MODELLING AND STUDY OF A RAILWAY WHEELSET WITH TRACTION

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Abstract

This paper studies the dynamic behaviour of a railway wheelset in different operating conditions, and investigates the effect of traction sub-system on the stability and performance of the wheelset. The system studied combines the conventional wheelset with a traction subsystem and a wheelset stabilisation control subsystem. The mathematical model of the wheelset including the two control sub-systems is presented. The issues of stabilising the wheelset passively or actively are discussed and the detail of the traction control is given. A comprehensive analysis is made to study characteristics of the wheelset and more importantly interactions between the wheelset control and the traction subsystem. One of the key conclusions from the study is that the conventional passive solutions for the wheelset instability is reasonably robust in the presence of tractive effort and traction dynamics, whereas some of the newly proposed active control approaches require more careful considerations.

1. Introduction

Suspensions of railway vehicles are deigned to give the best possible performances in terms of the stability, the curving and the ride comfort etc. Conventional passive suspensions rely on the optimisation of parameters such as masses, spring stiffness, dampers, geometrical properties etc to provide reasonably good responses to track inputs and other disturbances. More recently, many studies have shown that active suspensions can deliver the performance improvement far beyond what is possible with passive means [1]. The research into the active steering of the railway wheelset has shown that the more complexity offered by active control may be used to solve the difficult design trade-off between the high speed stability and low speed curving [2] and more importantly to enable mechanically simpler vehicle suspensions to be proposed [3].

On the other hand, the traction and braking control systems for railway vehicles have traditionally been considered as a subsystem somewhat unrelated to the vehicle dynamics, and largely developed independently [4]. However, the delivery of traction and braking forces is achieved through the contact patches between the wheels and the rail, which are also related to the stability and steering control of a railway vehicle. On modern conventional vehicles, all suspensions are formed using passive components, the interactions are less a problem and there have not been a great urgency for a systems approach. If the active steering is used to control the vehicle dynamics, both the traction and wheelset controls will all be electronically controlled and thus it will be possible to optimise the use of the contact patches through an integrated control system.

However the integration of the sub-systems (functions) is not guaranteed a simple process because of the complexities of the wheel-rail contact mechanics, which is even more difficult to deal with under the influence of traction. In this study, a basic configuration of the railway wheelset is investigated and the overall aim is to study the effect and interactions between the traction and stability control subsystems. Although a conventional railway vehicle consists of four wheelsets connected onto two bogies, which in turn connected to the body frame via suspension components, this paper is focussed on a single wheelset, which is the key element of a vehicle as far as the wheel-rail contact is concerned. The results can be readily extended and applied to complete vehicles. In this paper, a conventional solid-axle wheelset with the traction and the active control subsystems is defined and the mathematical model of the system is developed. A number of different cases are considered, including a rigid axle wheelset; a wheelset with a flexible axle and a wheelset combined with a traction subsystem. Both conventional passive stabilisation and more advanced active controls are addressed in the study in order to expose their effects on the wheelset stability of the wheelset. Significances of traction and transmission mechanics to performances of the wheelset are also revealed.

2. Modelling

The study is focussed on the railway wheelset, where primary/critical interactions between the active steering and traction sub-systems take place. On a complete vehicle, other components such as bogies and body frame will also have some influence via suspension connections. But those are not considered essential and will therefore only be included at a later stage of the study. Figure 1 shows the basic configuration of the wheelset used in the study.

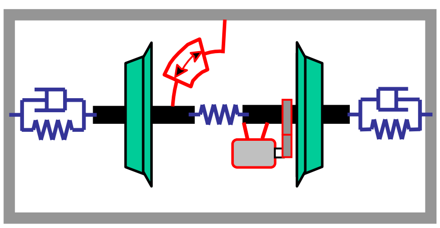

Figure 1. Wheelset configuration

The wheelset consists of two wheels mounted onto a common axle with a stiffness of k_t (typically to give the relative rotational mode of the two wheels of 40-60Hz). The traction subsystem consists of a DC motor and the traction transmission dynamics. Modern railway vehicles are typically equipped with AC motors for traction. However advances in high power switching devices as well as motor control methods have enabled an induction motor to behave very similarly to a separately excited DC motor in the range of frequencies of interest for this application. Therefore the complexity of the AC traction motor and its associated power electronics and control is substituted in this work by a DC motor with the separate excitation control of the torque and flux producing currents, with the reasonable expectation that no significant difference will be introduced in the dynamic behaviour of the wheelset. Two control approaches to stabilize the inherently unstable wheelset are included in the study. One is the conventional passive approach where longitudinal springs are used to ensure the stable running. The other is a novel active control technique, where a torque on the wheelset is provided by a controlled actuator to stabilise the vehicle [4]. More detail for the wheelset stabilisation issue is given in the next section.

The behaviour of a wheelset is dominated by creep forces developed at the contact points with the rail, which are largely proportional to the relative velocity between the two metal surfaces. This study is based on a linear creep law and the non-linearity due to wheel-rail profiles and the variation of creep coefficients may be considered using look-up tables as a non-linear function of the wheelset lateral movement and overall creep levels respectively. The creep force of the right hand wheel in the longitude direction can be represented as

$$F_{xR} = f_{11}[-1+\frac{L_g}{v}\dot{\psi}+\dot{\phi}_R(r_0+\lambda y)/v-\frac{L_g}{R}] \qquad (1)$$

where the four terms are caused by the forward speed, the yaw velocity, the rotation velocity, and the difference in length of the inner rail and the outer rail respectively.

A longitudinal creep force on the left hand wheel and a lateral creep force on the right/left hand wheel are similarly obtained,

$$F_{xL} = -f_{11}[1+\frac{L_g}{v}\dot{\psi}-\dot{\phi}_L(r_0-\lambda y)/v-L_g/R] \qquad (2)$$

and

$$F_y = -f_{22}(\dot{y}-v\psi)/v \qquad (3)$$

A set of equations for the whole system shown in Fig 1 can then be derived, where equations 4 and 5 represent the lateral and yaw motions of the wheelset. Equations 6 and 7 are for the rotational movements of the two wheels; equation 8 describes the longitudinal dynamics; and equations 9 and 10 represent the dynamics of the traction motor and gearbox.

$$m\ddot{y}+(2f_{22}/v+c_{lateral})\dot{y}+k_{lateral}\,y-2f_{22}\psi-mv^2/R=-mg\theta \qquad (4)$$

$$J\ddot{\psi}+2f_{11}L_g^{\ 2}\dot{\psi}/v+L_g f_{11}\lambda/v(\dot{\phi}_R+\dot{\phi}_L)+L_g f_{11}r_0\dot{\phi}_R/v-L_g f_{11}r_0\dot{\phi}_L/v-2f_{11}L_g^{\ 2}/v=-T_\psi \qquad (5)$$

$$I_w\ddot{\phi}_R + f_{11}r_0{}^2\dot{\phi}_R + k_t\phi_R - k_t\phi_L + r_0 f_{11}L_g\dot{\psi}/v + r_0 f_{11}\lambda/v\dot{\phi}_R y = r_0 f_{11}(1+L_g/R) \quad (6)$$

$$I_w\ddot{\phi}_L + f_{11}r_0{}^2\dot{\phi}_L - k_t\phi_R + (k_t + k_{gear})\phi_L - r_0 f_{11}L_g/v\dot{\psi} - r_0 f_{11}\lambda/v\dot{\phi}_L y - k_{gear}\theta_m/n = r_0 f_{11}(1-L_g/R) \quad (7)$$

$$(m + m_v/2)\dot{v} - f_{11}/v[\lambda y(\dot{\phi}_R - \dot{\phi}_L) + r_0(\dot{\phi}_R + \dot{\phi}_L)] = -2 f_{11} - F_{run} \quad (8)$$

$$L_a\frac{di_a}{dt} + R_a i_a + K_m\dot{\theta}_m = u_a \quad (9)$$

$$nI_m\ddot{\theta}_m + k_{gear}\theta_m/n = nK_m i_a + k_{gear}\phi_L \quad (10)$$

3. Control Strategies

The conventional wheelset for the railway vehicle is composed of two wheels rigidly fixed to a common axle. The wheels are profiled, typically with non-linear characteristics. In the development of control strategies, the concity is usually treated as a constant, hence the use of the linearised model. However to ensure the necessary robustness of the system, a range of the conicty values (e.g. 0.05-0.4) is normally examined in the design process to account for possible variations.

It is well known that the solid axle wheelset has the ability of natural centring and curving. A drawback of the arrangement is that when unconstrained the wheelset exhibits a sustained oscillation in the lateral plane often referred to as the "*wheelset hunting*". This is overcome on conventional railway vehicles using springs connected from the wheelset to the bogie or the body of the vehicle. However this added stiffness degrades the curving/centring performance of the wheelset [5].

Active controls, where actuators are used to replace the yaw stiffness, provide an opportunity to remove the trade-off issue between the stability and the curving performance. However, development of a suitable controller is not always a straightforward exercise, as a railway vehicle is dynamically very complex, highly inter-active and non-linear. A number of control schemes have been proposed for the active steering of the railway wheelset [2,6-8]. This paper uses an active yaw damping approach where an active control torque is set to be proportional to the lateral velocity of the wheelset to provide a stabilising action on the wheelset [8], and a phase lead compensator is used to improve the yaw damping. Obviously it is difficult to see how this principle can be implemented using the conventional passive means, but it is possible by measuring the lateral velocity/acceleration of the wheelset as the feedback and apply a torque via an actuator in the yaw (or longitudinal) direction.

For the specific situation considered, the controller is designed as

$$K_s = 1.05\times10^5 * 7.5\frac{s+37}{s+274}.$$

For the control of the traction sub-system, the most commonly used controller is the Proportional-and-Integral (PI) control, which has been proved sufficient in many industrial applications. The controller used for the traction motor in this study is given as:

$$K_t = \frac{0.5s+1}{0.5s}$$

The aim of this study is to develop the necessary and accurate model, with which the critical interactions between the wheelset control and traction control can be studied, which will include assessments for the effect of different control strategies on the interactions.

4. Simulation and Analysis

The contact patch between the wheel tread and rail head is the interface of interactions between the traction and active steering sub-systems. In general, the increase of creep tends to reduce creep coefficient.. The application of tractive effort can significantly increase creep in the longitudinal direction (upto 10% of the normal force and even more in braking), and the effect can be made much worse in bad wheel/rail contact conditions. Therefore it is necessary to examine how the wheelset stability is affected by the reduced creep coefficient especially in the region where the creep is near (or beyond) the point of traction slip.

Figure 2 shows how the eigenvalues of a passively stabilised solid axle wheelset move with the creep coefficients which is varied from 1 to 10 *MN*. The yaw stiffness seems to cope well with the lower coefficients in term of the kinematic mode of the wheelset, the damping of which can be actually improved. The frequency of the high-frequency mode varies almost proportionally with the creep coefficient, but the mode appears to be well damped.

The active yaw damping of a pure gain is less robust for the kinematic mode as shown in Figure 3. The wheelset becomes unstable when the creep coefficient is reduced to about 5 *MN*. The high frequency mode is always over-damped in this case, although the frequency changes with the coefficient.

The use of a phase lead compensator in additional to the pure gain for the active yaw damping will provide much improved damping for the kinematic mode, however the high frequency mode is now in danger of becoming unstable as demonstrated in Figure 4. This can become even worse when the mass (or the moment of inertia) of the traction subsystem and the axle flexibility are taken into account as given in Figure 5.

For time history simulations, a deterministic track input is used, which is a typical high-speed curve, including a curve track and two transitions connecting to straight tracks on either side. The track is canted through the curve to reduce the lateral acceleration experienced by the passengers, the resultant acceleration being referred to as "cant deficiency".

A traction demand nK_mI_a of 2.46 (*kN m*) and a running resistant force F_{run} of 5.47 (*kN*) are applied to the wheelset at time t=5 second simultaneously so that the dynamic effect of the traction can be investigated without the vehicle speed being significantly changed. It is a common practice that the speed should be kept more or less constant for the study of wheelset dynamics, which can vary significantly with speed.

Only results from normal wheel/rail contact conditions are presented below because of space and time constraints, but further findings will be reported in due course. Figure 6 shows the lateral displacement the wheelset with passive stabilisation and active control under the influence of traction. Figure 7 compares the control effort of the active steering with the suspension torque with passive control. Figure 8 presents the creep forces of the two control approaches.

As expected the active control provides a solution for the design conflict of the stability and curving which is not possible with the passive means. The lateral displacement of the wheelset follows more or less the pure rolling line - no longitudinal creep. Consequently the creep forces are smaller, which implies that more adhesion is available for the traction and/or braking. The control effort required for the active stabilisation is smaller (0 on constant curves) than the suspension force/torque in the passive system.

There are clear transient effects on all responses, which may be the consequence of the tractive effort exciting the kinematic mode of the wheelset.

It is found that the stiffness in the transmission dynamics of the traction sub-system has an adverse effect on the overall damping as well as transient performance of the wheelset. When the inappropriate stiffness is used, the traction dynamics may interfere with the wheelset kinematical mode and the phenomenon of a beating between two modes with similar frequencies as shown in Figure 9.

5. Conclusions

This paper has presented the modeling of a railway wheelset with the wheelset stabilization and traction control subsystems and it has studied the behavior of the system under the influence of traction.

It has been shown that, compared with passive means, the main advantages of an active control is that better curving and stability can be provided with much reduced creep forces and hence more scope for increasing traction and braking.

On the other hand, the traction sub-system can have a significant effect on the active control discussed in the paper in terms of the stability, the availability of traction performance and transient responses. There may even be some interference between the traction transmission dynamics and wheelset kinematical mode. Therefore greater attention is required to address the issue more carefully.

Further work is planned to study the effect of the non-linear contact law due to the profiled wheel and rail and to develop integrated solutions.

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Appendix: Symbols and Parameters

a	Parameter (7.5) of phase lead compensator
$c_{lateral}$	Damping per wheelset (24 *kN s/m*)
F_{run}	Vehicle running resistant (5.4 *kN*)
F_{xL}	Longitudinal creep force on the left wheel
F_{xR}	Longitudinal creep force on the right wheel
F_y	Lateral creep force on the wheels
F_{11}	Longitudinal creepage coefficient (10 *MN*)
F_{22}	Lateral creepage coefficient (10 *MN*)
I	Wheelset yaw inertia (600 $kg\ m^2$)
I_m	Motor inertia (11 $kg\ m^2$)
I_w	Wheelset rotation inertia (35 $kg\ m^2$)
i_a	Motor current
K_{gain}	Parameters of phase lead compensator (1.05e5)
K_m	Motor machine constant (4.28)
K_p	Parameter of PI control for traction motor (1)
k_t	Axle rotation stiffness so that corresponding axle torsion frequency is 40*Hz*
k_{yaw}	Yaw stiffness (4.7e6)
L_a	Motor inductance (4e-4*H*) displacement
L_g	Half gauge of wheelset (0.7 *m*)
m	Wheelset mass (1250 *kg*)
m_v	Vehicle mass (20,000 *kg*)
n	Gearbox ratio (5)
R	Radius of the curved track (2300 *m*)
R_a	Motor resistance (0.04 *ohms*)
r_0	Wheel radius (0.45 *m*)
T_ψ	Controlling torque for wheelset
u_a	Motor voltage
v	Vehicle travel speed (83.3 *m/s* or 300 *km/hour*)
y	Lateral displacement of wheelset
γ_{xL}	Longitudinal creepage on the left wheel
γ_{xR}	Longitudinal creepage on the right wheel
γ_y	Lateral creepage on the wheels
θ	Cant angle of the curved track (6^0)
θ_m	Motor rotation displacement
λ	Wheel conicity (0.2)
τ_i	Parameter of PI control for traction motor (0.05)
ϕ_L	Rotation displacement of the left wheel of wheelset
ϕ_R	Rotation displacement of the right wheel of wheelset
ψ	Yaw displacement of wheelset
ω_1	Parameter of phase lead compensator (37)
ω_2	Parameter of phase lead compensator (274)

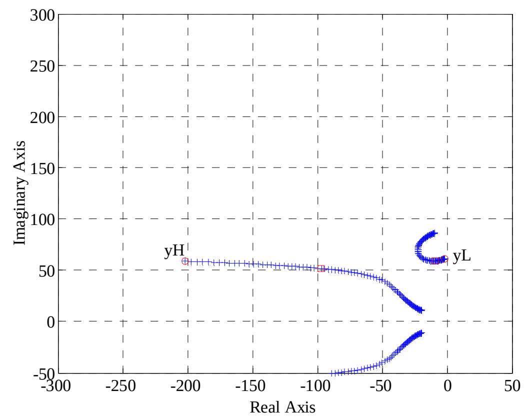

Figure 2 Wheelset modes (passives stabilisation)

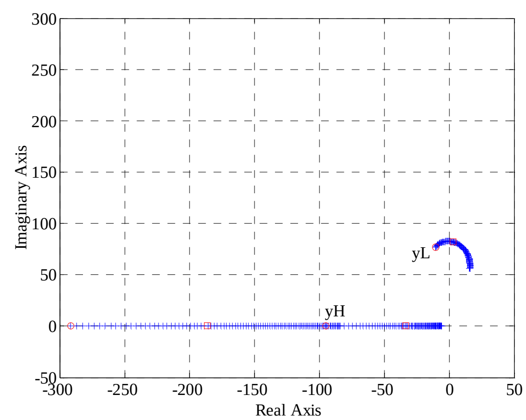

Figure 3 Wheelset modes (active yaw damping)

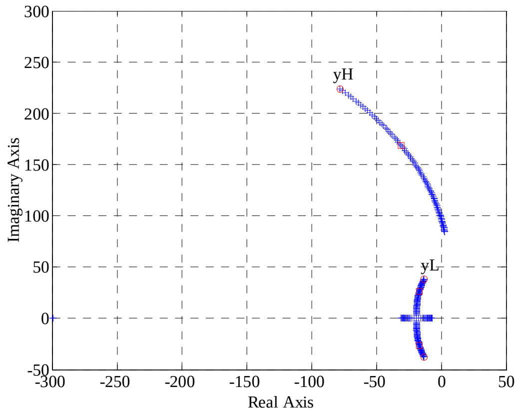

Figure 4. Wheelset modes (active yaw damping with phase compensation)

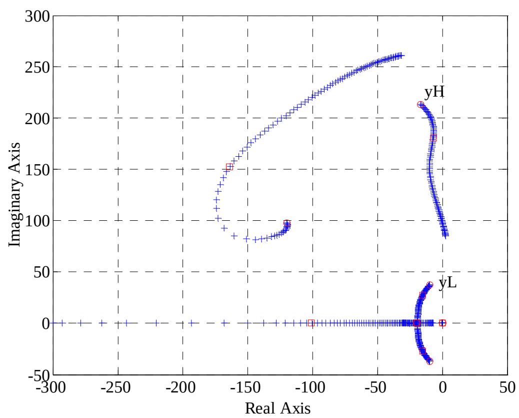

Figure 5 Wheelset modes (with axle flexibility)

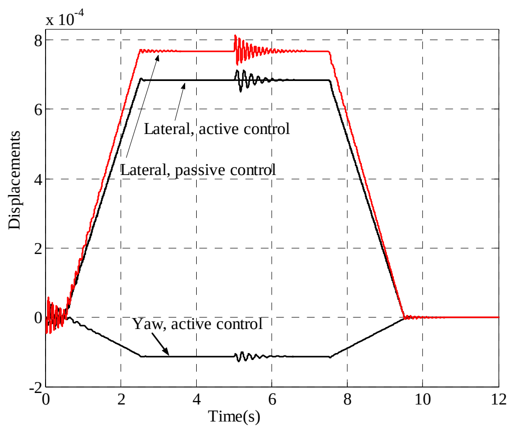

Figure 6. Lateral displacements of the wheelset

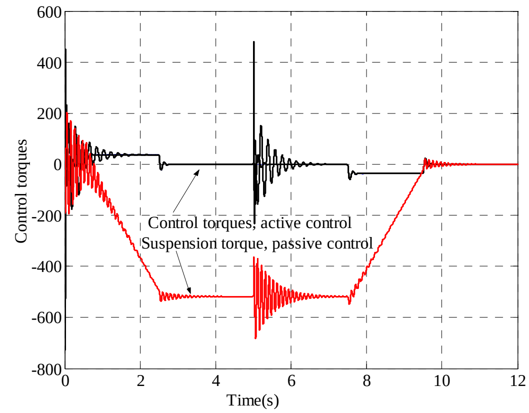

Figure 7. Control effort

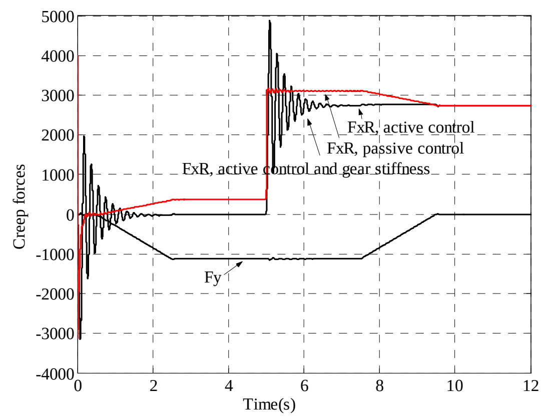

Figure 8 Creep forces

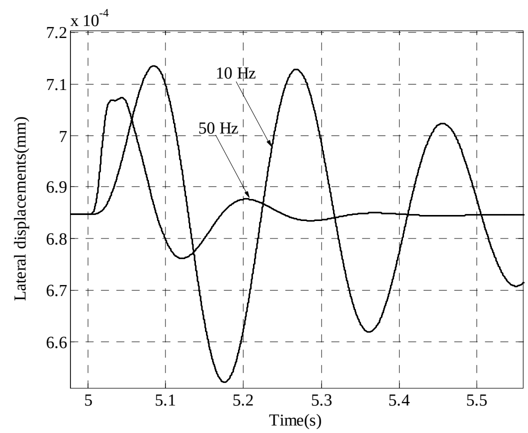

Figure 9. Transient effect of interaction