RE-ADHESION CONTROL BASED ON WHEELSET DYNAMICS
IN RAILWAY TRACTION SYSTEM

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Abstract: This paper demonstrates how wheelset dynamics is affected by wheel-rail contact nonlinearities. Based on vibration phenomena due to wheel slip, two detection methods are proposed, both utilizing wheel speed information. One uses a running FFT to detect spectrum variation of a particular frequency component while the other uses bandpass and low pass filters to obtain magnitude information of the frequency interest. A re-adhesion scheme which is based on slip detection is developed. The simulation results show that this re-adhesion control scheme can detect slip effectively and improve traction stability.

Keywords: re-adhesion control, wheelset dynamics, vibration, nonlinearity, stability

1. INTRODUCTION

The knowledge of adhesion force through wheel-rail contact is of primary importance in railway traction systems. Generally, adhesion force has nonlinear features which are related to the creep value and are strongly affected by wheel-rail contact conditions such as dry/wet, dust/leaves and so on. When the mechanical torque transmitted to the wheel is much higher than the maximum adhesion torque that can be obtained from wheel-rail contact, the wheel will lose adhesion, and slip will occur. Slip is harmful in traction operations since it will increase the wear of wheel and rail, increase mechanical stress in the system, affect stability and, furthermore, lead to poor traction performance. Therefore re-adhesion schemes are required to reduce the risk of slip.

Disturbance observers are often used to detect slip conditions (Kadowaki, et al., 2002; Woo-Seok Kim, et al., 1999). In these papers, traction torque is treated as a disturbance torque, and estimated either through zero/first-order observers (Kadowaki, et al., 2002) or through state observers (Woo-Seok Kim, et al., 1999) using motor speed and torque information. Besides, hybrid anti-slip methods, which used slip speed, wheel speed and acceleration information, were introduced in some papers (Don-Young Park, et al., 1999; Hyoun-Chul Choi and Suk-Kyo Hong, 2002). Those controllers have difficulty obtaining satisfactory performance, and also require accurate measurement of wheel slip. On the other hand, most disturbance observers are established on the assumption of rigid shaft connection, and the performances of such anti-slip schemes based on a disturbance observer are to a large extent affected by noise in the system.

In this paper, a wheelset model is established and its dynamics under stable and slip conditions are studied and compared. Based on this, novel slip detection methods are proposed. In addition, a torque control unit is adopted for a vector controlled three phase induction motor to achieve re-adhesion.

2. WHEELSET MODEL

In order to analyse slip phenomenon in the traction system, a wheelset model is established as shown in Fig.1. The right side driving wheel is driven by an inverter-fed induction motor and connected to the other wheel through a shaft, which is considered to have finite torsional stiffness.
The dynamics of the system are governed by equations (1) to (3). Equation (1) describes the dynamics of the right wheel, which is driven by an induction motor through a gearbox; (2) describes the motion of left wheel; (3) is the description of the torsional torque, and (4) gives the general form of the traction torque.

\[
T_m - T_r - T_v = J_r \frac{d\omega_r}{dt} \tag{1}
\]
\[
T_r - T_v = J_l \frac{d\omega_l}{dt} \tag{2}
\]
\[
T_v = k_t \int (\omega_r - \omega_l) dt + c(\omega_r - \omega_l) \tag{3}
\]
\[
T_v = \mu M_v g \cdot r \tag{4}
\]

where, \(T_m\) is the driving torque transmitted from the motor through a gearbox, \(s_T\) is the torsional torque, \(trT\) is the traction torque contributed by the right side wheel-rail adhesion, \(tlT\) is the traction torque contributed by the left side wheel-rail adhesion, \(J_r\) and \(J_l\) are moment of inertias of right and left side respectively, \(s_k\) is the torsional stiffness of the shaft, \(c\) is the viscous damping of the shaft, and \(\omega_r\) and \(\omega_l\) are the angular velocity of the right and the left wheel respectively. \(\mu\) is called the traction coefficient, and its maximum value is called adhesion coefficient. \(v_M\) is the equivalent vehicle mass of each wheel. In this paper it is assumed that each wheel shares vehicle mass evenly.

Since the damping of the system is very small, we assume that \(c = 0\). Then the state equation of this two inertia system is given by (5) where the state variables are chosen as \(x = [\omega_r \omega_l \theta_r]^T\) with \(\theta_r\) defined as \(\theta_r = \int (\omega_r - \omega_l) dt\):

\[
\begin{bmatrix}
\dot{\omega}_r \\
\dot{\omega}_l \\
\dot{\theta}_r
\end{bmatrix} =
\begin{bmatrix}
0 & 0 & \frac{k}{J_r} \\
0 & 0 & \frac{k}{J_l} \\
1 & -1 & 0
\end{bmatrix}
\begin{bmatrix}
\omega_r \\
\omega_l \\
\theta_r
\end{bmatrix} +
\begin{bmatrix}
-1 \\
0 \\
0
\end{bmatrix} T_v +
\begin{bmatrix}
0 \\
0 \\
-1 \frac{J_r}{J_l}
\end{bmatrix} T_v \tag{5}
\]

Then, the natural frequency \(f_n\) of the two inertia system is given by:

\[
f_n = \frac{1}{2\pi} \sqrt{\frac{k}{J_r} + \frac{k}{J_l}} \tag{6}
\]

In equation (10), the traction torques \(T_v\) and \(\Delta T_v\) have nonlinear properties and are determined by wheel-rail contact characteristic, which can be expressed as a group of slip curves as shown in Fig.2. Each slip curve gives a rule for how the traction coefficient varies with slip ratio, which is defined in equation (7) where \(\lambda\) is slip ratio, \(\omega_e\) is equivalent angular speed of the vehicle at the contact point.

\[
\lambda = \frac{\omega_e - \omega_l}{\omega_l} \tag{7}
\]

In Fig.2, curve I with the higher adhesion coefficient, can be considered as a dry condition which indicates a good contact, and curve II represents a very wet/snowy condition as a poor case. It can be seen that the traction coefficient varies nonlinearly with slip ratio. The left side of the maximum point is a stable region where the slip curve has a positive slope, and on the other side, the traction coefficient decreases as the slip ratio increases. Hence, a slip condition is associated with negative slope.

An approximation of traction torque can be obtained through linearization of the slip curves. Equation (8) and (9) give such approximation of the right and left hand side traction torque:

\[
\Delta T_v = k_1 \omega_e \tag{8}
\]
\[
\Delta T_{v'} = k_2 \omega_e \tag{9}
\]

here, \(k_1\) and \(k_2\) are values related to slopes at the operating points of the right and left wheel on their respective slip curves.

Using (8) and (9), the small signal model is given as:

\[
\begin{bmatrix}
\Delta \omega_r \\
\Delta \omega_l \\
\Delta \theta_r
\end{bmatrix} =
\begin{bmatrix}
\frac{k}{J_r} & \frac{k}{J_l} & 0 \\
0 & -\frac{k}{J_l} & \frac{k}{J_l} \\
1 & -1 & 0
\end{bmatrix}
\begin{bmatrix}
\Delta \omega_r \\
\Delta \omega_l \\
\Delta \theta_r
\end{bmatrix} \tag{10}
\]

It is clear from equation (10) \(k_1\) and \(k_2\) add extra damping to the system, and their values are closely related to the stability of the system. Generally, positive \(k_1\) and \(k_2\) values indicate a stable condition, and negative ones denote an unstable condition, which give rise to self-excited vibrations.

A simulation model was built in SIMULINK together with the SimPowerSystems toolbox to model the induction motor vector control unit as shown in Fig. 3.
3. SLIP PHENOMENON AND DETECTION METHOD

In this section, the results of simulation are given to study wheelset dynamics in slip condition, and based on that two detection methods are presented.

3.1 Slip phenomenon

In all simulations, the torque demand of motor $T_e$ was as shown in Fig. 4.

Fig. 4 Torque demand profile

At a specified time e.g. $t=4.5s$, the wheel-rail contact condition is changed from a dry case to a poor case to develop a slip condition. Fig. 5 gives the wheel-rail adhesion data used in the simulation. At $t=4.5s$ the contact curves of both wheels are switched from curve ① to curve ②.

Fig. 5 Contact curves used in Simulation

Table 1 gives the motor and vehicle parameters used in simulation. All the results are given and analysed with a shaft natural frequency of 40 Hz unless stated otherwise.

Table 1 Simulation parameters

<table>
<thead>
<tr>
<th></th>
<th>$T_{e,\text{max}}$</th>
<th>$J_r$</th>
<th>$J_i$</th>
<th>c</th>
<th>$f_s$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1500 Nm</td>
<td>133.2 kgm$^2$</td>
<td>62.8 kgm$^2$</td>
<td>0</td>
<td>40Hz</td>
</tr>
</tbody>
</table>

Fig. 6 shows speeds of both wheels and vehicle from the simulation results. It is obvious that after $t=4.5s$ when the contact condition is switched to a poor one, wheel speed rises much faster than that of vehicle, and slip happens.

Fig. 6 Angular speeds of wheels and vehicle

Fig. 7 shows the torsional torque. It can be seen that when slip occurs a vibration appears. The frequency of this vibration is 40Hz, which is the natural frequency of the two inertia torsional system.

Fig. 7 Torsional torque

From equation (10), it is known that $k_1$ and $k_2$ affect system stability. In the simulation, two sets of $k_1$ and $k_2$ were used. Table 2 gives one set of $k_1$ and $k_2$ values. They are obtained from the stable operating conditions in the simulation. The corresponding eigenvalues of matrix A1 in (10) are also given in the table. It is clear that all eigenvalues are real and negative, and this confirms that the system is stable.

Table 2 $k_1$ and $k_2$ values and eigenvalues (stable)

<table>
<thead>
<tr>
<th>$k_1$</th>
<th>$k_2$</th>
<th>eigenvalues</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.43e5</td>
<td>2.43e5</td>
<td>-22.39, -1813, -3835</td>
</tr>
</tbody>
</table>

Table 3 gives the other set of $k_1$ and $k_2$, and they are obtained from slip operation condition. One of the eigenvalues is positive real, and the others are complex conjugate with positive real part. Hence, the system is unstable. The oscillation frequency obtained from the imaginary part of complex eigenvalues is
40Hz, and this agrees with the natural frequency of the torsional system.

Table 3 $k_1$ and $k_2$ values and eigenvalues (slip)

<table>
<thead>
<tr>
<th>$k_1$</th>
<th>$k_2$</th>
<th>eigenvalues</th>
</tr>
</thead>
<tbody>
<tr>
<td>-2175.8</td>
<td>-2175.8</td>
<td>22.23, 14.38 ± 250.77i</td>
</tr>
</tbody>
</table>

In practice, when a vehicle is built, the natural frequency of wheelset axles is fixed. So this typical vibration can be used as an indication of slip condition.

3.2 Slip detection method using running FFT

When a torsional vibration occurs, the speeds of two wheels are directly affected. Such vibrations can be detected from speed information. Fig. 8 shows the difference between the right wheel and left one. It shows that the torsional vibrations affect the speeds, and that makes it possible to detect the slip.

In this section, a running FFT is used to detect the vibration from information of the speed difference between the right wheel and left wheel. The running FFT uses a 0.2s window to extract the information. From the FFT, the magnitude of the natural frequency component is determined. When the value exceeds a threshold, slip has been assumed to occur. Fig. 9 shows magnitude of the natural frequency component. When slip occurs, the magnitude increases due to torsional vibration. Setting a magnitude of 0.1 as the threshold, slip can be detected after 0.2s, which would be satisfactory in practical situation.

To demonstrate that frequency of the vibration when slip occurs is directly determined by the shaft parameters. The shaft natural frequency is changed to 60Hz. Fig.10 shows the speed difference of the two wheels. The result shows the vibrations of this frequency at slip condition.

In this section, another method is introduced to obtain vibration magnitude as shown in Fig.12. A band-pass filter is used to extract the signals of the shaft natural frequency. Then, the absolute values are obtained. After that a low pass filter with a very low cut off frequency is used to catch the magnitude augment due to torsional vibration.

3.3 Slip detection method using filters

In this section, another method is introduced to obtain vibration magnitude as shown in Fig.12. A band-pass filter is used to extract the signals of the shaft natural frequency. Then, the absolute values are obtained. After that a low pass filter with a very low cut off frequency is used to catch the magnitude augment due to torsional vibration.

Fig. 11 gives the magnitude of 60Hz component. And once again, the slip condition can be determined after 0.2s.
3. RE-ADHESION CONTROL

A torque control unit is added to correct the motor torque demand such that when slip condition is detected, the demand is reduced.

When re-adhesion is achieved which is indicated by the disappearance of vibrations, the present torque demand is kept to maintain this stable traction level. The amount of torque reduction is determined by applying the slip ratio which is obtained from equation (7) to a PI unit as shown in equation (11):

$$ T_{red}(s) = (K_p + \frac{K_i}{s})\lambda(s) $$  \hspace{1cm} (11)

Here, $K_p$ and $K_i$ are the proportional and integral parameters respectively, and need to be tuned appropriately.

Fig. 15 gives the wheels and vehicle speeds, and shows that rapidly increasing wheel speeds are drawn back to vehicle’s within 3 seconds due to the action of re-adhesion scheme, and then increase stably in the new contact condition. Fig. 16 gives the torsional torque, and shows that the torsional vibration disappears when re-adhesion is achieved.

Fig. 17 shows magnitude of the natural frequency component from the running FFT analysis results, and this is used to indicate slip and re-adhesion condition in the simulation. Fig. 18 shows the estimation of vibration magnitude using filter method and it agrees well with running FFT results.

4. CONCLUSION AND FURTHER WORK

In this paper, a wheelset model has been presented. Based on the study of its dynamics, it is observed that a typical self-excited torsional vibration appears in slip conditions. Then effective slip detection methods have been developed using this vibration information. The first method uses a running FFT to determine the spectrum of the speed difference at a given instance. The magnitude of the specified frequency component is then selected from the FFT data. The second method uses a combination of bandpass and low pass filters to obtain magnitude information of the specified frequency component. Simulation results show that both methods indicate the slip condition in a timely fashion. Based on the slip detection scheme, a torque control scheme was introduced to reduce the torque to enable the wheelset system to regain...
adhesion after a slip condition is indicated. The simulation results show that the re-adhesion scheme can bring the slipping system back to a stable condition within 3 seconds, and then exert appropriate traction effort after re-adhesion.

The planned future work will concentrate on two aspects: one is to study the possibility of detecting the shaft vibration from the motor operating parameters such as stator currents, and the other issue of interest is to try using vibration parameters directly to adjust the motor torque demand.

REFERENCES


