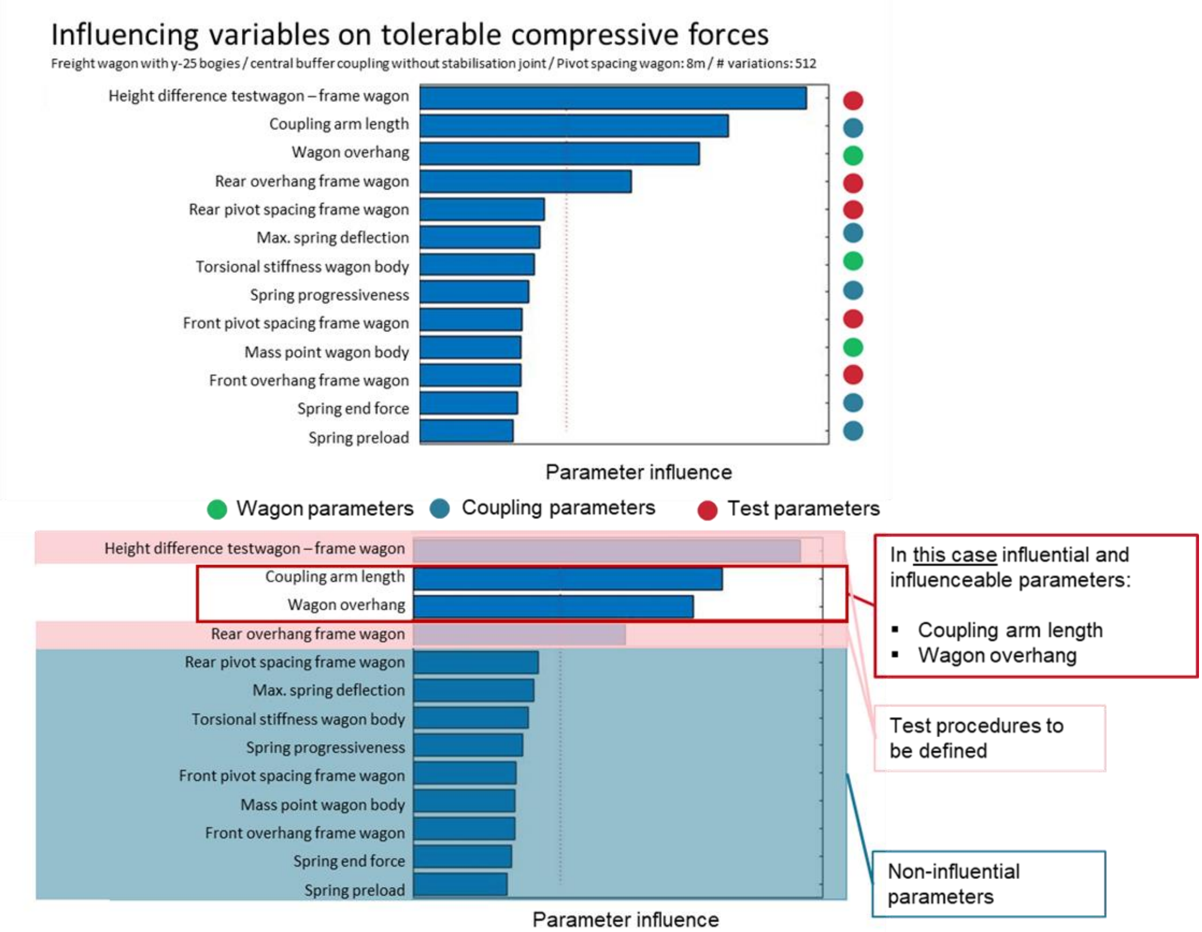



**Fig. 41: Identification of influencing variables on tolerable longitudinal compressive forces for the DAC without stabilisation joint**

When the influence of different parameter values on the tolerable longitudinal compressive forces are analysed (see Fig. 42:), it is immediately noticeable that the range of the results between the simulated scenarios is significantly lower. The maximum tolerable longitudinal compressive forces in the simulations for this case are approx. 800 kN. Furthermore, the influence of different characteristics can be seen much more clearly. This is probably due to the fact that the coupling without a stabilisation joint has a much more linear behaviour in the case of lateral, vertical and angular displacement between adjacent wagons. As seen in the previous analysis, a longer coupling arm leads on average to higher tolerable longitudinal compressive forces. The opposite is true for the wagon overhang, where a small value is advantageous for both the wagon under investigation and the frame wagons. The lower the difference in height between the two adjacent wagons, the higher the tolerable longitudinal compressive forces.

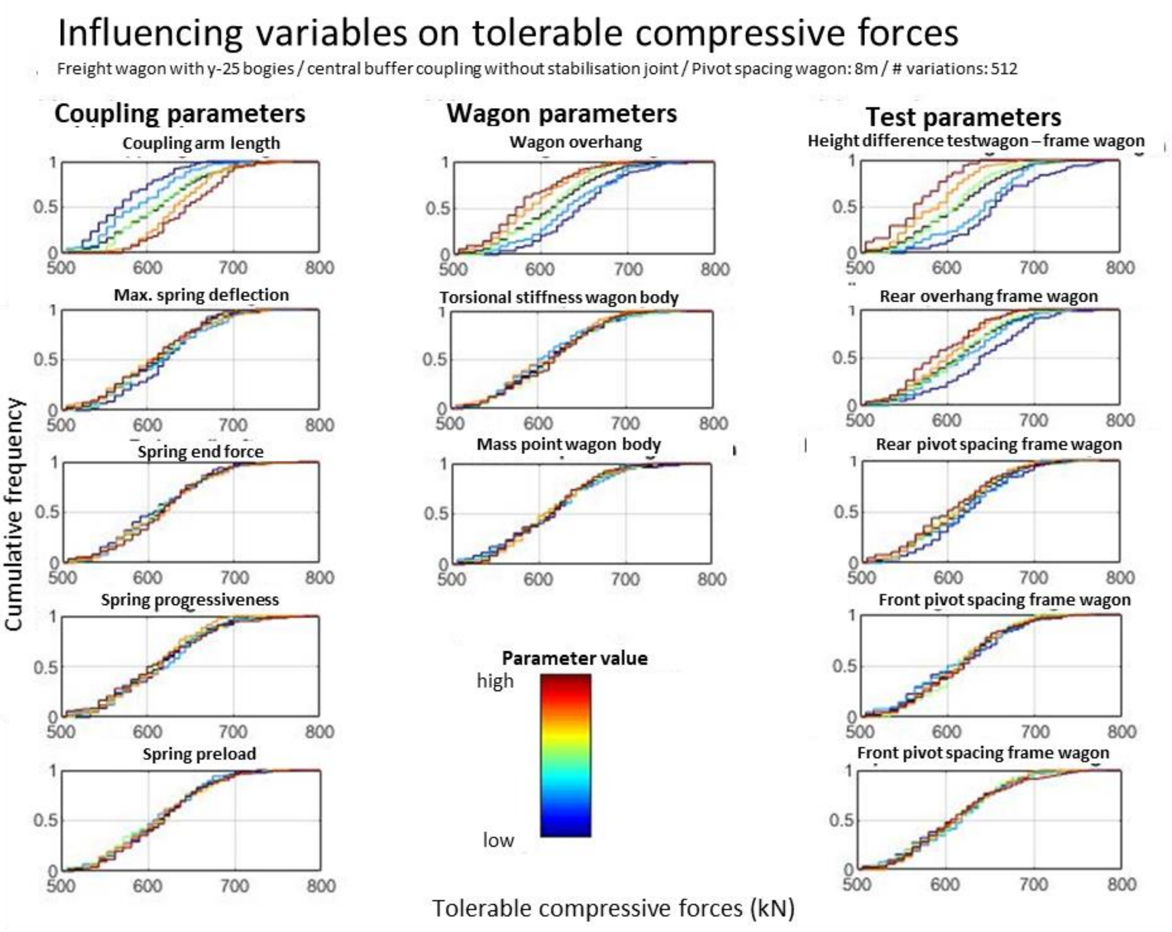



**Fig. 42: Parameter values of the tolerable longitudinal compressive forces f or couplings without stabilisation joints**

In the final step of this investigation, the tolerable longitudinal compressive forces between the coupling with and the coupling without a stabilisation joint are compared across all the considered scenarios. The results are presented in Fig. 43:. The comparison shows the effects described above – that a coupling with a stabilisation joint can, on average, withstand significantly higher longitudinal compressive forces. The maximum value occurring in the simulations is 1,400 kN, which is approx. 75% higher than 800 kN. Only approx. 12% of the scenarios for the coupling with a stabilisation joint lie below the minimum value of 600 kN required by UIC 530 1/2 , whereas for the coupling without a stabilisation joint this figure is approx. 40%. It is striking, however, that in both cases the minimum tolerable longitudinal compressive forces lie in a similar range of approx. 500 kN for some scenarios.

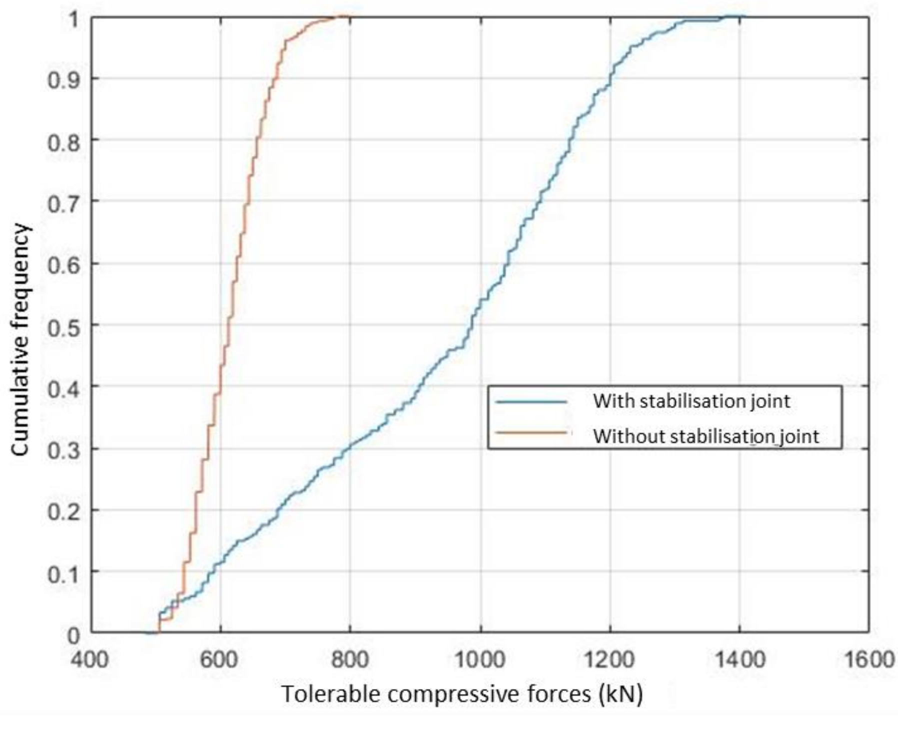



**Fig. 43: Comparison of the tolerable longitudinal compressive forces over all scenarios**

### 6.1.2 Interim conclusion

The investigation of the tolerable longitudinal compressive forces generally shows that the presence of a stabilisation joint has a major influence on the level of tolerable longitudinal compressive forces. On average, a stabilisation joint results in significantly higher tolerable longitudinal compressive forces, which is not surprising since this was the original reason for its introduction. Nevertheless, the simulations show that even with the inclusion of a stabilisation joint, there may be cases in which the tolerable longitudinal compressive forces fall below the value of 600 kN specified by UIC 530 1/2, i.e. wagons with an unfavourable combination of pivot pin spacing and wagon overhang.

In terms of the construction and design of the coupling, it can be seen that using the longest possible coupling arm has a positive effect on the tolerable longitudinal compressive forces.

When considering the coupling with and without a stabilisation joint, the dynamic parameters play only a minor role and are dominated by other influencing factors. Consequently, these parameters can be optimised with minimal consideration of the tolerable longitudinal compressive forces – the focus should be on longitudinal dynamics and shunting impacts.

The results show that the conditions under which the tolerable longitudinal compressive forces are determined have a major influence on the measurements obtained. In addition to the parameters shown here, this probably also applies to the geometry of the track on which the tests are conducted. No generally accepted test procedure has yet been defined for freight wagons with automatic couplings, since the associated standard DIN EN 15839 only refers to freight wagons with side buffers [33]. Moreover, it is by no means clear that the derailment criteria, which in practice represent the limit value for the tolerable longitudinal compressive forces, can simply be transferred from freight wagons with screw couplings and side buffers to freight wagons with DACs. The concept of tolerable longitudinal compressive forces is therefore somewhat vague and further research is required in this area for freight wagons with DACs.

A further qualification is also required: the results of this analysis primarily show tendencies, and specific values for individual scenarios are of secondary importance here. The analyses are based on simulations which cannot be compared with tests due to the lack of available references. In practice, the simulation uses laboratory conditions and this could have both negative and positive effects on the resulting tolerable longitudinal compressive forces. For detailed analyses, it would be desirable to be able to use reference results for comparison – based on a similar methodology, for a certain number of different combinations of wagon and coupling under various test conditions.

## 6.2 Longitudinal dynamics

### 6.2.1 Sensitivity analysis

Analogous to the consideration of the tolerable compressive forces, the objective of the sensitivity analysis for forces occurring due to longitudinal dynamics is firstly to identify the most influential parameters and secondly to determine favourable characteristics for these parameters. For various reasons, however, much greater differentiation is required. When considering forces caused by longitudinal dynamics, a distinction must first be made between the various basic scenarios, such as service braking, emergency braking or starting up. The different scenarios produce different longitudinal dynamic effects, so different factors have a different degree of influence depending on the scenario. In addition, a distinction must also be made relating to the lengths of the various basic trains: some parameters have a different effect on short trains than on long trains and vice versa. Similar considerations affect the different mass arrangements and mass distributions of the trains.

More generally, sensitivity analysis is also a question of the methodology used. As explained in Chapter 4.2, sensitivity analysis always refers to a specific output variable. When analysing longitudinal dynamics, however, there are various possibilities, e.g. the maximum value of the longitudinal compressive or longitudinal tensile forces, filtered or unfiltered, a corresponding mean value or a characteristic value for the dynamics occurring. Depending on the output variable on which the analysis is based, parameters can have varying degrees of influence.

JOBSTFINKE analysed this subject in detail in his dissertation [5]. This study uses his analysis and presents selected results in the following overviews.

**Operating scenario: Braking**

Fig. 44: shows the results of the sensitivity analysis for emergency braking of 750 m trains with the mass distribution 30/60/10 in a block train configuration with composite brake pads. The underlying output variable is the respective maximum longitudinal compressive force filtered over a period of 1s. The left section of the diagram shows the cumulative frequency of the maximum longitudinal compressive forces occurring over all simulations included in this scenario. This illustration shows that in approx. 40% of the simulated parameter combinations, the longitudinal compressive forces

occurring during emergency braking for this scenario are not greater than 250 kN. In slightly more than 90% of the simulations, the force is below 500 kN. For the remaining parameter combinations, maximum longitudinal compressive forces of up to 700 kN result.

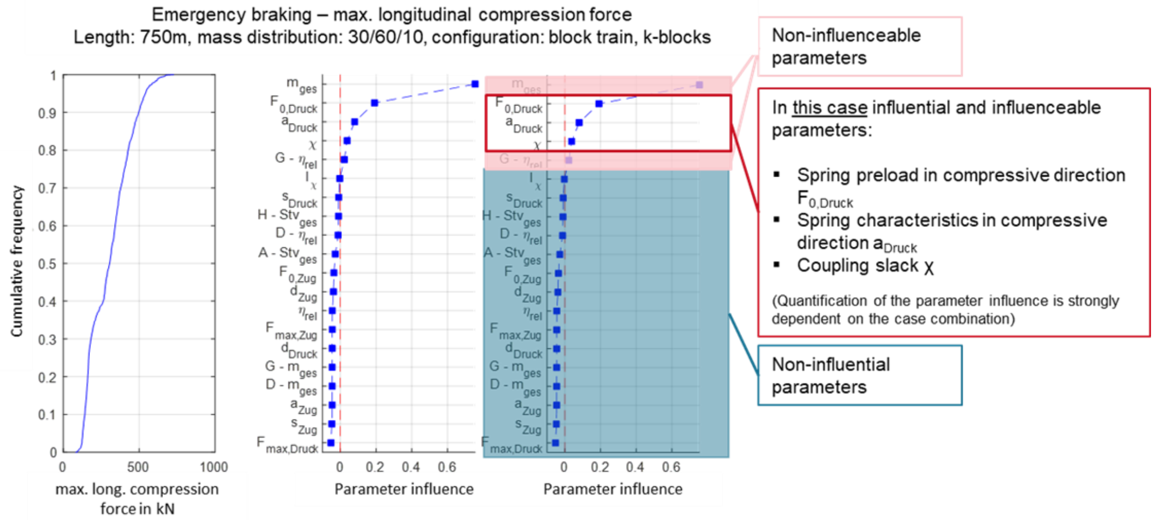



**Fig. 44: Identification of influential parameters for the longitudinal dynamics during emergency braking in a typical case**

The right section of the diagram shows the influence of the individual parameters. The parameters are graded according to their degree of influence and the red vertical line separates influential and non-influential parameters. By far the greatest influencing factor in this scenario is the total mass of the train, with higher masses tending to lead to higher forces. This parameter cannot be influenced, as the total mass cannot be limited to a specific value without operational restrictions. Another highly influential factor is the distribution of wagons with different brake linkage efficiency levels along the train. Again, this parameter cannot be influenced as the efficiency of the wagon brake rod is usually not known and depends, among other things, on the maintenance condition of the respective brake rod. For obvious reasons, this parameter cannot be taken into account when assembling trains, with the result that a wide variety of configurations can occur during normal operation.

However, the most influential parameters also include three parameters which affect the coupling design and can therefore be influenced: the spring preload and spring characteristics in the compression direction (degressive, linear or progressive characteristic) and the coupling slack. The effects of the respective values for these three parameters are examined below. For this purpose, the analysis is somewhat extended and considers the effects of the characteristics on both the maximum longitudinal compressive and tensile forces occurring for the underlying scenarios.

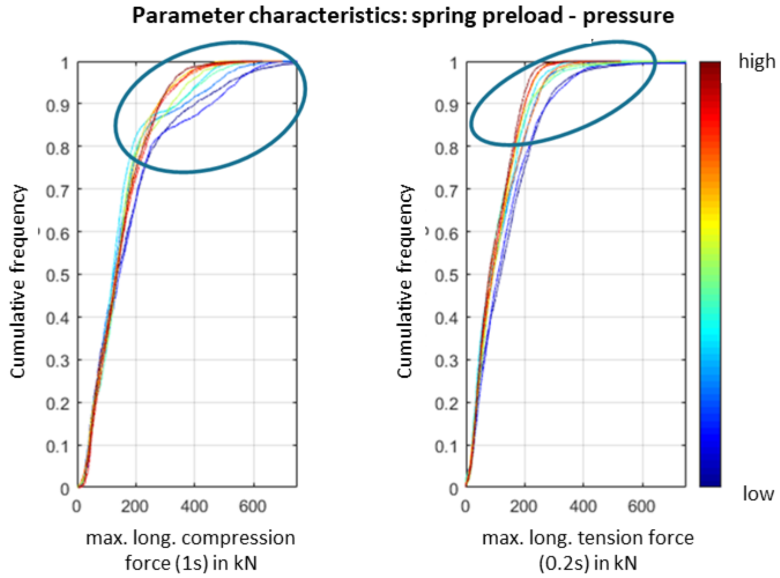



**Fig. 45: Parameter characteristics of the spring preload in compression direction during emergency braking**

Fig. 45: shows the parameter values of the spring preload in the compression direction over all simulated emergency and service braking operations across all scenarios. The influence of these parameters on the maximum longitudinal compressive forces is shown on the left and on the longitudinal tensile forces on the right. It can be clearly seen that the differences between different levels of preloading in the compression direction also remain low in scenarios where the overall force level is low. In the area of high forces, however, there are clear differences between the various specifications. Overall, using the highest possible preload in the compression direction has a positive effect on the forces in both the compressive and tensile directions.

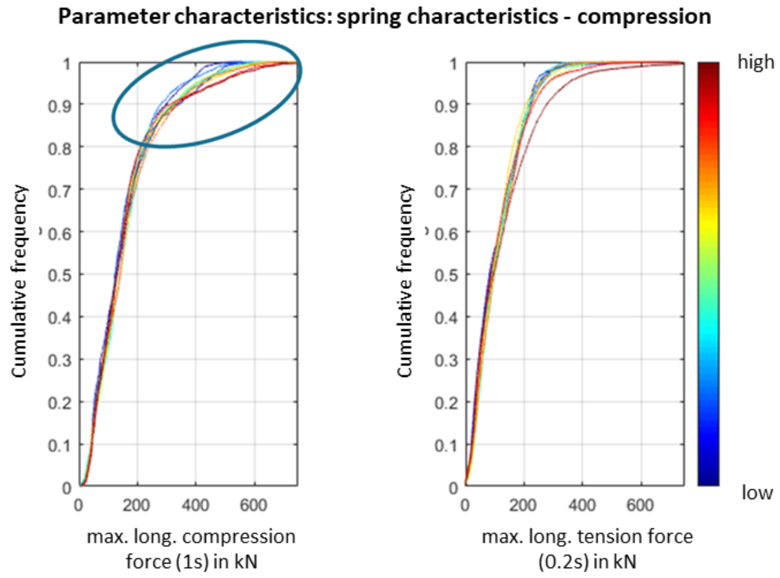



**Fig. 46: Parameter specifications of the spring characteristics in the compressive direction during emergency braking**

The parameter specifications for the spring characteristics in the compressive direction are listed in Fig. 46: . These parameter specifications are more difficult to interpret intuitively: a high value means a more progressive characteristic curve, low values stand for degressive to linear characteristic curves. As was the case for the spring preload, different specifications play only a minor role in cases where the overall force level is low. However, the influence is clear in the range of high forces. Here, it can be seen that low values lead to low forces and high values to high forces. Thus, linear or, in the best case, even degressive characteristic curves are generally advantageous in terms of both the longitudinal compressive and longitudinal tensile forces that occur.
In contrast to the other two influential parameters, which concern the design of the spring apparatus, the effect of the coupling slack is almost always independent of the prevailing force level in the scenarios. The influence of the parameter specifications is shown schematically in Fig. 47:. With the exception of very small and very large forces, where there are only slight differences between the various specifications, a low level of coupling slack consistently leads to lower longitudinal compressive and longitudinal tensile forces during braking.

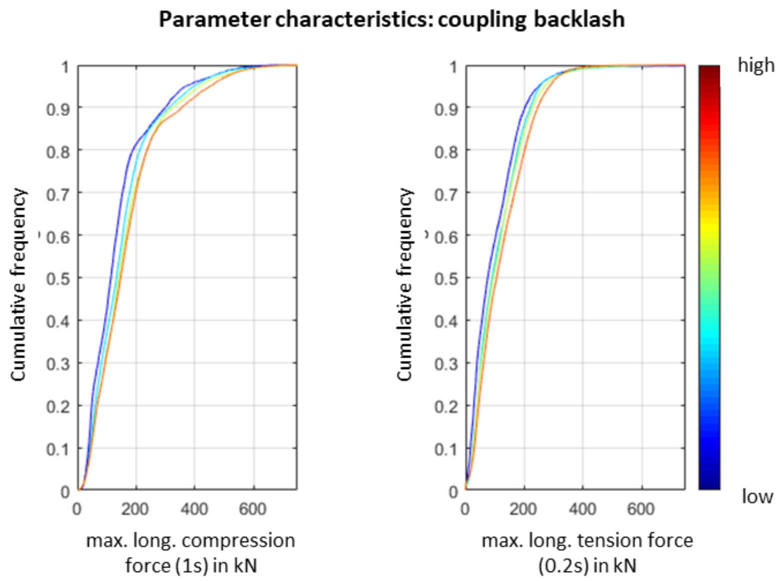



**Fig. 47: Parameter specifications of the coupling slack in the compressive direction during emergency braking**

### Operating scenario: Starting up

In the second step, the sensitivity analysis for the operating scenario “starting up” is performed. Fig. 48: presents the results using the same scheme as above for a basic train with a length of 500 m and a uniform mass distribution and arrangement. In contrast to the analysis of braking processes, a whole range of influential parameters must be considered here. These include the total mass, mass arrangement and differences in the masses of the wagons within the train. As explained above, these parameters must be considered beyond the designers’ control.

However, the most important effects are due to parameters that can be changed within the coupling design. These include the spring preload in both the compressive and tensile directions, damping in the tensile direction, maximum spring travel and the corresponding spring force in the tensile direction as well as the spring characteristics in the tensile direction and the coupling slack. It is thus reasonable to assume that intelligent selection of the design parameters can play a decisive role in reducing the forces that occur.

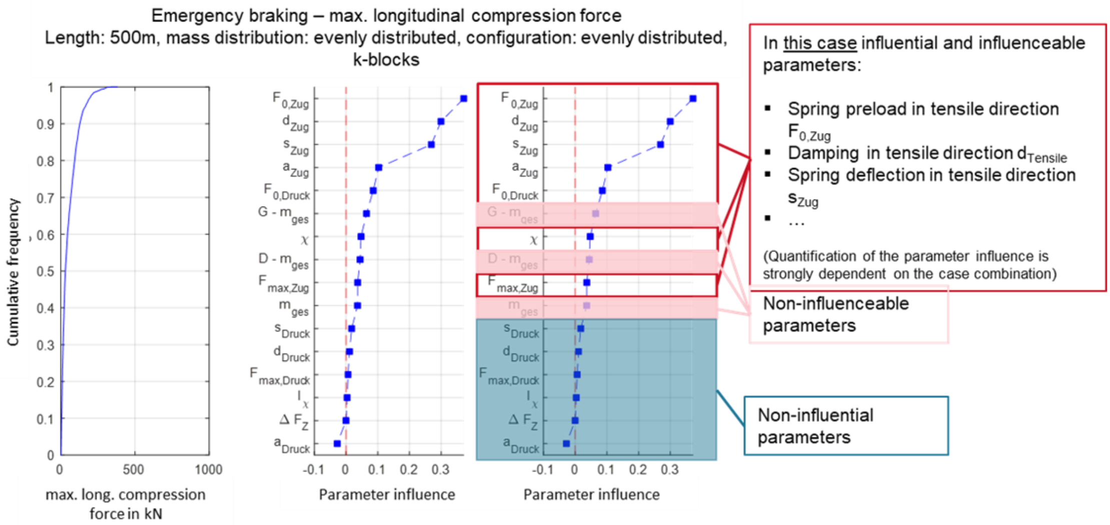



**Fig. 48: Identification of influential parameters for the longitudinal dynamics while starting up in a typical case**

The parameter specifications for the spring preload in the tensile direction for the scenario “starting up” are listed in Fig. 49:. Here it can be seen that a high preload reduces the forces in both directions. For longitudinal compressive forces, this is particularly evident in cases with an overall high force level; for longitudinal tensile forces, the specification of the preload in the tensile direction has a consistent effect on the maximum force that occurs.

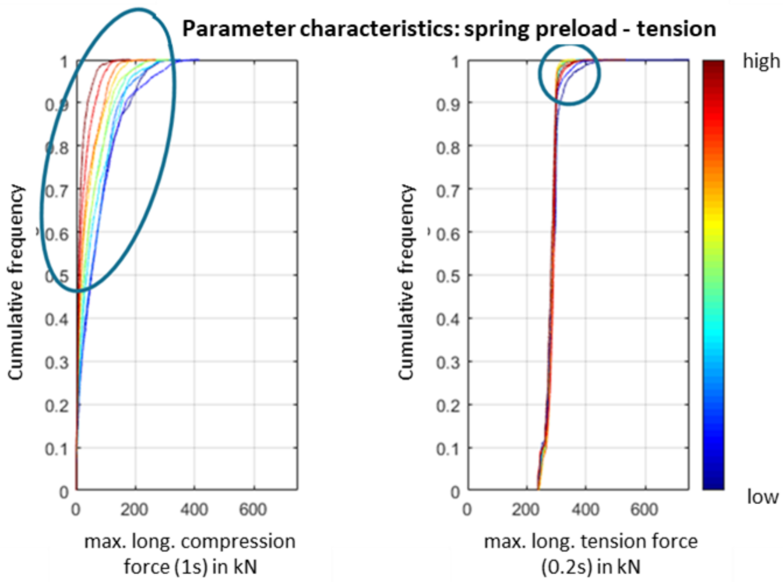



**Fig. 49: Parameter specifications of the spring preload in the tensile direction while starting up**

Damping in the tensile direction and maximum spring travel have a similarly strong influence on the longitudinal compressive forces that occur when starting up. With these two parameters, however, it is not possible to provide a general answer regarding their influence on the longitudinal forces that occur. Low damping in the tensile direction tends to have a positive influence on the longitudinal tensile forces at a high force level, but this is not observed for longitudinal compressive forces. Thus, it is impossible to make clear statements about advantageous specifications for this parameter. The influence of the spring deflection in the tensile direction has the reverse effect. Providing the longest possible spring deflection until the end force is reached has a positive effect on all the longitudinal compressive forces, while no clear statement can be made for the tensile forces based on these results. When selecting the spring characteristics in the tensile direction, the positive effects of a degressive or linear characteristic curve only apply in the compressive direction – as they do for the braking scenario. Another influencing factor – the end force in the compressive direction when the maximum spring stroke is reached – shows no clear tendency for either the maximum compressive or tensile forces.

### 6.2.2 Interim conclusion

With the aid of the sensitivity analysis, it was possible to sort the various parameters that were included in the investigation of the longitudinal dynamics according to their level of influence and to separate the influential from the non-influential ones. A distinction was made between the parameters that can be influenced by the DAC design and those that can be assumed to be variable during operation. The spread of these parameters means that the maximum forces to be expected during operation can vary greatly between different train configurations. This underlines, once again, that Monte Carlo methods are a useful tool for studying longitudinal dynamics. Based on the objectives of the study, the next step examined which specifications have a positive effect on the maximum longitudinal compressive and longitudinal tensile forces occurring for the parameters classified as influential and influenceable.

For the design of the draft gear, the results show that the following criteria should be applied for optimisation relating **exclusively** to the longitudinal dynamics, i.e. ultimately a reduction of the forces occurring:

- high spring preload
- degressive or linear characteristic curves
- short spring travel until the end force is reached
- low coupling slack.

For the longitudinal dynamics, it can therefore be concluded that, in general, a coupling behaviour that tends towards a rigid connection is positive. This optimisation is particularly effective with high force levels.

## 6.3 Shunting impacts

### 6.3.1 Sensitivity analysis

The sensitivity analysis procedure for identifying the most important parameters and specifications relating to shunting impacts is similar to the principle used for the longitudinal dynamics. Using GSA, the most influential parameters are first very generally identified and classified according to whether they can be influenced by a specific design. The effects of each parameter are then examined.

In this analysis, it is necessary to differentiate between the various impact speeds considered, because the influence of some parameters may be different at low impact speeds than at high impact speeds. A distinction is also made between the various mass distributions. The output variable under consideration – firstly, the maximum value of the compressive force and, secondly, the maximum acceleration of the wagon – may make a decisive difference. In the simulations, the maximum compressive force is capped at 2,300 kN, even though higher forces are possible in some of the scenarios examined. This is because serious damage to the wagon structure is to be expected, at the latest, when the wagon is subjected to forces of this level. Moreover, the MBS model is not able to adequately represent these forces in any case. In practice, these instances are also irrelevant because such an unfavourable choice of parameters must be avoided under all circumstances.

Fig. 50: shows the results of the sensitivity analysis for shunting impacts at a speed of 10 km/h and a mass distribution of 30/60/10 for the wagons running down the hump. The output variable being analysed here is the maximum compressive force. The diagram depicting the cumulative frequencies of the maximum compressive forces shows that forces of 500 kN or less occur in approx. 20% of cases and forces of ≥ 2,300 kN in approx. 12% of cases, i.e. there is quite a wide spread overall. The question here is to what extent this spread is caused by parameters that can be influenced by the spring design or whether they are factors that are beyond the designers’ control.

The right section of the diagram shows the individual parameters in ascending order of influence. In this case, the greatest influence on the maximum compressive forces during impact is the maximum spring travel until the end force is reached in the compressive direction. This is a design parameter for the draft gear. The end force itself is one of the most influential parameters and also subject to the manufacturer's design. Furthermore, the mass of the wagons involved in the impact plays a decisive role. Generally, it is not possible to influence the masses of the impacting or impacted wagons as these are dependent on the train configuration. The type of dynamic behaviour of the coupling (see 5.3.1) is another influential parameter which can be shaped during the design process. The implementation of the apparent wagon elasticity in the model (“WE type”) and the energy absorption due to the wagon design (“WE value”) are also shown to affect the results. Neither of these factors can be controlled. However, they must be considered uncertain in the context of this project due to the limited data available (see 5.3.1) and require further research.

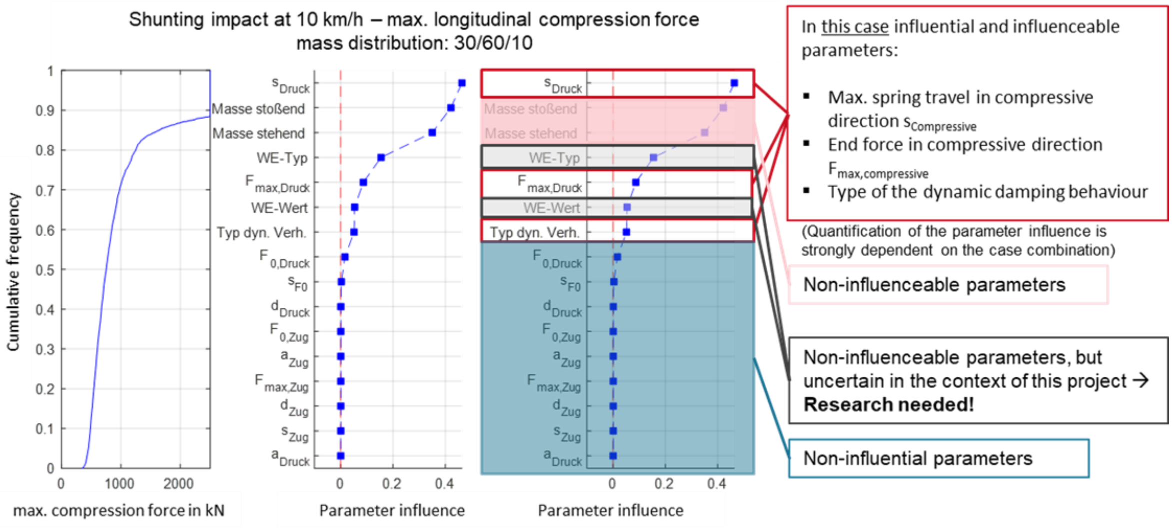



**Fig. 50: Identification of influential parameters for the maximum compressive force occurring in shunting impacts**

If the maximum wagon acceleration is considered instead of the maximum compressive force during impact, the structure of the most influential parameters shifts slightly (see Fig. 51:). The following example considers a scenario with an impact speed of 5 km/h with equally distributed masses. As in the previous example, the most important parameter for this scenario by far is the maximum spring travel in the compressive direction. This is followed by the spring preload in the compressive direction, the type of dynamic behaviour and the spring characteristics in the compressive direction. All these parameters affect the spring design. In addition, the wagon mass – in this case that of the stationary wagon – and the type of wagon elasticity play important roles, as was also the case above. The other parameters are considered to have little influence.

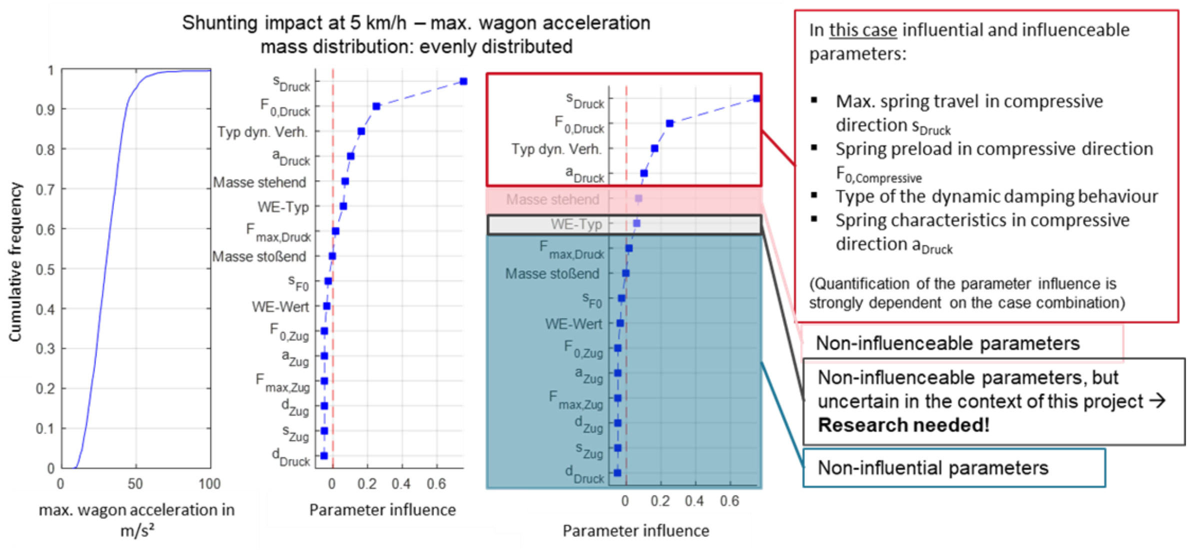



**Fig. 51: Identification of influential parameters for the maximum wagon acceleration occurring in shunting impacts**

The analyses presented as examples for the two cases show that the maximum compressive force and wagon acceleration are strongly dependent on the parameters used in the design of the draft gear, i.e. the force or acceleration level can probably be limited overall by careful selection of the parameters.

The next step now examines the effects of each parameter specification. The two output variables considered so far – maximum compressive force and wagon acceleration – are examined and the results compared.

Fig. 52: lists the parameter specifications for the maximum spring travel in the compressive direction, the most important parameter for both output variables. The longest possible spring travel is advantageous for both compressive forces and acceleration. Its influence can be seen very clearly in the wagon accelerations: while a value of less than 50 m/s² for a long spring travel occurs in more than 90% of the scenarios examined, this is only true for about 65% of the scenarios with a very short spring travel. Optimisation of this parameter for shunting impacts is effective throughout all scenarios but is particularly effective in cases with high force levels and high wagon acceleration.

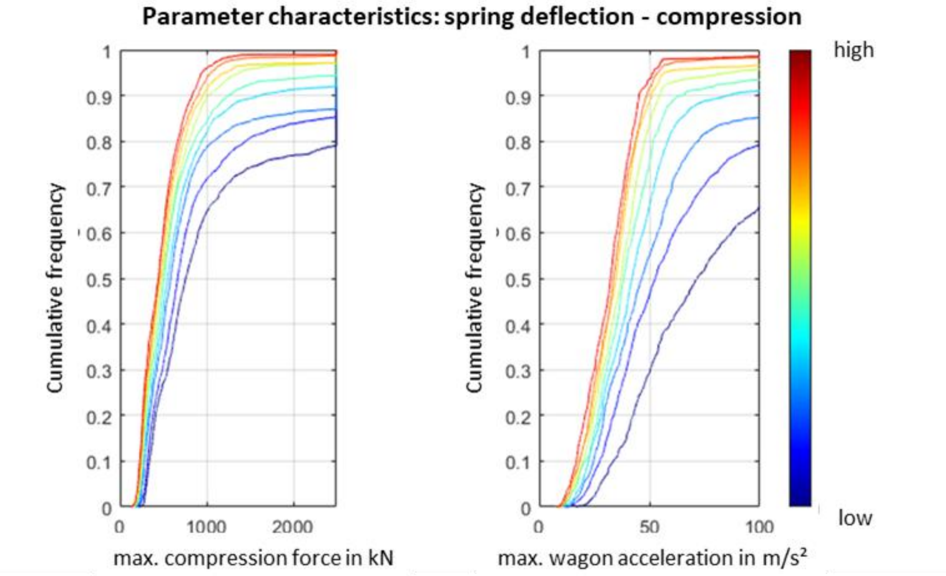



**Fig. 52: Parameter specifications for the maximum spring travel in the compressive direction in shunting impacts**

In the case of the spring preload – Fig. 53: shows the influence of various specifications for this in detail – an ambivalent behaviour can be observed that depends on the output variable being considered. For the maximum compressive force, a high preload tends to be advantageous, even if the differences are not large. For the maximum acceleration, it is necessary to differentiate between scenarios with accelerations of up to approx. 60 m/s² and scenarios with higher values. In scenarios with a low level of acceleration, a low preload is clearly advantageous. This advantage diminishes with increasing acceleration. In scenarios with extremely high accelerations, however, this effect is reversed, so that for the highest 20% of impacts, a high preload would again tend to be advantageous. Since it can be expected that the level of wagon acceleration will be limited by careful design of the coupling parameters, a low preload for shunting impacts can still be regarded as advantageous overall.

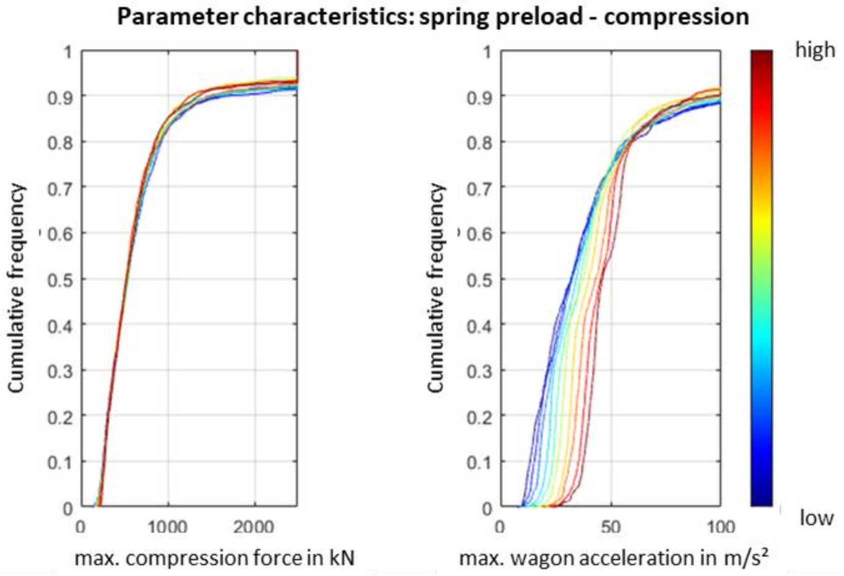



**Fig. 53: Parameter specifications for the spring preload in the compressive direction in shunting impacts**

Fig. 54: shows the influence of the parameter specifications for the spring characteristics in the compressive direction. A high value here stands for a progressive characteristic curve, a low value for a degressive to linear characteristic curve. Across all the scenarios considered, the trend shows that a characteristic curve that is as progressive as possible has a beneficial effect on both the compressive forces and the acceleration that occurs. The influence on the acceleration is even slightly greater than the influence on the compressive force. For extremely strong impacts, the influence of this parameter diminishes because the effect is overlaid by others.

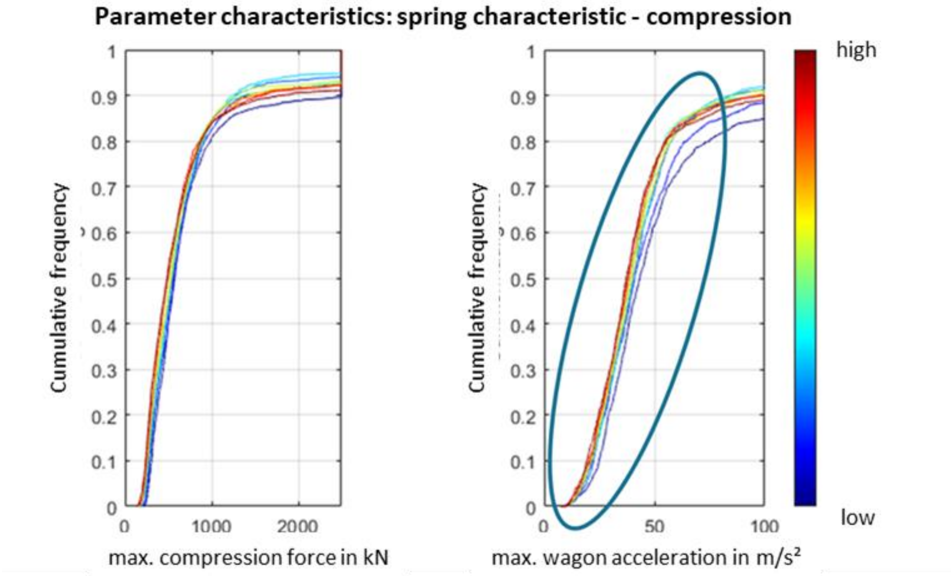



**Fig. 54: Parameter specifications of the spring characteristics in the compressive direction in shunting impacts**

Fig. 55: shows the influence of the specifications for the spring end force in the compressive direction. Here, too, there is a fundamental difference between low to medium and high impacts. For relatively low compressive forces and wagon accelerations, a high end force is slightly worse. However, it has clear advantages for strong impacts.

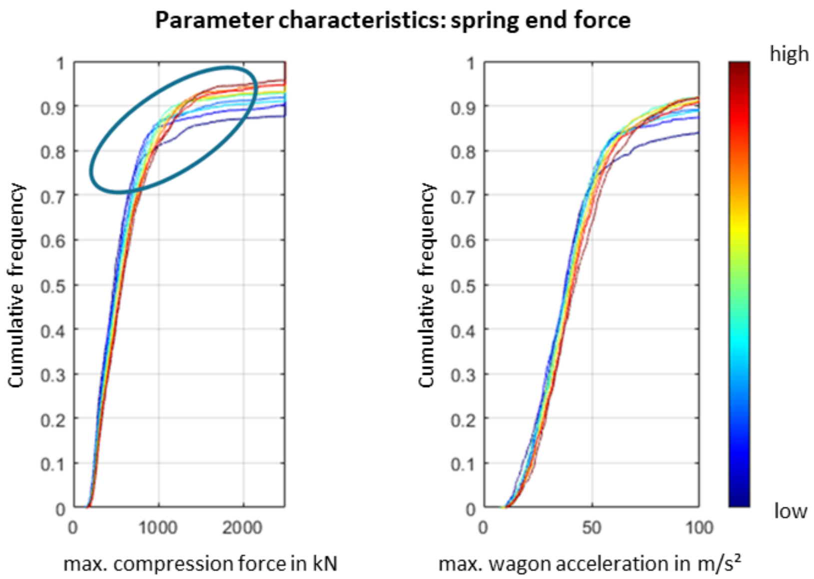



**Fig. 55: Parameter specifications of the spring end force in the compressive direction in shunting impacts**

### 6.3.2 Impacts with groups of wagons

As explained in 3.4, it can generally be assumed that the forces that occur in the impacting coupling are largely independent of the number of impacting or impacted wagons [1]. Thus, in the context of the sensitivity analysis, an analysis of the impact between only two wagons – one impacting and one impacted – is perfectly sufficient to identify the most important parameters.

However, the analysis must be extended somewhat for the investigation of operational stability. Due to the shunting process, a wagon's coupling is not only subjected to a force when the wagon runs over the hump and into stationary wagons. The coupling experiences a force even if the wagon is already in the sorting siding and further wagons are running into the stationary ones. This is because the impact energy is typically distributed over several wagons and couplings. Although this is smaller than the force on the impacting coupling on the wagon running down the hump, it cannot be ignored during the analysis of operational stability. In addition, the wagons do not always run down individually – they also roll over the hump in groups. In this case, more wagons are involved in the impact, so the total impact energy is potentially higher than it would be for a single wagon. On the other hand, a coupling may be located in the middle of the group of wagons rather than at the point of impact. In this case, the coupling in question would never be exposed to the full load from an impact during a single shunting operation.

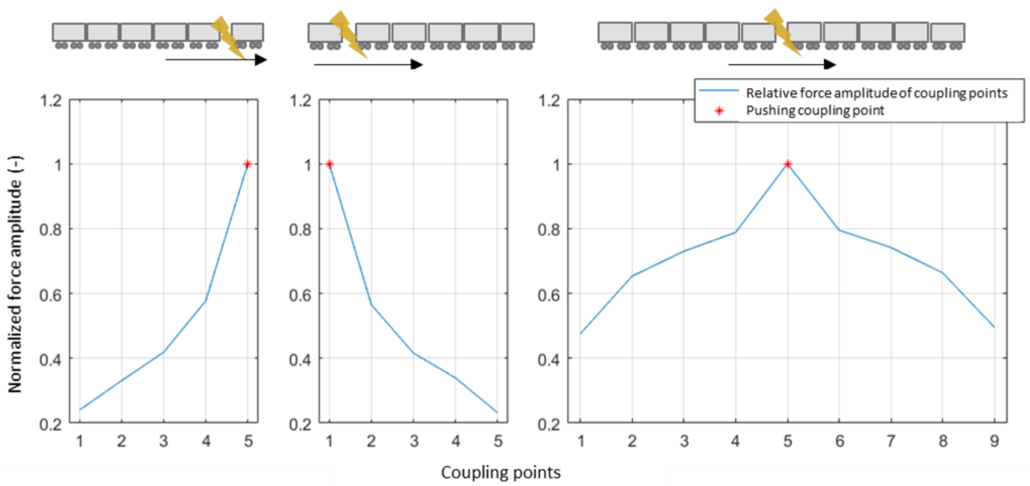



**Fig. 56: Relative force amplitude for shunting impacts over several coupling points**

For this reason, the distribution of impact energy during impacts between individual wagons and groups or even different groups of wagons is also investigated. Fig. 56: shows a comparison of three typical cases: impacts in which a group of five wagons runs into a single stationary wagon, impacts where a single wagon runs into a group of five stationary wagons, and impacts between two groups of wagons consisting of five freight wagons each. The investigation compares the maximum forces occurring in a coupling point during the impact. The representation is normalised to the force in the impacting coupling point, which is set at a value of 1. For the impacts between individual wagons and groups of wagons, similar behaviour is observed irrespective of the direction of the impact. The amplitude of the force at the coupling point nearest to the impacting coupling is only about 60% of the force amplitude at the impacting coupling itself. The amplitude is still about one fifth at the coupling point that is furthest away from the impacting coupling. A slightly different behaviour is seen in the impacts between groups of wagons. Since more vehicles, and therefore more couplings are involved, the impact energy is distributed differently over the couplings involved. The relative amplitude in the non-impacting couplings is therefore higher than in impacts in which fewer wagons are involved.

### 6.3.3 Interim conclusion

The influence of the dynamic parameters to be determined during the design process for the maximum compressive forces or wagon accelerations in the case of shunting impacts is complex. For some parameters, the most advantageous design specifications depend on whether weak or very strong impacts are being considered – with the opposite characteristics being advantageous in each case. Overall, it can be concluded that high energy absorption and the softest possible suspension is advantageous for limiting the forces and accelerations.

When designing the draft gear, the results show that the following criteria should be applied for an optimisation relating **exclusively** to the shunting impacts, i.e. ultimately a reduction of the forces and accelerations occurring, assuming impacts are in low to moderate range:

- low spring preload in the compressive direction

- highly progressive spring characteristic curve
- long spring travel
- high end force (optimised for *strong* shunting impacts).

Some of these parameters show an exactly opposite effect with particularly strong impacts. Firstly, however, it is assumed that the impact speeds will decrease overall due to the greater use of modern hump yards, which will reduce the number of strong impacts. Secondly, the aim of a targeted design for the coupling in its entirety should be to limit the force of any impacts that occur.

## 6.4 Effects of parameter optimisation on the forces occurring

Chapters 6.2 and 6.3 presented the parameter specifications, based on the sensitivity analysis, that would be considered optimal if the dynamic parameters were to be optimised **exclusively** for longitudinal dynamics or shunting impacts. For the longitudinal dynamics, the aim is to reduce the level of maximum longitudinal compressive and longitudinal tensile forces that occur. For shunting impacts, the focus is on reducing the compressive force caused by the impact and the resulting acceleration of the wagon. This section investigates the effects of the respective optimisation strategies on the forces generated by longitudinal dynamics and shunting impacts in various scenarios. These scenarios consider different train lengths, mass distributions and mass arrangements.

### 6.4.1 Overview of parameter optimisation

The optimal parameters for shunting impacts and longitudinal dynamics are diametrically opposed to each other. Table 15 provides a comparison of the advantageous specifications for three respective parameters – spring preload, spring characteristics and spring travel. For the longitudinal dynamics, a draft gear design that tends towards being a rigid connection is advantageous. In the extreme case, there would be little to no relative movement between the wagons and thus only minor longitudinal dynamic effects. The opposite is true for shunting impacts, where a soft suspension with high energy absorption is an advantage.

Table 15: Overview of parameter optimisation for longitudinal dynamics and shunting impacts

| Optimisation goal | Longitudinal dynamics | Shunting impacts |
|---|---|---|
| **Spring preload tension/ compression** | As high as possible | As low as possible |
| **Spring characteristics** | Degressive to linear characteristic curves | Highly progressive characteristic curve |
| **Spring travel to end force** | As short as possible | As long as possible |

In the design process, neither optimisation strategy leads directly to success. The question now arises as to how high the compressive force is during shunting impacts for a coupling optimised for longitudinal dynamics and how high the longitudinal compressive and longitudinal tensile forces are during various operational manoeuvres for a coupling optimised for shunting impacts. For the following investigations, the parameters listed in Table 15 are determined for the couplings optimised for longitudinal dynamics or shunting impacts up to the respective extreme specifications within the spread considered here (see 4.3).

### 6.4.2 Longitudinal dynamics

This section presents the effects of the respective optimisations on the longitudinal dynamics based on the operational manoeuvres already considered in this report. The focus here is particularly on emergency and service braking and the respective effects of optimisation on the longitudinal compressive and longitudinal tensile forces that occur. To achieve a greater significance, differentiation is made on several levels. Firstly, trains are differentiated according to their length. Secondly, the mass distributions and mass arrangements introduced in Fig. 5: are also considered. For clarity, the presentation uses the cumulative frequencies that have already been used previously. To aid comparison of the individual scenarios with the total for all simulations performed, a reference line is plotted in each case (shown in pink in Fig. 57: toFig. 61:). Enlarged versions of the following diagrams have also been provided in Annex **Fehler! Verweisquelle konnte nicht gefunden werden.** for better readability.

Fig. 57: shows the effects of parameters optimised for longitudinal compressive forces during emergency braking. Looking at the three train lengths considered here, it is

noticeable that the forces occurring depend very heavily on the train length and the mass distribution. This can be observed in all cases, regardless of the respective optimisation strategy. Implicitly, the total mass, which increases simultaneously with the train length and thus the respective brake position used, plays a major role. The train length emerged as the most influential parameter in the analysis of emergency braking. In all the scenarios considered, it is shown that the optimisation has a consistently positive effect on longitudinal dynamics. The longitudinal compressive forces occurring with the coupling optimised for longitudinal dynamics are consistently lower than those occurring with the coupling optimised for shunting impacts.

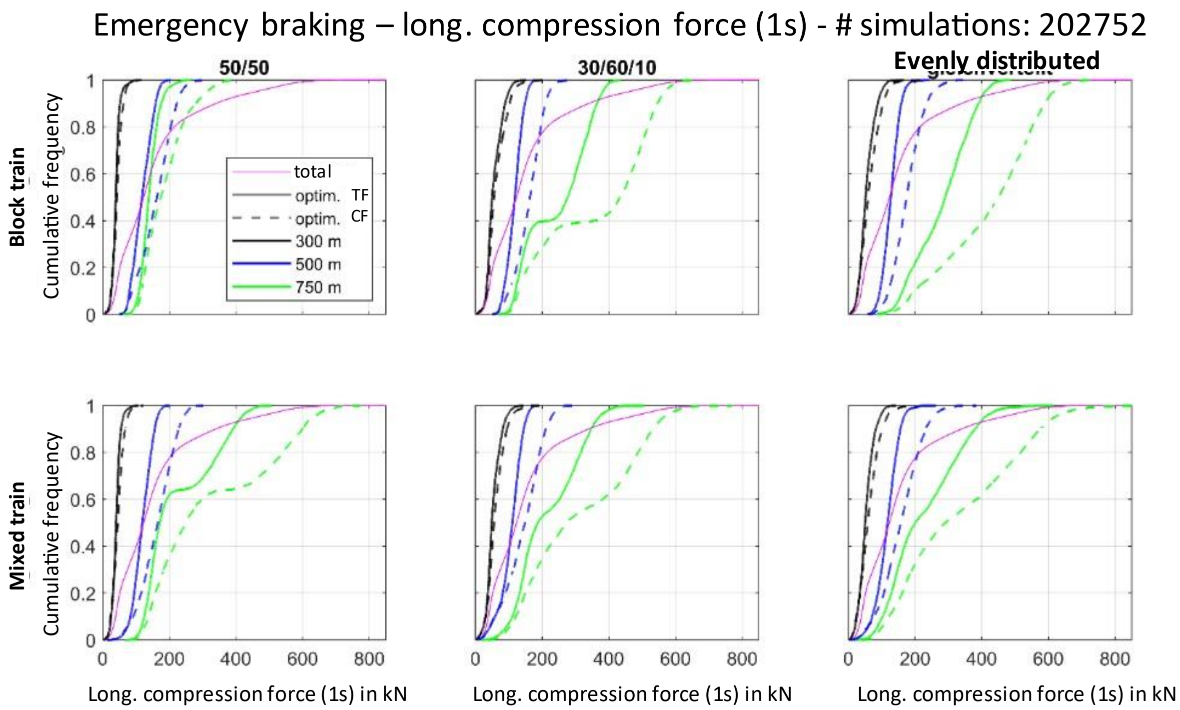



**Fig. 57: Effects of parameter optimisation on longitudinal compressive forces during emergency braking with composite brake pads**

This effect, however, is displayed to varying degrees in the various scenarios. While only small differences are observed for short trains, the optimisation has a very large effect on long trains. In absolute terms, the optimisation for the respective scenario shows the greatest effect at the highest forces.

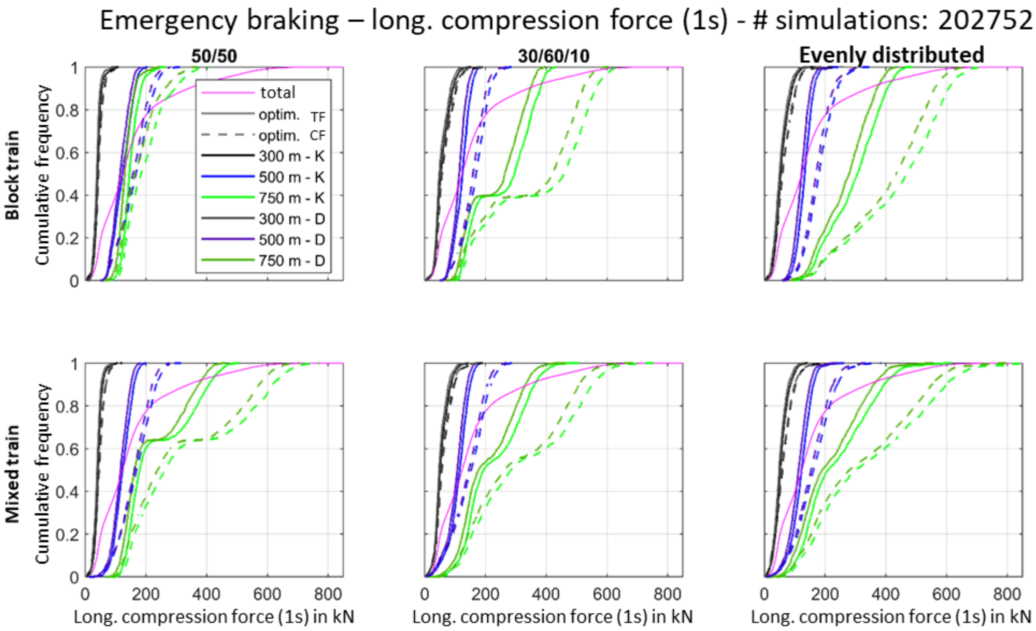



**Fig. 58: Effects of parameter optimisation on longitudinal compressive forces during emergency braking with various friction materials**

Using the same scenario as above, the influence of different friction materials is now investigated as well. This compares composite brake pads with disc brakes for all the trains considered. The results are presented in Fig. 58:. They show that the longitudinal compressive forces occurring with disc brakes are somewhat lower than with composite brake pads. In absolute terms, the effect is once again particularly strong in longer trains. Although the influence is clearly recognisable, it cannot be considered a dominant factor overall and does not change the trend, i.e. the respective optimisation of the coupling has a much greater influence than the friction material used.

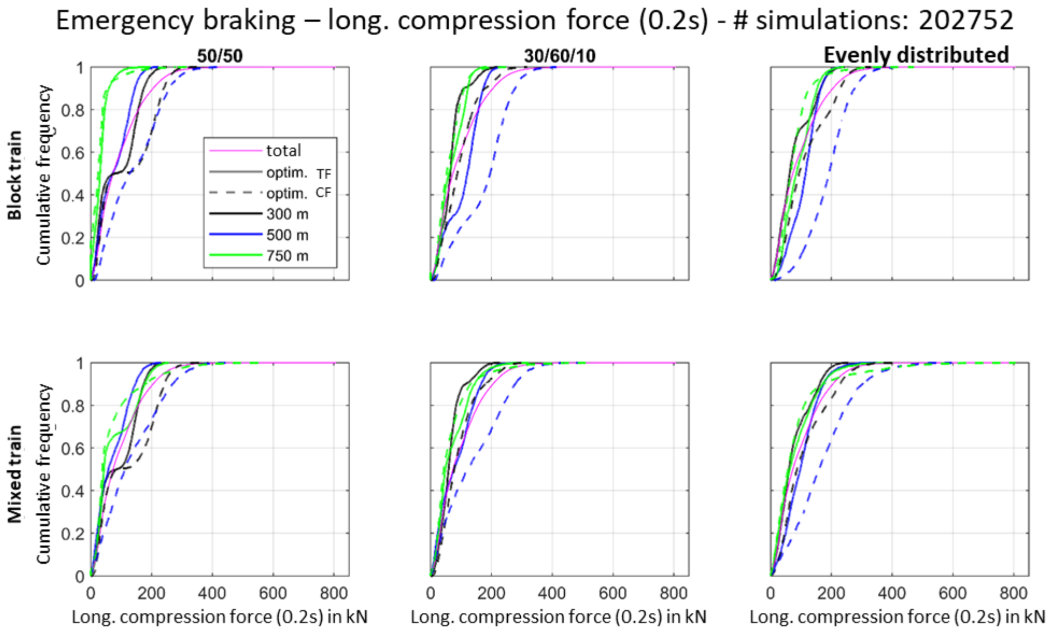



**Fig. 59: Effects of parameter optimisation on longitudinal tensile forces during emergency braking**

The final step of the analysis of emergency braking examines the effects of the optimisation on the longitudinal tensile forces for this operating scenario (see Fig. 59:). The effects of different train lengths manifest themselves differently here than in the case of longitudinal compressive forces, and the forces involved are lower overall. Nevertheless, the optimisation also clearly shows its effect in the tensile direction. Here, however, the selection of optimal parameters for the longitudinal dynamics tends to reduce forces more markedly in short and medium trains, while only a small effect is seen in long trains.

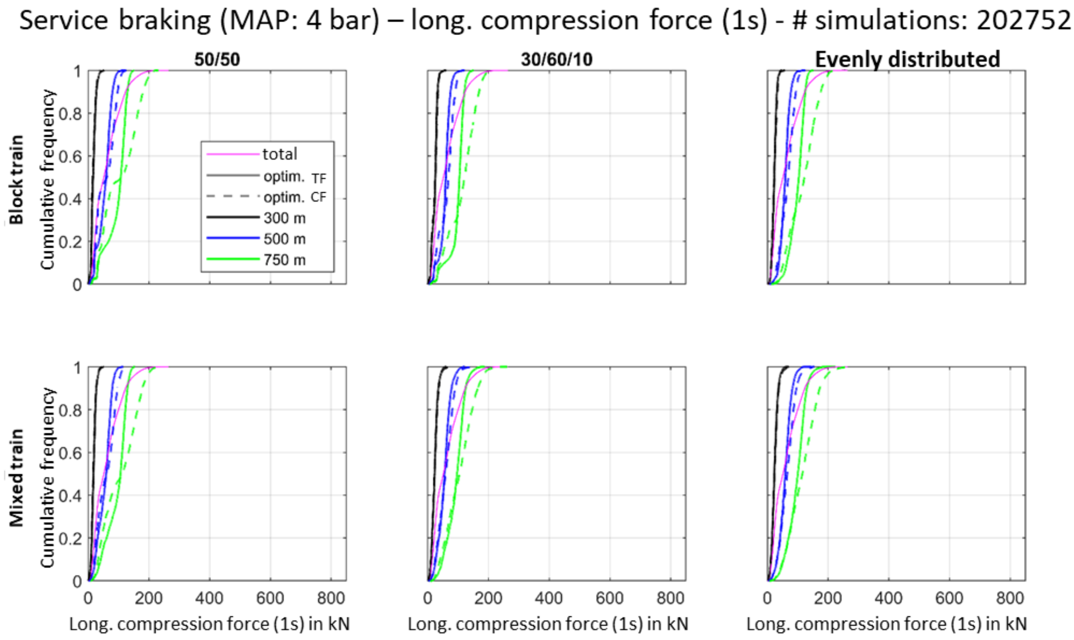



**Fig. 60: Effects of parameter optimisation on longitudinal compressive forces during service braking**

During service braking – defined in this study as braking operations where the pressure in the MBP is reduced to 4 bar – the longitudinal compressive forces are significantly lower than the forces that occur during emergency braking, as can be clearly seen in Fig. 60:. Nevertheless, the effects of optimisation for longitudinal dynamics are still evident at somewhat higher forces. As above, this effect is more obvious in long trains than in short ones. However, the opposite behaviour can be observed when the area of low forces is considered. Here, the parameters optimised for shunting impacts prove to be better in some cases than the parameters optimised for longitudinal dynamics. Overall, the effect is not particularly strong, but is more pronounced for mass distributions of 50/50 and 30/60/10 than for the mass distribution with an equally distributed probability of the parameters. Moreover, this tendency is more pronounced in block train mass arrangements than in mixed trains. In summary, however, the effect of the optimisation is apparent because higher forces tend to be more relevant with regard to the design than low forces.

Service braking (MAP: 4 bar) dynamic indicator force- # simulations: 202752

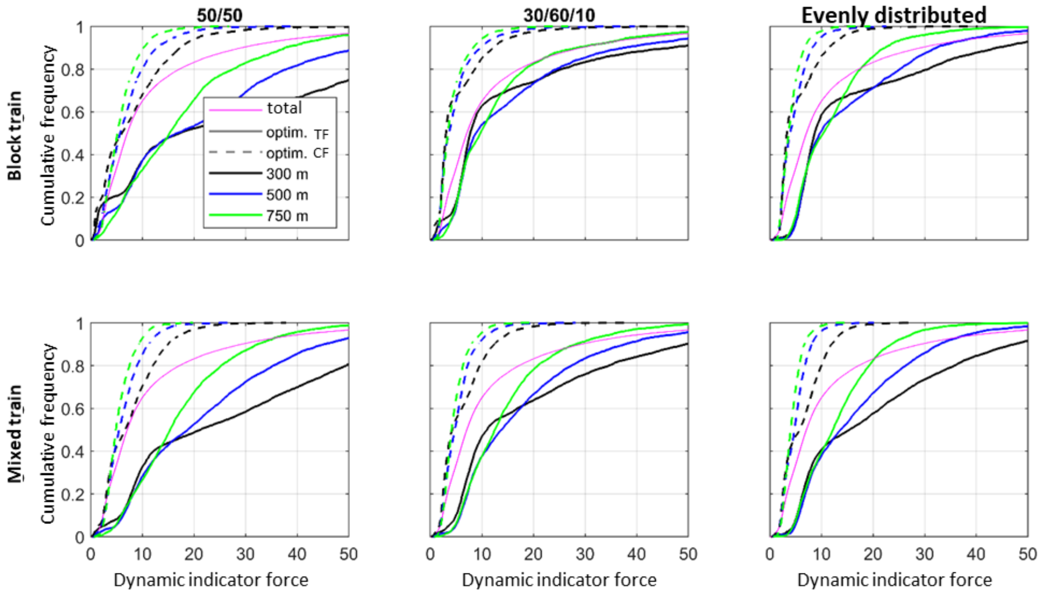



**Fig. 61: Effects of parameter optimisation on dynamics occurring during service braking**

Finally, the study examines the dynamics that occur using the example of service braking. To do this, the dynamic index, introduced by JOBSTFINKE in his dissertation, is used as an indicator. The dynamic index expresses the duration of an oscillation process, i.e. ultimately how many oscillation cycles occur until a longitudinal oscillation subsides after an operational manoeuvre, such as service braking. The higher the dynamic index, the more fatigue cycles occur as a result of an event [5]. The fatigue cycles that occur cause the coupling to vibrate, so this value is particularly important when considering operational stability. As explained in 4.5, both the amplitude of forces and the frequency with which they occur is important here.

The results of the investigation of the dynamic index are presented in Fig. 61:. Across all the scenarios considered, it can be seen here that the dynamics are significantly higher when the coupling is designed to be very stiff, as is the case with the version optimised for longitudinal dynamics. The vibrations decay much faster with the parameters optimised for shunting impacts, i.e. for an overall softer suspension and higher energy absorption. In general, this effect becomes stronger as the train in question becomes shorter. It follows from this that optimising the parameters for longitudinal dynamics, with the aim of reducing the maximum longitudinal compressive and tensile forces, reduces the force amplitude but results in more oscillations with smaller amplitudes. The effect of this behaviour on operational stability still requires investigation.

### 6.4.3 Shunting impacts

The second step examines the effects of the respective optimisation on forces caused by shunting impacts. The procedure is similar to that used in the analysis of longitudinal dynamics, but different train lengths and mass arrangements are not considered for obvious reasons. However, it replaces these by differentiating according to the speed distribution of the shunting impacts in the marshalling yards. As in 6.3, the output variables investigated are the maximum compressive force and the wagon acceleration. In addition, this step considers the maximum tensile force that occurs as a result of the coupling process. To aid comparison of the individual scenarios with the total for all simulations performed, a reference line is plotted in each case (shown in black in Fig. 62: to Fig. 64:).

The effects of the respective parameter optimisations on the maximum compressive forces occurring for shunting impacts are shown in Fig. 62: based on their cumulative frequency in the underlying simulations. Firstly, it is noticeable that the amplitudes of the forces for the different velocity distributions vary widely. Depending on the scenario, the respective amplitudes are separated by factors of 2 - 4. In addition, it can be seen that the mass distribution influences the distribution of the force amplitudes. These have different effects in different force ranges. In the comparison of the curves for the two optimisation strategies it is noticeable that optimising for impacts is advantageous in the area of compressive forces, reducing the respective amplitude independently of the scenario and speed distribution. The absolute reduction in the force occurring is of a similar magnitude in each case. Overall, the magnitude of the forces occurring is very clearly dominated by the assumed speed distribution, giving this parameter enormous significance with regard to operational strength.

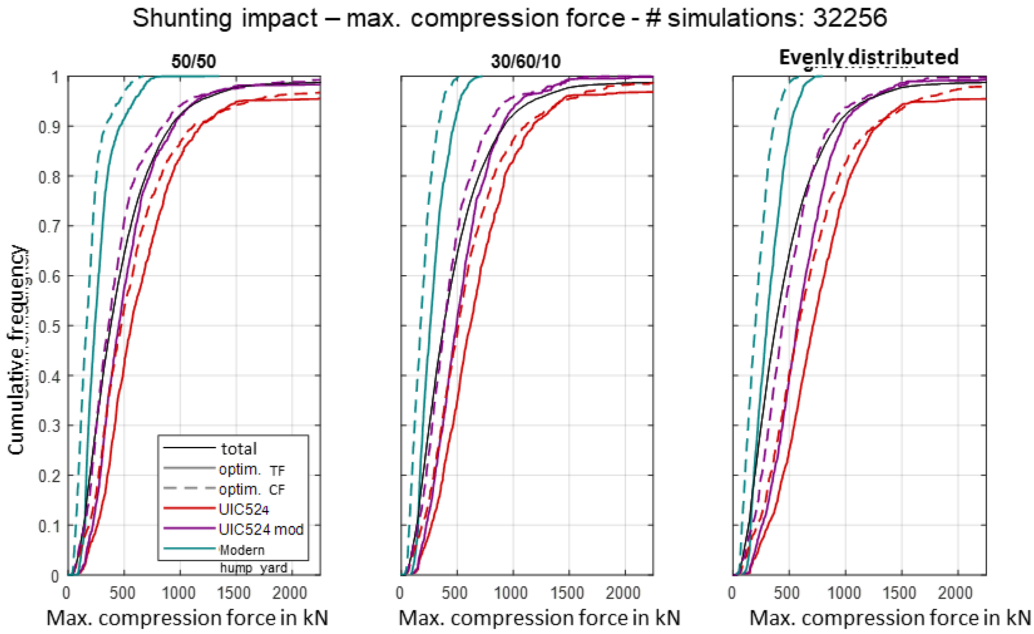



**Fig. 62: Effects of parameter optimisation on the maximum compressive force in shunting impacts**

The large spread in the forces due to speed distribution can also be seen in the analysis of tensile forces in Fig. 63:. Here, too, the positive effect of the optimisation on shunting impacts is confirmed across all scenarios and speed distributions. In contrast to the analysis for compressive forces, however, it is evident here that the reduction in the maximum force becomes much greater as the mean value of the running speed down the hump increases. Thus, the optimisation has a greater effect if a greater number of strong impacts tend to occur. On the other hand, in the analysis of the tensile force, the total force occurring is determined somewhat less by the different speeds and more by the optimisation of the parameters. In Fig. 62:, this can be seen, among other things, by the fact that the graphs for the designs optimised for longitudinal dynamics and shunting impacts always lie next to each other for the respective speed distribution. However, this is no longer the case in the consideration of tensile forces in Fig. 63:. For example, lower forces occur with a coupling optimised for shunting impacts with the speed distribution defined by UIC 524 than with a coupling optimised for longitudinal dynamics in accordance with the modified distribution, which has a lower mean value for the impact speed.

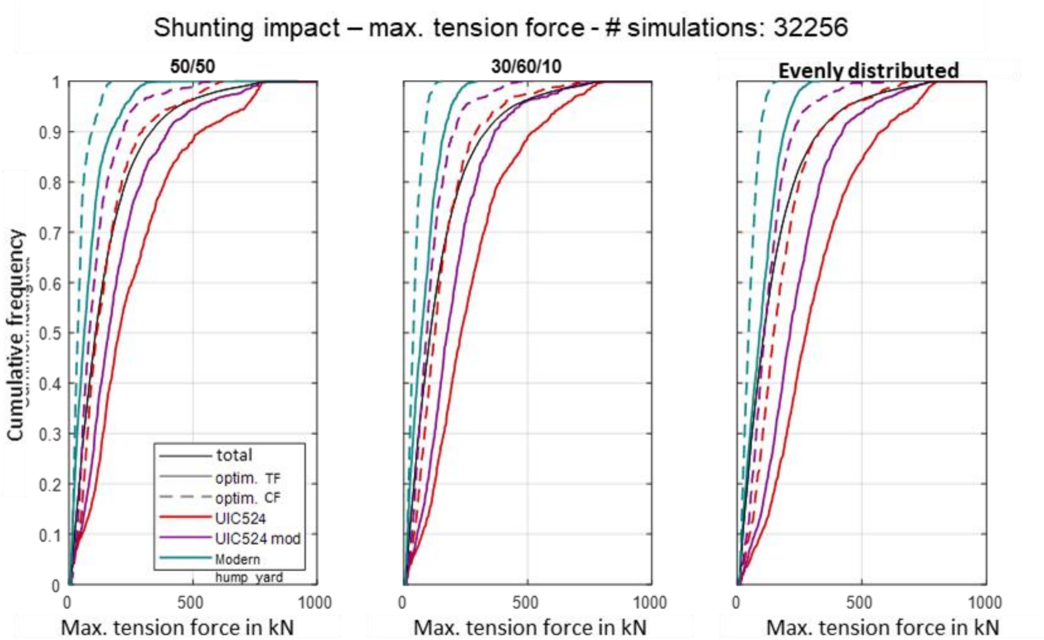



**Fig. 63: Effects of parameter optimisation on the maximum tensile force in shunting impacts**

The effects of optimisation on shunting impacts become particularly clear in the analysis of the maximum wagon acceleration (see Fig. 64:). Here, too, the influence of the different speed distributions is immediately obvious, and this also ensures a large spread of the accelerations. However, the effect of the optimisation strategy is visible across all scenarios and is particularly evident in scenarios with acceleration amplitudes up to approx. 30 m/s² for all mass distributions. Here, even with the extreme speed distribution defined by UIC 524, the acceleration amplitude for the optimised coupling is below that of the coupling optimised for longitudinal dynamics in the case of a modern hump yard. In absolute terms, the reduction in the respective force occurring lies in a similar range independently of the speed distribution.

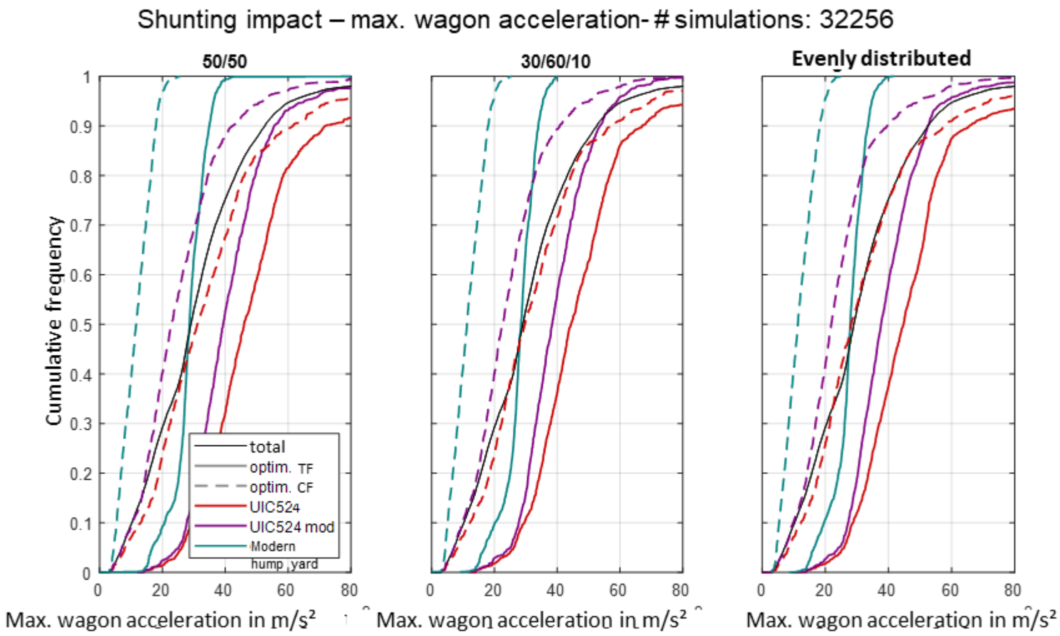



**Fig. 64: Effects of parameter optimisation on the maximum wagon acceleration in shunting impacts**

Based on these results, it can be concluded that an optimisation of the dynamic parameters for shunting impacts is advantageous for the wagons in terms of the forces and wagon accelerations occurring, independently of the speed distributions and mass distribution.

### 6.4.4 Interaction of parameter optimisation and EP brake

The conflict of objectives between optimisation of the dynamic parameters for longitudinal dynamics and optimisation for shunting impacts cannot be resolved under the limiting technical and operational conditions considered so far. A compromise between the two strategies is required. In both cases, the respective optimisations show themselves to be effective. A one-sided optimisation to benefit longitudinal dynamics may be possible under certain circumstances if technical or operational measures can be introduced to achieve further significant reductions in the forces caused by shunting impacts at the speed distribution in modern hump yards. However, this is probably unrealistic because freight wagons are still likely to pass through a high proportion of old hump yards in Europe. This optimisation would, of course, be possible in principle for wagons that never pass through hump yards.

On the other hand, one-sided optimisation for shunting impacts could be possible, provided that the forces caused by longitudinal dynamics can be reduced significantly. The following section examines the extent to which this may be feasible through the introduction of electro-pneumatic brakes (EP brakes) in rail freight transport. Compared to purely pneumatic brakes, EP brakes have the advantage that they can be applied and released synchronously with potentially significant reductions in the longitudinal compressive forces [56]. The following section therefore compares purely pneumatic and EP brakes to examine how the longitudinal compressive forces behave in a coupling optimised exclusively for shunting impacts.

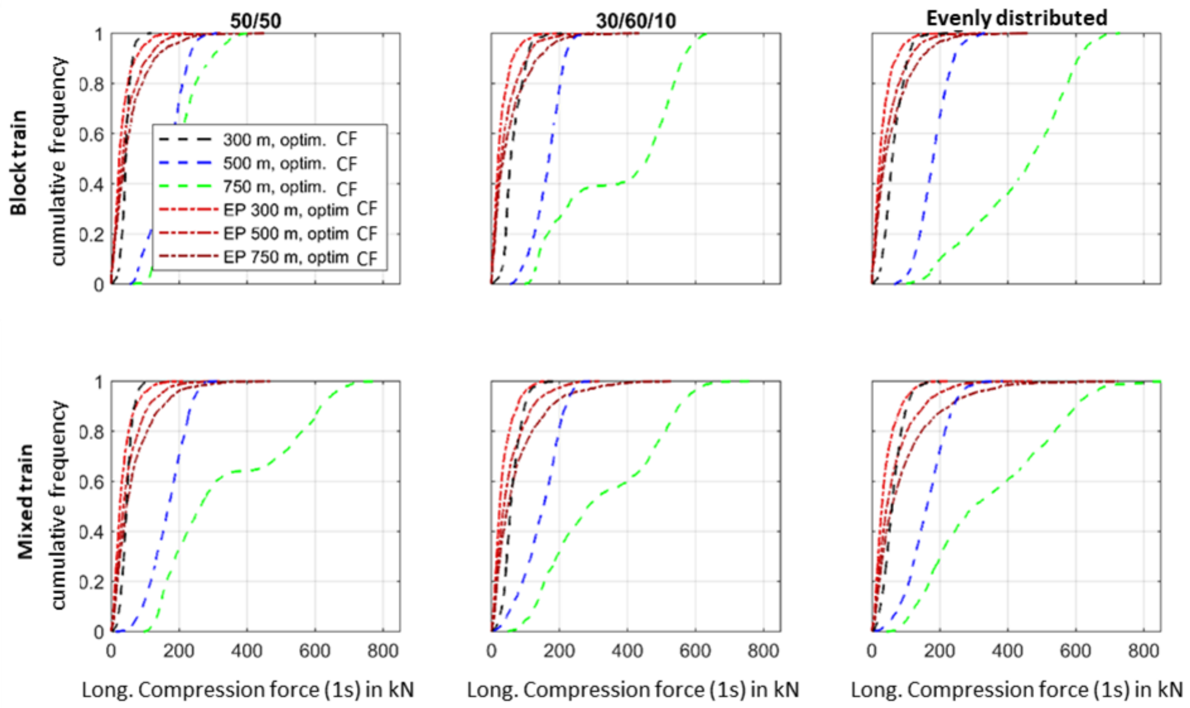



**Fig. 65: Comparison of purely pneumatic and EP brakes: Longitudinal compressive forces during emergency braking**

Fig. 65: shows a comparison of the longitudinal compressive forces occurring due to longitudinal dynamics using the example of emergency braking. The analysis is again divided according to various train lengths, mass arrangements and distributions. It clearly shows that the use of EP brakes can reduce the longitudinal forces occurring by orders of magnitude, especially in long trains. While the maximum values for different train lengths vary enormously with the purely pneumatic brake, the synchronous control provided by EP brakes ensures much smaller differences in the forces

occurring. Despite the parameters being optimised for shunting impacts, the use of EP brakes consistently results in significantly reduced longitudinal compressive forces during emergency braking. In the same way, this also applies to the dynamics, i.e. the number of fatigue cycles that result from braking. Although not shown here, this is also reduced by the use of EP brakes.

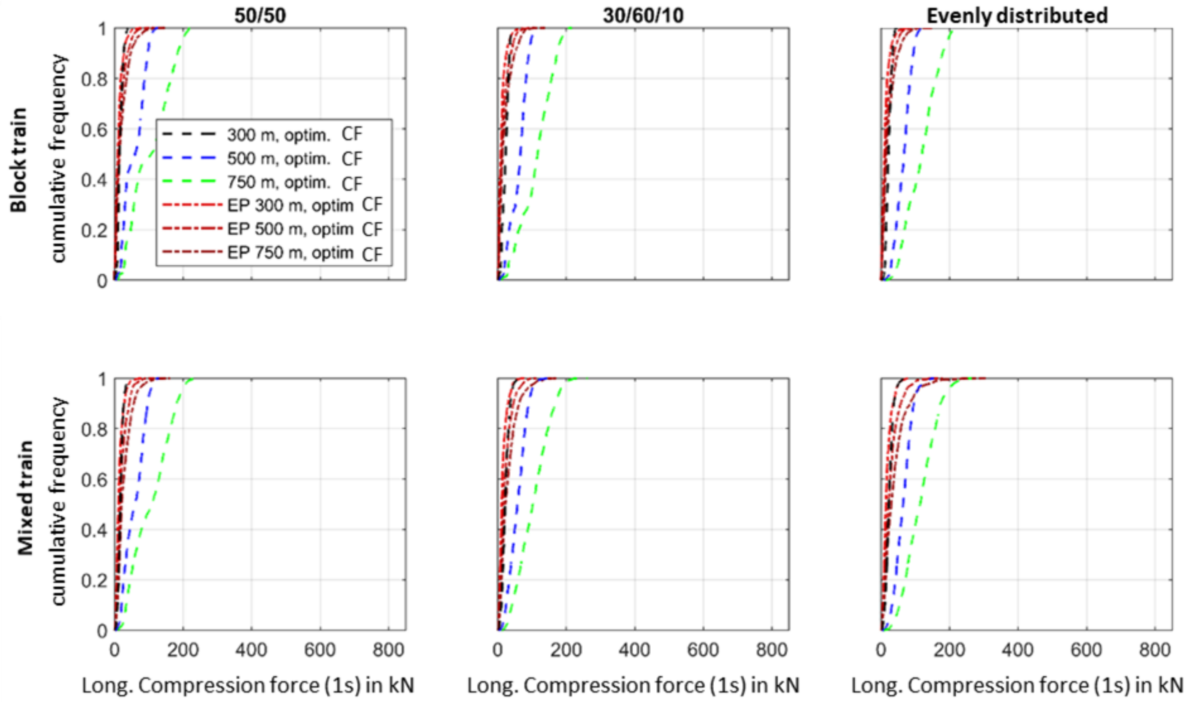



**Fig. 66: Comparison of purely pneumatic and EP brakes: Longitudinal compressive forces during service braking**

Fig. 66: extends the investigation to service braking, where the force level is lower than in emergency braking. The positive effects of EP brakes are also clear during service braking. The force level decreases significantly over all train lengths, mass arrangements and mass distributions considered, despite the parameters being optimised for shunting impacts.

### 6.4.5 Interim conclusion

The analyses of the forces occurring/wagon accelerations for the respectively optimised dynamic parameters show very clearly that the optimisations in both directions are effective in achieving their objectives. A coupling optimised for longitudinal dynamics significantly reduces the longitudinal compressive and longitudinal tensile forces arising from starting up and braking processes, especially at high force levels. The effects of optimisation increase as trains become longer. In low force ranges, however, the values for the coupling optimised for the longitudinal dynamics of the train may sometimes be lower than those of the coupling optimised for shunting impacts.

A coupling optimised for shunting impacts has a consistently positive effect during manoeuvring, regardless of the speed distribution under consideration. On the one hand, the optimisation reduces the compressive forces occurring over the entire force range and reduces the accelerations occurring at the wagons involved in the impact even more significantly. The results also show that the speed distribution at the hump has a major influence on the impact forces. For subsequent investigations, these distributions must therefore be determined as accurately as possible and the range limited as well as possible.

The only clear statement that can be made is that – with the technical solutions currently implemented for the DAC draft gear – a coupling with optimum behaviour for both longitudinal dynamics and powerful shunting impacts is not possible. The overall optimum parameters are probably a compromise between the two strategies. The possible common optimisation objective is outlined in more detail in Chapter 6.4 on the investigation of operational stability. The fact that the coupling parameters influence the result distributions means that the force collectives to be determined also depend to some extent on the coupling design and not only on the limiting operational conditions.

The use of an EP brake in freight traffic reduces the forces resulting from longitudinal dynamics very significantly. This effect is also evident when using a coupling with parameters optimised for shunting impacts. Several possible conclusions can be drawn from these results, depending on the objective and possible developments in RFT. Assuming a continuation of existing train and operational concepts with a train length generally restricted to 740 m and thus also limited maximum train masses, the EP brake allows the dynamic parameters of the DAC to be optimised purely for shunting impacts. On the other hand, a significant reduction in the force level caused by starting up and braking processes opens the way for using new traction and operating concepts that were previously impossible for reasons of running safety due to the longitudinal compressive forces that occur. These include, e.g. the use of longer trains and thus higher maximum train masses [56].

## 6.5 Operational stability

The investigations on operational stability are based on the results presented above for longitudinal dynamics and shunting impacts. Ultimately, the objective is to estimate the stresses and strains to which a DAC will be exposed during its service life. These depend partly on the operational limiting conditions which influence, e.g. how often trains brake, accelerate or pass through marshalling yards. These factors affect the forces acting on the coupling due to longitudinal dynamics or shunting impacts. In the level below, these basic conditions can be broken down even further, e.g. by considering different speed distributions. Longitudinal dynamic effects are not only triggered by driving manoeuvres, but also to a small extent by changes in the inclination of the track or fluctuations in speed during steady-state running. Different mass arrangements and distributions must also be considered. In addition, the effects of different brake pad materials are investigated as technical parameters of the wagons.

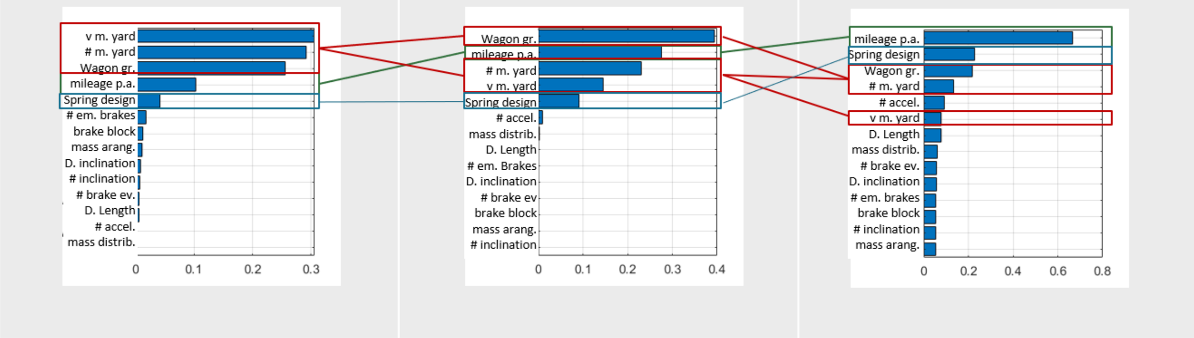
In the previous chapter, it was shown that the spring design itself also has a clear influence on the forces occurring and thus potentially also on the operational stability of the coupling. For this reason, the effects of the optimisation on longitudinal dynamics and shunting impacts are also studied in the investigation of operational stability. The procedure is divided into two steps: In the first step, a sensitivity analysis using GSA is performed to find out which parameters have a particularly large influence on the operational stability. For these parameters, the study seeks to identify which parameter characteristics have a positive or negative effect on operational stability. In the second step, the analysis is differentiated according to various RFT segments in order to determine the traffic types for which parameter optimisation has an effect on the operational stability.

### 6.5.1 Sensitivity analysis

The aim of the sensitivity analysis is to determine which parameters from the groups mentioned above have a particularly important influence on operational stability. Here, too, it is necessary to identify the most appropriate variable to use as a reference point for the analysis. In the absence of a standard procedure for comparing different force collectives, this study uses the cumulative difference of the force collectives (CDFC) as explained in 4.5.4. The collective for a given scenario is compared with the reference collective, which was formed as a minimum from the totality of the collectives for all scenarios considered. In principle, the respective numbers are subtracted class by class and the differences added up.
However, the result also depends on how the differences are weighted. This is particularly relevant for forces that occur only rarely but with a high amplitude. For this purpose, three different procedures for producing the respective CDFCs are investigated. The method that seems obvious at first glance is the summation of the absolute difference, with all classes of force amplitudes being given an equal weighting. A stronger weighting of higher forces is achieved by adding up the square roots of the respective differences. The weighting of high forces is even higher if the sum of the logarithmic differences of the frequency per class is used. The sensitivity analysis is performed for all three methods, the results are listed in Table 16. For each weighting method, the parameters are listed in the order of their influence according to GSA. To simplify classification of the results, a distinction is made between parameters that concern only the longitudinal dynamics, only the shunting impacts, both cases or the coupling itself.

**Table 16: Comparison of different methods for weighting the cumulative difference of the force collectives**

| Strong weighting of high forces | Medium weighting of high forces | Equal weighting of all forces |
|---|---|---|
| $A_{Diff} = \sum_{i=1}^{n_{Klassen}} \log\left[n_{Sz}(i) - n_{ref}(i)\right]$ | $A_{Diff} = \sum_{i=1}^{n_{Klassen}} \sqrt{n_{Sz}(i) - n_{ref}(i)}$ | $A_{Diff} = \sum_{i=1}^{n_{Klassen}} n_{Sz}(i) - n_{ref}(i)$ |




Parameters affecting shunting impacts

Parameters affecting the coupling itself

Parameters affecting shunting impacts and longitudinal dynamics

The different methods result in clear differences between the rankings of the most important parameters. Thus, the weighting method also influences the results of the sensitivity analysis. However, almost the same parameters prove to be the most influential in all three methods. A comparison of the rankings shows the following: the higher the weighting of high forces, the more dominant the role of factors involved in shunting impacts. However, the annual mileage is also important. If, on the other hand, all forces are equally weighted and only the absolute number of amplitudes occurring is important, the parameters of the longitudinal dynamics are the dominant factor – especially the annual mileage.

For the following analysis, the method based on logarithmic differences and a strong weighting of high forces is selected. This method is most comparable to the Wöhler method in the investigation of stress collectives , which also uses logarithmic frequencies. The results are presented in detail in Fig. 67:. The influence of the various parameters on the force collectives is shown in the right of the diagram. The three most important parameters are all related to shunting impacts in marshalling yards, the speed distribution during shunting, the number of marshalling yard stops (MY stops) per 100 km and the average size of the groups on the hump. These results reflect the findings for forces occurring due to shunting impacts, where the speed distribution has a decisive influence on the amplitudes. The number of stops at marshalling yards influences the frequency with which force effects with potentially high loads on the coupling occur due to impacts. The average number of wagons running down the hump together influences the frequency with which the coupling is actually loaded with the full impact force during an impact. Another important parameter is the annual wagon mileage. In addition – and confirming previous assumptions – the spring design itself also plays a role. The other parameters are not completely without influence but should be regarded as playing a minor role in comparison to the above-mentioned variables under the conditions of this analysis.

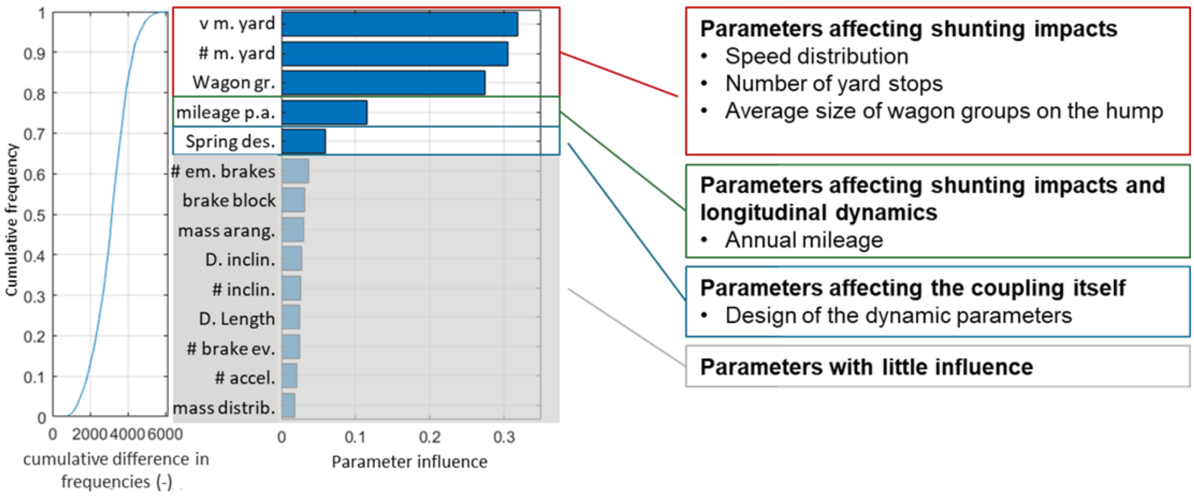



**Fig. 67: Identification of influential parameters for the operational stability of the DAC**

At this point it must be noted that the sensitivity analysis refers to the parameter ranges identified and to the chosen method for determining the cumulative difference of the areas of the force collectives in relation to the reference collective. Some parameters in this analysis, such as the speed distribution, have very large ranges. Limiting certain parameters to a smaller range or shifting the ranges of individual parameters also changes the results of the sensitivity analysis. The more precisely the ranges can be limited, the more accurate the statements that can be made about the influence of this parameter as well as the other parameters.

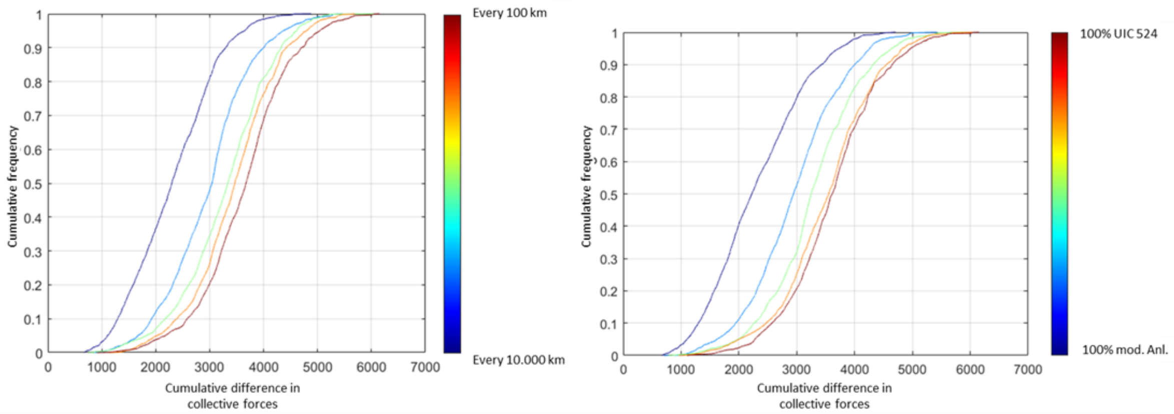



**Fig. 68: Parameter specifications of the number of MY stops per 100 km (left) and speed distribution on the hump (right)**

The next step now examines the advantageous characteristics of the most important parameters based on the cumulative frequencies. Here too, the parameter values are clustered into five graduated groups.

Fig. 68: provides a breakdown of the effects of the parameter values for the number of MY stops and the speed distribution on the humps. The more often wagons pass through marshalling yards, the greater the deviation of the force collective from the reference collective and thus the greater the overall frequency or amplitude of the loads on the couplings. The higher the proportion of marshalling yards equipped with modern gravity marshalling systems that the wagons pass through, the lower the frequency and amplitude of the forces. The two variables are closely related, again underlining the results of the investigation of shunting impacts.

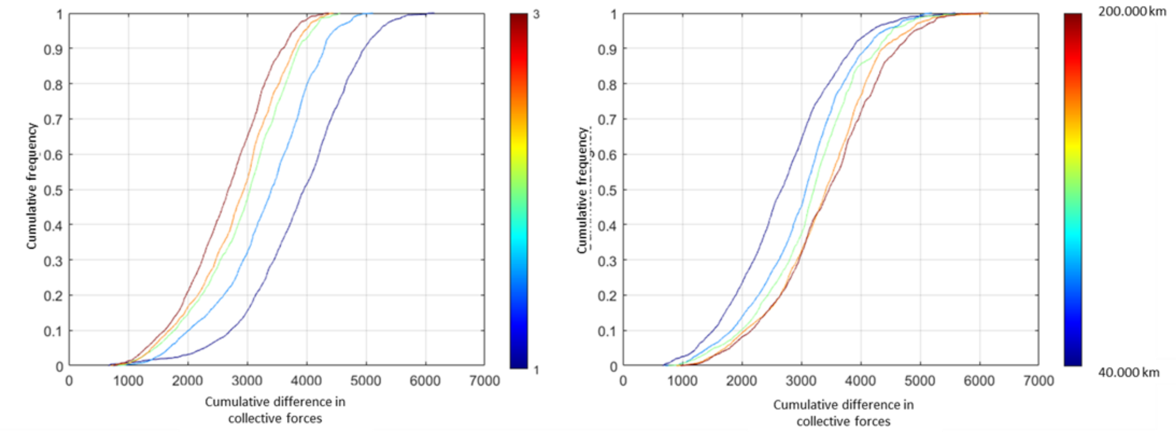



**Fig. 69: Parameter specifications of the average size of the wagon groups on the hump (left) and annual mileage (right) in relation to the operational stability**

Fig. 69: shows the influence of the parameter specifications for the average size of the wagon groups on the hump and the annual mileage. It shows that the amplitudes and frequencies of forces decrease as the average size of wagon groups on the hump increases and the annual mileage decreases.

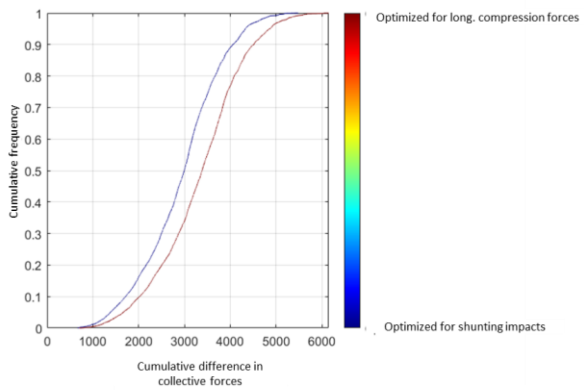
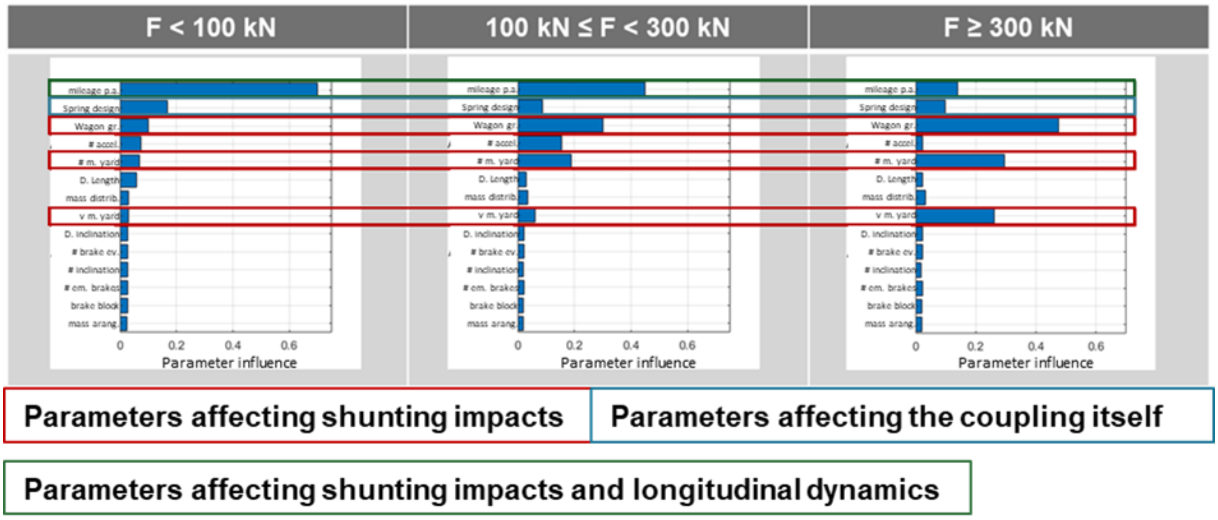



**Fig. 70: Parameter specifications of the spring design in relation to the operational stability**

The effects of the different spring designs on the CFDC are listed in Fig. 70:. In contrast to the previous parameters, this parameter is not continuous, i.e. only the respective optimised designs were examined (as described above). Across all the simulated cases, a coupling that tends to be optimised for shunting impacts proves to be somewhat more advantageous. This is hardly surprising, since the most important parameters for the force collectives concern the shunting impacts. However, this result must be qualified: here again, it is important to note the sometimes wide ranges, especially for the most influential parameter – the speed distribution. Since the spring design, as shown in the previous chapters, influences not only the force amplitude but also the frequency and number of fatigue cycles, the analysis must make a greater differentiation when considering a design optimised for operational stability. This analysis is looked at again in 6.5.2 and the results broken down according to different operating scenarios or RFT segments.

In the final step of the sensitivity analysis, the dominant parameters are now examined separately for different force ranges. Forces below 100 kN are classified as low forces, forces between 100 kN and 300 kN are recorded above this, and forces above 300 kN in the third group. The results of the individual analyses are listed in Table 17. The parameters were plotted here in the same order.

**Table 17: Identification of influential parameters for operational stability in various force ranges**




The results clearly show that the annual mileage plays the most important role for forces in the range up to 300 kN. In the low force range, the factors for shunting impacts play only a minor role. In the medium force range, they play a somewhat larger role and in the high force range they are clearly dominant. The spring design plays a significant role across all force ranges, although it is not the most important factor.

### 6.5.2 Evaluation of sample operating scenarios

Based on the results of the sensitivity analysis, the influence of different parameters is illustrated by means of some sample scenarios for specific collectives. The scenarios are roughly based on different production forms in RFT, which result in potentially different force collectives and thus different requirements for the design of the DAC. The underlying assumptions are listed in Table 18. These are used to derive the specific parameters. Within each of the three scenarios, further differentiations are made to resolve the effects of the parameter values in greater detail.

**Table 18: Assumptions for the operating programme of various production forms in RFT**

| Production form | Intermodal trains | Bulk goods block trains | Single wagonload traffic |
|---|---|---|---|
| Properties | • High mileage<br>• Usually block trains<br>• Rarely in MYs<br>• Rarely completely empty or completely full | • Medium mileage<br>• Block trains<br>• Often completely empty or completely full | • Low mileage<br>• Usually mixed trains<br>• Very different load states<br>• Frequent MY stops |

The collectives shown refer to an assessment period of one year and are based on the results of the previously performed simulations.

#### Intermodal trains

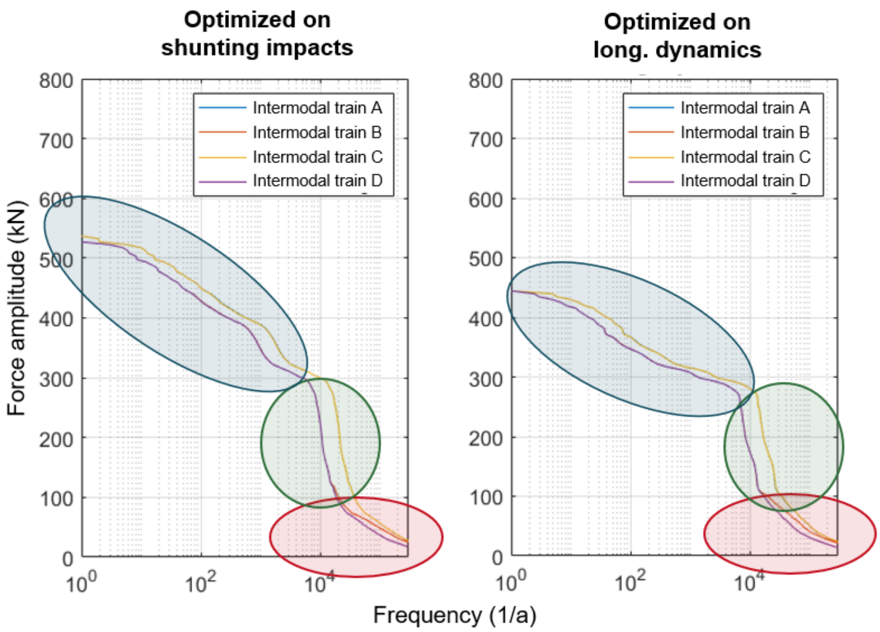
For intermodal trains, four different operating programmes are examined as examples. In the following, these are referred to as intermodal trains A to D. Some assumptions here apply to all trains, e.g. the distribution of the load states and the mass arrangement in the train. The annual mileage of the wagons, the relative number of braking and starting up operations based on IFW data and the brake pad material were varied. For shunting impacts, it was assumed in all scenarios that the trains pass through a marshalling yard every 10,000 km and that the impact speeds correspond to the values expected for a modern hump yard.

**Table 19: Examples of operating programmes investigated for intermodal trains**

| | Intermodal train A | Intermodal train B | Intermodal train C | Intermodal train D |
|---|---|---|---|---|
| Type of wagon | R or S | R or S | R or S | R or S |
| Distribution of load states | 30 % empty, 60 % half full, 10 % full | 30 % empty, 60 % half full, 10 % full | 30 % empty, 60 % half full, 10 % full | 30 % empty, 60 % half full, 10 % full |
| Arrangement of masses in the train | Stochastically distributed (with groups) | Stochastically distributed (with groups) | Stochastically distributed (with groups) | Stochastically distributed (with groups) |
| Annual mileage of wagons | 200,000 km | 200,000 km | 200,000 km | 100,000 km |
| Frequency of starting/braking | IFW | IFW · 0.5 | IFW | IFW |
| Brake pad materials | Disc brake | Disc brake | Mean value composite brake pads | Disc brake |
| Number of marshalling yard stops | every 10,000 km | every 10,000 km | every 10,000 km | every 10,000 km |
| Impact speeds in MY | Values for a modern hump yard | Values for a modern hump yard | Values for a modern hump yard | Values for a modern hump yard |
| | Reference | Halved frequency of braking/accelerations | Other brake pad material | Halved mileage |

Fig. 71: shows a comparison of force collectives for the four intermodal trains. The analysis distinguishes between the force ranges already broken down in Table 17. This is because the differences between the four scenarios manifest themselves in different ways. The comparison shows the collectives for a coupling optimised for shunting impacts and a coupling optimised for longitudinal dynamics. For presentation purposes, the frequency is scaled logarithmically.

In the low force range (red frame), it can be seen that the frequency of the forces occurring is reduced by halving the annual mileage compared to the reference collective and reducing the number of braking and acceleration processes. This confirms the results of the sensitivity analysis, which showed that the parameters affecting longitudinal dynamics are particularly important in this force range. However, halving the mileage (Intermodal train D) has a stronger effect because the vibration processes during steady-state running also influence the frequencies of the forces in the low force range. The selection of a different friction material makes almost no difference to the force collectives and is dominated by other factors. A comparison of the frequencies of the two optimised couplings shows that optimisation for longitudinal dynamics has an effect, albeit to a lesser extent.




| Intermodal train A | Intermodal train B | Intermodal train C | Intermodal train D |
|---|---|---|---|
| Reference | Halved frequency of braking/accelerations | Other brake pad material | Halved mileage |

**Fig. 71: Comparison of force collectives for intermodal trains**

In the force range between 100 kN and 300 kN (green frame), there are hardly any differences between the intermodal trains B and D. The lower number of operational manoeuvres that influence the longitudinal dynamics is also clearly shown in the comparison with the reference collective. Clearly, within this force range, forces due to steady-state running no longer have any, or only a very small, effect. No differences due to the use of different brake pad materials are observed here either. A comparison of the collectives between the two optimisation strategies also shows slight advantages for the coupling optimised for longitudinal dynamics.

With force amplitudes above 300 kN (blue frame), however, optimisation of the dynamic parameters has a very clear effect on the longitudinal dynamics. Both the overall maximum force amplitude and the frequencies of the forces in the high force range are significantly reduced. As already seen in the medium force range, halving the mileage and halving the number of braking and acceleration processes have almost the same effect, while the choice of brake pad material is essentially irrelevant.

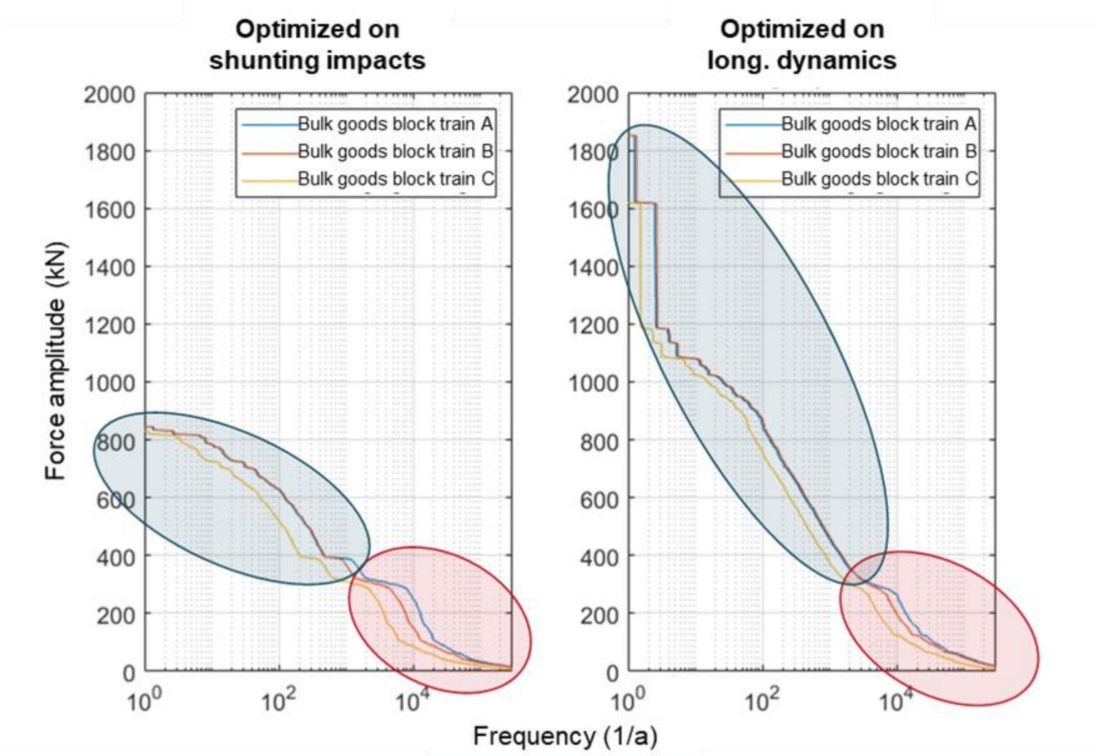
## Bulk goods block trains

In a second step, the force collectives for three examples of operating programmes for bulk goods block trains are investigated (see Table 20). It is assumed that these trains run 50% of their mileage empty and 50% fully loaded in the block train configuration. All wagons are assumed to have been fitted with composite brake pads. It is expected that the trains will pass through marshalling yards every 1,000 km and that the distribution of impact speeds will correspond to that of a modern hump yard. Example B considers a halved frequency of braking and acceleration processes and example C considers a halved annual mileage compared with the reference collective.

**Table 20: Examples of operating programmes investigated for bulk goods block trains**

| | Bulk goods block train A | Bulk goods block train B | Bulk goods block train C |
|---|---|---|---|
| Type of wagon | E and/or F | E and/or F | E and/or F |
| Distribution of load states | 50 % empty, 50 % full | 50 % empty, 50 % full | 50 % empty, 50 % full |
| Arrangement of masses in the train | Block train (all uniform) | Block train (all uniform) | Block train (all uniform) |
| Annual mileage of wagons | 100,000 km | 100,000 km | 50,000 km |
| Frequency of starting/braking | IFW | IFW · 0.5 | IFW |
| Brake pad materials | Mean value composite brake pads | Mean value composite brake pads | Mean value composite brake pads |
| Number of marshalling yard stops | every 1,000 km | every 1,000 km | every 1,000 km |
| Impact speeds in MY | Values for a modern hump yard | Values for a modern hump yard | Values for a modern hump yard |
| | Reference | Halved frequency of braking/accelerations | Halved mileage |

The results for the three collectives are presented in Fig. 72:. The analysis is divided into the force range up to approx. 300 kN (red frame) and the force amplitudes above this level (blue frame). Analogous to the results for intermodal trains, halving the mileage and reducing the number of braking / acceleration processes per 100 km significantly reduces the frequency of force amplitudes in the force range up to 300 kN. Optimisation for longitudinal dynamics has a positive effect in the range from 100 kN to 300 kN. In the range of higher forces, the opposite picture emerges. Here, there are hardly any differences between the reference collective and the collective with a reduced number of braking and acceleration processes. This clearly shows that, in this range, the forces resulting from shunting impacts almost exclusively affect the collective. In the medium to high force range, it is therefore also very clear that optimisation for shunting impacts greatly reduces both the maximum amplitude and the frequency of forces.




| Bulk goods block train A | Bulk goods block train B | Bulk goods block train C |
|---|---|---|
| Reference | Halved frequency of braking/accelerations | Halved mileage |

**Fig. 72: Comparison of force collectives for bulk goods block trains**

Over the entire collective, halving the mileage has an effect on the frequency of the force amplitudes occurring, but not on the amplitude. The reason for this is that, in this analysis, both the number of operational manoeuvres and the frequency of stops in marshalling yards are linked to mileage.

### Single wagon trains (wagonload traffic)

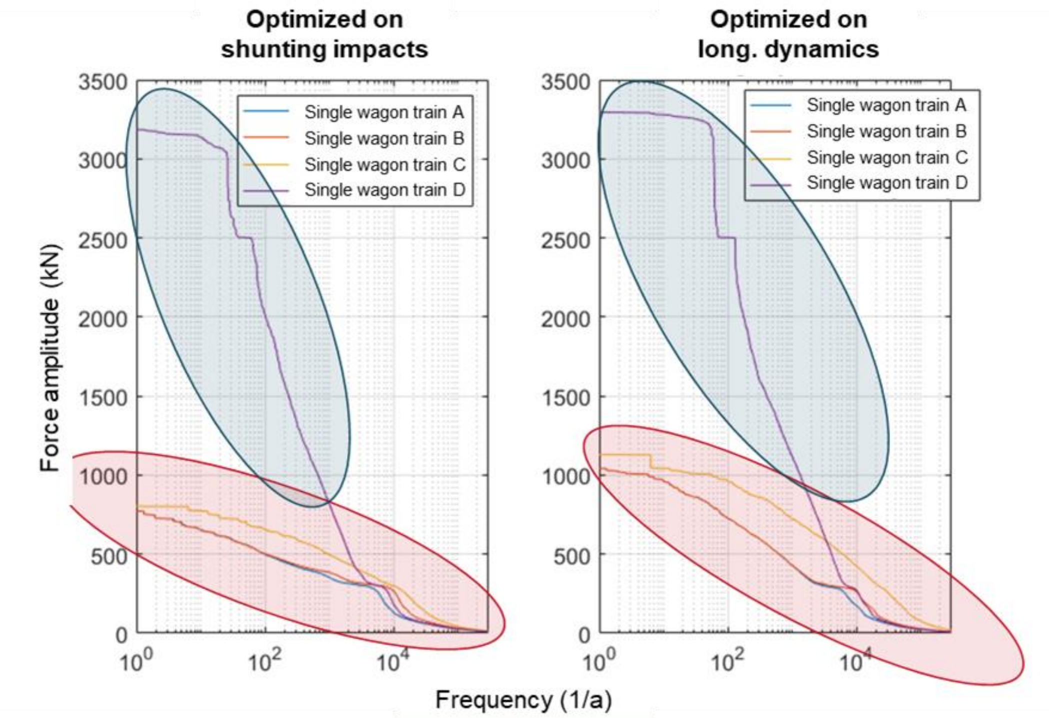
In contrast to intermodal and bulk goods block trains, it is not possible to make generalised statements about typical load states and arrangements of masses for trains in wagonload traffic – in the following referred to as single wagon trains. It is therefore assumed that all wagon load states – from empty to full – are equally probable in theory and that the distribution of wagons with different masses within trains is stochastically distributed. However, this also takes into account that there are often groups of wagons with similar loads. For single wagon trains, the analysis distinguishes between four scenarios, in which the number of acceleration and braking processes, the frequency of MY stops and the speed distribution of the impacts differ from the reference case. In single wagon train D, it is assumed that the speed distribution of the impacts is constant, i.e. in principle for each marshalling yard the train passes through, in accordance with UIC 524. An overview of the four scenarios can be found in Table 21.

**Table 21: Examples of operational programmes investigated for trains in wagonload traffic**

| | Single wagon train A | Single wagon train B | Single wagon train C | Single wagon train D |
|---|---|---|---|---|
| Type of wagon | E and/or F | E and/or F | E and/or F | E and/or F |
| Distribution of load states | All load states equally probable | All load states equally probable | All load states equally probable | All load states equally probable |
| Arrangement of masses in the train | Stochastically distributed (with groups) | Stochastically distributed (with groups) | Stochastically distributed (with groups) | Stochastically distributed (with groups) |
| Annual mileage of wagons | 50,000 km | 50,000 km | 50,000 km | 50,000 km |
| Frequency of starting/braking | IFW | IFW · 2 | IFW | IFW |
| Brake pad materials | Mean value composite brake pads | Mean value composite brake pads | Mean value composite brake pads | Mean value composite brake pads |
| Number of marshalling yard stops | every 1,000 km | every 1,000 km | every 100 km | every 1,000 km |
| Impact speeds in MY | Values for a modern hump yard | Values for a modern hump yard | Values for a modern hump yard | UIC 524 |
| | Reference | Doubled frequency of braking/accelerations | More frequent MY stops | Different speed distribution |

The comparison of the force collectives in Fig. 73: shows major differences between the four scenarios. Compared to the intermodal and bulk goods block trains considered above, these require an upward extension of the force range under consideration. A comparison of A, B and C shows that in the range up to 300 kN, a doubling of the acceleration and braking processes has a significant effect on the frequency of the forces occurring. Above 300 kN, only slight to no differences are visible. More frequent MY stops increase the frequencies over all force amplitudes. Optimisation for shunting impacts significantly reduces the effects (amplitudes/frequencies) of more frequent MY stops in the high force range, but also yields consistent advantages in the low force range.

Compared to the other three scenarios, the single wagon train D stands out clearly. In the range up to 300 kN the differences are less noticeable. However, above this level changing the speed distribution from that of a modern hump yard to a distribution according to UIC 524 has a dramatic effect. This confirms the findings presented in Chapters 6.3 and 6.4, i.e. that the speed distribution has a very large influence on the forces occurring. Optimising for shunting impacts can reduce the frequency of the forces but only achieves minor reductions in the maximum amplitudes that occur.




| Single wagon train A | Single wagon train B | Single wagon train C | Single wagon train D |
|---|---|---|---|
| Reference | Doubled frequency of braking/accelerations | More frequent MY stops | Different speed distribution |

**Fig. 73: Comparison of force collectives for trains in wagonload traffic**

Finally, the investigation of the operating scenarios compares examples of collectives for the three RFT segments under consideration. For this purpose, one train is selected as an example in each case under the condition that the three examples should differ in as many parameters as possible. This is in order to show the bandwidth. The intermodal train C, the bulk goods block train A and the single wagon train C were selected for this purpose. The three examples differ in terms of their mass distribution and arrangement, annual mileage and frequency of MY stops. The standard number of braking and acceleration processes per 100 km is selected in accordance with the IFW data. In all three cases, the impact speeds are based on the distribution for a modern hump yard. An overview is provided in Table 22.

**Table 22: Comparison of trains for intermodal, bulk goods and wagonload traffic**

| | Intermodal train C | Bulk goods block train A | Single wagon train C |
|---|---|---|---|
| Type of wagon | R or S | E and/or F | E and/or F |
| Distribution of load states | 30 % empty, 60 % half full, 10 % full | 50 % empty, 50 % full | All load states equally probable |
| Arrangement of masses in the train | Stochastically distributed (with groups) | Block train (all uniform) | Stochastically distributed (with groups) |
| Annual mileage of wagons | 200,000 km | 100,000 km | 50,000 km |
| Frequency of starting/braking | IFW | IFW | IFW |
| Brake pad materials | Mean value composite brake pads | Mean value composite brake pads | Mean value composite brake pads |
| Number of marshalling yard stops | every 10,000 km | every 1,000 km | every 100 km |
| Impact speeds in MY | Values for a modern hump yard | Values for a modern hump yard | Values for a modern hump yard |

The force collectives are presented in detail in Fig. 74:. The results differ widely depending on the parameter optimisation and the force range considered. If the coupling optimised for shunting impacts is first considered in the force range up to 300 kN, it is noticeable that the bulk goods block train has the lowest frequencies in this force range. With intermodal trains and single wagon trains, the higher mileage and the higher frequency of MY stops compensate for each other. Above 300 kN, the picture is different, with intermodal and bulk goods block trains initially showing a similar pattern. However, for forces greater than 400 kN, the more frequent MY stops and more frequent journeys in the fully loaded state become noticeable for the bulk goods block train. The frequencies and the maximum amplitude are significantly higher than those for the intermodal train. As the force amplitude increases, the results for the bulk goods block train and single wagon train start to converge. The maxima of the forces are in a similar range.

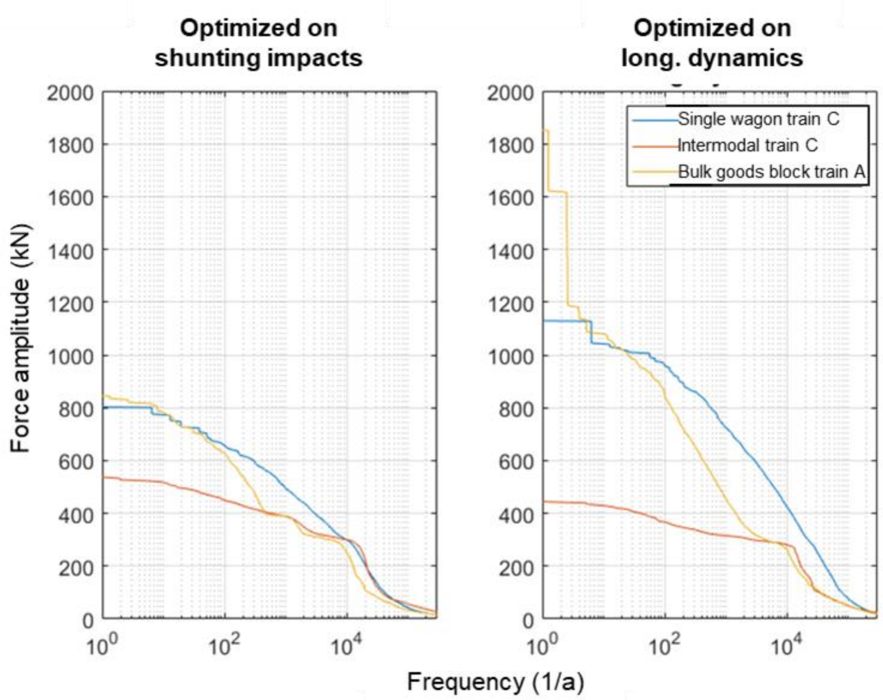



**Fig. 74: Comparison of the force collectives for intermodal, bulk goods and single wagon trains**

In comparison, the picture is somewhat different for a coupling optimised for longitudinal dynamics: in the force range up to 300 kN, the intermodal train with the higher mileage and the bulk goods block train with the higher frequency of MY stops show a similar pattern. The single wagon train is processed much more frequently and the effects of this are already very noticeable here. The frequencies are significantly higher than those in the other two scenarios. Above 300 kN, the behaviour is similar to that of the coupling optimised for shunting impacts: the collectives for intermodal and bulk goods block trains clearly diverge. There is a large difference between the maximum amplitudes. The collectives for the single wagon train and the bulk goods block train converge with increasing amplitude and at the maximum value, the bulk goods block train is even higher than the single wagon train. This is probably due to the type of mass distribution. Wagons in bulk goods block trains are fully loaded much more frequently, resulting in higher impact energies and thus higher forces during shunting impacts.

### 6.5.3 Interim conclusion

The investigations into operational stability identified the parameters which have the greatest influence on the form of the force collectives, i.e. on the amplitudes and frequencies of forces occurring. The following parameters have the greatest influence:

- Speed distribution during shunting impacts
- Frequency of MY stops
- Average group sizes when running down the hump
- Annual mileage of the wagons
- Design of the dynamic parameters

This underlines that both the limiting operational conditions in RFT and the design of the coupling itself have an influence on the force collectives. Train parameters (train length, mass arrangement etc.) play only a minor role compared to the factors mentioned above. Likewise, the routing – in this case the inclination and number of changes in inclination – is not particularly influential by comparison. Thus, the dynamic parameters can be designed independently of the location in which the wagons are used. The results show that reducing the impact speeds, e.g. by retrofitting modern equipment, would lead to significantly different and more favourable assumptions for the operational strength of the couplings.

However, if a factor is classified as having little influence in this global view, this does not mean that it cannot have greater significance in more specific cases. The sensitivity analysis relates to the presented ranges of the parameters. Limiting certain parameters to a smaller or larger range or shifting the ranges of individual parameters also changes the results. The more precisely the ranges can be limited, the more accurate the statements that can be made about the influence of this parameter as well as the other parameters

To draw conclusions about the optimisation of the dynamic parameters from these investigations, it is necessary to take a somewhat more differentiated view. Since the limiting operational conditions for the use of the DAC generally play a dominant role in the sensitivity analysis, a differentiation is made between the various RFT segments, which differ significantly in terms of their limiting conditions. Intermodal trains, bulk goods block trains and single wagon trains were analysed as examples based on assumptions for the respective operating programmes.

The results of the sensitivity analysis are clearly confirmed by the sample scenarios. At low force amplitudes, the wagon mileage and frequency of stops in marshalling yards have a major influence. At high and very high amplitudes, the number of stops in marshalling yards and the speed distribution play the dominant role. Optimisation of the spring parameters for the conditions found in certain types of traffic/production forms brings clear advantages and significantly reduces both the amplitudes that occur and the frequencies of forces. Optimisation for longitudinal dynamics has a clearly positive effect on traffic that rarely or never passes through a marshalling yard. In contrast, optimisation for shunting impacts has a very positive effect on traffic that passes through a marshalling yard regularly or very frequently.

In the three segments considered as examples, the differences in the limiting operational conditions are reflected in clear differences in the resulting force collectives. Consequently, it makes little sense to try to identify a design that can be used in all freight wagons regardless of the operating conditions. On the contrary, differentiating between several types of traffic when designing the dynamic parameters, as well as the operational stability of the coupling as a whole, offers great advantages. The more accurately the respective ranges of the parameters can be limited, the greater the scope for optimisation of the coupling with regard to its service life or weight/costs for the individual segments. The three segments presented here should be understood as examples. In a further analysis, it would make sense to evaluate as much data as possible on the limiting conditions that occur for individual wagons during operation. In this way, it may be possible to achieve a somewhat different, but certainly more targeted, differentiation for the interpretation of the DAC.

# 7. Final review

## 7.1 Conclusion

Within the framework of this study, the dynamic parameters of the DAC, which characterise the behaviour of the draft gears for the coupling, were examined from several viewpoints. The effects of the draft gear design were investigated with regard to the tolerable longitudinal compressive forces as an indicator of the running safety, on longitudinal dynamics of freight trains, on impacts during shunting processes and on the operational stability of the coupling. Multi-body simulation and Monte Carlo methods as well as sensitivity analysis were used for this purpose. The aim was to determine the significance of the dynamic parameters in the respective sub-investigations and identify which designs prove positive in which cases.

In accordance with the framework conditions of the study, the sub-investigation into the tolerable longitudinal compressive forces described in Chapter 6.1 focuses on four-axle freight wagons with Y25 bogies. The results show that the dynamic parameters overall have only a minor influence on the tolerable longitudinal compressive forces. However, the geometric parameters of the wagon and coupling play an important role, with the length of the coupling arm being particularly significant. The longest possible coupling arm is generally advantageous for the tolerable longitudinal compressive forces. Geometric combinations of specific wagons with specific coupling arm lengths will require investigations of the tolerable longitudinal compressive forces in each individual case to ensure adequate running safety. On average, the presence of a stabilisation joint for the DAC leads to significantly higher tolerable longitudinal compressive forces. Nevertheless, the investigation also shows cases with an unfavourable combination of the geometric parameters of coupling and wagon, where the tolerable forces are below the 600 kN required by UIC 530 1/2. Conversely, however, tolerable forces above 600 kN can also be achieved for a coupling without a stabilisation joint by selecting reasonable geometric parameters for the wagon and the coupling arm length. No generalised recommendation for or against a stabilisation joint can be derived from the results.

The longitudinal dynamics of the freight train and shunting impacts in marshalling yards are the decisive factors when designing the dynamic parameters for the DAC, as shown in Chapters 6.2 to 6.4. Under this objective, the two sub-investigations cannot be considered independently of each other. Buffers and draw gear optimised **exclusively** for longitudinal tensile dynamics would tend towards a rigid connection. This would include a high spring preload in both the tensile and compressive directions, ideally a degressive characteristic curve, short spring travel distances and the minimum degree of coupling slack. Buffers and draw gear optimised **exclusively** for powerful impacts between wagons in marshalling yards would provide very soft suspension and a high degree of energy absorption. This results from a low spring preload, a highly progressive characteristic curve in the static case and long spring travel distances. For high energy absorption at high dynamics, correspondingly steep increases in the dynamic characteristic curve are required. Under some circumstances, the highest possible end force is positive. The respective parameter values are thus diametrically opposed and there is a conflict of objectives between a coupling optimised for longitudinal dynamics and a coupling optimised for shunting impacts.

When the couplings with the respective optimised parameters are analysed in different scenarios, it is clear that the respective optimisations achieve the desired effect. In the case of the longitudinal tensile dynamics, the positive effect of the optimised coupling increases with the length of the train. The values of the forces occurring for the coupling optimised for longitudinal tensile dynamics are not always lower than those of the coupling optimised for impacts. However, the advantages become apparent with at high force levels, e.g. in the context of emergency braking. In the investigation of shunting impacts, the positive effect of the optimisation is particularly evident during wagon acceleration. For both weak and strong impacts, the advantages of the corresponding optimisation of the dynamic parameters are evident. However, the limiting conditions also play a decisive role: the speed distribution at the hump has a major influence on the impact forces. Due to the lack of operational data, a very wide range was considered here – with the result that this distribution is at least as important as the optimisation. For future investigations into the coupling design, it is essential that this distribution be determined on the basis of measured impact speeds from real-world operations.

The fact that the coupling parameters influence the result distributions means that the force collectives to be determined also depend to some extent on the coupling design. However, it is already clear that there cannot be **one** optimal design and that different conditions require different designs. The challenge is to identify the goal and limiting conditions under which this optimisation should take place. One important limiting condition is running safety, which must be proven based on a specified limit value. The design of the DAC must not result in a high probability of this value being exceeded significantly more often in operation. The analysis of shunting impacts reveals that, if the dynamic parameters are chosen unfavourably in conjunction with unfavourable framework conditions, such as very high impact speeds and high masses of the vehicles involved, forces occur which are very likely to lead directly to damage of the coupling, wagon and load. These cases must be excluded as far as possible by operational and technical measures.

As long as these two limiting conditions are guaranteed, operational stability plays the decisive role in the design. The objectives are either the longest possible service life at a defined weight/cost of the coupling, or the lightest possible (and thus more cost-effective) coupling design, assuming a certain service life. The investigation of the operational stability is based on the previous investigations of shunting impacts and longitudinal dynamics.

Another way to solve the conflict of objectives between shunting impacts and longitudinal dynamics is to change the technical and/or operational framework conditions in rail freight transport. One possibility here is to use the EP brake, which significantly reduces the longitudinal compressive forces that occur. This applies both to couplings optimised for longitudinal dynamics and couplings optimised for shunting impacts. The reduction in longitudinal compressive forces allows either the dynamic parameters to be optimised for shunting impacts, or the development of new train and operating concepts with longer, heavier trains and the same degree of running safety.

The sub-investigation into operational stability described in Chapter 6.5 focuses on the operational conditions in rail freight traffic and their influence on the force collectives to which the coupling is exposed during operation. The conflicting optimisations for longitudinal dynamics or shunting impacts are investigated here. The parameters with the greatest influence on the force collectives are the speed distribution of the shunting impacts, the number or frequency of marshalling yard stops, the average size of wagon groups running down the hump, the annual mileage of the wagons and, last but not least, the spring design itself. Since the limiting operational conditions are of paramount importance, the key force collectives for the design also change depending on the operating programme under consideration. In particular, a very broad spectrum of speed distributions was considered for shunting impacts – ranging from modern hump yards to data from UIC leaflet 524, which is more than 40 years old. The frequencies and amplitudes of the forces occurring can be significantly reduced by optimising the dynamic parameters of the DAC to the requirements of specific types of traffic. In terms of operational stability, this allows the design to be tailored either for a longer service life or a lighter weight, depending on the goal. However, optimisation requires that the limiting operational conditions can be determined as accurately as possible. The wider the range of certain parameters – especially if they are not based on representative values – the less effective the parameter optimisation will be. It is therefore extremely important to determine the limiting operational conditions that actually arise during operation as preparatory work for the design, e.g. the distribution of impact speeds in marshalling yards. The more assumptions that can be supplemented or replaced by operational data within the framework of this study, the more precisely an optimisation can be carried out with regard to operational stability.

However, a targeted delimitation or clustering into several RFT production segments, in which the boundary conditions are as similar as possible, should be undertaken. The range for these individual segments can be more precisely defined compared to the overall range in rail freight traffic. This allows optimisation of the dynamic parameters to be targeted more effectively – and will result in several variants of the DAC and/or its draft gears. However, later when the wagons are in operation, it will not be 100% necessary to guarantee that the wagons are always coupled with other couplings of the same design. The use of wagons fitted with couplings of different classes is not considered critical for short- and medium length trains. Further investigation may be required to assess the use of different couplings in long trains. This study examined three RFT segments: intermodal transport, bulk goods block trains and wagonload traffic. The differences in their respective limiting conditions are also reflected in their clearly different force collectives. Here, the advantages of the respective optimisation of the DAC are particularly evident: in intermodal transport, where wagons have a high annual mileage and rarely pass through marshalling yards, a coupling that tends to be optimised for longitudinal dynamics has proven advantageous. In wagonload traffic, where wagons are processed many times per year and tend to have a rather low mileage, optimisation for shunting impacts has clear advantages.

The force collectives shown in Fig. 71: to Fig. 74: can be regarded as an indication of the force collectives to be expected in operation. The force collectives shown here provide a reference point and highlight trends, but are not sufficient for use in design specifications. Force collectives in general should be supplemented with measurement data from actual operations, especially if they are based on simulation data [35]. The path to identifying the most realistic force collectives possible should incorporate a combination of measurement data and simulation technology. Simulation technology cannot foresee every operational situation. Conversely, not every conceivable case that can be investigated by simulation can be measured in operation. Combining this with the transfer from force to stress collectives and the associated possibility of quantifying damage to the coupling using the forces and amplitudes that occur not only enables optimisation for existing traffic, but also for other future scenarios.
The results of this study indicate a need for further research in various fields, which will be explained in more detail in the following chapter.

## 7.2 Outlook and further research

### 7.2.1 Collection of statistical operational data to enhance the analysis of operational stability

The data on braking and acceleration processes used in the investigation of operational stability were obtained within the framework of the project “Construction and testing of innovative freight wagons”. The analysis of the impacts that occur was essentially based exclusively on assumptions. Consequently, for example, the speed distribution of the impacts had to cover a large range. To arrive at realistic force collectives for designing the DAC, representative operational data should be systematically collected and evaluated for various types of traffic, wagons and countries. The focus should be on the most important parameters identified in this study. The more data that can be combined from different sources, the more accurate the overall picture that can be created using statistical methods together with an overview of relationships in the rail freight sector. For this reason, it is not essential to collect all data at once. The more data that are available, the better the approximation of real conditions will be. These data can be used to refine the simulation tool still further, since only a fraction of the cases that can be simulated can be also measured in reality. This combination of measurement and simulation makes it possible to define force collectives, and indirectly also stress collectives, ever more precisely. Based on the parameters identified as being most important and other limiting conditions (e.g. sensitive loads), trains can be divided into classes that require different interpretations of the dynamic parameters.

### 7.2.2 Transfer from force to stress collectives based on specific technical implementations of the DAC

Statements about the actual operational stability or service life can only be made based on stress collectives. For stress collectives, various standard procedures are available for calculating the damage caused by a collective (damage accumulation hypotheses). The calculation of stress collectives enables the collective to be reduced to an equivalent alternating load. This, in turn, can be used to perform a targeted optimisation of the dynamic parameters within a defined range of operational parameters with the aim of achieving the longest possible service life or lightweight construction/low costs [34, 35]. For this purpose, it is necessary to know specific design parameters of the coupling, such as the relevant cross-sections or materials. This investigation requires a large number of new simulations. Building on the models and simulation controls developed in this study, however, this can be done with relatively little work since the simulations can be largely automated. Based on these simulations, the analysis can be transferred from force to stress collectives.

### 7.2.3 More detailed analysis of tolerable longitudinal compressive forces of freight wagons

The original motivation for the introduction of the stabilisation joint was, among other things, the lower tolerable longitudinal compressive forces of two-axle freight wagons compared to bogie freight wagons [57]. In many European countries, two-axle freight wagons still account for a significant proportion of rolling stock[16]. In order to be able to make robust statements on the necessity of a stabilisation joint, the investigation of tolerable longitudinal compressive forces should therefore be extended to two-axle wagons. This requires a new MBS model for two-axle freight wagons with leaf springs

("UIC-Link"). The simulation environment and coupling models from this study can be used for this purpose.

The overriding question, however, is which test method should be used in future to determine the tolerable longitudinal compressive forces. The simulations also show that the choice of frame wagons and the track geometry used in the tests have a major influence on the results. Wherever possible, the test procedure should reflect conditions that may actually occur for a particular wagon in operation and not an unrealistic superimposition of worst-case scenarios. This is dependent on the longitudinal compressive forces caused by the longitudinal dynamics of freight trains, with emergency braking being of particular importance. The investigations of longitudinal dynamics for trains with conventional, purely pneumatic brakes show that maximum forces of less than 600 kN occur in 99% of the simulated scenarios for emergency braking. Here, only trains that correspond to the current limiting conditions of rail freight traffic in Europe in terms of train length and maximum mass are considered. In this case, the limit value seems to be well chosen. However, if the results using an EP brake are considered, the longitudinal compressive forces that occur during emergency braking are less than half of this level in 99% of cases. One possible conclusion to be drawn from this, while maintaining the current general conditions, is that a reduction in the requirements for running safety with regard to the tolerable longitudinal compressive forces may be appropriate.

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# Annex

## A Presentation of cumulative frequencies

Information on cumulative frequencies is often used for the presentation of results in the context of this study. This form of presentation from the descriptive statistics allows statements to be made about how a certain parameter affects a certain result variable. The type of presentation is shown in Fig. 75: using the example of the parameter *spring preload* in the compressive direction for a specific operating situation, in this case emergency braking.

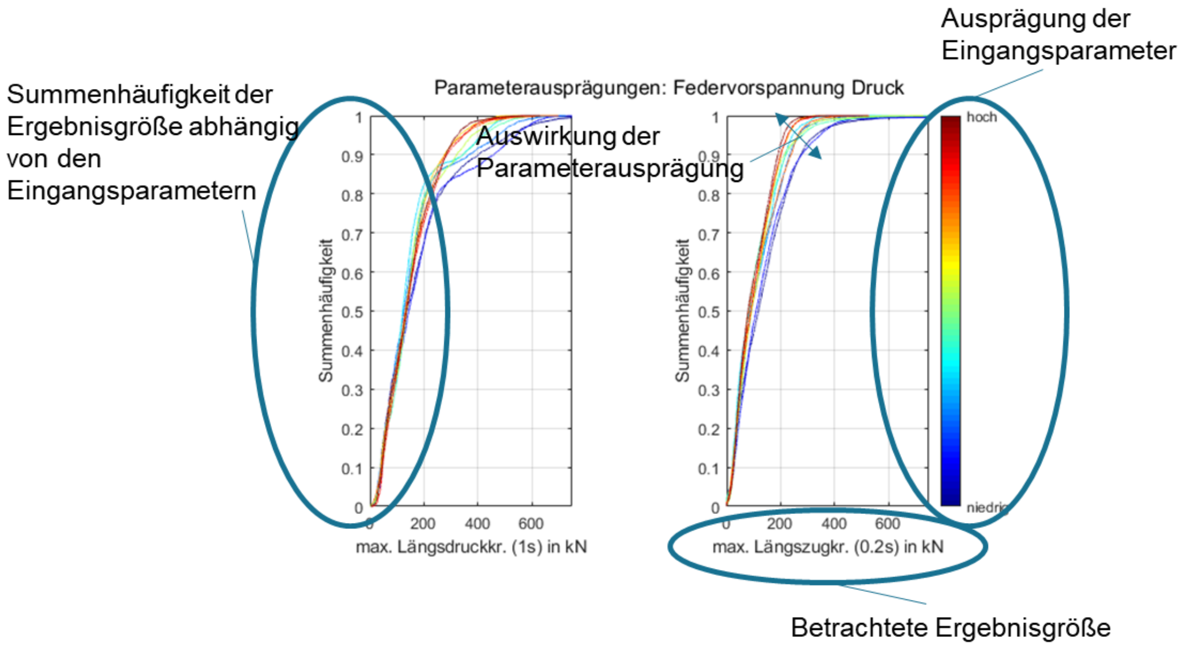



**Fig. 75: Sample presentation of cumulative frequencies**

The graphs of the cumulative frequencies show which proportion of the input values (in this case the simulated scenarios for the operating scenario emergency braking) leads to a result (in this case the maximum longitudinal compressive force or longitudinal tensile force) that is less than or equal to a certain value. Accordingly, the cumulative frequency for the largest occurring value is 1 or 100%. With an additional differentiation of the input values into defined classes, it is also possible to see which characteristic of a certain parameter affects the output variable in which way. However, only one parameter of a scenario can be considered at a time, e.g. the spring preload in the example here. The scenarios were divided into nine different classes based on the characteristics of this parameter. The lowest class contains those scenarios in which the value of the parameter is in the lower ninth of the value range for this parameter. Conversely, the highest class contains the scenarios in which the value belongs to the highest ninth of the value range. The remaining scenarios are also classified according to their level. The cumulative frequencies are now plotted by classification as shown in Fig. 75:. Using these graphs, it is possible not only to make statements about the importance of a parameter, but also about particularly advantageous characteristics. If the graphs of the different classes are far apart, it can be said that this parameter has a major influence on the distribution of the results. In an extreme case where all the graphs are superimposed, the parameter has no influence on the results. To evaluate the characteristic, we now look at how the graphs lie in relation to each other. The question here is which characteristic is positive in the particular context. In the example, these are the lowest possible longitudinal compressive and longitudinal tensile forces, i.e. a value for the spring preload becomes more advantageous the further to the left the corresponding graph lies in the diagram. In individual cases, however, graphs can also cross over. This means that a certain characteristic is advantageous up to a certain force level, while a different characteristic is advantageous at another force level.

# B High resolution images

**Fig. 57**:

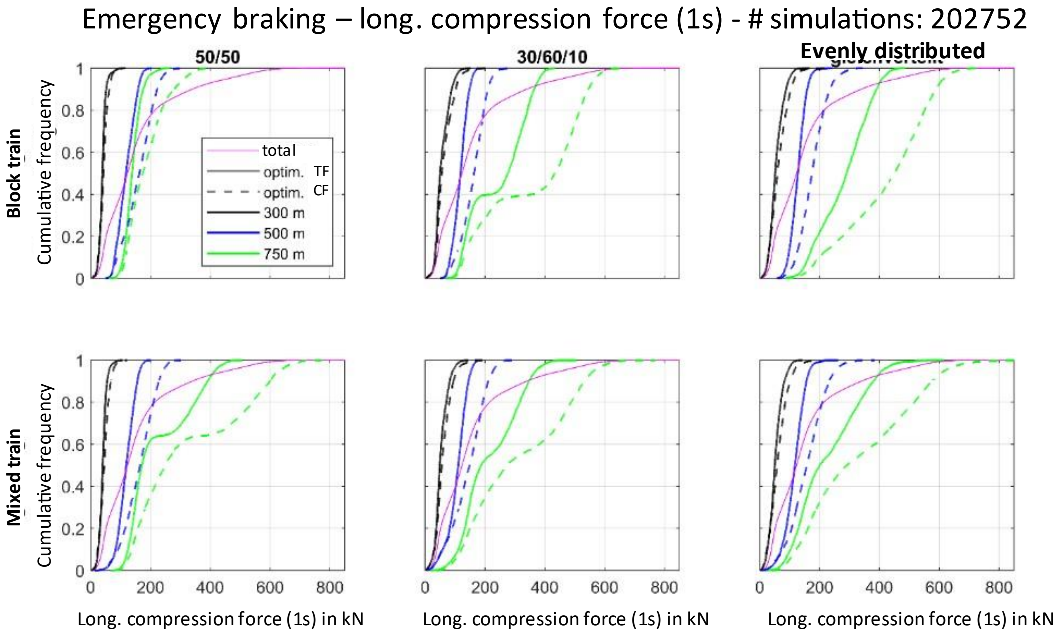



**Fig. 58**:

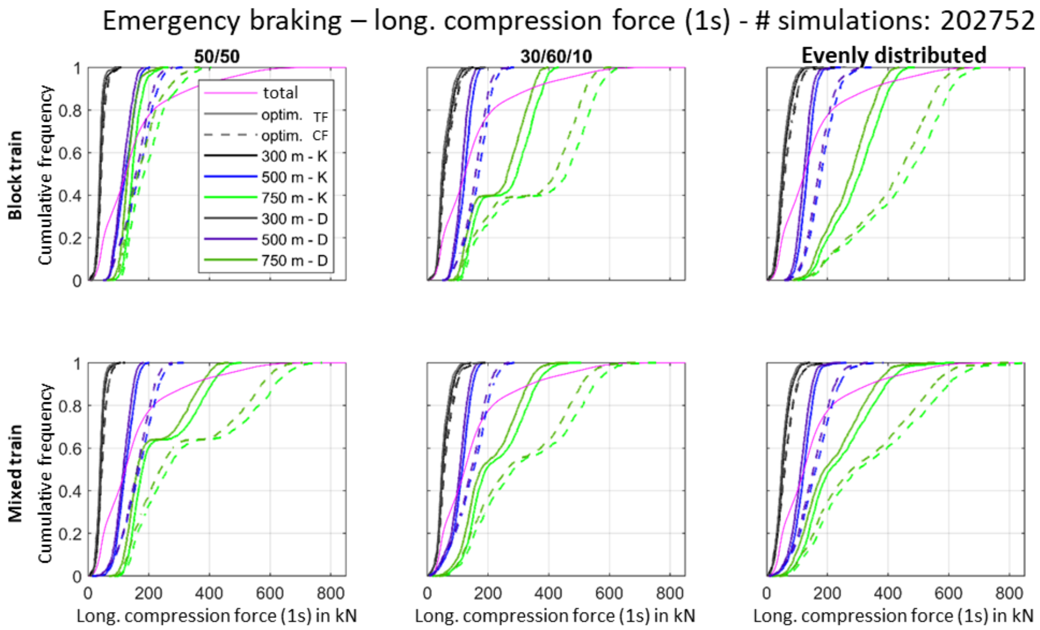
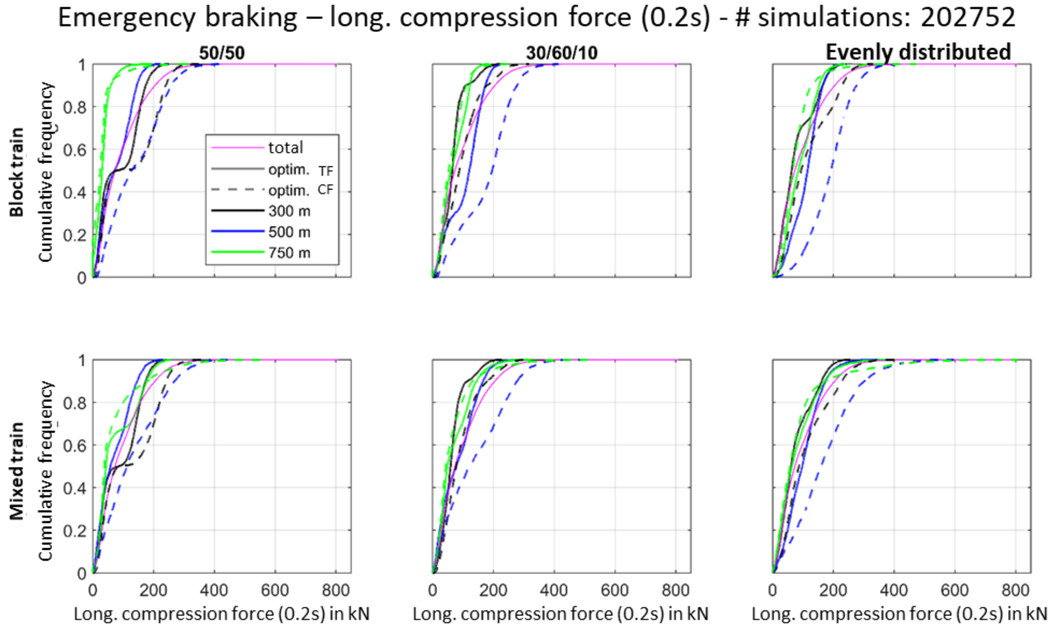
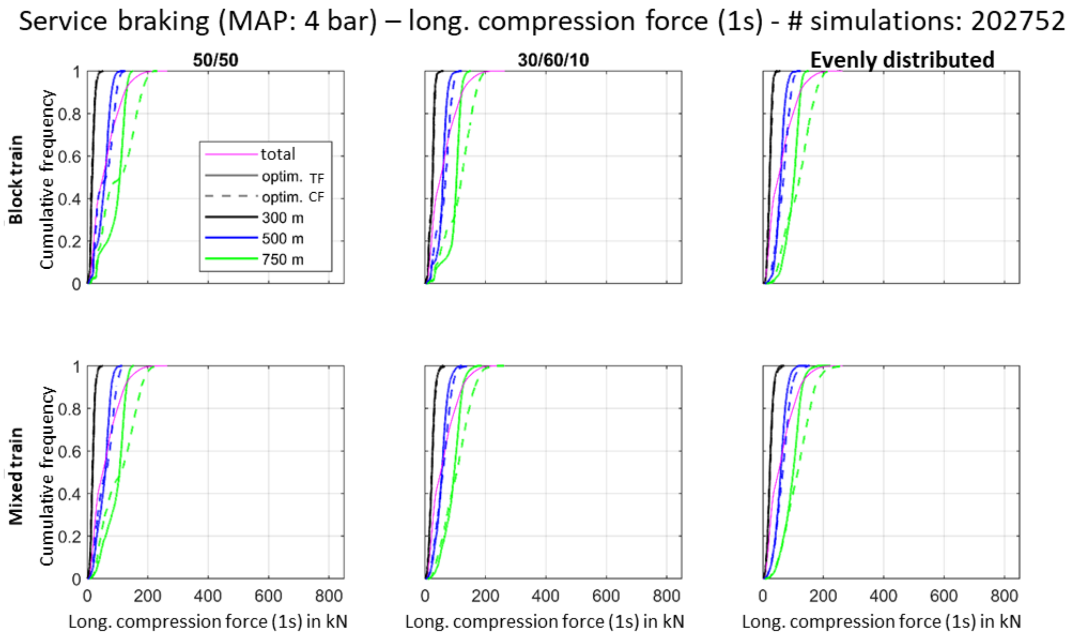


**Fig. 59**:



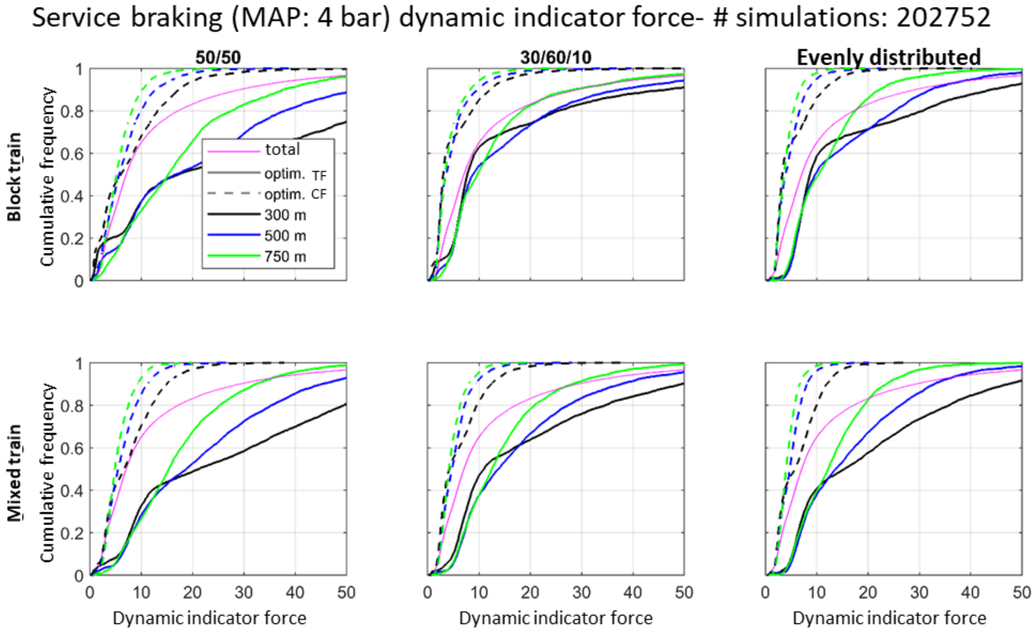
Emergency braking – long. compression force (0.2s) - # simulations: 202752
50/50
30/60/10
Evenly distributed
Block train
Mixed train
Cumulative frequency
total
optim. TF
optim. CF
300 m
500 m
750 m
Long. compression force (0.2s) in kN


**Fig. 60**:



Service braking (MAP: 4 bar) – long. compression force (1s) - # simulations: 202752
50/50
30/60/10
Evenly distributed
Block train
Mixed train
Cumulative frequency
total
optim. TF
optim. CF
300 m
500 m
750 m
Long. compression force (1s) in kN

**Fig. 61**:


Service braking (MAP: 4 bar) dynamic indicator force- # simulations: 202752
50/50
30/60/10
Evenly distributed
Block train
Mixed train
Cumulative frequency
total
optim. TF
optim. CF
300 m
500 m
750 m
Dynamic indicator force