Numerical simulation of the drive performance of a locomotive on straight track and in curves

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Summary: This paper introduces two sophisticated numerical locomotive models. The first is for the simulation of the traction performance on straight track. The mechanical structure is built with ADAMS/Rail and the control part is created by MATLAB/SIMULINK. The second has been set up for the simulation of the traction performance in curves and both the mechanical and the control part are created with MATLAB/SIMULINK.

The contact mechanics for both models is described analytically without using ADAMS/Rail features since the drive performance is mainly determined by contact forces acting in rolling direction and therefore no complete contact theory is necessary. Furthermore, measurements in the field strongly suggest a negative gradient of the coefficient of friction curve at relatively high slip velocities which cannot be described by common contact theories.

Both locomotive models are discussed and numerical simulation results are validated against measurements. It reveals that simulated results are in satisfactory correspondence with measured results if a theoretical coefficient of friction curve is adjusted to measured slip velocities and traction forces. It is also shown that the typical decrease in traction force while the locomotive is entering a curve could be explained by lubrication and different running lines in curves while the influence of kinematical effects proved to be rather small.

1. INTRODUCTION

Numerical simulation is increasingly used to improve the design process of technical systems. To date several numerical tools have been developed to address the needs of engineers. For an electrical locomotive the design process has to focus on the mechanics, control and electric. At present these three parts are separately developed using different numerical tools. This way, mechanics, control and electric can be optimized. However, it is rather unlikely that the combination of these three optimized parts results in the optimal locomotive.

If the interaction between electric, control and mechanics is neglected during the design process, unexpected effects caused by this interaction occur during prototype test runs. Usually, changes in the locomotive parameters are necessary due to these interactions. This causes delay in the design process and increases the development costs.

This paper introduces two sophisticated numerical locomotive models where the interaction between control and mechanics is considered.

The first one is for the simulation of the driving performance on straight track. The mechanical structure is built with ADAMS/Rail, a Multi-Body-System program, and the control part is represented by MATLAB/SIMULINK, a numerical simulation tool frequently used by control engineers.

The second locomotive model has been set up for the simulation of the driving performance in curves and both the mechanical and the control part are created with MATLAB/SIMULINK. The modeling of the mechanical part is therefore rather simple: Only rotational degrees of freedom for the drive and one translational degree of freedom for the whole vehicle are considered. This simplification has two advantages: The numerical simulation is less time consuming than using a sophisticated ADAMS/Rail model and different curving effects can be investigated separately.

In the first part of the paper the control system is discussed. The second part deals with the modeling of the mechanical structure and the third part shows how the contact between wheel and rail is described. In the fourth part the coupled mechatronical locomotive model is introduced. This model is validated against measurements in the fifth part.

2. MODELING 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.

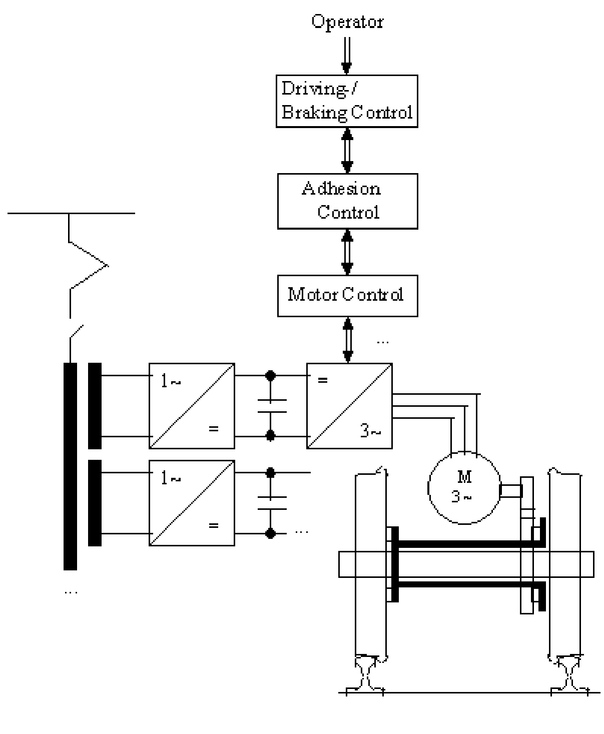

FIG. 1: Simplified illustration of the electrical part of the locomotive (taken from [3])

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.

The basic concept of the adhesion controller is illustrated in Figure 2. The speed sensor measures the response of the angular velocity of the rotor to the motor torque plus a sinusoidal test signal added to the motor torque. Based on the filtered sinusoidal rotor speed response the gradient of the coefficient of friction curve (cf. Figure 5) is computed. The actual gradient is compared to a desired gradient. If the difference between both is too large a new desired rotational rotor speed is calculated. This is the input to the Speed Controller which determines a new desired motor torque.

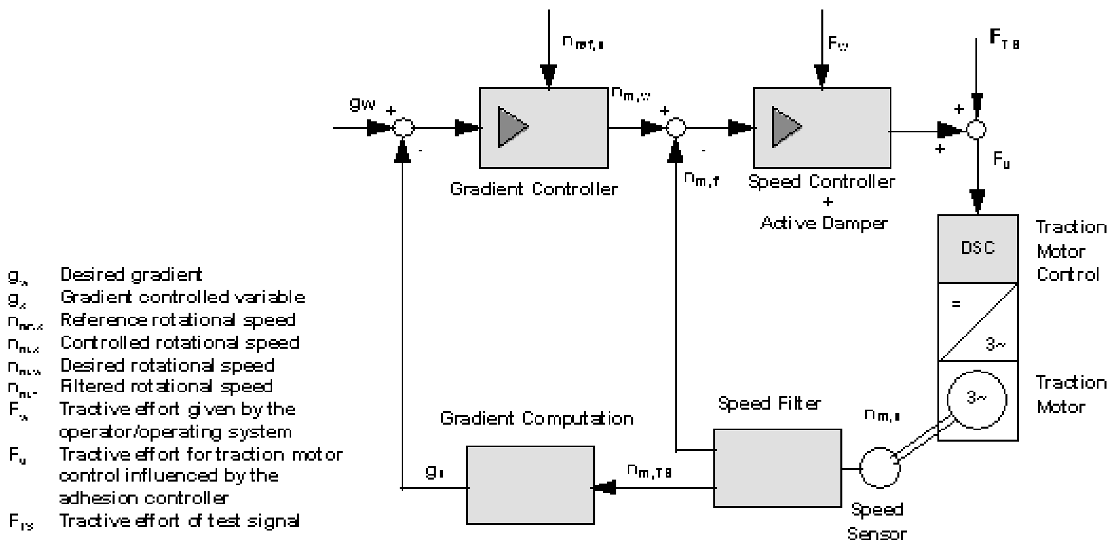

FIG. 2: Simplified illustration of the adhesion controller (after [3])

3. MODELING OF THE MECHANICAL STRUCTURE

In this report two different approaches to the modeling of the mechanical structure of the locomotive have been made. The first one is for straight track, where the mechanics is represented by a very detailed ADAMS/Rail model of the whole locomotive. The second one is for curved track sections. This is a rather simple model, where only torsional degrees of freedom of the drive and one translational degree of freedom of the movement of the whole locomotive in rolling direction is considered. The more simplified description of the locomotive has been chosen since it is felt that the drive performance in curves is mainly determined by the contact mechanics, control and drive dynamics. Oscillations of bogie and car body are neglected.

3.1 ADAMS/Rail model for straight track

For the investigation of the locomotive on straight track the mechanical structure is represented by a sophisticated ADAMS/Rail model with 91 degrees of freedom (cf. Figure 3): The car body is a simple rigid box, supported by the secondary springs and dampers between car body and the two bogies. At the end of the car body there is a hook which is the connection to the rest of the train. The rest of the train is represented by a single mass which has only a longitudinal degree of freedom.

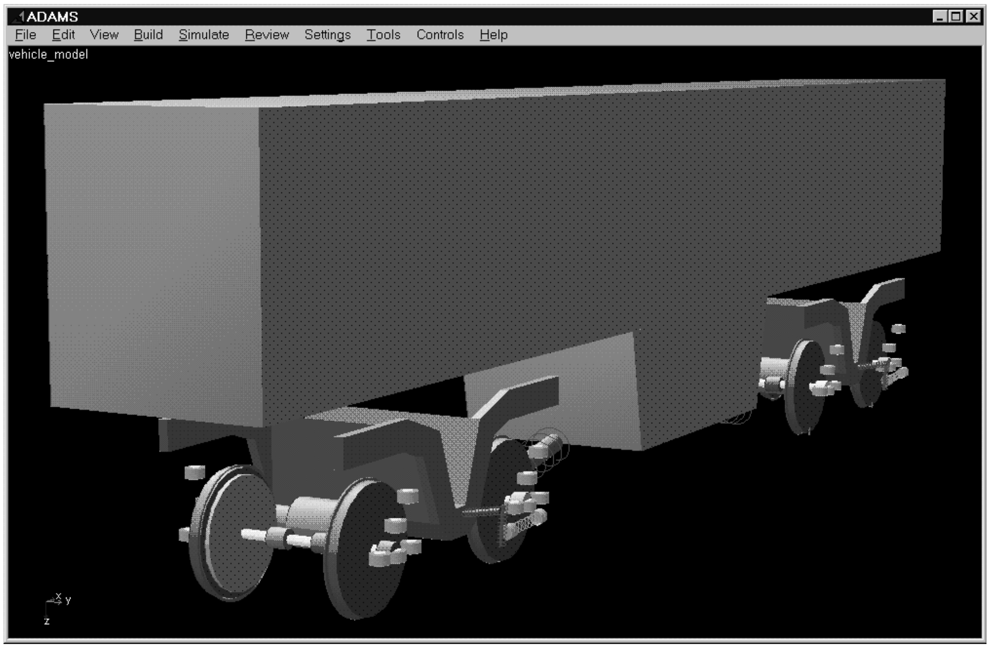

FIG. 3: Mechanical ADAMS/Rail model of the locomotive on straight track

The secondary support comprises the springs and dampers between the car body and the two bogies. In general, secondary springs and dampers might have non-linear characteristics. In this model, however, all springs and dampers are assumed to be linear. Each bogie is connected to the car body by the secondary suspension and to every wheelset by the primary suspension. The primary support consists of springs and dampers between bogie and 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. 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.

3.1 MATLAB/SIMULINK model for curved track

The MATLAB/SIMULINK model of the locomotive is represented by state-space matrices of the drive model illustrated in Figure 4. The motor torque is transmitted via rotor shaft, gearwheel and coupling to a hollow shaft which is connected to one wheel of the wheelset. The drive model has five rotational degrees of freedom: Rotation of the shaft/gearwheel, rotation of the hollow shaft, rotation of the first wheel and rotation of the second wheel. The rotation of shaft and gearwheel is described by one degree of freedom, and the rotation of the hollow shaft is represented by two degrees of freedom to account for the torsion of the shaft.

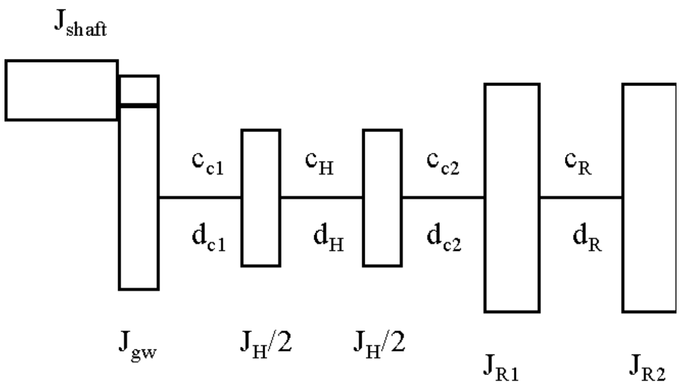

FIG. 4: Mechanical model of the locomotive in curves

4. MODELING OF THE CONTACT BETWEEN WHEEL AND RAIL

For two reasons the contact mechanics for both straight track and curved track is described analytically without using ADAMS/Rail features. First, the drive performance is mainly determined by contact forces acting in rolling direction. Therefore, no complete contact module comprising longitudinal forces, lateral forces and spin momentum is necessary. Second, the traction force mainly depends on the coefficient of friction characteristic $\mu(\Delta v)$ (cf. Figure 5). Measurements in the field strongly suggest a negative gradient of μ at relatively high slip velocities Δv. However, none of nowadays existing contact theories describes this effect.

For the locomotive model on straight track the contact forces are calculated in ADAMS while for the model in curves the contact forces are determined in MATLAB/SIMULINK.

4.1 Contact on straight track

For the modeling of the normal contact on straight track 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

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

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 assumed for the investigation of the adhesion control performance is illustrated in Figure 5, where μ is plotted at various relative velocities Δv. The dashed line describes the coefficient of friction on a dry and the solid line on a wet rail. The dynamical wheel load $N(t)$ follows from the force in the spring between wheel and ground in vertical direction.

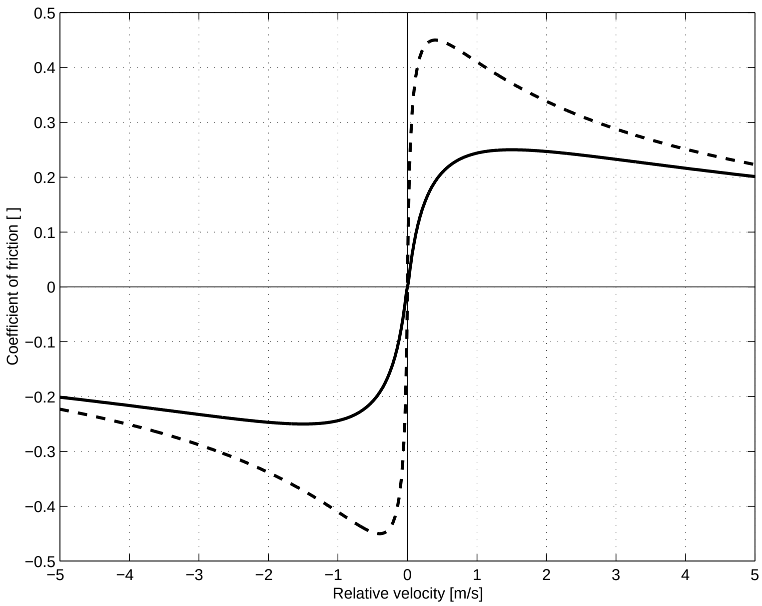

FIG. 5: Coefficient of friction at various relative velocities for dry rail (dashed) and wet rail (solid)

4.1 Contact in curves

In the following, effects which typically occur in curves are illustrated and approximate solutions to describe these effects are given.

Different translational velocities of inner and outer wheel

Both wheels of a rigid wheelset are rolling with the same angular velocity. The translational velocities of inner and outer wheel must therefore be different in curves. They are

$$\begin{aligned} v_{outer} &= \omega_{trans}\left(R_{track} + e_0\right) \\ v_{inner} &= \omega_{trans}\left(R_{track} - e_0\right) \end{aligned}$$

and the velocity of the train is

$$v_T = \omega_{trans}\, R_{track} \tag{2}$$

with

v_{outer}	- Translational velocity of the outer wheel
v_{inner}	- Translational velocity of the inner wheel
ω_{trans}	- Angular velocity of the train in the curve
R_{track}	- Radius of the curve
e_0	- Half of the distance between inner and outer wheel

It follows

$$
\begin{aligned}
v_{outer} &= \frac{v_T}{R_{track}} \left(R_{track} + e_0\right) \\
v_{inner} &= \frac{v_T}{R_{track}} \left(R_{track} - e_0\right)
\end{aligned}
$$

and for the relative velocities between wheel and rail for the inner and outer rail we have

$$
\begin{aligned}
v_{outer} &= \Omega_{wh} R_{wh} - v_T \left(1 + \frac{e_0}{R_{track}}\right) \\
v_{inner} &= \Omega_{wh} R_{wh} - v_T \left(1 - \frac{e_0}{R_{track}}\right) .
\end{aligned}
$$

where R_{wh} is the wheel radius and Ω_{wh} is the angular velocity of the wheel. The radius of the track R_{track} is measured, Ω_{wh} and v_T are calculated during the simulation and e_0 and R_{wh} follow from the geometry of the wheelset.

Different rolling radii of inner and outer wheel

In comparison to straight track sections, where the same rolling radius is assumed for all wheels (cf. [2]), the rolling radius of wheels vary in curves. This mainly depends on three parameters:

1. Traction force: The more traction force is produced the worse the self-adjustment mechanism of the wheelsets works. The reason for this is that if the produced forces are nearly maximum only small additional forces for the self-adjustment mechanism can be produced.

2. Lateral acceleration: The yaw angle and the lateral displacement of a wheelset may be different for different lateral accelerations. The lateral acceleration in a curve can be negative due to the cant angle or positive due to the vehicle speed. The combination of both results in an effective acceleration which is given in equation (3).

   Effective acceleration:

   $$a_q = \frac{v_T^2}{R_{track}} - g \, \frac{e_{track}}{2e_0} \tag{3}$$

   with

e_{track}	- elevation of the track
g	- gravity constant.

   For a given track and wheelset geometry the lateral acceleration varies with vehicle speed.

3. Geometry of the curve: For the same effective acceleration and traction force the position of the wheelsets in the curve can be different for different cant angles and different rail radii.

Generally, the first wheelset of a bogie tends to move laterally to the direction of the outer rail, so that the rolling radius of the outer wheel increases while the rolling radius of the inner wheel decreases. At relatively high vehicle speeds the position of the second wheelset of a bogie is comparable to the position on straight track. At small velocities the second wheelset moves laterally to the direction of the inner rail.

A change in rolling radius produces a change in slip velocity

$$\Delta v_{\Delta R} = \Omega \left(R_{wh} + \Delta R_{wh} \right) - v_T \,. \tag{4}$$

The change in rolling radii has been determined by MEDYNA. In Figure 6 the lateral displacement of all wheelsets of the locomotive calculated by MEDYNA are given.

Fz[kN]	v[km/h]	uy_r1[mm]	uy_r2[mm]	uy_r3[mm]	uy_r4[mm]
100	10	5.4	-4.2	6.0	-2.1
100	20	5.5	-3.8	6.0	-1.9
100	30	5.6	-2.9	6.0	-1.1
100	40	5.8	-1.4	6.0	0
100	50	5.9	-0.5	6.0	0.5
100	60	6.0	-0.1	6.0	0.6
100	70	6.0	-0.3	6.0	0.9
100	80	6.0	-0.5	6.0	-0.1
100	90	6.0	-0.3	6.0	0.
125	10	5.1	-4.6	6.0	-2.5
125	20	5.1	-4.5	6.0	-2.3
125	30	5.2	-4.1	6.0	-1.7
125	40	5.2	-2.9	6.0	-0.5
125	50	5.5	-1.7	6.0	0.2
125	60	5.7	-1.0	6.0	0.3
125	70	5.9	-1.3	6.0	0.3
125	80	6.0	-1.0	6.0	-0.5
125	90	6.0	-0.3	6.0	-0.3
150	10	4.8	-4.8	5.8	-3.2
150	20	4.8	-4.7	5.9	-2.9
150	30	4.9	-4.6	5.9	-2.5
150	40	5.0	-4.3	6.0	-1.4
150	50	5.0	-3.0	6.0	-0.5
150	60	5.1	-2.2	6.0	-0.2
150	70	5.2	-2.7	6.0	-0.3
150	80	5.6	-1.4	6.0	-1.0
150	90	5.8	0.0	6.0	-0.6
175	10	-0.5	-5.3	5.2	-4.2
175	20	-0.3	-5.2	5.3	-3.9
175	30	0.1	-5.2	5.4	-3.5
175	40	0.5	-5.1	5.6	-2.6
175	60	4.6	-3.1	5.9	-0.9
175	70	4.8	-2.4	6.0	-0.9
175	80	5.0	0.4	6.0	-1.5
175	90	5.1	3.8	6.0	-0.8

FIG. 6: Lateral wheelset displacements calculated by MEDYNA

The calculation has been performed for different traction forces F_z, different vehicle speeds v and different radii of the curves. With these displacements and the MEDYNA module RSGEO, the rolling radii can be determined for different lateral displacements.

Influence of lateral creepage

Typically, in curves lateral and spin creepage between wheel and rail occur. This influences the traction force. If spin creepage is omitted the effective tangential slip velocity depends on longitudinal and lateral slip velocity. The effective tangential slip velocity is

$$\Delta v = \sqrt{\Delta v_x^2 + \Delta v_y^2} \,. \tag{5}$$

The coefficient of friction depends on the effective slip velocity of equation (5). The coefficient of friction multiplied by the wheel load equals the traction force which acts in the direction of the effective slip velocity. To find the traction force which acts in the rolling direction this value must be transformed. After the transformation the traction force of one wheel in rolling direction is

$$T_\xi = \mu \left(\sqrt{\Delta v_x^2 + \Delta v_y^2} \right) N_\zeta \, cos \left(arctan \left(\frac{\Delta v_y}{\Delta v_x} \right) \right) . \tag{6}$$

Furthermore, we assume that the lateral creepage only depends on the misalignment angle $\Delta\varphi_z$ between wheel and rail. The lateral slip velocity is then

$$\Delta v_y = \Delta\varphi_z \, v_T \, , \tag{7}$$

where the sign of Δv_y depends on the definition of $\Delta\varphi_z$.

Influence of different running lines

In curves the yaw angle and lateral displacement of the first and second wheelset of a bogie are in general different. This effect influences the traction force.

On straight track and if the lateral displacement of the hunting motion is small the running lines of all wheelsets are more or less the same. If the rail surface is wet or greasy the first wheelset experiences a smaller coefficient of friction than the following wheelsets. The first wheelset "cleans" the rail surface which improves the traction conditions. This "cleaning effect" occurs because dirt or water is removed or because the water is vaporized due to the heating process which takes place while the wheel is rolling over the track and which leads to high temperatures in the vicinity of the contact point. This changes in curves where the running line of the front wheelset is different to the running line of the rear wheelset. Therefore, both wheelsets of the first bogie of the locomotive experience the same bad traction condition while all following wheelsets are running more or less on "cleaned" running surfaces.

Lubrication of the flange

If the wheelset is running in the flange high frictional work is produced and the wheelset is heated. To avoid this the flange is lubricated. The lubrication mechanism can be different for different locomotives. Some locomotives lubricate after a fixed driving distance while other locomotive types lubricate only in curves. Lubrication decreases the coefficient of friction and therefore influences the traction performance in curves: As soon as the running surface of the outer wheel of the front wheelset of a bogie approaches the flange the coefficient of friction and thus the traction force is reduced.

4. Setting up the mechatronical system

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

- Modeling of the control part within ADAMS
- Modeling of the mechanical part within MATLAB/SIMULINK
- Export of state-space matrices from ADAMS to MATLAB/SIMULINK

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

For the investigation of the locomotive on straight track we co-simulate the mechatronical system by using both the ADAMS/Rail model of the mechanical structure comprising the non-linear description of longitudinal contact and the non-linear MATLAB/SIMULINK model of the control. During the co-simulation ADAMS/Rail and MATLAB/SIMULINK are running at the same time exchanging input and output values after predefined time steps. This way, non-linearities in the MATLAB/SIMULINK model and in the ADAMS/Rail mechanics model are considered.

For the investigation in curves control and mechanics are coupled simply by modeling the whole locomotive in MATLAB/SIMULNK.

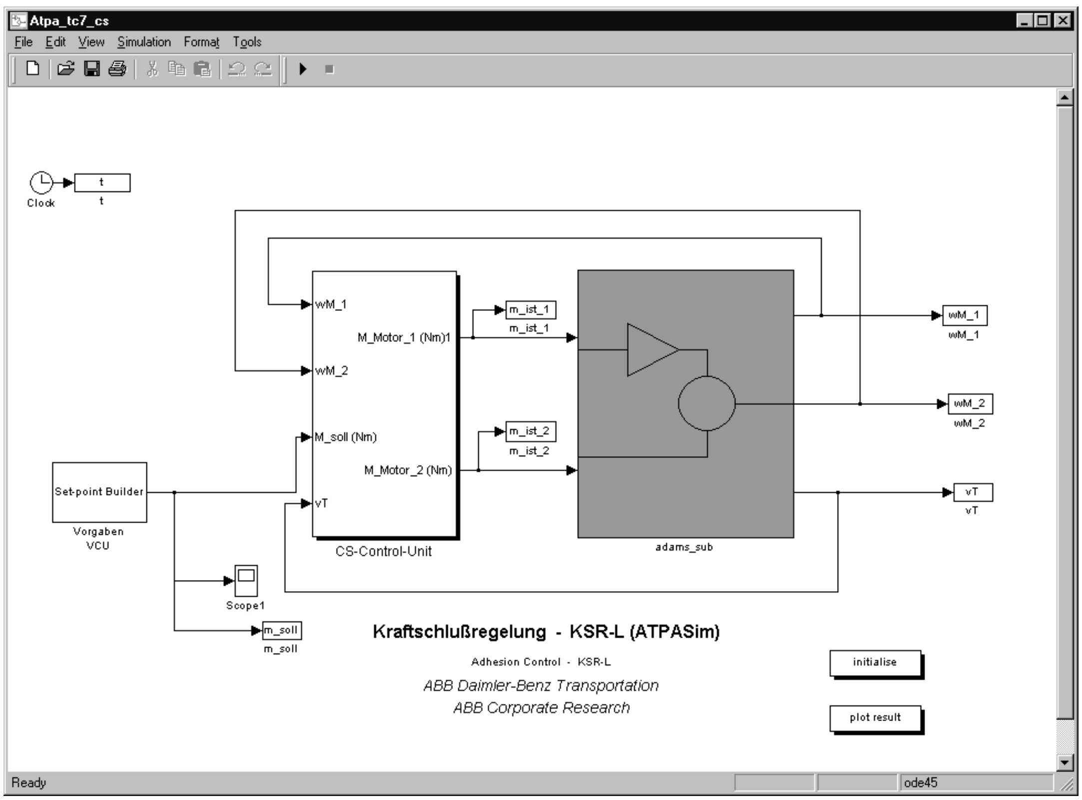

FIG. 7: The mechatronical simulation model

In Figure 7 the mechatronical MATLAB/SIMULINK model of the locomotive is shown. The colored block represents the mechanical part and is created either by ADAMS/Rail or 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.

5. COMPARISON WITH MEASUREMENTS

Now numerical results are compared with measurements made in the field. For this comparison there exists a basic problem: The coefficient of friction which characterize the longitudinal contact between wheel and rail during the measurements is 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 real and 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.

5.1 Results for running on straight track

The measurements on straight track 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 particular 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 5 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 5 by 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 5 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.

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 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 5. In Table 2 the factors are given for the wheel/rail contact after the wheelsets have been watered.

In Figure 8 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 9 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 5 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:

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

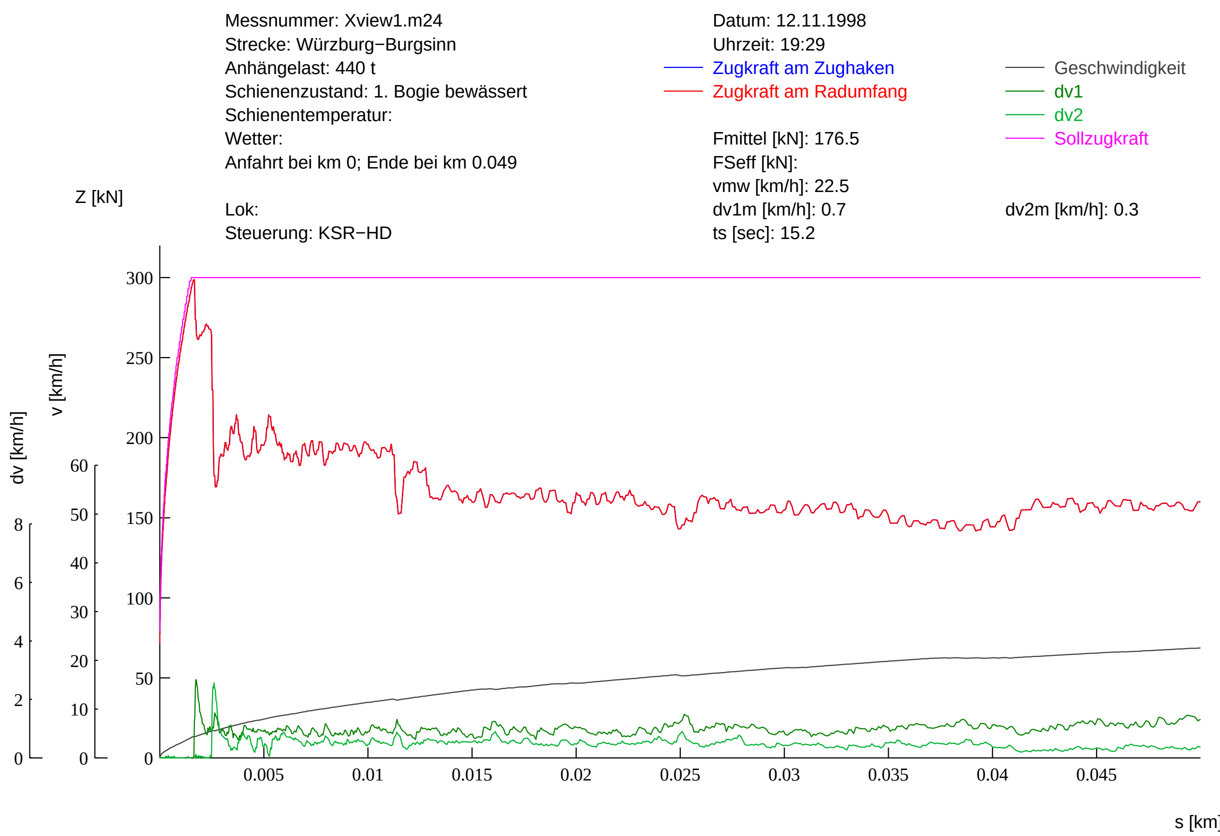

FIG. 8: Measured results on straight track

The comparison between measured and calculated results shows that the differences in mean velocity v_{mw} and time t_s after a distance of 50 m are about 8.4 % and 13.2 %. Probably, these differences can be 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 t_s becomes smaller and the velocity v_{mw} is higher.

The effect of watering is clearly visible and is similar to what is shown in Figure 8. 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.

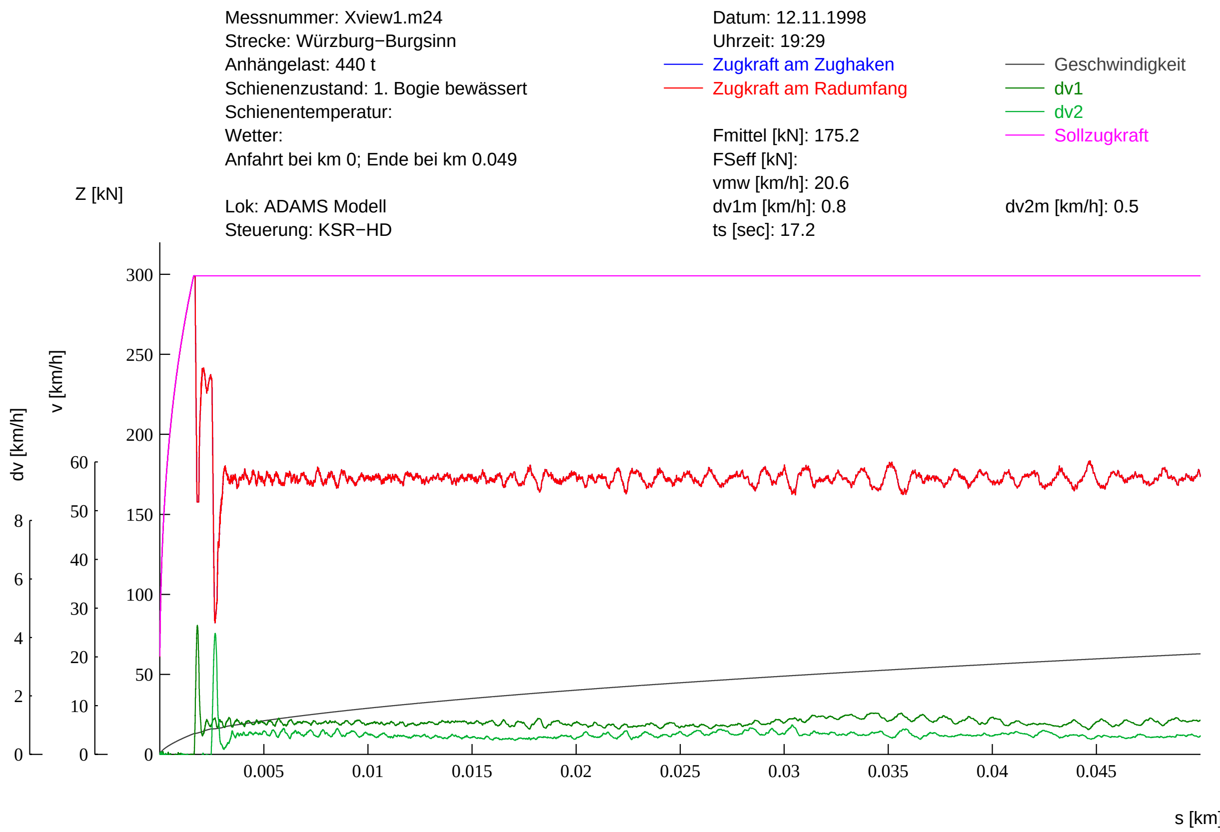

FIG. 9: Simulated results on straight track

5.1 Results for running in curves

In Figure 10 the measured results for a rail section with a wet and unconditioned rail surface are shown. On an unconditioned rail surface the coefficient of friction between wheel and rail is higher than on a conditioned rail surface. The load of the rest of the train is 584 t. The locomotive starts on straight track and enters a right curve with a radius of 300 m after approximately 50 m. After the curve a small straight section follows and then a 385 m curve is entered. The track slope is about 27 /1000 and constant during the measurement. The wet rail surface is produced by steadily watering the first axle of the locomotive.

The measured traction force shows a typical decrease while entering the curve. The force increases while leaving the curve. The longitudinal slip velocity of the first axle remains constant for both the straight and the curved track section. The slip velocity of the second axle rapidly decreases after the locomotive has entered the first curve and remains comparatively small during the rest of the measurement.

The outcome of the numerical simulation is plotted in Figure 11. For the simulation all effects described in Section 4 are considered except for the influence of lateral creepage which is likely to be very small.

It is assumed that due to different running lines the ratio of coefficient of friction at the second wheelset and the first wheelset is 1.22 on straight track and 1 in the curve. This causes the first drop in the traction force at about $s = 15.37$ km.

It is further assumed that the first wheelset is running in the flange if the radius of the track becomes smaller than 3000 m. The ratio of coefficient of friction if the wheel is not running in the flange and if it is running in the flange is 0.7. The first wheelset starts to run in the flange at $s = 15.43$ km and causes a sudden decrease in the traction force.

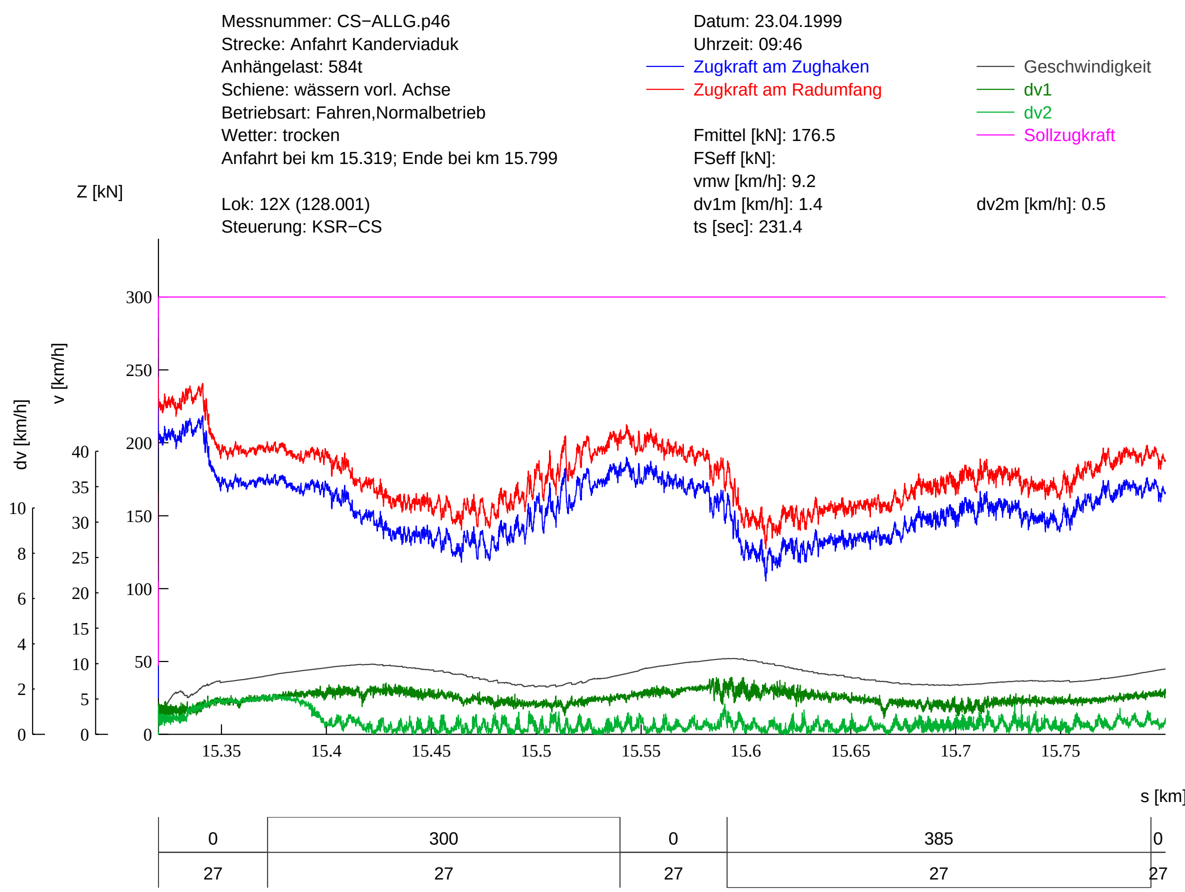

FIG. 10: Measured results in curve

Although the traction force in Figure 11 is in good correspondence to the traction force in Figure 10 we still have significant differences in the slip velocity. With the model presented it is not possible to simulate the sudden decrease in the slip velocity of the second wheelset. However, the typical decrease and increase in traction force while entering and leaving the curve could be explained by the model presented.

Based on the result it can be concluded that lubrication and the effect of different running lines have a strong impact on the traction performance in curves. The influence of different rolling radii and different translational velocities for inner and outer wheel proved to be rather small in this model.

6. CONCLUDING REMARKS

Two numerical models of an electrical locomotive have been presented. One for the simulation of the traction performance on straight track and one for the simulation in curves. Both models are mechatronical systems where mechanics and control of the locomotive are coupled.

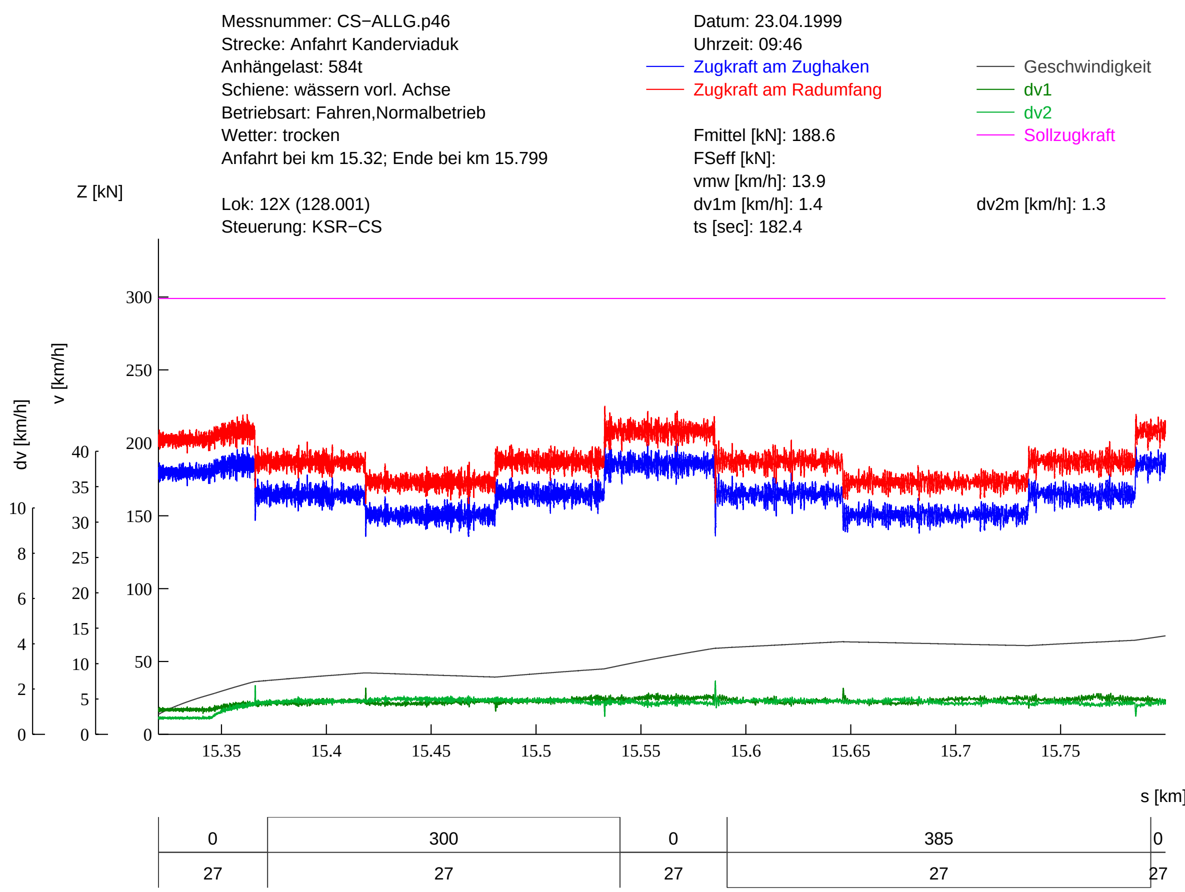

FIG. 11: Simulated results in curve

For the model on straight track the mechanical part has been created by ADAMS/Rail and the simulation was performed by co-simulating the control part in MATLAB/SIMULINK and the mechanical part in ADAMS/Rail. During the co-simulation both programs are running at the same time, exchanging input and output values after predefined time steps. The computing time for the co-simulation is relatively high. However, the advantage is that both the ADAMS/Rail model and the MATLAB/SIMULINK model can be non-linear.

The approach to modeling the locomotive in curves is different. The control part is the same as on straight track, but the mechanical structure is represented by torsional degrees of freedom of the drive and one translational degree of freedom of the movement of the whole locomotive in rolling direction. This more simplified description of the locomotive has been chosen since it is felt that the drive performance in curves is relatively independent of oscillations of bogie and car body.

The contact mechanics for both straight track and curved track is described analytically without using ADAMS/Rail feature since the drive performance is mainly determined by contact forces acting in rolling direction and therefore no complete contact theory is necessary. Furthermore, measurements in the field strongly suggest a negative gradient of the coefficient of friction curve at relatively high slip velocities which cannot be described by common contact theories.

Both locomotive models have been discussed and numerical simulation results have been validated against measurements. From the comparison between measured and simulated results the following is concluded:

- For all simulations we have the basic problem, that we do not know the exact coefficient of friction. Therefore, we had to calibrate a theoretical coefficient of friction curve based on the measured results. Without measurements only qualitative results can be obtained.
- Numerical results for starting the locomotive on straight track and watering at different axles during operation are in very good correspondence to measurements.
- Lubrication and the effect of different running lines have a strong impact on the traction performance in curves.
- In curves the influence of different rolling radii and different translational velocities at inner and outer wheel on the traction performance proved to be rather small in this model.
- With the model presented only some effects like the general decrease in traction force while entering a curve can be explained. Other effects which occurred in curves could not be simulated.

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