Development of Traction Control for Hauling Locomotives*

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Abstract
As heavy haul trains continue to push the limits of train size and mass and maximise locomotive performance, improving control of adhesion and slip will continue to be in demand. In considering the need to realise maximum adhesion forces for a rail vehicle, it is important to provide the development of new algorithms for traction control in a proper way that takes into account the need to avoid rail and track damage. This paper presents a strategy based on the Polach contact model for the detection of maximum adhesion force, and this strategy also includes slip compensation. The proposed traction control system has been verified by means of a co-simulation approach between the Gensys multibody code and the Simulink package.

Key words: Traction Control, Hauling Locomotive, Friction, Adhesion, Wheel/Rail Contact

1. Introduction
The modern solutions in the field of the development of new control systems for mechatronic systems of running gears allows the possibility to improve the interaction between wheels and rails for different modes of movement for rail vehicles.

One such system is the traction and braking control system, also called the adhesion control system, which is needed to adjust torque applied to the vehicle wheels. Many different strategies can be used in order to reach optimum adhesion, and most of these are based on variables that depend on vehicle velocity and friction conditions.

The classical method based on a speed comparison between all wheelsets is very convenient but still has small errors in speed measurement and is not failsafe. A more significant issue is that this method cannot optimise adhesion performance because it does not consider real-time changes in wheel load distribution or drive train resonance, as well as real-time changes in friction at each wheel rail contact site.

The traction control system of an electric motor coach uses a first-order disturbance observer constructed on a direct torque feedback control strategy. This method is very progressive, but for correct operation it needs exact creep curves, which require knowledge of variations of friction coefficients between wheel and rail.
The adhesion control system\(^{(2)}\) has been developed using vibration information for wheelset dynamics, and its control actions apply to reduce the tractive effort until the torsion oscillations disappear. However, this method gives adhesion forces just below the maximum adhesion available in the stable region of the creep force curves.

The control of adhesion force\(^{(3),(4)}\) has been performed based on the usage of beam and bristle models. The verification of the models has been done using an adhesion test rig. Unfortunately, this work does not provide any method for determining the maximum friction coefficient of real vehicles running on the track.

An adhesion control method without speed sensors\(^{(5)}\), focused on the current of each motor, is based on the behaviour of motor current when a slip occurs. It seems that this method is very good for implementation, but it does not guarantee reaching maximum adhesion force due to the signal noise related to the natural vibration modes.

Investigations\(^{(6),(7)}\) have detailed attempts to improve adhesion based on the estimation of the creep coefficient and monitoring the wheelset dynamic behaviour. These approaches can be very useful for new rail vehicles. However, wheel and rail profiles change significantly after a period of time, and so the method can lead to the wrong estimation. In addition, suspension damping may also change as the equipment ages.

All these strategies can be realised for passenger trains as well as hauling locomotives. However, in the latter case, the situation is more complicated if AC traction is applied (high adhesion > 35\%). Where these locomotives are used at high levels of adhesion, questions are sometimes raised regarding rail damage and the strength of the rail to sleeper connections. The traction control in this case is very important, but the information about the algorithms which are used in hauling locomotives is usually confidential and, therefore, cannot be accessed.

The traction control system\(^{(8)}\) includes wheel slip/creep and its algorithm is based on the monitoring of absolute acceleration of each axle. It is a very reliable approach and provides a significant increase in adhesion, but the achievement of maximum adhesion is still not realised.

A re-adhesion control system of an electric locomotive\(^{(9)}\) has been proposed based on taking into account the re-distribution of axle weights due to the wheel torques. It provides better adhesion performance than classical approaches, but it does not mean a realisation of maximum possible adhesion.

Analysing the different traction strategies, it is possible to see that the first issue of velocity or acceleration detection is almost solved apart from parameter measurement, but the second issue of the determination of contact characteristics between a wheel and a rail at any instant of a railway vehicle’s movement has no answer at the present time due to the large number of uncertainties. One such uncertainty is to understand the difference between friction and adhesion coefficient\(^{(10)}\). For the rolling mode with slip, the maximum value of slip-friction coefficient must be higher than the adhesion coefficient for the same contact conditions. In theory, the adhesion (traction) coefficient can be defined as traction force divided by wheel load. Therefore, both of these coefficients describe almost the same thing, which determines the ratio of the tangential to normal forces. However, the slip-friction coefficient only depends on the physical state of rail and wheel surfaces, while the adhesion coefficient also depends on the design characteristics of rail tracks and railway vehicles. These design characteristics can be modified by the unaccounted slipping motions, the difference between wheel diameters of wheel pairs, conicity and eccentricity of wheels, track curvature, reallocation of loads between wheels, irregular loading of wheels for the wheel pair, the bogie, the car, the train, vibrations, etc. All of these make the task of the detection of maximum adhesion coefficients very complex. Some preliminary solutions in this field are proposed. For example, the estimation of the friction coefficient for rail vehicles based on neural network and computational methods\(^{(11)}\) is one such solution.
Unfortunately, the information about the application of the proposed technology has not yet resulted in any implementations for design of control systems. This is probably due to problems achieving validation. The other example is the adhesion control system\textsuperscript{(12)} which proposes the use of noise spectrum analysis for friction coefficient detection. This paper therefore aims to further research the use of noise spectrums for adhesion control and its application for hauling locomotives.

2. Wheel-Rail Noise and Adhesion

The aspects of the usage of wheel-rail noise for the control system based on the previous experimental and theoretical investigation in this field have been discussed in literature\textsuperscript{(12)}. The recent research\textsuperscript{(13)}\textsuperscript{--}(15) performed on full and scaled test rigs shows that wheel-rail noise emitted from the contact zone depends on many factors (friction coefficient, angle of attack, speed etc).

The preliminary results for the further development of a strategy that enables the prediction of adhesion conditions of rail-wheel contact based on rolling noise analysis are presented in recent work\textsuperscript{(14)}. The obtained experimental results from the scaled test rig data shows that the amplitude of the signals in the frequency ranges of 300-400 and 2600-2800 Hz depend on the friction conditions in the contact zone. A change of amplitude, however, appears in a third frequency range when the investigation has been made under dry and wet friction conditions with velocities of 1.5, 2.0 and 2.5 m/s. In the case of the velocity of 2.5 m/s, the third range appears between 3400 and 3700 Hz.

The same tendency can be seen in investigations performed for the automotive industry. Investigation\textsuperscript{(16)} focuses on dynamic adherence force and noise generation in the tyre/road contact. The obtained results show that the dynamic adherence force, which depends strongly on load, load duration and unloading rate, is a fairly good measure for predicting the total sound pressure level, while more information is needed to predict levels in 1/3-octave bands.

As a consequence, it is necessary to do further research in this field for a more detailed study on the dependence of the parameters mentioned above, and relating these to noise in rail-wheel contact for different values of creepage (slip), velocity, wheel load and the wheels’ positions on track. The obtained results would allow further development of the concept of a control system based on noise spectrum analysis.

3. Traction Control System

The traction control system is based on the direct torque feedback control strategy with the compensation gain based on the slip difference. The proposed control system is presented in Fig. 1 with the following variables and parameters: $T_{\text{ref}}$ - reference torque; $T_{\text{ref}}^*$ - reference torque generated by the control system; $T_{\text{in}}$ - input motor torque; $T_{\text{opt}}$ - optimal motor torque; $T_{\text{wheels}}$ - traction torque applied to the wheelset; $\Delta T$ - torque compensation signal; $y$ - lateral displacement of the wheelset; $Q$ - the wheel load; $\omega$ - the real angular velocity of a wheelset; $\mu$ - maximum adhesion coefficient dependent on friction conditions, and $V$ - the locomotive velocity.

The inverter and traction motor dynamics including the gearbox characteristics can be written as:

\[
T_{\text{wheels}} = \frac{1}{\tau} T_{\text{in}} \cdot i
\]  \hspace{1cm} (1)

where $\tau$ is a time constant and $i$ is a conversion ratio of the gearbox.
Three assumptions for the control system are required:

- friction coefficient can be estimated by means of noise spectrum analysis;
- the contact area dimensions can be determined from lookup tables depending on the lateral position of the wheelset (for example, these dimensions can be updated and predicted based on systematic measurements of the wheel and rail profiles obtained from the railway operators); and
- the wheel load can be measured from sensors.

The Polach model for the low adhesion estimation has been described in literature\(^\text{(17)}\). In our case, this model is also needed to find the optimal adhesion force that can be defined by the observer based on the Polach model\(^\text{(18),(19)}\):

\[
F = \frac{2Q\mu}{\pi} \left( \frac{k_A e}{1 + (k_A e)^2} + \arctan(k_A e) \right), \quad k_A \leq 1 \quad (2)
\]

\[
e = \frac{2C\pi a^2 b}{3Q\mu} s \quad (3)
\]

\[
s = \sqrt{s_x^2 + s_y^2} \quad (4)
\]

\[
F_x = F \frac{s_x}{s} \quad (5)
\]

\[
\mu = \mu_0 \left[(1 - A)e^{-B\lambda} + A\right] \quad (6)
\]

where \(\mu_0\), \(k_a\), \(k_b\), \(A\) and \(B\) are model parameters for different friction conditions, \(a\) and \(b\) are the length of the semi-axes of the elliptical contact patch, \(s_x\) and \(s_y\) are the longitudinal and lateral creepages, \(F_x\) is the longitudinal creep force, \(Q\) is the wheel load and \(C\) is a proportionality coefficient characterising the contact shear stiffness.

In order to find the optimal adhesion force for each wheel, the special calculation and search algorithm has been realised. At the first stage, the value of the adhesion force is calculated for the longitudinal creepage range from zero to 1, and then the maximum value
of the calculated force and its longitudinal creepage are found by the search function. It is necessary to mention that the optimal adhesion force is slightly less than the value of the maximum adhesion force in order to be in the stable slip zone (Fig. 2).

\[ T_{\text{opt}} = F_{x1} r_1 + F_{x2} r_2 \]  

where \( F_{x1} \) and \( F_{x2} \) are the left and right longitudinal creepage forces respectively, and \( r_1 \) and \( r_2 \) are the left and right wheel radii at the contact point respectively.

The optimal longitudinal creepage can be found as:

\[ s_{\text{opt}} = \frac{s_{x1} + s_{x2}}{2} \]  

The estimated longitudinal creepage can be calculated as:

\[ s_{\text{est}} = \frac{W \cdot r - V}{W \cdot r} \]  

The slip controller is a PI controller, which uses the difference between optimal and estimated creepages as the input. The controller provides a compensation of the torque value, which is adjusted to suit the complex non-linear phenomena of the friction process.

The switch block (Fig. 1) then corrects the value of reference torque (that in a real system is selected by the locomotive driver to suit the current operating conditions) to a value of the optimal torque (that which can be practically applied).

4. Simulation and Results

For this simulation process, the client interface in Matlab/Simulink has been developed for co-simulation with Gensys multibody code. The methodology of the development of such an interface for the Windows 32 environment is described in our recent work\(^{20}\) and is based on the data exchange between these software tools by means of TCP/IP. This interface has been modernised in order to use it in both products running under OS Ubuntu 10.10. The traction control system has been realised in Simulink and the locomotive model in Gensys.
4.1. Locomotive Model in Gensys Software

The vehicle model of a hauling locomotive has the design and characteristics presented in Table 1 and shown in Fig. 3 (a). The vehicle has a gross mass of 134 metric tons. The locomotive has two bogies with six traction axles (Co-Co). Each of these axles is equipped with an AC motor.

The locomotive was modeled with standard gauge railway bogies with the gauge distance of 1435 mm. The locomotive body, two bogies and six wheelsets are modeled as rigid bodies and have six degrees of freedom each. Normally when creating a railway vehicle in Gensys, two coordinate systems are used. One is called the Euler system (esys), which handles large displacements and rotations, and the vehicle speed is defined in this system. One esys is created for each vehicle and it is generally located in the middle of the vehicle. The other is the linear coordinate system (lsys), which is related to the esys. The locations of vehicle body, bogies and wheelsets are defined in this system. This system allows separate coordinate systems for all masses in the vehicle along the track, because all masses are in different places along the track having different cant, curvature, vertical lift etc. The track is modeled as lumped masses under wheelsets, and joins wheelsets through the wheel-rail contact zones. Each track mass has three degrees of freedom (lateral and vertical translations and roll rotations). Figure 3 shows the locomotive and track modelling:

- four vertical coil spring elements;
one spring element and one damper to provide flexibility for the traction rod in the direction specified by the coupling's attachment points.

**Table 1** Common parameters for the model of the hauling locomotive

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel Spacing</td>
<td>2.38</td>
<td>m</td>
</tr>
<tr>
<td>Bogie Spacing</td>
<td>14.68</td>
<td>m</td>
</tr>
<tr>
<td><strong>Vehicle body</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Centre of gravity, vertical</td>
<td>1.930</td>
<td>m</td>
</tr>
<tr>
<td>Mass</td>
<td>87180</td>
<td>Kg</td>
</tr>
<tr>
<td>Moment of inertia, roll</td>
<td>168550</td>
<td>Kg.m^2</td>
</tr>
<tr>
<td>Moment of inertia, pitch</td>
<td>3610410</td>
<td>Kg.m^2</td>
</tr>
<tr>
<td>Moment of inertia, yaw</td>
<td>3590650</td>
<td>Kg.m^2</td>
</tr>
<tr>
<td><strong>Bogie frame</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Centre of gravity, vertical</td>
<td>0.733</td>
<td>m</td>
</tr>
<tr>
<td>Mass</td>
<td>14860</td>
<td>Kg</td>
</tr>
<tr>
<td>Moment of inertia, roll</td>
<td>6520</td>
<td>Kg.m^2</td>
</tr>
<tr>
<td>Moment of inertia, pitch</td>
<td>45370</td>
<td>Kg.m^2</td>
</tr>
<tr>
<td>Moment of inertia, yaw</td>
<td>50300</td>
<td>Kg.m^2</td>
</tr>
<tr>
<td><strong>Axles</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Centre of gravity, vertical</td>
<td>0.565</td>
<td>m</td>
</tr>
<tr>
<td>Mass</td>
<td>2850</td>
<td>Kg</td>
</tr>
<tr>
<td>Moment of inertia, roll</td>
<td>1789</td>
<td>Kg.m^2</td>
</tr>
<tr>
<td>Moment of inertia, pitch</td>
<td>1200</td>
<td>Kg.m^2</td>
</tr>
<tr>
<td>Moment of inertia, yaw</td>
<td>1789</td>
<td>Kg.m^2</td>
</tr>
<tr>
<td><strong>Primary Suspension</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lateral Position</td>
<td>1.012</td>
<td>m</td>
</tr>
<tr>
<td>Vertical Stiffness</td>
<td>730</td>
<td>KN/m</td>
</tr>
<tr>
<td>Vertical Damper</td>
<td>5</td>
<td>KN.s/m</td>
</tr>
<tr>
<td>Longitudinal Stiffness</td>
<td>24.0</td>
<td>MN/m</td>
</tr>
<tr>
<td>Longitudinal Damper</td>
<td>25</td>
<td>KN.s/m</td>
</tr>
<tr>
<td>Lateral Stiffness</td>
<td>12.0</td>
<td>MN/m</td>
</tr>
<tr>
<td>Lateral Damper</td>
<td>15</td>
<td>KN.s/m</td>
</tr>
<tr>
<td><strong>Secondary Suspension</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Vertical Stiffness</td>
<td>1.069</td>
<td>MN/m</td>
</tr>
<tr>
<td>Vertical Damper</td>
<td>40</td>
<td>KN.s/m</td>
</tr>
<tr>
<td>Longitudinal (traction rod) Stiffness</td>
<td>25</td>
<td>MN/m</td>
</tr>
<tr>
<td>Longitudinal Damper</td>
<td>100</td>
<td>kN.s/m</td>
</tr>
</tbody>
</table>

The connections (the primary suspensions) between one bogie frame and three wheelsets (see Fig. 3 (b and c)) include:
- six stiffness and six damping connections (connecting axle boxes to bolster frame) with the length zero, which can not be shown in the modelling figures; and

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1 The model depicts the traction motor mass shared between bogie and axles.
at each connection, the longitudinal, lateral and vertical stiffness and damping coefficients are taken into account.

The wheel and rail profiles used are standard new wheel S1002 and rail UIC60 profiles. In the wheel-rail contact modelling, three different wheel-rail contact surfaces can be in contact simultaneously. The wheel contacts are modelled with three spring elements normal to three wheel-rail contact surfaces, which allow normal wheel-rail contact forces to be determined. The calculations of tangent creep forces at the wheel-rail contact zone are made using the contact model based on Kalker theory.

4.2. Results

The estimation of the control system has been done by means of simulation for the cases: constant linear speed 20 km/h for the locomotive and acceleration. The locomotive is running on a straight track. The modeling has used track with no track alignment errors.

The discrete time solver with a fixed time step size has been used in Simulink. The heun_b integrator has been used in Gensys, which works according to a modified Heun's method. The integrator has variable steps, and the length of the step is calculated based on how fast the error increases or decreases between two consecutive time steps. The output time for Gensys was chosen equal to the fixed time step in Simulink and it was equal to 0.001 second.

For the adjustment of the discrete PI controller (the slip controller), it was necessary to choose correct values of proportional and integral coefficients. Using the variables P and I, it is possible to choose how fast the vehicle shall respond to changes in the speed adjustment. However, taking into account that the control driving torque generated by a controller should always be limited to the motor maximum characteristics as described in the paper(21). In this simulation, the characteristics have been defined based on the characteristics for a high adhesion locomotive described in the publication(22).

The simulation strategy used in this work is intended to simulate dry, wet, low and very low adhesion (friction) conditions. Table 2 contains all required parameters for the Polach model. The switch between the adhesion conditions has been done for the constant speed test in the following order: low – dry – low – wet – very low. The switch between different adhesion conditions happens every 10 seconds. For the acceleration test, the movement on the track with the wet adhesion condition has been simulated.

Table 2 Parameters in the Polach model for simulation of different adhesion conditions(17)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Very Low</th>
<th>Low</th>
<th>Wet</th>
<th>Dry</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k_a$</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>$k_s$</td>
<td>0.40</td>
<td>0.40</td>
<td>0.40</td>
<td>0.40</td>
</tr>
<tr>
<td>$\mu_o$</td>
<td>0.03</td>
<td>0.06</td>
<td>0.30</td>
<td>0.55</td>
</tr>
<tr>
<td>$A$</td>
<td>0.40</td>
<td>0.40</td>
<td>0.40</td>
<td>0.40</td>
</tr>
<tr>
<td>$B$</td>
<td>0.10</td>
<td>0.20</td>
<td>0.20</td>
<td>0.60</td>
</tr>
</tbody>
</table>

For the estimation of the performance of the proposed system, the parameter called the traction coefficient has been used. This parameter can be defined as:

$$\mu_t = \frac{F_w}{Q_w}$$

where $F_w$ is the total longitudinal creepage force for both wheels, $Q_w$ is the axle load.

Figures 4 and 5 show the calculated values of traction coefficients and longitudinal
creepages for the first wheelset of the leading bogie in the time-domain. In addition, Fig. 6 presents information about distance and vehicle velocity during acceleration tests.

![Graph showing longitudinal traction coefficient and longitudinal creepage over time](image)

Fig. 4 The results obtained for the hauling locomotive running with a constant speed 20 km/h under different adhesion conditions (blue solid line – optimal parameter, green dash line – estimated parameter)

The test with constant speed shows that the system recovers very quickly after changing the friction conditions between wheels and rails. The main output of the proposed system is a traction coefficient. The usage of longitudinal creepage as the additional output is needed to verify that the system is optimized for both parameters. In order to get a perfect graph for longitudinal creepage, it needs to switch a system very quickly from traction to braking mode. From the practical point of view, this is strongly not a desirable action. However, from the simulation point of view it can be smoothed by means of introduction of a special torque function in the similar way as described in the publication [1]. Besides it, these errors depend on the characteristic of electrical equipment, which are not described with a full model in this paper. Based on the above and taking into account wheel and rail damage issues, which are also very significant during traction process, it is better to represent a worst case scenario, which can be obtained from our numerical investigation. As a result, the control provides a very good optimisation between applied traction torque and longitudinal creepage in order to reach the maximum value of the traction coefficient.
Fig. 5 The longitudinal traction coefficient and creepage results obtained for the hauling locomotive running in the acceleration mode under different adhesion conditions (blue solid line – optimal parameter, green dash line – estimated parameter)

Fig. 6 The distance and linear velocity results obtained for the hauling locomotive running in the acceleration mode under different adhesion conditions
The second test for acceleration modes also shows good results, but some instability for small values of linear velocities up to 3 m/sec can be seen. There are two reasons for this simulation behavior. The first one is connected with the usage of small speed values inside of multibody codes and the second one, which is more important, is the limitations of the PI controller. The latter can be solved by means of the introduction the flexible PI control (for example, PI-fuzzy logic controller).

Based on the obtained results, the satisfactory working of the proposed system is observed in both test modes.

5. Conclusions and Further Research

Advanced traction control system designs for hauling locomotives opens the possibilities of both high adhesion and wear reduction. In this paper, such a system has been proposed and simulation results show that the goal can be reached, if sufficiently accurate determinations of friction parameters can be developed. More detailed investigation on this system is required for when the vehicle runs on curves and to incorporate the many variations of passive and active bogie steering systems. The work of the traction system also should be verified in the real-time mode.

The future plan also includes experimental investigation of the noise emitted from the wheel/rail contact and the possibility of developing an algorithm. The investigation will focus on the discrete-time mode and return the values of the amplitudes for the definite frequencies. These values as well as data obtained from the sensors (angular velocity, locomotive velocities and etc) will then to be used as input data for the second algorithm, which searches a database to get the dependence between the noise characteristics and the value of the estimated contact parameters.

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