Abstract

It has been shown that active steering control can improve steering performance in two axle bogies whilst also improving high speed train stability. In freight operations passive steering three axle bogies have been produced in large numbers for reported benefits including improved adhesion performance and reduced wheel wear. However, passive steering bogie designs have been shown to perform similarly to straight rigid frame bogies under high traction forces due to the loss of longitudinal creep forces. These factors have not been modelled for hauling locomotives and published in public domain though commercial testing by locomotive manufacturers has occurred. A gap therefore exists in the knowledge of steering bogie performance in freight locomotive operations particularly for three axle bogies.

The following thesis explored traction curving operations for hauling locomotives considering, and in particular, the effect of high adhesion levels, rail-wheel contact friction and lateral components of coupler forces. Both passive and active designs were analysed and new designs were proposed. Methods of active control and data measurement were also compared and discussed. The following three axle bogies were evaluated and compared.

- Bogies with passive steering: Rigid, yaw relaxation, self steering, forced steering.

- Bogies with actively controlled designs: Actuated Wheelset Yaw (AWY) bogies with creep force control; steering angle precedence control; yaw angle precedence control.

- Bogies with new designs developed as part of this thesis: Actuated Yaw Force Steered (AY-FS) bogies with creep force control; precedence control; and curve estimation control; Actuated Yaw Variable Steering (AY-VS) bogies with precedence control and curve estimation control.

Active bogie designs were also analysed with idealised control systems to provide a datum for comparison.
The following thesis obtained the following results:

- A new definition for a condition designated as “ideal steering” under traction which is distinct from previous published criteria that only consider steering at low traction levels;

- Confirmation that passive steering bogie curving performance deteriorates to rigid bogie performance levels for yaw relaxation and self steering bogie designs as traction saturates the friction limit of the wheel rail interface;

- Passive forced steered bogies achieve partial steering in traction curving even under traction saturated friction conditions;

- It is shown that the control of active bogies must be independent of the creep forces to retain function in traction curving.

- New bogie designs have been developed to actively control the bogie yaw position and wheelset steering angle with independence from wheel rail creep forces.

- A new actively controlled bogie which includes forced steering linkages was proposed and patent submitted. This bogie is called AY-FS (Actuated Yaw Forced Steering).

- A further improvement of the AY-FS bogie was proposed and patent submitted. The improved design includes variable steering. This bogie is called AY-VS (Actuated Yaw Variable Steering).

- The analysis of the new bogie designs and controllers shows there is always a compromise between the design requirements for curve transition steering and with the requirements for bogie stability. The investigations showed that some further benefits from active control systems could be achieved with switching systems to allow optimal control for both straights and curves.
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Research Thesis:

THREE AXLE LOCOMOTIVE BOGIE STEERING, SIMULATION OF
POWERED CURVING PERFORMANCE PASSIVE AND ACTIVE
STEERING BOGIES.

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Date: December 2009
Certificate of Authorship and Originality of thesis (Declaration)

The work contained in this thesis is a direct result of original work conducted by myself and has not been submitted for the award of a degree or diploma at any other tertiary institution in Australia or internationally.

Signed: ___________________ Dated: 9/12/2009

Scott Andrew Simson

Author

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Scott Andrew Simson
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Published Work


Submitted and Unpublished

1. Introduction

1.1. Overview

Conceptually active steering control is a well established technology with numerous technical papers and a major book chapter. The published investigations of active steering have focused on high speed passenger train configurations. The simulation completed of these ‘active’ bogies is limited to bogies with long wheel base, two axles operating without traction, without coupler forces and using either highly linear or fully linear suspension characteristics. The simulations in some cases use linear conicity models and rarely investigate curving situations of flange contact or curves where the wheel rail conicity is insufficient for the radius of curve. These studies have shown that active controlled bogies can give improved steering performance whilst improving high speed train stability.

 Freight locomotives used in heavy haul operations have significant differences. Heavy haul operations have high numbers of three axle tractive bogies, operating with high traction loads in curves on mountain grades. The suspension arrangements of freight locomotives often feature friction damping. Coupler loads can be extreme with multi locomotive push and pull train consist. The curve radius seen on the heavy hauling mountain operations can also be far tighter then the minimum radius for which the wheel rail effective conicity can self steer.

 Freight operations since the 1993 have seen large numbers of passive steering three axle bogies enter service following the release by EMD of a self steering 3 axle bogie with the mid axle disconnected from the steering. The EMD self steering bogie reported benefits including improved adhesion performance and reduced wheel wear (Sections 2.3.5). However, passive steering bogie designs have been shown to perform similarly to straight frame bogies under high traction forces due to the loss of longitudinal creep forces at high adhesion demands. These factors have not been modelled for hauling locomotives and
published in public domain though commercial testing by locomotive manufacturers has occurred.

Recent studies on bogie rotation friction management [1] have shown a high sensitivity to the cant deficiency and coupler forces in the steering performance of bogies in general. Available literature on locomotive steering performance is generally focused on cant equilibrium curving despite most train operations being cant deficiency. The exception to this occurs in heavy haul operations on steep grade operations where cant deficiency can vary due to the locomotive power being insufficient to maintain maximum track speeds.

Development with active steering designs has been limited. Shinkansen bogies have been made with active secondary suspensions controlling yaw and lateral movements of the bogie [2], [3]. Control design for these bogies is only for the stability. Active two axle steering designs have been in prototype development in Italy [4] Switzerland [5] and Japan [2], [5]. Designs have been either yaw actuation of the secondary suspension or steering angle actuation of the wheelsets. All of these developments have focused on high speed or tilting passenger train operations.

Several gaps therefore exist in the knowledge of steering bogie performance as applied to freight locomotive operations particularly for three axle bogies.

1.2. Objective of Project

The objective of the PhD program is:

- Develop a new active steering design for hauling locomotives to minimise creep forces and thus improve adhesion during curving.

The enabling objectives include:

- Defining curve steering ideals as relevant to hauling locomotives that minimises creep forces.
- Develop active steering control systems based on achievable sensing technologies (accelerometers, gyroscopes, displacement transducers or load cells) and applying forces to the bogie in response.

- Assessment of the performance gains of passive steering bogie and active steering bogies using curve sensing controllers and precedence controllers in tractive curving for hauling freight locomotives.

- Assessment of hunting stability of rigid, passive steering, and active steering bogies using curve sensing controllers and precedence controllers.

### 1.3. **Scope**

The plan for the research program was to start with a focus on passive steering mechanisms with later recourse to active steering designs. Simulation modelling was limited to the current commercial vehicle dynamics package VAMPIRE® that is provided by CQU. Refinements eg: Polach model for creep saturation (section 2.2.4), wheelset angular momentum modelling and traction motor dynamics are not supported. The listed simulation refinements would require use of an alternative commercial package or CQU development of its own advance vehicle dynamics package. Upgrades to VAMPIRE® software occurred during the life of the project but have not included these simulation capabilities.

Activities in order that they appear in the thesis were:

1. Review of all published literature, products and patents. (see Chapter 2)

2. Determination of the steering task for quasi static curving and requirements of perfect or ideal steering for locomotive performance. The influence of coupler forces on steering in locomotive positions: head end of train, at second head locomotive, and pusher locomotives. (see Chapter 3)

3. Transition curve steering task requirements (see section 3.4)

4. Steering performance evaluations of passive steering three axle bogie designs for locomotives, designs of: rigid; floating mid axle rigid; yaw relaxation; Self steering
bogies: three axle self steer, two axle self steer with floating mid axle, half radial
three axle self steer; Force steered bogies: three axle force steered, two axle with
floating mid axle force steered. (see Chapter 4)

5. Alternative bogie designs, three axle locomotive bogies. (see Chapter 5)

6. New bogie steering performance evaluation. (Chapter 6)

7. New bogie steering control algorithm design for active control. (see Section 6.2)

8. Caparison performance evaluation of active steering bogies to the new bogie design.
   (see Chapter 7)

9. Control algorithms and performance evaluations for advanced new bogie designs (see
   Chapter 8)
1.4. Glossary

The following terms, acronyms and notations laid out here are used in the text.

1.4.1. Terms

Articulated Steering: bogies with linkages connecting yaw movements of adjacent vehicle bodies to the yaw and steering movements of the bogie.

Bogie: British terminology for a railway truck is a sub frame including wheelsets located under the vehicle body and able to yaw relative to the vehicle body assisting curving.

Bogie Hunting: lateral instability of bogie can occur in multiple forms either yaw angle oscillation of the entire bogie typically involving steering angle oscillations of the wheelsets or otherwise the wheelsets oscillate in warp angle with only lateral motion of the bogie.

Bogie yaw oscillation mode: Bogie hunting, a vibration of the bogies yaw angle position.

Cant: the difference in the level of the two rails or crosslevel. Cant is applied to the rail to reduce lateral curving forces.

Cant Deficiency: The amount of cant by which the prescribed cant is insufficient to equalise the centrifugal acceleration for a track horizontal curve at the sign posted speed.

Creepage: is the movement occurring between the wheel and rail surfaces during rolling and produces related creep forces due to friction.

Critical radius: This term is used in this thesis to refer to the radius of curvature at which a wheel profile and diameter can negotiate without flange contact. See section 3.3.3.

Excess cant: occurs when a vehicle travels slowly around curve such that the cant on the rail is excess to balancing the centrifugal acceleration.

Forced Steering: A passively steered bogie 2 or 3 axle configuration that links the bogie yaw to the steering angle of the bogie.
Hunting: On going oscillatory motions of the wheelsets generated by the wheel conicity and creep forces between the wheelset and rail. On going oscillations would normally require five near equal or increasing cycles.

Lateral: the direction level with the rail track and at right angles to the track.

Longitudinal: the direction parallel to the track.

Perfect steering: A concept of idealised steering developed at British Rail Research where by longitudinal creep is kept zero or equal and lateral creep forces are used to balance lateral forces and maintain the required lateral position on the wheel rail contact profiles.

Pitch: rotational movement about the lateral axis.

Precedence control: Control application based on prior knowledge of the track alignment [3]. Thus the controller has prior knowledge of transitions.

Radial Steering: Steering such that the bogie wheelsets are radially aligned with the track curvature. Thus the wheels have same angle of attack to the rail.

Roll: rotation about the longitudinal axis which is parallel to the direction of the track.

Self Steering: A passively steered bogie 2 or 3 axle configuration that cross links the wheelset yaws to limit parallel movements of the axles (warp) and allow steering.

Spring Point: The point at which tangent track changes to constant curvature for transition-less curves.

Transition: the part of the track where the cant or cross level and curvature changes from tangent straight track to canted curved track.

Two point contact: Wheel rail contact occurring where by two separate points on the wheel contact the rail, typically one contact on the wheel tread and one contact on the wheel flange.
Variable Steering: Control of the steering angle in an advance steering bogies that permit the steering angle to be controlled independently to the bogie yaw.

This is a new design proposed in the thesis.

Wear index: Wear index is a measure wear rate per unit length, it is the creep force multiplied by creepage.

Wear Energy: is an indicator for wear rate. Wear energy is the creep force by the creepage integrated over the distance travelled.

Yaw: rotation about a vertical axis

Yaw Relaxation: A passive steering bogie with soft primary suspension stabilised by damping elements in the primary suspension.

1.4.2. Acronyms

AWY Actuated Wheelset Yaw, active steering of the wheelset at the primary suspension

AY-FS Actuated Yaw Force Steered bogie, a new bogie concept proposed in the thesis.
Passive forced steering with actuated yaw control.

AY-VS Actuated yaw variable steered bogie, a new advance steering bogie concept proposed in the thesis. The AY-VS bogie is an advanced version of the AY-FS bogie with direct control of steering angle and yaw angle of the bogie.

AY-VS Lock: A vehicle model equivalent to AY-FS but including the extra weight used for the AY-VS model. The AY-VS vehicle with steering actuation locked by high stiffness spring to match forced steering.

Cd Creep force dependent control

FS Force Steering bogie

IC Idealised Control

iFS Imitation Force Steering control, or steering angle control

HS-3 Self steering three axle bogie where the lateral movement of the middle axle is proportional to the half steering angles of the end axles.

R Rigid bogie or conventional bogie suspension
1.4.3. List of Notation

- \( a_f \): Centrifugal acceleration of the moving co-ordinate system
- \( A \): Ratio of static and dynamic friction, (used by Polach)
- \( A_i \): Bogie pivot semi spacing, vehicle \( i \)
- \( B \): Unit of exponential decay in friction with slip (used by Polach)
- \( B_i \): Coupler centre (or draw pin centre) semi spacing, vehicle \( i \)
- \( C \): Curvature at zero distance in the reference frame
- \( C_i \): Coupler length, vehicle \( i \)
- \( C_z \): Locomotive centre mass height above rails
- \( D \): Draft gear connection length
- \( D_i \): AAR notation for a curve radius vehicle cord perpendicular distance
- \( f \): Creep coefficient, creep force per unit creep or creep force per unit normal load
- \( f_x \): Longitudinal creep force or adhesion force per unit normal load
- \( f_0 \): The static friction coefficient, used in creep coefficient calculations
- \( f_\infty \): The dynamic friction coefficient at infinite slip, used in creep coefficient calculations
- \( F_i \): AAR notation for a curve radius vehicle cord coupling pivot distance
- \( F_{ijc} \): Longitudinal creep force at the wheelset \( i \), contact \( j \)
- \( F_{ijy} \): Lateral creep force at the wheelset \( i \), contact \( j \)
- \( F_y \): Wheel rail lateral creep force for the vehicle
- \( g \): Gravitational acceleration
- \( \Delta g_i \): Change in lateral position, High or low rail contacts
- \( g_y \): Lateral component of gravity acceleration due to track cant
$h$  Wheelset semi spacing of the bogie

$H$  Hardness of the material

$H_c$  Coupler height above rails

$Hv$  The portion of load on the high rail

$i$  identifier 1 or 2 for vehicles or couplers, L or H for low and high rail

$I$  Yaw moment of inertia of the bogie

$k$  Wear coefficient

$K(x)$  Curvature a function distance along the track

$l$  The half distance between wheel contact points on each rail

$L$  Lateral force of the vehicle on the rails

$L$  In simplified couple angling method, cord length between adjacent bogie centres

$\Delta L$  Change in lateral force between the front bogie and rear bogie

$m$  Mass of the vehicle or bogie

$M$  Change in curvature per unit length

$M_{Di}$  Creep force yaw moment difference on bogie i

$M_{dp}$  Sum of coupler yaw moments on the vehicle

$M'_{zwi}$  Yaw moment on the vehicle from longitudinal creep forces for limit case i

$M_{Whk}$  Creep force yaw moment for wheelset on wheelset k

$N$  Normal contact force

$P$  Tractive effort or net coupler load of the locomotive

$P_i$  Coupler load at each end of the vehicle $i$, (compression negative)

$P_y$  Summed lateral coupler force on the vehicle

$\Delta Q/Q$  Wheel unloading

$r_0$  Wheel radius at the tap line

$\Delta r_i$  Change in wheel radius, high or low rail contacts

$R$  Radius of curvature of a track, AAR use $R_o$

$R_c$  Curving radius, as calculated from contact data.
\( R_{ijy} \) Lateral component of the wheel rail contact reaction force, wheelset \( i \), contact \( j \) 
\( R_y \) Lateral component of the wheel rail contact reaction force for the vehicle 
\( s \) sliding distance of the contact patch 
\( V \) Either vertical force of the vehicle on the rails or the velocity of the vehicle or volume of material 
\( W_c \) Locomotive wheel centre spacing 
\( x \) distance along the track 
\( y \) Wheelset lateral movement from the centre position 
\( Y_{rel} \) Wheelset lateral position relative to the track centre line 
\( \alpha_i \) Cord angle of the vehicle bogie centres for vehicle \( i \) 
\( \beta_i \) Vehicle angle to the curve centre at draft gear centres for vehicle \( i \) 
\( \gamma \) couple angle to the curve centres 
\( \gamma_{iy} \) Longitudinal creepage at the wheelset \( i \), contact \( j \) 
\( \gamma_{iy} \) Lateral creepage at the wheelset \( i \), contact \( j \) 
\( \delta_i \) Contact patch \( i \) (H = high rail or L = low rail) contact angle 
\( \Delta y_i \) Variation in the lateral position of the middle wheelset of bogie \( i \) 
\( \Delta \phi \) Change in the half steering angle of the bogie 
\( \lambda \) Linear or equivalent conicity of the wheel rail contact 
\( \Lambda \) Klingel wave length of kinematic oscillation for conical wheels, Equation 1 
\( \mu \) Friction coefficient between the wheel and the rail 
\( \phi_i \) Coupler \( i \) (1 = front or 2 = rear) angle to the vehicle body 
\( \phi \) Half steering angle of the bogie 
\( \phi_{ak} \) Measured half steering angle of the bogie \( k \) 
\( \phi_k \) Half steering angle of the bogie \( k \) 
\( \psi \) Yaw angle of the bogie 
\( \psi_i \) Yaw angle of the wheelset or bogie \( i \) 
\( \psi_{rk} \) Relative yaw angle of the wheelset or bogie \( k \)
\( \psi_{tk} \) Target yaw angle of the bogie \( k \)

\( \chi \) Warp angle or Lozenge of the bogie

\( \omega_{sk} \) Angular yaw velocity of vehicle body

\( \omega_{ak} \) Angular yaw velocity of bogie \( k \)

\( \omega_{rk} \) Angular yaw velocity of bogie \( k \) relative to the vehicle body
2. Current Literature

2.1. Fundamentals of Rail Vehicle Dynamics

A good treatment of vehicle dynamics modelling is described in the text produced by Wickens [7]. The general approach used for modelling railway vehicles is a non linear multi-body dynamics model of the vehicle components coupled to a non-linear contact mechanics model of the wheel rail contact patch. The models generally use a moving co-ordinate system based on the vehicle body [7] though there are exceptions [8]. Some packages also remove the wheelset pitch degree of freedom and assume the wheelset angular velocity is to be matched to the velocity of the moving co-ordinate system [9].

2.1.1. Early History of Railway Dynamics

Wickens, [7] explains that railway vehicle dynamics starts with the wheelset, Figure 1. Two wheels connected on an axle resting on two parallel rails such that both wheels rotate with a common angular velocity. Flanges are used on the inside edge of each wheel with the clearance between the wheel flange and the rail gauge face being ~10 mm. Although flat wheel treads are used in some railway systems, the predominant design is cone shaped treads as shown in Figure 1, typically with tapers in the range 1/20 to 1/40. Wheels begin service with conical or near conical treads, however wear results in a curved tread shape and severe wear results in concave treads, commonly described as a hollow wheel wear.

The first theoretical analysis on wheelset behaviour was done in the 19th century by Redtenbacher and Klingel, [10]. Klingel gave the first mathematical analysis of the kinematic oscillation determining a wavelength $\Lambda$ from the wheelset conicity $\lambda$ and the wheel radius $r_o$ and the lateral distance between the contact points $2l$, as:

Equation 1: Klingel equation for wavelength of kinematic oscillation

$$\Lambda = 2\pi \times \left(r_o \times \frac{l}{\lambda}\right)^{\frac{1}{2}}$$
Redtenbacher gave a formula for the rolling of a wheelset with conical tread profiles on a curve which is shown in Figure 2 where \( R \) is the curve radius, \( y \) is the wheelset’s lateral offset, \( r_o \) is the wheel radius, \( \lambda \) is the wheel conicity and \( 2l \) is the lateral distance between the contact points. Mackenzie did the first essentially correct analysis of a vehicle curving in 1883, [7]. Mackenzie’s work and much of what followed ignored the effect of conical wheel profiles but considered lateral friction forces at the wheels that where yawed to the rails and the effect of flange forces. This approach culminated in the work conducted by Heumann 1913, and Porter 1934-5, [10].

Problems with the stability of rail vehicles limit train speeds. Initially the main problems were concerned with the dynamic balancing of the locomotive connecting rod systems. Later, lateral instability of wheelsets due to the self centring or Klingel effects became a focus. Experience told designers that symmetric designs suffered inherent instability and most early locomotives used non-symmetrical designs (differing wheel diameters at front and rear) unless the vehicle was limited to low speed operations. Recent studies on uni-directional three piece bogie wheel profiles [11], has shown that similar symmetry effect on stability can be derived from non-symmetric wheel profiles, (differing wheel profiles front and rear).
The concept of creep forces and contact patch modelling came later with Carter with the first realistic model for lateral dynamics [10]. Carter was an electrical engineer and applied Routh’s work on stability conditions to railway vehicles extending the “Hertz theory of elastic contact”, [10]. Carter derived equations for the motion of a rigid bogie which are given in Equation 2. In Equation 2 $y$ and $\psi$ are lateral and yaw displacements of the bogie, $m$ and $I$ are the mass and yaw moment of inertia, $f$ is the creep coefficient (creep force per unit creep), $h$ and $l$ are semi wheel base and the semi wheel contact spacing, $V$ the forward velocity, with $Y$ and $G$ being the lateral force and yaw moment on the bogie.

**Equation 2 Carter’s equations for bogie motion**

$$m\ddot{y} + 4f\left(\frac{\dot{y}}{V} - \dot{\psi}\right) = Y$$

$$4f\lambda l\dot{y}/r_o + I\ddot{\psi} + 4f\left(l^2 + h^2\right)\dot{\psi}/V = G$$

Vehicle dynamics studies were also driven by train derailment investigations. In particular flange climb derailments lead to the development of derailment safety criteria. Derailment safety criteria have been used for over 100 years. The first criterion to receive wide spread acceptance was that by Nadal in 1908 [7], [10]. The assumption made by Nadal is that the wheel is in two point contact with the rail. One contact on the tread one on the flange with the flange contact point leading the tread contact. The derailment then occurs when the friction at the flange contact overcomes the vertical load. The L/V criterion is given below in Equation

Figure 2 Redtenbacher formula for wheelset curving

$$y = r_o l / R \lambda$$
3, where \( L \) is the lateral load on the wheel, \( V \) the vertical load on the wheel, \( \delta \) is the flange contact angle and \( \mu \) is the friction coefficient.

**Equation 3 Nadal’s derailment criteria**

\[
\frac{L}{V} = \frac{\tan\delta - \mu}{1 + \mu \tan\delta}
\]

The criterion links an allowable lateral to vertical load ratio to the flange angle and the rail friction coefficient. Nadal’s criterion is only valid for flange contact at high angles of attack as experienced in curving, [7], [12]. In recent years Nadal’s criterion has been modified by various researchers to take into consideration the time or distance spent with high lateral to vertical load ratio [12] and there are reductions to the criterion for angles of attack under 5 mrad. Dynamic wheel unloading is very significant to derailment as vertical load reductions reduce the \( L/V \) ratio. Track irregularities and body motion dynamics of bounce, pitch and roll can all contribute to wheel unloading.

### 2.1.2. Early History of Radial Steering

Innovations for improved steering have a long history beginning with cross braced wheelsets and articulated vehicles [10]. Linz-Budweis Railway used cross bracing directly between wheelsets in 1827 [10]. Today such designs are known as self steering bogies which allow opposing yaw movements but stop any parallel yaw movements (i.e. warp or lozenge) of the wheelsets. The first articulated locomotive was designed by Horatio Allen in 1832 [10]. The first Allen locomotive had a short life and was followed by a succession of designs with arrangements of two vehicle bodies articulated on three axles. Three axle vehicles (three axles for the full vehicle not just a single bogie) have had numerous innovations for improved steering. The Linz-Budweis Railway used a three axle vehicle in which the lateral displacement of the central axle steered the outer wheelsets through linkages in 1826 [7], [10]. Another form of steering (usually known as forced steering) used the angle between the bogie and the vehicle body to steer the wheelsets relative to the bogie frame using linkages [10]. Forced steering can be applied for either two or three axle bogie arrangements. Few of
these early innovations achieved widespread adoption as they generally gave an increased number lateral instability mode when compared with more conventional designs [10].

Self steering, force steering and articulated bogies were not successfully applied until Liechty successfully developed forced steering bogies in the 1930’s and other designs where developed successfully in the 1970’s, notably the Scheffel self steering freight bogie [10]. Linear dynamics analysis of these steering bogie designs were undertaken in the 1970’s and early 1980’s notable by Wickens [7], [10]. See section 2.3 for discussion on current steering bogie developments.

2.2. Wheel Rail Contact Geometry and Creep

Rolling contact between the wheel and the rail has been the subject of considerable research ever since railway began. Today there is a major international conference dedicated to just wheel rail contact mechanics. The concept of creep forces or friction between rolling contacts was defined by Carter and is critical to the curving and lateral stability behaviour of railway vehicles.

2.2.1. Hertzian Contact

The normal load between the wheelset and the rail causes elastic deformation of the surfaces. Hertz showed that for contact between wheels and rails the contact patch is elliptical in shape with the distribution of the normal pressure being semi-ellipsoidal [7] for pure rolling motion. Pure rolling occurs when the relative velocity of the wheel $V$ equals the wheel radius $r_0$ by the angular velocity of the wheel $\omega$. If we consider the case of a longitudinal traction force applied by the wheelset to the rails, the resulting longitudinal force causes a deviation to the pure rolling motion. In the longitudinal traction case the relative velocity of the wheel divided by the forward velocity is the longitudinal creepage $\gamma_x$ and the elliptical patch distorts. The Equation 4 below defines longitudinal creepage.
Equation 4 Longitudinal Creepage

\[ \gamma = \frac{\omega \cdot r_0}{V} \]

Similarly lateral creepage is the relative lateral velocity divided by forward velocity and relative angular velocity between the wheel and the rail is defined as spin creepage. If all the creepage is small, it is accommodated by elastic strains. As the wheel rotates the material entering the contact patch is unstrained and the tangential strain on the surfaces increases as the contact patch move forwards, see Figure 3. Eventually the tangential stresses exceed the normal stresses multiplied by the coefficient of friction and wheel slip takes place. As a result the contact patch is split into an area of adhesion where the surfaces are locked together and an area of slip. As larger tangential forces are applied to the contact patch the region of slip increases until eventually the entire surface is in the slip condition and the creep forces reduce to those that can be supported by the dynamic coefficient of friction. Note the dynamic coefficient of friction for wheel rail contacts is influenced by third body lubrication and temperature change and is not a constant.

Figure 3 Longitudinal creep stresses showing the locked region and the slip region of contact.

At low creepages both longitudinal and lateral creepage generate forces that are directly proportional to the corresponding creepage. At high creepages the creep force saturates due to the limitations of surface frictions. The maximum resultant creep force for lateral and longitudinal creepages is limited by the friction coefficient and the vertical contact load. In
most cases creep force saturation occurs when the creepage approaches 1%. When spin creepage is involved the pattern of elastic strain is more complicated the relative spin velocity is directly proportional to the distance from the centre of the contact region. For the case of running on straight track the strain on the contact surface is built up as the wheel passes. The contact strains build such that the spin creepage applies a lateral force due to the mismatch in strain between an unstrained front of the contact patch and the highly strained rear of the contact patch. The spin creepage is generated by the conicity of the wheel at the contact patch with outside edge of the contact patch having a slower speed than the inside edge causing a relative angular velocity between the wheel and rail surfaces. As creepages increase and the slip region increases the lateral forces from spin creep reduce to zero.

When the wheelset is laterally displaced towards flange contact the change in contact angles of the wheels changes the lateral components of the normal surface contact forces. This effect is illustrated in Figure 4 and is known as the gravitational stiffness. This provides a self correcting force to centre the wheelsets. However the previously mentioned spin creep effect generates lateral forces that largely negate gravitational stiffness. An exception to this cancellation of the gravitational stiffness occurs for very large creepages with the large creepages causing creep force saturation. Thus gravitational stiffness effect occurs with a high contact angle of the flange face or in the presence of large lateral or longitudinal creepages.