Simulation of a locomotive as a mechatronical system

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Summary: The representation of the mechanical behaviour of a railway vehicle by rather simple analytical MATLAB/SIMULINK drive models proved to be insufficient for the numerical simulation of the control part of a locomotive. For some problems which occur during operation these models cannot give satisfactory explanations. Therefore, a sophisticated numerical ADAMS/Rail model has been developed which describes the structural dynamical behaviour and the contact between wheel and rail in more detail. The control part created by MATLAB/SIMULINK and the mechanical part created by ADAMS/Rail can be coupled and this mechatronical simulation model can be used for a more reliable control unit design.

1. INTRODUCTION

For the design of the control part of a locomotive the simulation tool MATLAB/SIMULINK is used which assists in developing and optimising a control concept before the first prototype is built and helps in understanding and investigating problems which occur during operation. For the simulation with MATLAB/SIMULINK the dynamical behaviour of the masses and the contact mechanics between wheel and rail are represented by a mechanical model [1]. Up to now, the mechanical models used have been rather simple and the design process has focused on the modelling of the control part. Problems which occurred during operation, however, could not always be explained by these simulation models. This indicates that the mechanical behaviour must be described in more detail to get a better representation of the real locomotive.

It was decided to take one locomotive type as an example to demonstrate how the representation of the mechanical behaviour within a MATLAB/SIMULINK model can be improved. The simple mechanical model considered only torsional degrees of freedom of the drive and was described by state-space matrices within MATLAB/SIMULINK. This model is now replaced by a sophisticated ADAMS/Rail model [2].

In the first part of the paper the control system is discussed. The second part deals with the modelling of the mechanical structure and the third part shows how the contact between wheel and rail is described. In the fourth part the mechatronical ADAMS/RAIL - MATLAB/SIMULINK model is introduced. This model is used for comparison with measurements in the fifth part.

2. MODELLING OF THE CONTROL SYSTEM

In Figure 1 the basic control concept of the drive is illustrated. In general, the investigated locomotive has one Vehicle Control Unit (VCU) and one Drive Control Unit (DCU) for each bogie. The VCU controls the applied momentum necessary for accelerating and braking the locomotive. The DCU comprises the adhesion control, motor control and net converter control.

The alternating current of the net, which is characterized e.g. by 15 kV and 16 2/3 Hz, is modified by the transformer. After that, the net converter changes the alternating current to direct current, which is the input of the intermediate circuit. It follows the transformation from direct current to phase current. This is the final input of the motor which combined with the conditions given by the motor control drives the wheelset.
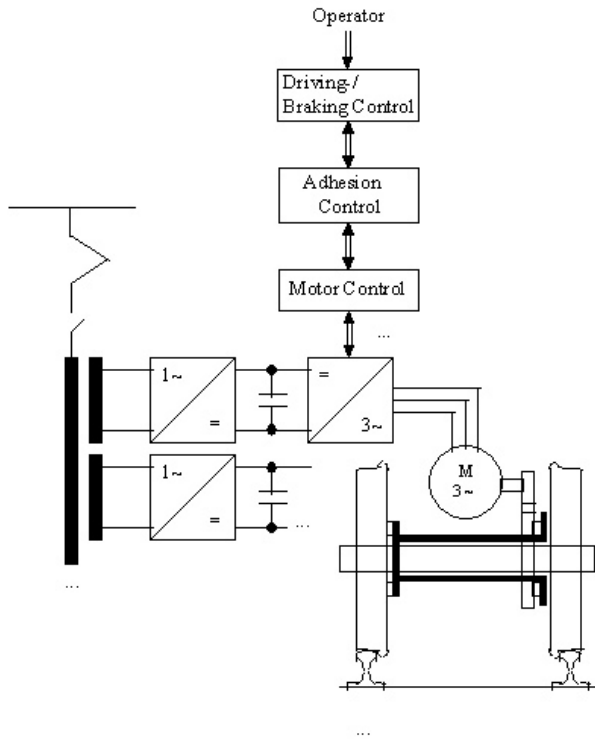

FIGURE 1   ADAMS/Rail model of the locomotive (taken from [3])

3. MODELLING OF THE MECHANICAL STRUCTURE

In contrast to former analytical MATLAB/SIMULINK state-space models the model presented considers not only the drive but also the car body, the secondary support, the bogies and the primary support. Thus, it is possible to take into account e.g. bounce, pitch and yaw modes which are likely to have an influence on the control performance and which had been omitted in former models. Most of the data of the mechanical structure was taken from an existing ME-DYNA model. The torsion dynamics of the drive was not considered in the MEDYNA model but is considered here. The ADAMS/Rail model is shown in Figure 2.

3.1 Car body and rest of the train

The car body is a simple rigid box, supported by the secondary springs and dampers between car body and the two bogies. Within ADAMS the car body has been created by the extrusion tool. At the end of the car body there is a hook which is the connection to the rest of the train. The hook is a spring element with stiffness and damping coefficients. The rest of the train is represented by a single mass which has only a longitudinal degree of freedom. The centre of mass of the car body has six independent degrees of freedom. Geometry and mass properties are symmetric with respect to the x-axis (longitudinal direction) and y-axis (lateral direction).
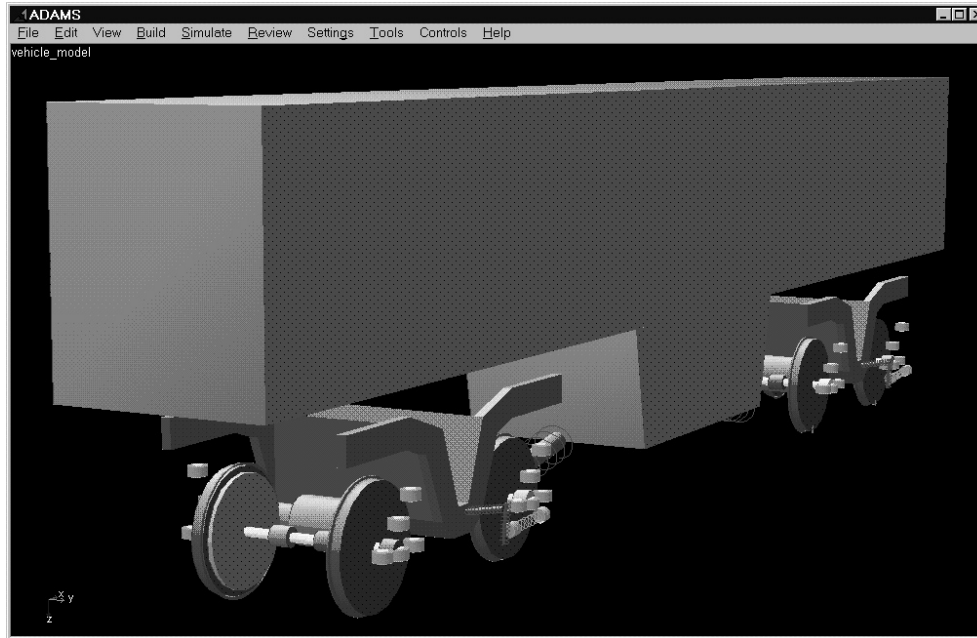

FIGURE 2   ADAMS/Rail model of the locomotive

3.2 Secondary support

The secondary support comprises the springs and dampers between the car body and the two bogies. At each side of a bogie there are flexicoil springs, lateral and vertical dampers. The flexicoil springs are represented within ADAMS by bushing elements, the dampers are modelled by a spring where the stiffness coefficient is set to zero and the damping coefficient is not equal zero. Furthermore, we have rolling dampers which are represented by Single-Component-Forces. These forces have been created by the Damper Element option within the Utility toolbox in the Rail-Start panel and describe a serial connection of a spring and a damper. The drawbar of each bogie is fastened to the car body and is represented by a linear spring . In general, secondary springs and dampers might have non-linear characteristics. As a first step, however, all springs and dampers are assumed to be linear.

3.3 Bogie

Both bogies are rigid bodies with six independent degrees of freedom. Each bogie is connected

to the car body by the secondary suspension and to every wheelset by the primary suspension. Within ADAMS the bogie has been created by the extrusion tool. Geometry and mass properties are symmetric with respect to the x-axis.

3.4 Primary support

The primary support consists of springs and dampers between bogie and wheelset. In vertical direction wheel and bogie are connected by large steel springs, all of which produce forces for vertical, lateral and longitudinal relative displacements and lateral relative velocities between bogie and wheelset. The longitudinal wheel steering is modelled by a spring/damper element and the vertical relative displacement between wheel and bogie is attanuated by vertical dampers.

3.5 Wheelset

For the investigation of the drive dynamics the torsional elastic behaviour of the wheelset axle must be taken into account. An appropriate wheelset model should consist at least of two single rigid wheel disks which are connected by a rotational spring. Thus, a wheelset has six independent degrees of freedom and an additional degree of freedom which describes the torsion of the wheelset axle. Since the wheelset library of ADAMS/Rail do not offer a wheelset with a rotational elastic axle, in the model presented one wheelset is represented by two rigid wheelsets taken from the ADAMS/Rail wheelset library. These both rigid bodies are connected by a revolute joint and a torsional spring and the mass and the momentum of inertia of both are half of the values of the real wheelset.

3.6 Drive

The drive comprises the motor box, the rotor, the gear wheel, an elastic coupling between the gear wheel and a hollow shaft, the hollow shaft and an elastic coupling between the hollow shaft and a wheel. The whole drive is connected to the bogie by linear springs and dampers. The torque produced by the motor is transferred to the wheelset by torsional degrees of freedom. To account for the torsional stiffness of the hollow shaft the shaft is divided into two parts which are connected be a linear torsional spring. The mass of each part is half of the mass of the whole hollow shaft.

## 4. MODELLING OF THE CONTACT BETWEEN WHEEL AND RAIL

ADAMS/Rail offers different levels for the modelling of the contact between wheel and rail [4]. One of them is Level IIb which considers only longitudinal contact forces while lateral and vertical contact forces are neglected. For this level, however, it is not possible to take into account the torsion of the wheelset axle which is an important natural mode for the simulation of the control performance. Therefore, it was descided to decribe the contact analytically without using ADAMS/Rail contact features. This analytical contact model is described in the following.

For the modelling of the normal contact the wheels are connected to the ground by springs which represent the serial stiffness of the Hertzian contact spring and the vertical track stiffness and by springs which represent the lateral track stiffness.

The longitudinal contact force is calculated by

$$T_\xi = \mu\left(\Delta v\right) N\left(t\right) \tag{1}$$

where μ is the coefficient of friction which depends on the relative velocity between particles of wheel and rail in the contact patch. The general characteristic of μ which is assumed for the investigation of the adhesion control performance is illustrated in Figure 3, where μ is plotted for various relative velocities Δv. The dashed line describes the coefficient of friction on a dry and the solid line on a wet rail, both of which are represented by a table of discrete values in an ASCII file. This table can be imported to ADAMS where two splines for each curve have been created. During the simulation with ADAMS Δv is calculated which determines μ by evaluating one of the two splines. The dynamical wheel load $N\left(t\right)$ follows from the force in the spring between wheel and ground in vertical direction.
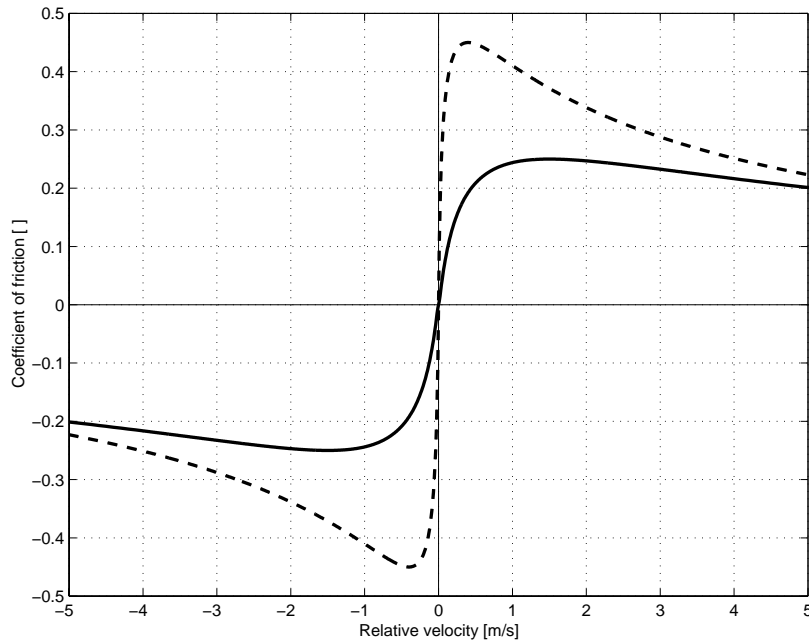
FIGURE 3   Coefficient of friction at various relative velocities for dry rail (dashed) and wet rail (solid)

4. CO-SIMULATION

Once the mechanical part and the control system have been modelled both models must be coupled. There are three opportunities:

- Export of state-space matrices from ADAMS to MATLAB/SIMULINK

- Export of state-space matrices from MATLAB/SIMULINK to ADAMS

- Co-Simulation

The export of state-space matrices is restricted to models with linear or linearized characteristics. However, the control system of a locomotive includes many non-linear features. A representation by state-space matrices is therefore not possible. Non-linearities of the mechanical model of the locomotive may result from non-linear spring-damper characteristics, the contact between wheel and rail and large displacements. For the drive control design the non-linearity of the contact must be considered while the others can be neglected in a first approach.

For the co-simulation both the mechanical and the control model can be non-linear. The contact mechanics including the non-linear coefficient of friction characteristic in Figure 3 can then be modelled using ADAMS/Rail. Input and output values of the ADAMS/Rail model must be defined and the model must be exported. This way a MATLAB/SIMULINK block is created which represents the ADAMS/Rail model and which can be built into the MATLAB/SIMULINK control model.

In Figure 4 the mechatronical MATLAB/SIMULINK model of the locomotive is shown. The adams-sub block represents the mechanical part and has been created by ADAMS/Rail while the CS-Control-Unit represents the control system and has been created by MATLAB/SIMULINK. The input values of the ADAMS/Rail model are the torques of the two motors of the front bogie. The output is: angular velocities of the rotors of both motors and vehicle speed. During the co-simulation of this mechatronical system ADAMS/Rail and MATLAB/SIMULINK are running at the same time exchanging input and output values after predefined time steps.
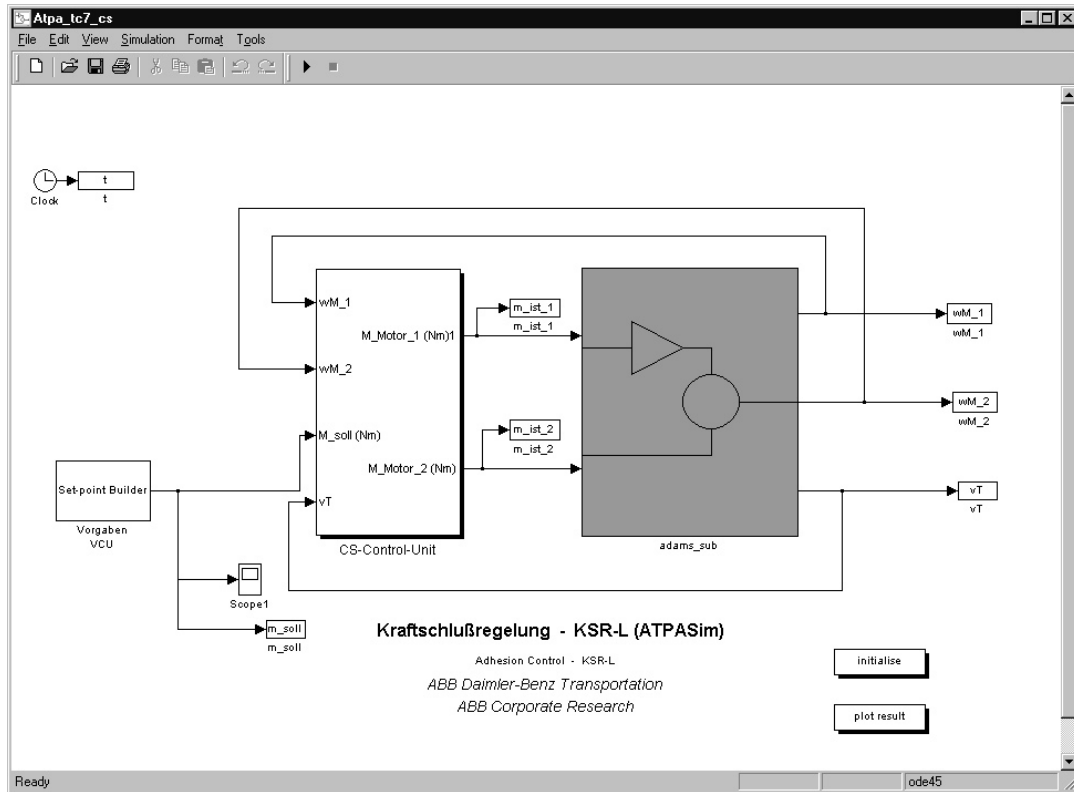

FIGURE 4   The mechatronical simulation model

5. COMPARISON WITH MEASUREMENTS

Now numerical results obtained by the ADAMS/Rail model are compared with measurements made in the field. For this comparison there exists a basic problem: The coefficients of friction which characterized the longitudinal contact between wheel and rail during the measurements are unknown. For the simulation we therefore have to determine equivalent coefficients of friction involving the measured relative velocities between wheel and rail and the measured traction forces. Furthermore, the control system used for the numerical simulation is slightly different to the control system of the real locomotive. During the measurements all four motors of the real locomotive were controlled, thus the resulting traction forces at each wheelset were different. For the numerical simulation, however, only the two wheelsets of the front bogie can be controlled to date. This difference between the real and the simulated locomotive was taken into account by reducing the mass of the simulation model. It is assumed that the traction forces produced by the wheelsets of the front bogie are more or less the same as the forces at the wheelsets of the rear bogie. Therefore, the mass of the locomotive and the mass of the trailer in the simulation model were reduced by half.

The measurements are characterized as follows:

- Measurements were made during the starting process of the locomotive.

- The trailer load is 440 t.

- The slope of the track was comparatively small.

- The locomotive was rolling on straight track.

- The first two axles were watered.

- Measurements show when watering started at the first and second axle and when the third and fourth axle approached the wet track section.

In the following the desired traction force and the real traction force is always the sum of the traction forces produced by all two wheelsets of the locomotive. Since in the simulation model only the first two wheelsets produce traction forces, for the comparison the sum of these traction forces is multiplied by two.

For the numerical simulation we have to calculate equivalent coefficient of friction characteristics at each wheelset. For this reason we determine from the measurements the traction force and the relative velocity at a certain time. The static wheel load is then calculated using the ADAMS/Rail model. The normal load divided by the traction force results in an equivalent coefficient of friction value. On the other hand, we have the coefficient of friction characteristic in Figure 3 which describes the longitudinal contact in the ADAMS/Rail model. Since we want to guarantee that the equivalent coefficient of friction for the numerical simulation and for the real locomotive is the same at least at one relative velocity we multiply the curve for a dry rail in Figure 3 by a factor which follows from the ratio

$$\frac{T_\xi(\Delta\nu_i)/N_\zeta}{\mu(\Delta\nu_i)} \, .$$

The coefficient of friction characteristic in the ADAMS/Rail model is then represented by the curve for dry rail in Figure 3 multiplied by this factor which can be different for each wheelset.

At the time before and after watering started two different equivalent coefficient of friction characteristics are determined. As long as the time is smaller than 4.17 s no wheelset is watered and the maximum traction force which can be produced is higher than the desired force 75 kN at each wheelset. The corresponding relative velocities between wheel and rail are very small. At t = 4.17 s the first wheelset is watered, at t = 4.795 s the second wheelset is watered.

In Table 1 the factors for the equivalent coefficient of friction characteristics are given as long as the wheelsets are not watered. For the numerical simulation the factors given are multiplied by the curve for dry rail in Figure 3. In Table 2 the factors are given for the wheel/rail contact after the wheelsets have been watered.

Table 1   Calculation of an equivalent coefficient of friction characteristic (1)

	Axle 1 (t < 4.17 s)	Axle 2 (t < 4.795 s)
$\Delta \nu_i$ [m/s]	0.01	0.028
$T_\xi(\Delta \nu_i)$ [kN]	80	80
N_ζ [kN]	207.3	209.6
$\mu(\Delta \nu_i)$ [ ]	0.39	0.38
Factor [ ]	1.77	1.73

Table 2   Calculation of an equivalent coefficient of friction characteristic (2)

	Axle 1 (t > 4.17 s)	Axle 2 (t > 4.795 s)
$\Delta \nu_i$ [m/s]	0.22	0.22
$T_\xi(\Delta \nu_i)$ [kN]	39	40
N_ζ [kN]	207.3	209.6
$\mu(\Delta \nu_i)$ [ ]	0.19	0.19
Factor [ ]	0.437	0.475

In Figure 5 the measurements during a 50 m passage are shown. The first and second axle are watered right after the desired traction force reaches the maximum, which is indicated by the clear decrease in the real traction force. After that, both rear wheelsets reach the wet track section and we have an additional but not so clearly visible decrease in the real traction force. A wet rail causes increasing relative velocities between wheel and rail which are forced by the control to be smaller than 1.08 km/h for the first and 0.72 km/h for the second wheelset as long as the driving speed is smaller than 6 km/h. The measurements show a sharp increase in the relative velocities when the wheelsets are watered.
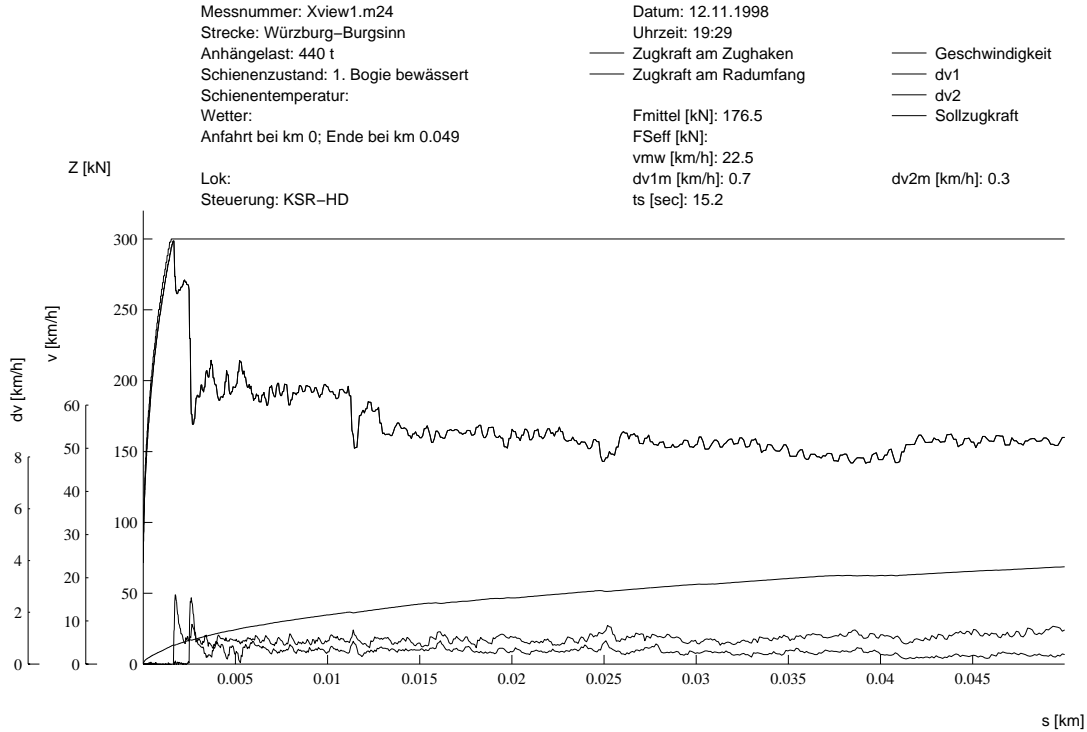
In Figure 6 the results of the co-simulation over a distance of 50 m are shown. For this calculation the coefficient of friction curve for dry rail in Figure 3 was modified. In addition

to the multiplication by the factors given in Table 1 and 2 the maximum had to be shifted to smaller relative velocities to obtain the same relative velocities as shown in the measurements. For the modified curves we assume for wheelset 1 and 2:

$$
\begin{aligned}
\mu_1^{modified} &= \mu_1(1.2\,\Delta\nu)\\
\mu_2^{modified} &= \mu_2(2\,\Delta\nu)
\end{aligned}
$$

The comparison between the measured and calculated results shows that the differences in velocity and time after a distance of 50 m are about 8.4 % and 13.2 %. These relatively large differences can be probably clearly reduced if the MATLAB/SIMULINK Control Unit is improved so that it is possible to consider that the traction forces at the wheelsets of the rear bogie are higher than the forces at both front wheelsets until the rear wheelsets reach the wet track section. If this is taken into account it can be suspected that the time used becomes smaller and the velocity at the end of the distance is higher.

Messnummer: Xview1.m24
Strecke: Würzburg–Burgsinn
Anhängelast: 440 t
Schienenzustand: 1. Bogie bewässert
Schienentemperatur:
Wetter:
Anfahrt bei km 0; Ende bei km 0.049

Datum: 12.11.1998
Uhrzeit: 19:29
Zugkraft am Zughaken
Zugkraft am Radumfang

Fmittel [kN]: 176.5
FSeff [kN]:
vmw [km/h]: 22.5

Geschwindigkeit
dv1
dv2
Sollzugkraft

Lok:
Steuerung: KSR–HD

dv1m [km/h]: 0.7
ts [sec]: 15.2

dv2m [km/h]: 0.3

FIGURE 5   Measured results

The effects of the watering are clearly visible and are similar to what is shown in Figure 5. The decrease in the real traction force when both rear wheelsets reach the wet rail can not be simulated since it is assumed that the sum of the traction forces of the two front and the two rear wheelsets are the same.

6. CONCLUDING REMARKS

A sophisticated numerical model of an electrical locomotive has been presented. It comprises a non-linear mechanical part created by ADAMS/Rail and a non-linear control part created by MATLAB/SIMULINK. Both models have been discussed and it has been shown how they are coupled for numerical simulations. Results of the co-simulation showed a satisfactory agreement between the model presented and measurements made on straight track. Future work will focus on simulations in curves.
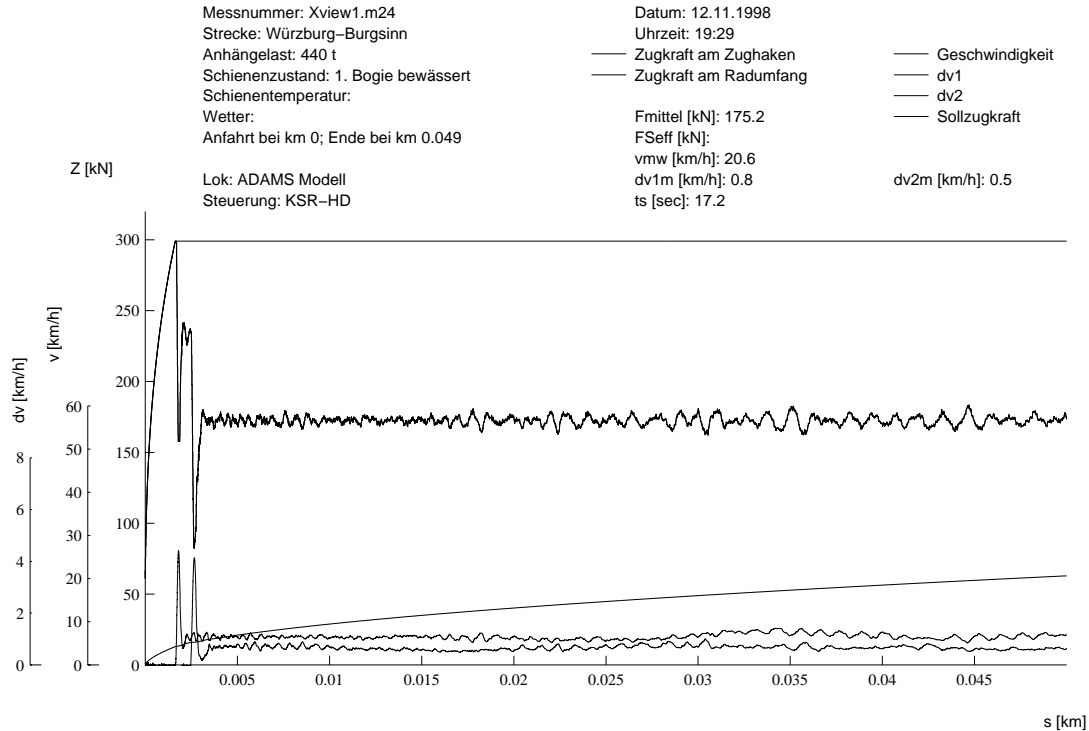

Messnummer: Xview1.m24
Strecke: Würzburg–Burgsinn
Anhängelast: 440 t
Schienenzustand: 1. Bogie bewässert
Schienentemperatur:
Wetter:
Anfahrt bei km 0; Ende bei km 0.049

Lok: ADAMS Modell
Steuerung: KSR–HD

Datum: 12.11.1998
Uhrzeit: 19:29
Zugkraft am Zughaken
Zugkraft am Radumfang

Fmittel [kN]: 175.2
FSeff [kN]:
vmw [km/h]: 20.6
dv1m [km/h]: 0.8
ts [sec]: 17.2

Geschwindigkeit
dv1
dv2
Sollzugkraft

dv2m [km/h]: 0.5

FIGURE 6   Simulated results

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