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Creation of synthetic motor torques and brake forces for determination of design loads for railway vehicle bogies

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ABSTRACT

To develop light and safe components of railway vehicle bogies, precise design loads are required. These loads are calculated using multibody simulation. For the simulation, different time signals including speed, motor torque or brake force are used. Usually, every required signal can be taken from measurements of railway vehicle. However, motor torque and brake force strongly depend on the vehicle. For the simulations of new vehicles it is not possible to use these time signals. Therefore, an approach for the creation of synthetic motor torques and brake forces is presented in this contribution. To this end, correlations between acceleration and motor torque respectively brake force are identified. The calculation of synthetic signals of motor torque and brake force is based on these correlations. It is possible to calculate the corresponding motor torque and brake force value for every arbitrary acceleration. A deceleration is achieved using motor and brake individually or combined. Hence, it is not possible to decide which components are used for the deceleration. Thus, a relative frequency distribution is created. In this distribution it is counted how often the different deceleration possibilities were used. To generate signals in the deceleration sections as well, the described distribution is used. To test the methodology, two simulations are carried out using the same multibody model of a railway vehicle. For the second simulation, the measured motor torque and brake force are replaced by the synthetic signals. The simulation results are evaluated and compared.

Keywords: Railway Vehicle, Design Loads, Mutibody Simulation.

1 INTRODUCTION

Rail transport can help to reduce the CO_2-Emissions in traffic [1]. A reduction of CO_2 and energy consumption can be achieved using light weight design for rail vehicles. A requirement for safe and light constructions are precise design loads. Design loads should be known as early as possible during the design phase. The loads depend on physical quantities of the vehcile such as mass, stiffnes or damping coefficients. These quantities are different for different vehicles. Therefore, the use of already measured loads is limited. Measurements and calculations are part of the validation process for bogie components following DIN EN 13749 [2]. Nowadays, multibody simulation (MBS) is used for the calculations and for the determination of precise design loads [3]. Using MBS for determining the design loads is also suggested in VDV152 [4].
In order to conduct the simulations a simulation scenario together with the MBS-model is required. A scenario generally consists of track geometry, track irregularities and time functions such as motor torque or brake force [5]. Scenarios can be created using a methodology based on measurements. This methodology is presented in [6] and [7]. In opposite to the track geometry and track irregularities, motor torques and brake forces depend on the vehicle itself. Hence, only

the track geometry can be used directly for the simulations of new rail vehicles. The approach for the calculation of these time functions is presented in this paper. In chapter 2 the methodology is described. Subsequently, in chapter 3 the methodology is validated. For this purpose a commuter train with a maximum velocity of 160 $\frac{km}{h}$ is used. The paper ends with the conclusion and outlook in chapter 4.

2 METHODOLOGY

The creation of synthetic motor torques and brake forces is based on quantified relationships between acceleration, speed and motor torque or brake force. To identify these relations measured data are required. It is crucial that these measurements (M1) were conducted with a vehicle which has similar physical characteristics as the vehicle to be designed. Modern rail vehicles can use besides the brake, the motor and both components in combination for deceleration. Thus, the mathematical relationships have to be set up for five different driving states.

- *Acceleration*
- *Constant Speed*
- *Deceleration* with motor (*Retardation*)
- *Deceleration* with brake (*Braking*)
- *Deceleration* with motor and brake (*Hybrid Braking*)

The time functions of synthetic motor torque and brake force are calculated with given acceleration and speed signals. These accelerations signals depend on the simulation scenarios for the design loads. According to chapter 1 these scenarios are created using measurements. In this publication, these measurements will be called M2. With the given accelerations of M2 it is only possible to identify the driving states *Acceleration*, *Constant Speed* and *Deceleration*. A specific identification of the driving states *Retardation*, *Braking* and *Hybrid Braking* is not possible. Therefore, the decisicion with which components the deceleration will be realized is based on a frequency distribution of the different deceleration states. This distribution is called braking history and is created using M1.
Figure 1 shows the flow chart diagram of the methodology.

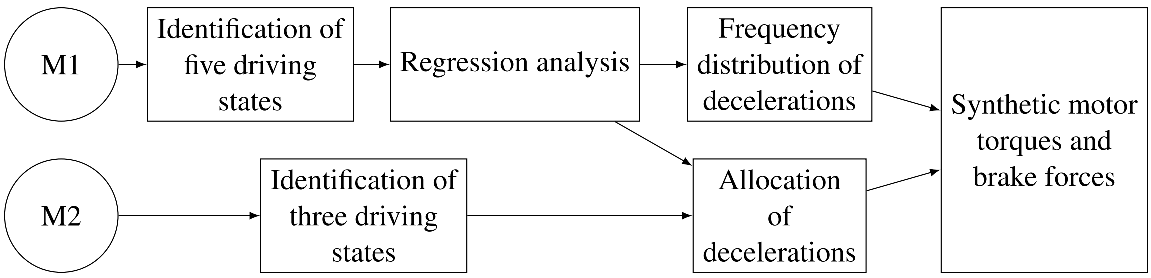

Figure 1. Flow chart diagram of the methodology

2.1 Driving states and regression analysis

The first step of the methodology is the identification of the driving states. To identify these states the accelerations of M1 are taken into account. A positive acceleration signal is an *Acceleration*. For the state *Constant Speed*, the acceleration has to be roughly zero. If the acceleration signal is negative the motor torque signal and the brake force signal have to be analysed. If motor torque and brake force are not equal to zero, the state is *Hybrid Braking*. If only one of the signals motor torque or brake force is zero during *Deceleration*, it is *Retardation* respectively *Braking*. The measurements of M1 were conducted in positive and negative driving direction. The explanation above is based on a positive direction. In case of a negative direction the signal of every time signal will be multiplied with -1. Consequently, an *Acceleration* can also have a negative acceleration and motor torque signal. This is also true for the other driving states. As an example, the identified states of acceleration, motor torque and acceleration of one test ride of measurements M1 are shown in figure 2. The driving direction of this test ride is positive.

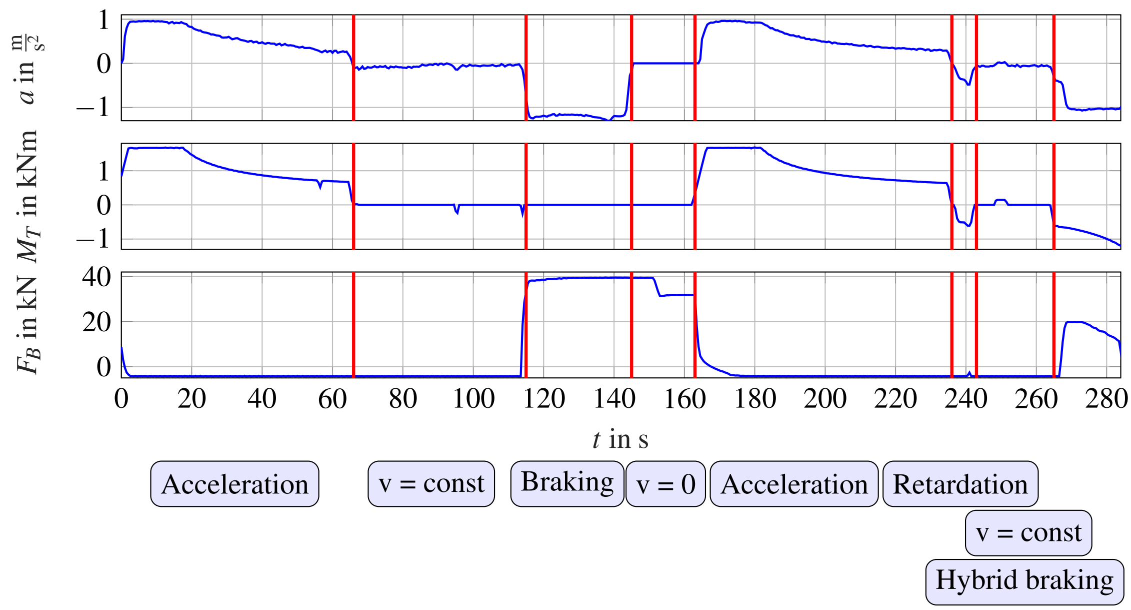

Figure 2. Identified driving states for acceleration, motor torque and brake force

To set up the relationships between acceleration, motor torque and brake force the driving states cannot be used directly, because the states have different lengths. In this case, very short driving states would have the same weight as longer states. To avoid this, the driving states are subdivided into windows with a length of 0.5 seconds. For every window the mean is calculated. It is assumed, that the relation between acceleration and motor torque respectively brake force is linear. Thus, the Bravais-Pearson coefficient is calculated in order to quantify the linear correlation [8].

Scatter plots, the Bravais-Pearson coefficient R and linear regression functions for the driving states *Acceleration*, *Retardation*, *Braking* and *Hybrid Braking* are shown in figure 3. The linear correlations between acceleration, motor torque and brake force are clearly identifiable. Very strong correlations have the driving states *Acceleration* and *Retardation* with $R = 0.99$ and $R = 0.98$. Strong correlations have *Braking* and *Hybrid Braking* with $R = 0.78$, $R = 0.89$ and $R = 0.81$. Furthermore, for *Braking* it can be seen that there are less data points. This indicates that the motor and the motor and brake in combination are used more often for *Deceleration* compared to the brake.

The last driving state which has to be desribed quantitatively is *Constant Speed*. For this driving state a relative frequency distribution is created (see figure 3). The motor torque is used in order to hold the required speed. It can bee seen that for every speed class the holding torque is often very small.

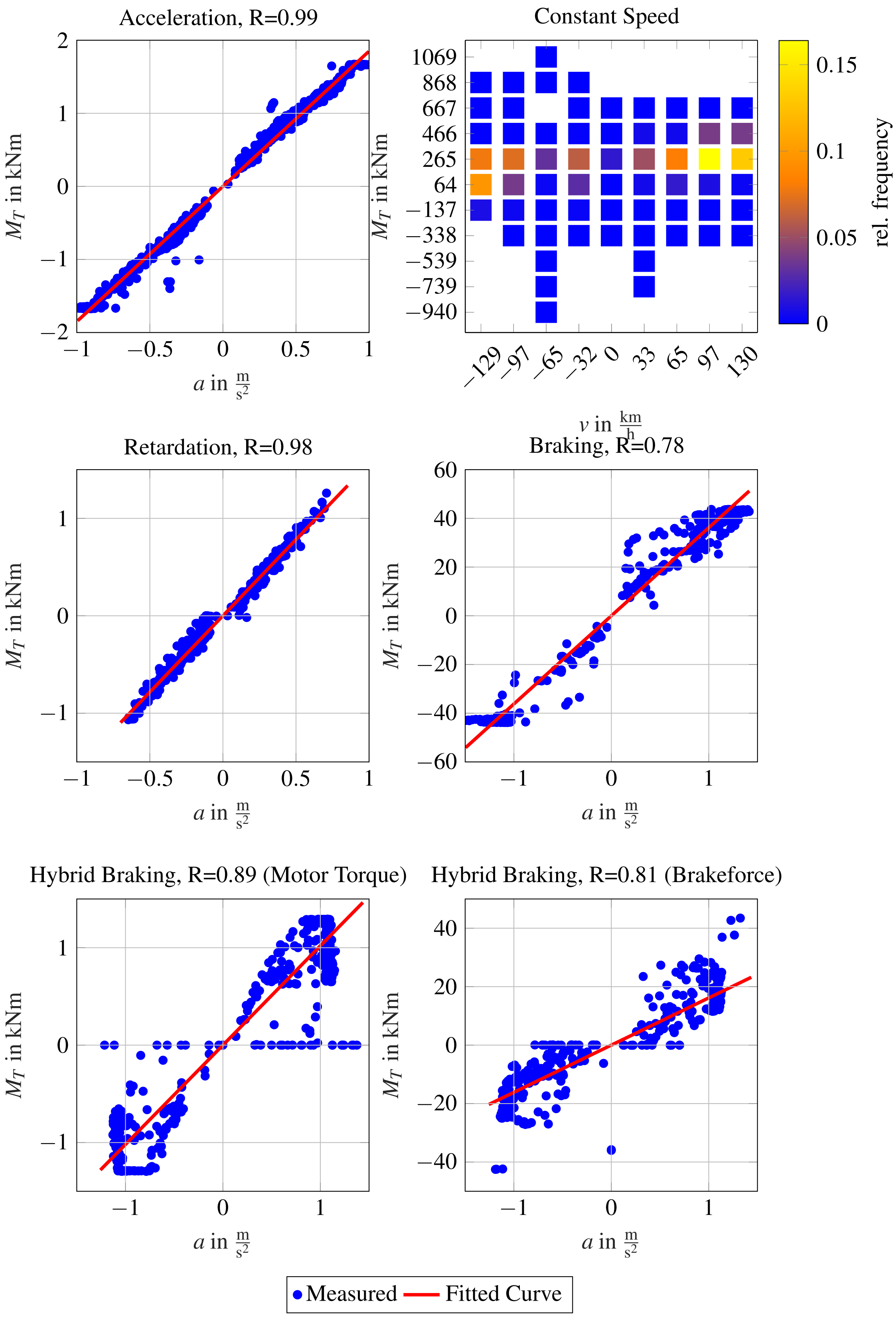

Figure 3. Scatter plots and relative frequency distribution for the driving states

2.2 Braking history

With the synthetic motor torques and brake forces the train should reach the given acceleration signals of measurements M2. It is not possible to identify which deceleration state was used only with the acceleration signal. Therefore, the braking history is created (see figure 4). The classification of the maximum absolute deceleration of each *Deceleration* state is on the x-axis. In total there are ten classes for the deceleration between $0\,\frac{m}{s^2}$ and $1.5\,\frac{m}{s^2}$. For each deceleration class the numbers of the states are counted. It can be seen, that for lower decelerations (class 1 to class 5) the deceleration is mostly realized by *Retardation*. For medium decelerations between class six and eight the hybrid braking have the highest share. In classes nine and ten the relative frequency of the *Braking* increases.

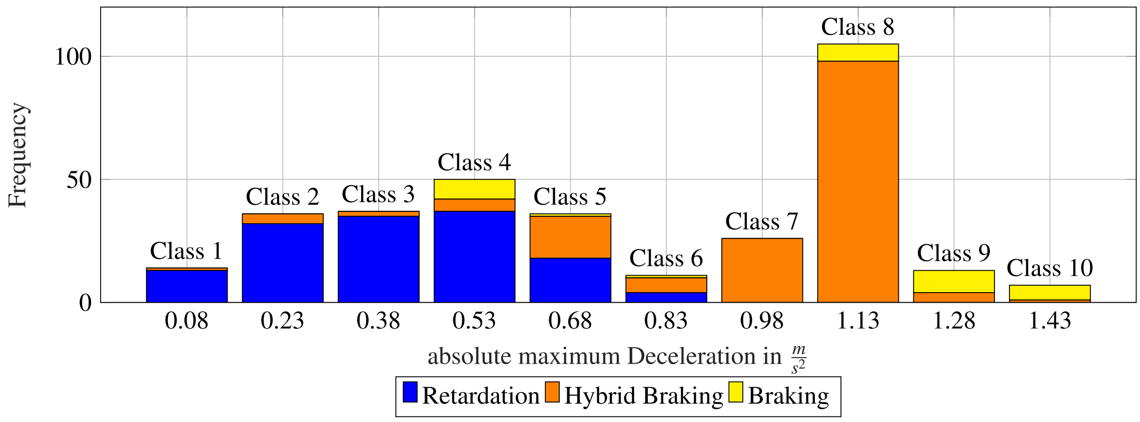

Figure 4. Braking history

2.3 Calculation of synthetic motor torques and brake forces

To calculate the signals of motor torque and brake force according to figure 1, the driving states *Acceleration*, *Constant Speed* and *Deceleration* have to be identified for the required accelerations of measurements M2. The identification is analogous to the identification presented in chapter 2.1. Furthermore, the three deceleration states *Retardation*, *Hybrid Braking* and *Braking* have to be allocated. Hence, the distribution of figure 4 is taken into account. It means, that for instance the most *Decelerations* in class 4 will be *Retardation*. Nevertheless, the states *Hybrid Braking* and *Braking* will also be allocated in this class.
After the allocation, the calculation of synthetic motor torque brake forces can be started using the linear regression functions and the relative frequency distribution of figure 3. With the linear regression functions the motor torques respectively brake forces can be caluclated easily for the driving states *Acceleration*, *Retardation* and *Braking*. In case of *Constant Speed*, the relative frequeny distribution is used (see figure 3).
For *Hybrid Braking* equations 1 to 4 have to be calculated. The gradient of the linear regression functions is called m. At first, the motor torque $M_{T,Hyb}$ is calucalted using the required acceleration a and $m_{T,Hyb}$ (equation 1). This motor torque is used to calculate the deceleration $a_{MT,Hyb}$ in case of *Retardation* (equation 2). With this theoretic deceleration generated by the motor torque the required deceleration generated by the brake force $a_{FB,Hyb}$ is determined (equation 3). The last step is the calculation of the brake force $F_{B,Hyb}$ for *Hybrid Braking* (equation 4).

$$M_{T,Hyb} = a * m_{T,Hyb} \tag{1}$$

$$a_{MT,Hyb} = \frac{M_{T,Hyb}}{m_{MT,Ret}} \tag{2}$$

$$a_{FB,Hyb} = a - a_{MT,Hyb} \quad (3)$$

$$F_{B,Hyb} = a_{FB,Hyb} * m_{FB,Hyb} \quad (4)$$

In figure 5 the time signals of measured (blue) and synthetic (orange) motor torque and brake force can be seen. In general, the synthetic motor torque is very accurate compared to the reference. The synthetic brake forces show in most sections a good agreement as well. However, there are still some differences between the brake force signal which have to be explained. Peaks in the reference brake force with forces higher than $25\,kN$ are sections where the speed is zero. In this cases the brake is used as a park brake. These time sections are not relevant for the simulation results. In the middle of the time signals are two sections where differences between reference and synthetic signal can be seen. In this sections the value of synthetic brake force is roughly $5\,kN$. At the same time the motor torque is not zero. Consequently, these two sections are the driving state *Hybrid Braking*. Especially the measured brake force has significantly higher values than the synthetic signal.

Figure 5 confirms the results from figure 3. Especially, the correlation of the states *Acceleration* and *Retardation* have stronger correlations than *Hybrid Braking*. This can be seen aswell in figure 5.

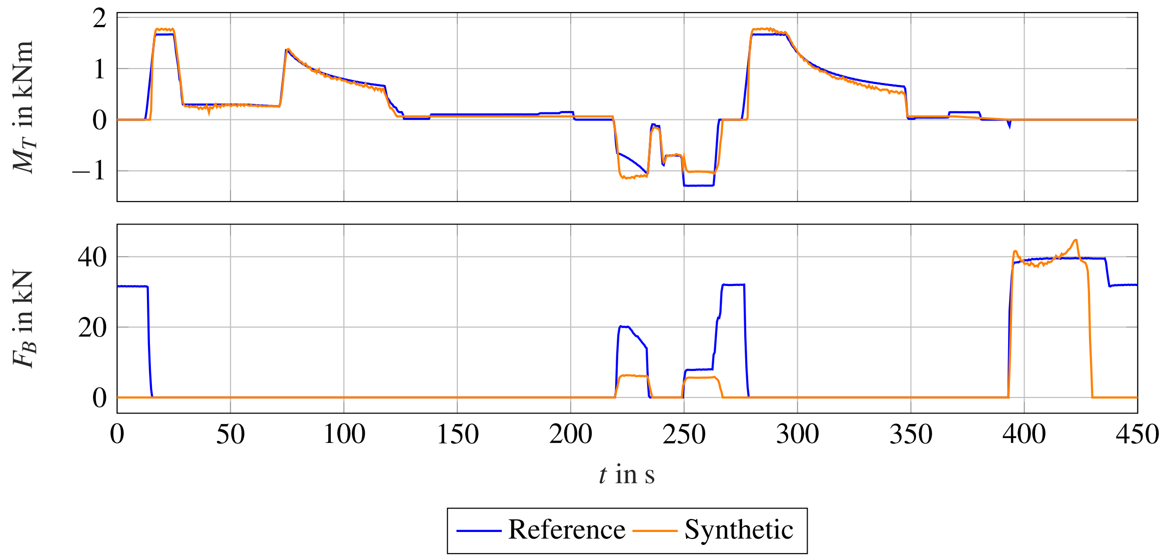

Figure 5. Time signals of measured and synthetic motor torque and brake force

3 RESULTS

In general, the presented time signals have a strong agreement. However, for design loads it is not sufficient to use only time signals. The methodology has to be tested in terms of damage as well. To take the damage into account, vibration amplitudes of several force elements are important. To this end, rainflow counting is used to identify the vibrations [9]. The vibrations and their corresponding frequencies can be displayed in the load spectra. For a better overview, the load spectra can be reduced to the damage equivalence amplitude. In terms of damage the equivalence amplitude is exactly the same as the corresponding load spectra [10].

To test the methodology reference and validation simulations were conducted. The scenarios of these two simulations are completely the same in terms of speed, acceleration, track layout and track irregularites. The only difference is the use of synthetic motor torques and brake forces in the validation simulations. Therefore, any differences in the results have to be explained with synthetic motor torques and brake forces. The simulations were conducted with a MBS-model of a regional train. The total distance of the simulations is $550\,km$.

In figure 6 the equivalence amplitudes of both simulations and the relative difference of the amplitudes can bee seen. For this figure the force elements of primary spring, primary damper, yaw damper, lateral damper, torque support and anti rollbar were analysed. In general, the relative differences are very small. The absolute maximum difference shows the yaw damper with $-2.3\,\%$. Their main task is to damp the rotation about the vertical axis between coach and bogie. Nevertheless, the yaw damper is not influenced by motor torque or brake force.

The torque support has a direct relation to the motor torque. This element supports the gear box on the bogie frame so that the complete motor torque can be transmitted to the wheelset. For that reason, any change in motor torque will result in a change in the torque support force. However, the relative differences of torque support 1 and 2 are very small with $1\,\%$ and $0.1\,\%$.

Nevertheless, the equivalence amplitude is only one value. Thus, the load spectra of the torque support forces are taken into account aswell. The load spectra can bee seen in figure 7. In the lower parts the loads spectra are roughly the same. Higher amplitudes show only small differences between reference and validation. Due to the very small differences in load spectra and the equivalence amplitudes the functionality of the methodology is confirmed.

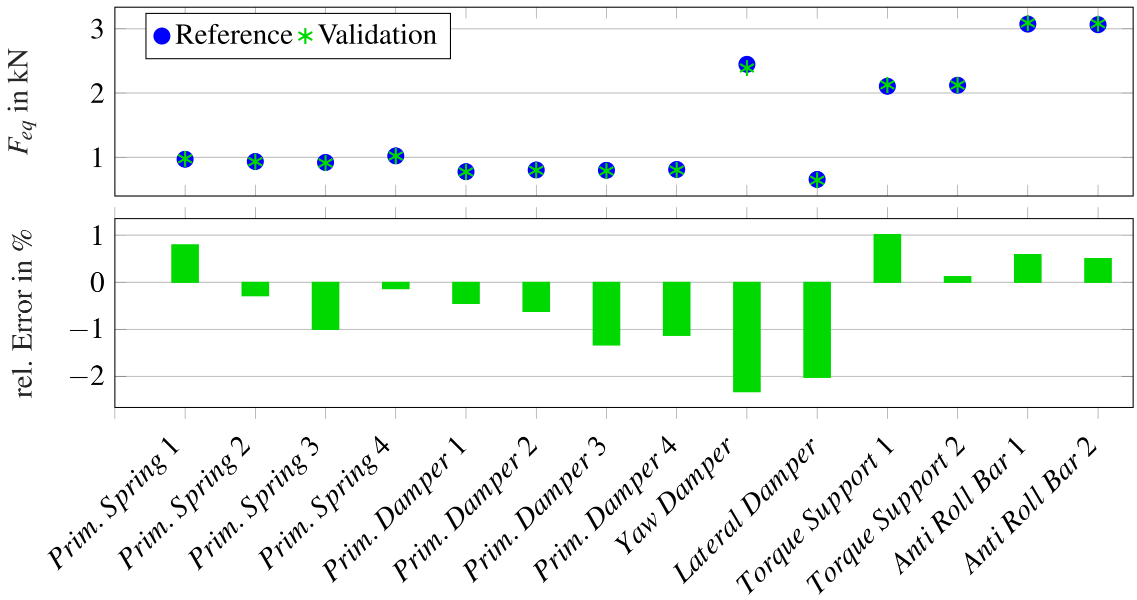

Figure 6. Equivalence amplitudes of reference and validation simulations

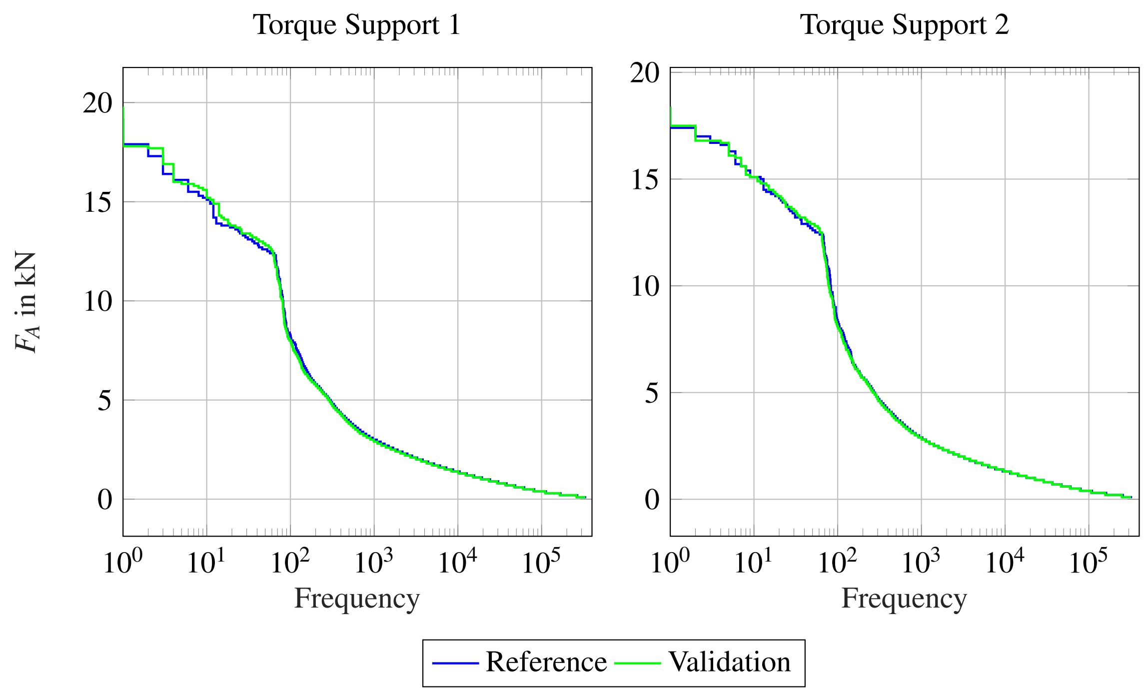

Figure 7. Load spectra of torque support forces 1 and 2

4 CONCLUSION AND OUTLOOK

In this paper a methodology for the creation of synthetic motor torques and brake forces has been presented. With linear regression functions and a relative frequency distribution, mathematical relationships between speed, acceleration, motor torque and brake force have been created. These relationships were used to calculate motor torques and brake forces. Modern rail vehicles can decelerate in three different states. It can be decelerated with motor, brake or motor and brake in combination. In order to decide which possibility has to be used, another frequency distribution (braking history) was created. The methodology was validated with reference and validation scenarios.

The creation of synthetic motor torques and brake forces should be extended to high speed trains. In this case the longitudinal force links have to be taken into account aswell. This force elements transmit the longitudinal force between bogie and coach. Therefore, motor torque and brake force will have a direct influence on these elements.

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