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Wheel and Rail Profile Development

GUIDELINE

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Appendix Contents

1 Introduction

1.1 Purpose

The objectives of the Guideline for Wheel and Rail Profile Development are to:

- Provide good practice guidance on assessing wheel and rail profiles theoretically and in the field (in service).
- To provide a change management process for developing, testing and implementing new wheel and/or rail profiles.

1.2 Scope

This Guideline is applicable to all Australian rail networks, including narrow, standard and broad gauge railways.

It is not intended to cover light rail networks that may have different requirements.

1.3 Definitions

For the purposes of this Guideline the definitions given in the Australian Guideline of Practice – Glossary of Railway Terminology (Reference 2) shall apply. The following definitions are specific to this guideline.

Term	Definition
Conformal contact	The general condition where the wheel and rail profiles have similar shapes resulting in a large contact area.
Closely conformal contact	The condition where the wheel and rail profiles have such similar shapes that the gap between the unloaded wheel and rail profiles is about 0.1mm or less. Once loaded, the elastic deformation of the wheel and rail should close the gap such that there is a wide contact band around 25-40mm in width.
Conicity	Conicity is a measure of the effective cone angle of the wheelset on the rails. For example, a wheel with a coned profile that has a slope of 1:20 that is sitting on rails with a convex head would be expected to have a conicity of 0.05 (i.e. 1/20). Mathematically, the conicity is calculated as one-half of the slope of the graph of rolling radius difference versus wheelset lateral shift.
Contact stress (P_0)	Maximum wheel/rail contact stress in the direction normal to the plane of contact.
Creepage	Relative movement between the wheel and rail with longitudinal, lateral and spin components.
Creep forces	Forces associated with longitudinal, lateral and spin creepage.
False flange	Raised portion of the wheel towards the outer edge of the wheel tread.

Flangeway clearance	The lateral movement available to the wheelset to permit it to move from flange contact on one rail, to a central position between the rails, to flange contact on the other rail. Care is needed when expressing a value for flangeway clearance. For example, let us consider a wheelset that is centrally between the rails with 8mm of clearance between the left hand wheel flange and rail and 8mm of clearance between the right hand wheel and rail: • Rolling stock Engineers tend to refer to this case as having a flangeway clearance of 8mm since it has ±8mm of clearance from the central position; • Track Engineers tend to refer to this case as having a flangeway clearance of 16mm since that is the total clearance available between wheelset and rails. Therefore, this is a point for clarification if numerical values of flangeway clearance are to be used.
Hunting	Sinusoidal motion of a wheelset along the track, limited only by flange contact on the rail.
In-plane force	Wheel/rail contact force in the plane of contact, longitudinal and lateral.
Non-conformal contact	Contact between the wheel and rail that results in a small contact area.
Normal force	Wheel/rail contact force in the direction normal to the plane of contact.
Radial steering index	For an isolated wheelset the radial steering index is defined as q_E, the smallest curve radius where radial steering (no flange contact) is possible (R_E) divided by the actual curve radius (R), i.e.: $q_E = \frac{R_E}{R}$
Rail cant (or rail inclination)	Rail cant (or rail inclination) is the angle between the vertical axis of one rail and the vertical axis of the track. Generally, rails are laid such that the vertical axis of each rail is inclined towards the track centreline at an angle of 1 in 20 to the track vertical axis. However, the angle of inclination can vary and the rails may be laid vertically.
RCF	Rolling contact fatigue.
RRD	Rolling radius difference between two wheels on an axle for a given value of wheelset lateral shift.
Single-point contact	Contact between the wheel and rail that results in a single point of contact.
Two-point contact	Contact between the wheel and rail that results in two points of contact.
Turnouts and other special trackwork	Points (or switches) and crossings.

2 THEORETICAL ASSESSMENT OF WHEEL AND RAIL PROFILES

Wheel and rail transverse profiles have a significant effect on railway costs, safety, and potentially operations. Apparently small changes in profiles, either by design or through natural wear, can lead to large changes in wheel and rail maintenance costs and service life, vehicle ride quality, and derailment risk. Done well, profile design changes can give major benefits; done poorly they can lead to problems that are costly to remedy. For example, hollow worn wheels can give rise to vertical split head in rails. The cost of remedial work is especially high in the case of rails where the application of inappropriate profiles can take many years to put right, usually by expensive and intrusive grinding, and cause potential accelerated wheel wear during this time.

For all these reasons, the development of new wheel or rail profiles is a specialist subject that needs to be done with clear objectives by engineers with considerable experience in contact mechanics and vehicle dynamics and associated software packages, and with good knowledge of relevant standards. It also needs to be stressed that wheel and rail profiles together work as a system. For a given vehicle the optimum wheel profile will depend on the existing rail profiles (from new to worn), and vice-versa. To complicate matters further, both the optimum wheel and rail profiles will depend on factors such as the vehicle suspension design and the range of track curvatures.

It is recommended that prior to considering a change to existing wheel or rail profiles, consideration is given to whether good wheel and rail monitoring and maintenance practices are in place, including wheel reprofiling and rail grinding. Correction of poor maintenance practices could more quickly and easily overcome existing problems without the need for changes to wheel and rail profiles.

When developing wheel and rail profiles a number of issues are key, relating to safety, ride quality, curving performance, wheel and rail wear and rolling contact fatigue (RCF), and rolling resistance. These issues are considered in Sections 2.1 to 2.5. It is also important to consider ease of application of new profiles and potential collateral effects and other factors. These topics are addressed in Sections 2.6 to 2.7.

Wheel and Rail Profile Expertise;

Wheel Rail Profile Development is the subject of considerable research, development and innovation. Expert knowledge of current research and development is presented at IAVSD conferences (International Association of Vehicle System Dynamics), held bi-annually) and CM conferences (International Conference on Contact Mechanics), held tri-annually.

There are two major text references that have come from these conferences:

- “Wheel-Rail Interface Handbook” (Reference 2); and
- “Handbook of Railway Vehicle Dynamics” (Reference 3).
- Another useful text reference from the International Heavy Haul Association is:
- “Guidelines to Best Practices for Heavy Haul Railway Operations: Wheel and Rail Interface Issues” (Reference 4).”

Also, the Railway Technical Society of Australasia (RTSA), a joint technical society of Engineers Australia (EA) and Institution of Professional Engineers New Zealand (IPENZ), hold a bi-annual CORE conference (Conference on Railway Excellence) that often has relevant papers (Reference 5).

2.1 Flange-climb derailment

Flange-climb derailment occurs when the forces in the wheel/rail contact patch cause the wheel to climb on to the top of the rail. Vehicle factors that increase the risk of derailment include wheelset misalignments and twisted bogie frames, high vehicle torsional stiffness, high bogie rotational resistance, suspension faults, and high primary yaw stiffness. Track factors include track twist and lateral and vertical misalignments that contribute to wheel unloading.

With respect to the effect of profiles on flange-climb derailment, the primary factor is the maximum gauge face angle allowed by the wheel/rail contact. This angle is commonly referred to as the wheel flange angle and is shown as α in Figure 1, which also illustrates the vertical wheel load (V), the applied lateral load (L), the reaction force (R), and the friction force (F). This friction force arises from the friction (coefficient μ) between the wheel flange and the rail gauge face.

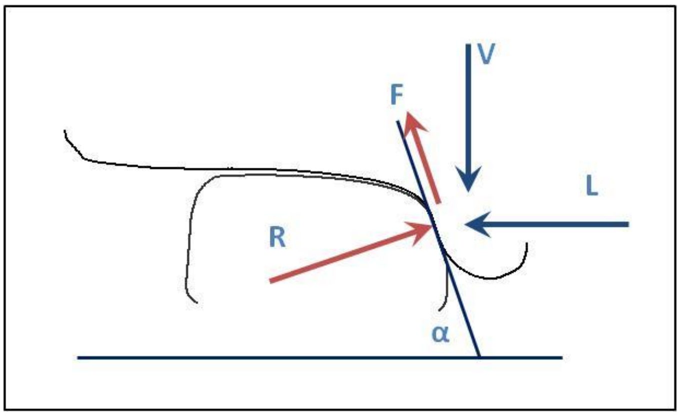

Figure 1: Illustration of forces in flange-climb derailment

Simple considerations show that the risk for the wheel flange to climb up the rail gauge face increases as L increases or μ increases, and as α decreases. Nadal's formula (Reference 6) gives the critical limit of individual wheel L/V in terms of the coefficient of friction and maximum gauge face angle (see equation 1). This equation is shown graphically in Figure 2 for coefficients of friction of 0.1, 0.3, and 0.5.

$$\frac{L}{V} = \frac{\tan(\alpha) - \mu}{1 + \mu.\tan(\alpha)}$$ *Equation 1*

Where: α = wheel flange angle

μ = friction coefficient between wheel flange and rail gauge face

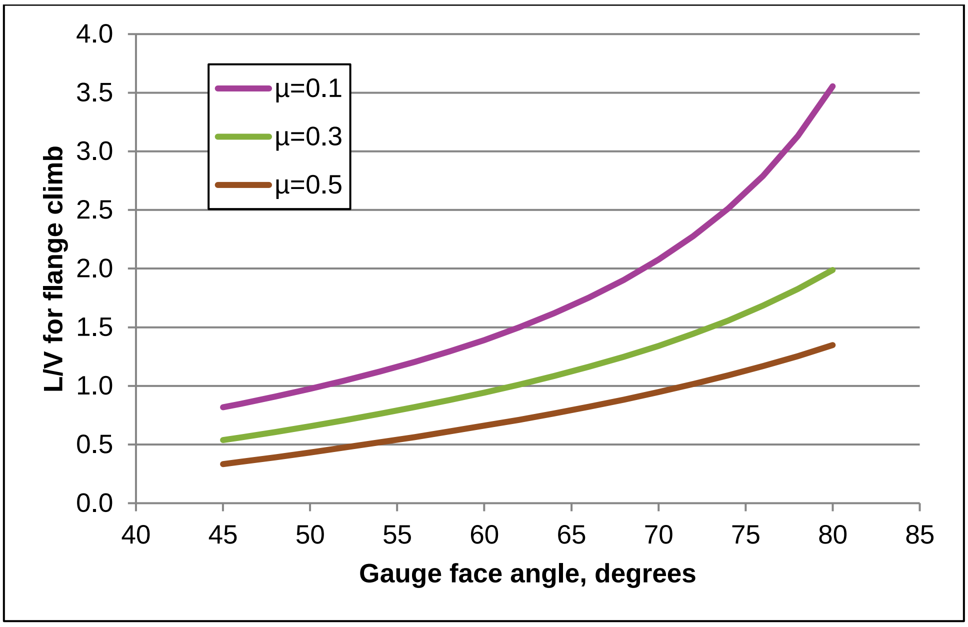

Figure 2: Relationship of L/V limit to gauge face angle for friction coefficients of 0.1, 0.3, and 0.5

Common practice is to aim for flange angles in the range 65 to 75 degrees, as can be seen in Table 1 which gives recommended or specified design wheel flange angles for a number of countries. During wheel profile design it is important to ensure that the detailed shape in the flange area leads to a flange angle greater than 65 degrees, and preferably higher for highly curved track.

Country	**Standard**	**Minimum flange angle, degrees**	**Comments**
Australia	AS7514 (2010)	65-70	All vehicles
New Zealand	NRSS/6	73	All vehicles
USA	APTA SS-M-015-06 (2007)	72	Passenger vehicles
USA	AAR MSRP Section G-II	75	Freight vehicles
European Union	High-Speed Rolling Stock Tech. Standard for Interoperability (2008)	67	Passenger vehicles
UK	RSSB GM/RC2494 (2010)	68-70	All vehicles

Table 1: Typical minimum flange angles given in standards

Australian Standard AS7509 'Railway Rolling Stock – Dynamic Behaviour' (Reference 7) mandates a safe limit for L/V of 1.0. For a flange angle (α) of 70 degrees, substitution into Equation 1 shows that the specified limit of 1.0 is applicable for a wheel-rail friction coefficient (μ) of 0.47.

However, this is a conservative limit for L/V for the following reasons:

- Nadal's formula is an approximation assuming a large angle-of-attack of the wheelset to the rail.
- Natural wheel/rail friction rarely reaches a level of 0.5, and lubrication is often applied to the wheel flange or gauge side of the rail on curves where flange contact is likely to occur, to give friction levels down to 0.1 to 0.2.
- Natural wheel wear tends to increase the wheel flange angle to steeper than the design angle.
- These high levels of L/V need to be sustained for some time or distance (rule of thumb is 50 msec, which can be translated into distance given the vehicle speed) to allow the wheel to climb up onto the rail top, and the increase in the effective radius of the climbing wheel tends to steer the wheelset back into normal running.
- Check rails may be used in sharper curves where high angles-of-attack are likely.

2.2 Ride quality

Wheel and rail profiles influence the ride quality of a vehicle by the effect they have on a parameter termed conicity. This is a measure of the effective cone-angle on the wheel tread, but the true measure of conicity is a function of the profiles of both wheels on the wheelset and both rails in track.

Figure 3 illustrates a wheelset running on a pair of rails. The wheelset is shifted to the right, such that the effective radius of the right rail is larger than that of the left rail. Since both wheels are fixed on the axle, this increase in radius causes the right wheel to roll further in one revolution that the left wheel leading the wheelset to steer back towards a central position. Generally, because of inertia, a wheelset will oversteer, and so tend to follow a sinusoidal path along the track. For any lateral position of the wheelset, the difference in radius between the two wheels is called the rolling radius difference or RRD.

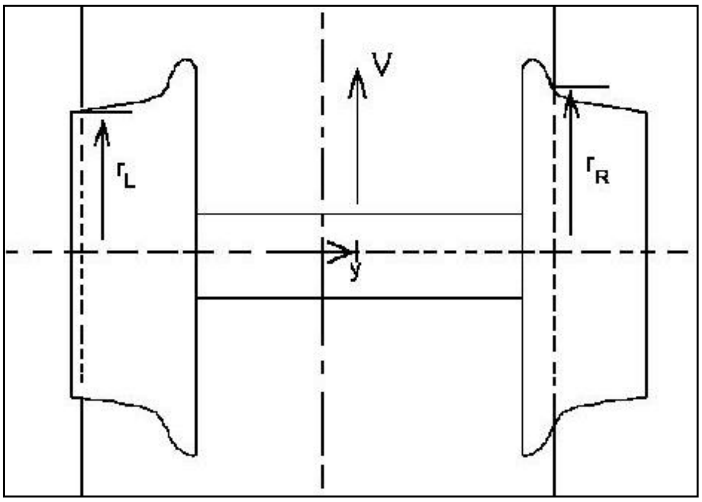

Figure 3: Illustration of a wheelset running on a pair of rails

The effective cone angle of the wheelset on the rails is referred to as the 'conicity' or "effective conicity". Mathematically, the effective conicity is calculated as one-half of the slope of the graph of rolling radius difference versus wheelset lateral shift:

- For example, a wheel with a coned profile that has a slope of 1:20 that is sitting on rails with a convex head would be expected to have a conicity of 0.05 (i.e. one half of [1/20+1/20] = 1/20 = 0.05).
- For new wheels with a variable slope and for worn wheels, the conicity is defined as one-half of the rate of change of RRD with wheelset lateral shift when the wheels are in tread contact on the rails. In this case it is common to use the term effective conicity.

For the case where the RRD changes in a highly non-linear way with lateral wheelset shift (for example with worn wheels on worn rails), various national and international standards (such as UIC 519, Reference 8) contain mathematical methods for the calculation of conicity for varying amplitudes of wheelset lateral shift. Conicity calculated by these mathematical methods is generally referred to as "equivalent conicity." When working with conicity values, and when presenting results, it is important to make clear how conicity is defined and calculated.

Figure 4 illustrates graphically how RRD changes with wheelset shift for three wheel profile cases and three rail profile cases. The x-axis shows the lateral wheelset shift (positive to the right) and the y-axis shows the associated RRD (right wheel minus left wheel).

Case Study:

WPR2000 wheel profile on Interstate routes:

The WPR2000 wheel profile was developed in conjunction with a corresponding RTG2000 rail profile intended to provide benefits to the wheel/rail contact on Interstate routes.

The WPR2000 profile was introduced onto rolling stock over a 1-2 year period. Limited testing had been undertaken over higher speed straight track sections of the Interstate routes. The corresponding RTG2000 rail profile was slower to implement.

The WPR2000 wheel profile on existing rail profiles resulted in the contact patch being close to the gauge corner of the rail. This caused high contact stresses leading to shelling, flaking and squats in the rail. Also, the wheel contact close to the flange root of the wheel leads to higher effective conicity and hunting of rolling stock at higher speeds.

The three wheel cases are shown in Figures 4 to 6 respectively and are:

- New WPR2000 wheel profile;
- New ANZR1 wheel profile; and
- Worn ANZR1 wheel profile.

The three rail cases are shown individually in Figures 4 to 6 respectively and are:

- New AS1085 60kg rail inclined at 1 in 20;
- New RTG2000 profiled rail inclined at 1 in 20; and
- Worn rail inclined at 1 in 20.

For the new AS1085 60kg rail case shown in Figure 4, the WPR2000 wheels show a conicity of about 0.09 for small lateral shifts, increasing to about 0.12 at about 6mm of lateral shift. The new ANZR1 wheels have a coned profile and show a constant conicity of about 0.05. The sudden increase in RRD at large wheelset shifts reflects the onset of contact between the wheel flange and rail gauge face. The large change in RRD that occurs when one wheel flange meets the rail gauge face can have a significant effect on lateral stability. The worn ANZR1 wheels show a changing conicity of about 0.05 for small lateral shifts, increasing to about 0.06 at about 7-8mm of lateral shift.

It is generally found that as wheels and rails wear they develop more conformal contact that leads directly to an increase in conicity.

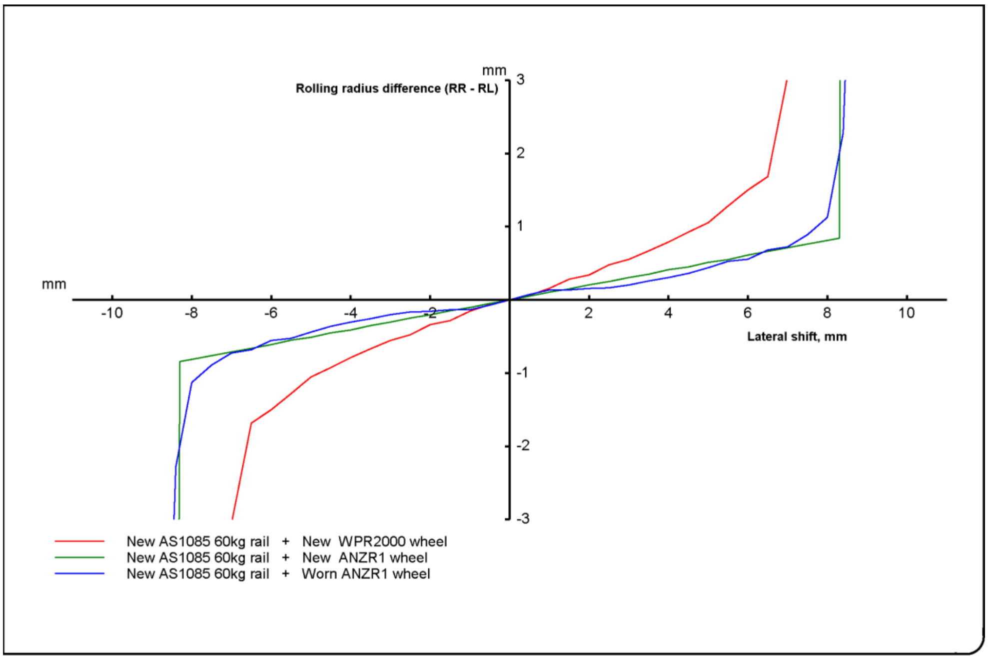

Figure 4: Illustration of rolling radius difference graph – New AS1085 60kg rail

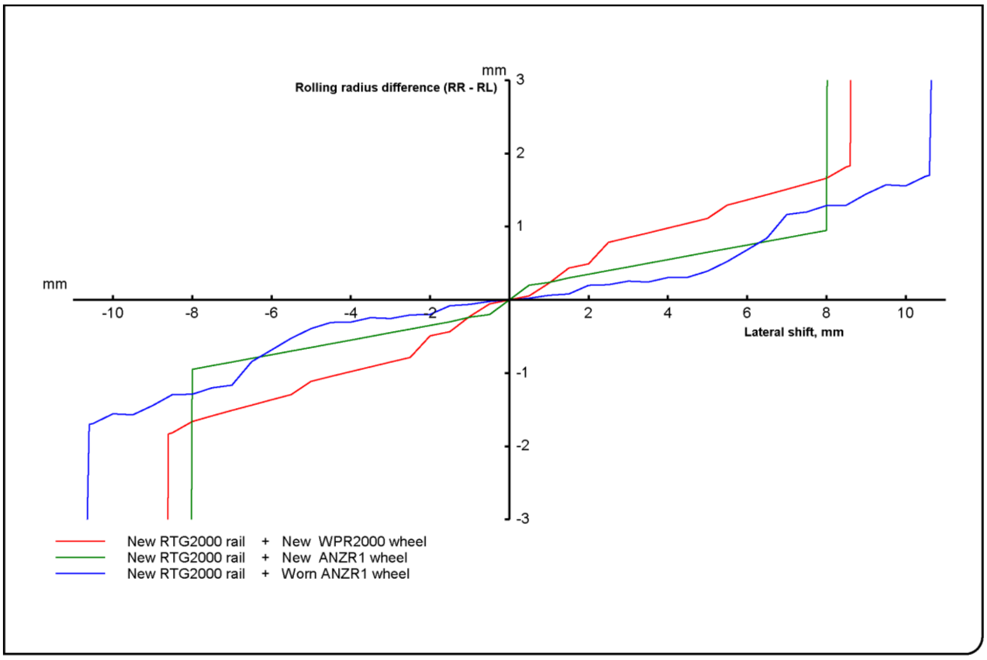

Figure 5: Illustration of rolling radius difference graph – New RTG2000 profiled rail

For the new RTG2000 rail profile case shown in Figure 5, the WPR2000 wheels show a more constant conicity with lateral shift of about 0.11. The coned tread of the new ANZR1 wheels continue to show a constant conicity of about 0.05. The worn ANZR1 wheels show a changing conicity of about 0.05 for small lateral shifts, increasing to about 0.09 at about 7mm of lateral shift.

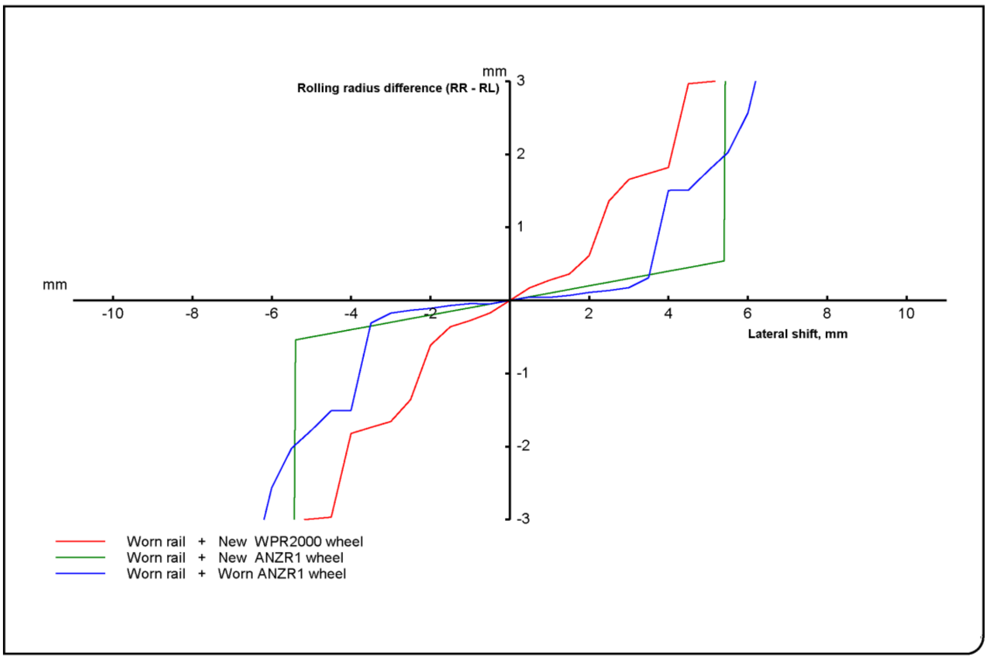

Figure 6: Illustration of rolling radius difference graph – Worn rail

The worn rail case in Figure 6 has reduced gauge caused by material flow from the rail head to the gauge face of the rail, giving reduced flangeway clearance for all wheel profile cases, as illustrated in Figure 7.

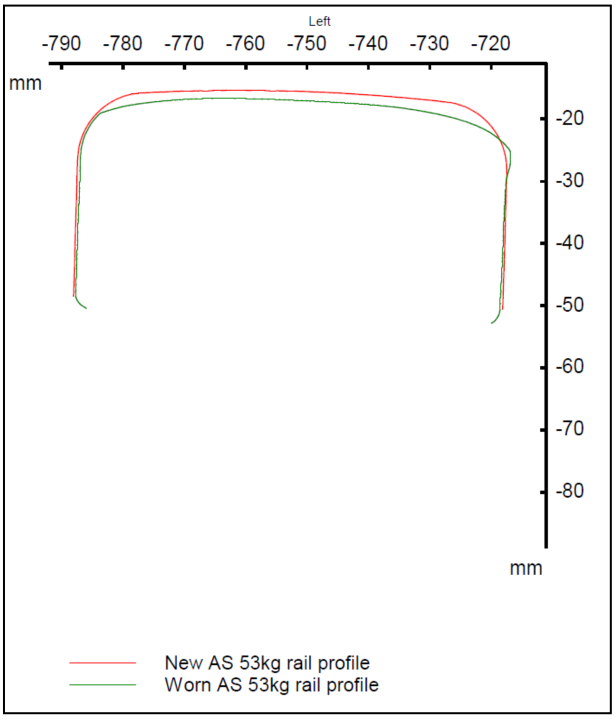

Figure 7: Comparison of new and worn rail profiles

The worn rail shape changes to the contact point of the wheel on the rail from a position close to the centre of the rail head to a point closer towards the gauge side of the rail, and towards the flange root of the wheel. This change in contact point leads to an increase in effective conicity. The WPR2000 wheels show a changing conicity, but the effective conicity is relatively high for a new wheel at around 0.23 at 4mm of lateral shift. The coned tread of the new ANZR1 wheels continue to show a constant conicity of about 0.05. The worn ANZR1 wheels show a changing conicity of about 0.05 for small lateral shifts, increasing rapidly to about 0.18 at about 4mm of lateral shift.

Figures 4 to 6 illustrate the sensitivity of the effective conicity of wheel and rail profile pairs to changes in wheel or rail profile. If changes in track gauge were included, this would show further variations.

The importance of conicity is that it influences the amplitude of the wheelset's sinusoidal motion along the track, especially in straight track. Other key factors in this sinusoidal motion are vehicle speed and the vehicle design. Track gauge can also be important, since track laid tight-to-gauge can also lead to high conicity. When a wheelset travelling in straight track and with small lateral oscillations is given a lateral impulse, say from a misaligned weld, then the oscillations increase in magnitude. Whether these oscillations decay (to give stable running) or increase (to give lateral motion limited only by flange contact on the rail gauge face, often referred to as hunting) depends on conicity, speed, and suspension.

Generally, for a given bogie and suspension design and operational speed a critical conicity exists – above which unstable running occurs. Therefore during wheel and rail profile design it is important to understand and calculate this relationship between vehicle design (primarily suspension design), operating speed, and conicity, and to ensure that the system enables stable running to speeds at least 10% above the maximum operating speed. It is also important to consider how wear may affect the detailed profile shapes and so affect conicity and the speed at which unstable running begins. These calculations require use of sophisticated vehicle dynamics simulation software.

Some operating regimes, such as hopper wagons that only operate in one direction of travel, can generate unusual wheel profile shapes that have an effect on stability.

2.3 Curving performance

While low conicity promotes vehicle stability at speed in straight track, higher conicity is an aid when vehicles have to negotiate curves. In a curve, the outer rail is longer than the inner rail, and therefore the outer wheel has to travel further than the inner wheel. Since both wheels on a wheelset rotate at the same rate, the outer wheel needs to run on a larger rolling radius than the inner wheel. The wheelset therefore needs to shift laterally outwards in the curve to produce this rolling radius difference. This is shown schematically in Figure 8. However, the maximum lateral shift is limited by the flangeway clearance. This clearance is the maximum amount of lateral movement the wheelset can achieve (moving from flange contact on one rail to flange contact on the other rail), and is a function of the wheel and rail profiles, track gauge, and back-to-back spacing.

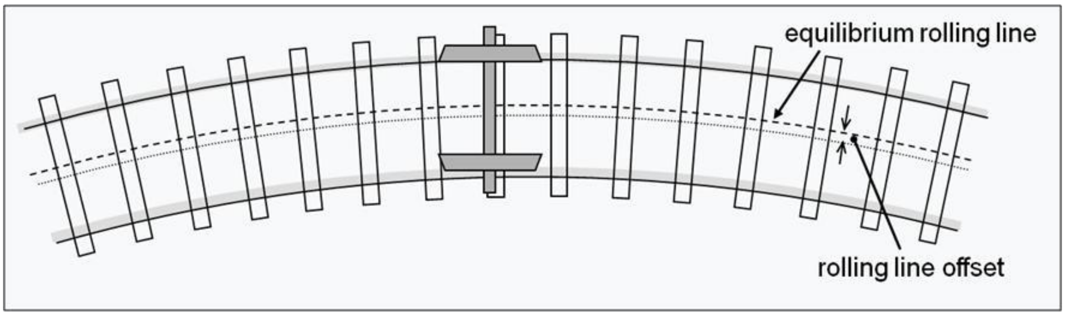

Figure 8: Schematic showing lateral shift of wheelset in curve

The amount of lateral shift needed to negotiate a curve depends on the conicity. The wheelset has to shift further laterally at low values of conicity than at high values; hence low conicity gives greater risk of the outer wheel flange meeting the outer rail gauge face, leading to accelerated wear of the flange and gauge face. Higher conicity enables a vehicle to negotiate smaller radius curves before flange contact occurs.

Figure 9, which is based on purely geometric considerations, shows the effect of conicity (in the range 0.05 to 0.20) on the ideal lateral shift needed to negotiate curves of varying radius. It shows that, for a given curve radius, the amount of wheelset lateral shift needed to negotiate a curve reduces as conicity increases.

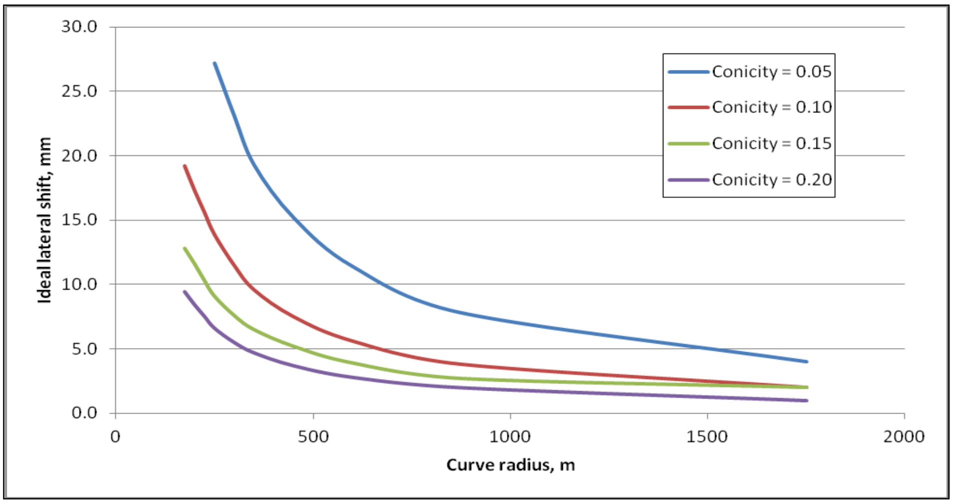

Figure 9: Effect of conicity on lateral shift needed to negotiate a curve

Figure 8 assumes that conicity is constant over the full range of lateral wheelset shift before flange contact occurs. This can be the case, but more often the conicity varies with the amount of lateral shift, and can increase in the few millimetres before flange contact occurs. An example of this is shown in Figure 9.

When this situation occurs, a useful parameter to consider is the radial steering index defined in EN 14363 (Reference 9). For an isolated wheelset the radial steering index is defined as q_E, the smallest curve radius where radial steering (no flange contact) is possible (R_E) divided by the actual curve radius (R):

$$q_E = \frac{R_E}{R} \qquad \textit{Equation 2}$$

R_E is calculated using the parameter Δr_E, the rolling radius difference at a point 1mm before onset of flange contact (shown in Figure 10), the nominal wheel radius (r_0), and the distance between the two wheel/rail contact points (b):

$$R_E = \frac{b.r_0}{\Delta r_E} \qquad \textit{Equation 3}$$

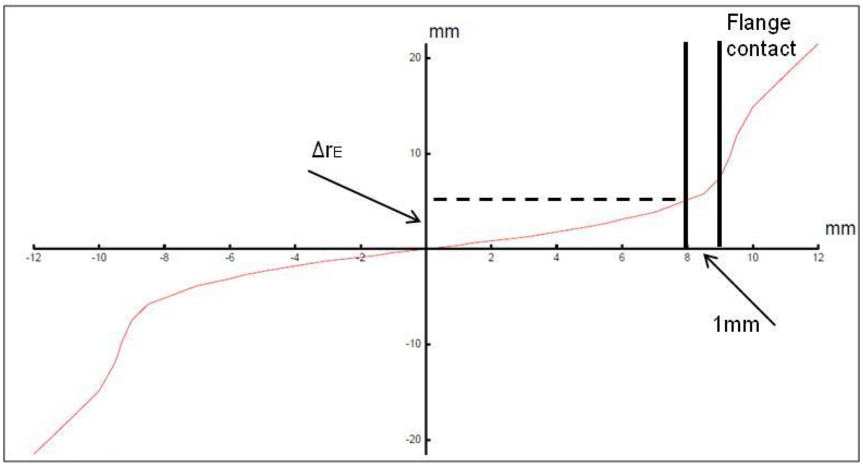

Figure 10: Example of rolling radius difference graph showing increase in conicity as flange contact approaches

When q_E is less than or equal to 1 then radial steering is possible. Therefore, if possible, wheel/rail profiles should be designed such that q_E is less than or equal to 1 for all main line curves. This design may need to consider the effect that large wheelset angles-of-attack can reduce flangeway clearance and affect RRD in tighter curves. The bogie design will have a large influence on the wheelset curving behaviour, but a bogie design that minimises changing the wheelset behaviour from that of a free wheelset should provide best curving performance.

During profile design there is therefore a trade-off with respect to conicity. Low conicity favours stable running at higher speeds, high conicity gives better curving performance. In practice, good vehicle suspension design can allow both stability at higher speed and good curving performance.

2.4 Wear and rolling contact fatigue (RCF)

Wheel/rail wear and RCF are influenced by many factors including the properties of the steels of which they are made, the profiles adopted, vehicle and route characteristics (such as track curvature), friction conditions, and operating details (such as traction and braking rates). Wear and RCF also interact, in that high wear rates can lead to a reduction in RCF as cracks are worn away faster than they can grow, and vice-versa.

With respect to steel properties, for conventional pearlitic steels, from which almost all wheels and rails are made, resistance to wear and resistance to RCF both increase with hardness. A newer type of steel, termed bainitic, is now available for wheel and rail production and has seen limited implementation. With this steel, wear resistance also increases with hardness; but experience indicates that, for a given hardness value, pearlitic steels give less wear than bainitic steels. With respect to RCF, the picture is less certain. Wheel and rail trials indicate that bainitic steels can be much less susceptible to RCF than pearlitic steels, but the reasons for this are far from clear.

Regarding the effect of wheel hardness on rail performance, and vice-versa, the situation is confused. In part this is because of the difficulty in undertaking full-scale trials to establish any effects. However, small-scale tests and the limited theoretical understanding of wear and RCF indicate that:

- There is little clear evidence that harder rails wear wheels more, or vice-versa. If any effect is present, it is likely to be small unless wheel and rail hardness differs considerably.
- From theoretical considerations (see Sections 2.4.2, 2.4.3) there is no reason to believe that rail hardness has a major effect on wheel wear, and vice-versa.
- To reduce system wear, harder steel grades should be used for both wheel and rail.

A discussion of the relative effect of wheel/rail hardness is given in Reference 10, which also details results from tests on the effects of wheel and rail hardness on wear and RCF.

With respect to profiles, both wheel and rail profiles have an effect on wear and RCF in three ways. First, by how they interact to spread contact relatively evenly over the wheel and rail surfaces. Second, by how they interact to control contact stress. Third, by how they interact to influence the energy dissipated in normal running.

2.4.1 Evenly spread contact

Figure 11 illustrates two types of contact. The top illustration shows a wheel and rail profile and how they contact each other as the wheelset is shifted laterally with respect to the rail. In this case, it is seen that contact is spread relatively evenly over the wheel and the rail; this is termed conformal contact. (Contact in fact is spread even more evenly than the figure implies since the connecting lines connect the centres of the wheel and rail contact patches, which are in general 5-10 mm wide.) The contact in this top illustration will lead to even wear over the wheel and rail surfaces, and this wear will lead to even more conformal contact. The bottom illustration shows what happens when the wheel and rail profiles are poorly matched. In this case the wheel and rail can contact only in the flange root/gauge corner areas; this is termed single-point contact. This will lead to initially very high wear at these two locations.

For this reason it is generally advisable to aim for wheel and rail profile designs that give reasonably conformal contact. However there are other factors that need to be considered. First, the designed wheel/rail contact should not lead to equivalent conicity values that lead to unstable running (see Section 2.2). Second, for situations where RCF can occur on the rail gauge corner, it may be necessary to develop wheel/rail profiles that reduce contact in this area.

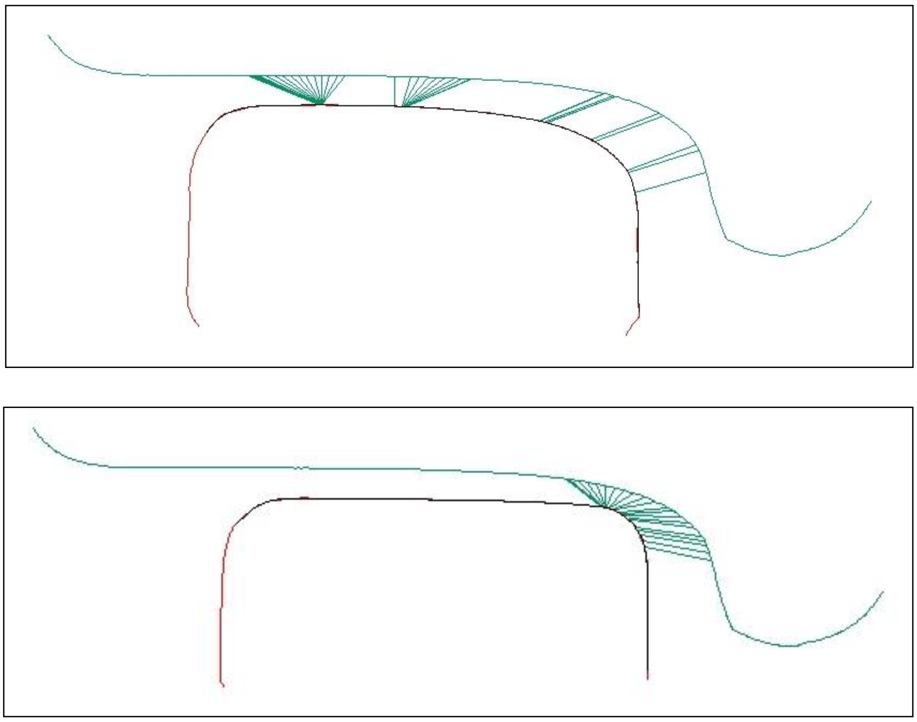

Figure 11: Example of conformal (top) and single-point (bottom) contact

2.4.2 Contact stress

Laboratory experiments have indicated that the level of contact stress influences the amount of wear and RCF on the wheels and rails. To a first approximation, wheel and rail wear is proportional to contact stress (and also proportional to creep). Also to a first approximation, RCF is proportional to the contact stress raised to a power greater than 1. Hence, wear and RCF can both be reduced by applying wheel and rail profiles that give lower contact stress.

In its simplest form, contact stress is controlled in Australian Standard AS7508 (Reference 11) by the 'P/D' ratio, where P is the static vertical load (kN) on one wheel, D is the wheel diameter (m) and the ratio P/D should not exceed 125kN/m. The P/D ratio is only applicable to wheel and rail profiles that are non-conformal.

The contact stress (P_0) between wheel and rail is commonly found by approximating profiles by a series of circular arcs and using the Hertz equations (Reference 12). Other methods are available, for example using interpenetration or finite element techniques. In the Hertz analysis, the contact patch is elliptical, and P_0 is the stress at the centre of the contact patch in the normal direction. The P_0 stress is compressive, and is the largest stress in the contact patch. All the other stresses (tensile/compressive in the longitudinal and lateral directions, and all the combinations of shear stress) are scaled by the P_0 stress. Hence, the P_0 contact stress gives a good characterisation of the overall stress state between wheel and rail.

In the Hertz calculation, P_0 is given by equation 4:

$$P_0 = \frac{3P}{2\pi ab} = \left(\frac{6PE^{*2}}{\pi^3 R_e^2}\right)^{1/3} .[F_1(R'/R'')]^{-2/3} \qquad \textit{Equation 4}$$

P is wheel load; πab is the ellipse contact area. R_e depends on the longitudinal and transverse rail radii, and the circumferential and transverse wheel radii (equations 5, 6). E* depends on wheel and rail elastic modulus and Poisson's ratio (equation 7). F_1 is a function of the wheel and rail radii and values are available graphically in Reference 12. To a first approximation, $F_1^{-2/3}$ is equal to 1.

$$R_e = \left(R^{/}.R^{//}\right)^{1/2} \qquad \textit{Equation 5}$$

$$\frac{1}{R^{/}} = \frac{1}{R_1^{/}} + \frac{1}{R_2^{/}}\ , \qquad \frac{1}{R^{//}} = \frac{1}{R_1^{//}} + \frac{1}{R_2^{//}} \qquad \textit{Equation 6}$$

$R_1^{/}$ is the wheel radius, and $R_1^{//}$ is the transverse wheel profile radius. $R_2^{/}$ is the rail radius along the rail, and $R_2^{//}$ is the transverse rail profile radius.

$$E^* = \left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)^{-1} \qquad \textit{Equation 7}$$

ν_1 and ν_2 are the wheel and rail material Poisson's ratio (about 0.3). E_1 and E_2 are the wheel and rail material Young's modulus (about 207,000 MPa).

In equation 4 the contact stress is proportional to the cube root of wheel load, and so contact stress changes very little with wheel load. In contrast, contact stress is very sensitive to changes in wheel and rail transverse profile radius. Small changes in wheel and rail profile can lead to a doubling or tripling of contact stress. Thus control of rail and wheel profiles is critical for control of contact stress and hence control of wear and RCF.

A common way of considering the effect of contact stress on wheel/rail deformation (and hence wear and RCF) is by using what is known as a "shakedown" diagram (Reference 13). An example is shown in Figure 12. In this figure the x-axis is the traction coefficient – defined as the in-plane wheel/rail contact force divided by the normal contact force. The in-plane force mainly consists of steering forces arising from creepage (see Section 2.4.3), but can also contain tractive and braking forces. The normal force is effectively the wheel load. The y-axis is the ratio of the P_0 contact stress divided by the shear yield strength (k) of the wheel or rail steel.

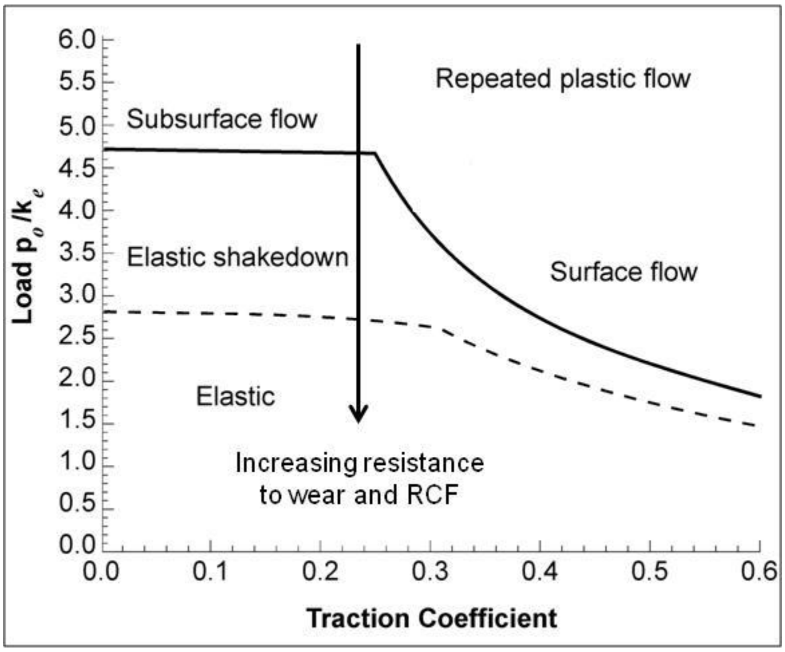

Figure 12: Shakedown graph, showing areas of elastic and plastic deformation

This shakedown graph is divided into three areas. Below the dotted line, wheel/rail contact gives only elastic deformation and wear and RCF should be minimal. Above the solid line, wheel/rail contact gives plastic deformation and in this case wear and RCF will be significant. In between these two lines contact initially gives plastic deformation, but the generation of residual stress eventually leads only to elastic deformation. The wheel or rail steel is said to “shake down” to elastic behaviour.

Applying Figure 12 quantitatively is difficult, in large part because in the difficulty in estimating the value of k. For virgin material k can be related to the yield strength in tension by equation 8. However, the steel at the surfaces of wheels and rail in service is highly strained and with highly anisotropic properties. Physically it little resembles the virgin material.

$$k = \frac{S_y}{\sqrt{3}} \qquad \textit{Equation 8}$$

It is fairly clear qualitatively from Figure 12 that wear and RCF will be reduced if in-plane forces are lowered (for example by lubrication), if the shear yield strength is increased (by use of harder steel), and if contact stress is reduced (by the design of improved wheel/rail profiles). For this reason it is good practice in the design of wheel and rail profiles to aim to reduce contact stress as far as practicable, by producing conformal contacts. However, there is a constraint in that although conformal contacts give reduced contact stress they increase conicity and introduce the risk of unstable running (see Section 2.2). During development it may well be necessary to examine the trade-offs between contact stress and conicity, considering operating speed and vehicle design. Also, for situations where RCF can occur on the rail gauge corner, it may be necessary to develop wheel/rail profiles that reduce contact in this area. One means of achieving this is to grind different rail profiles on straights and curves to avoid contact in the rail gauge corner on curves.

2.4.3 Dissipated energy

The energy dissipated in wheel/rail contact is called the wheel/rail rolling resistance, and it depends on the contact creepage and traction forces. Creepage is a measure of the relative motion of the wheel with respect to the rail. It has three components: γ_1 - longitudinal, γ_2 – lateral and ω_3 – spin (around the normal to the wheel/rail contact). Each of these creepages has an associated creep force: T_1 - longitudinal, T_2 - lateral and M_3 - spin moment.

For given vehicles running on given track, these creepage and force values can be predicted using vehicle dynamics simulation software. The instantaneous energy dissipated at a given wheel/rail contact (E_D) is given by the sum of the products of these creepages and forces:

$$E_D = T_1.\gamma_1 + T_2.\gamma_2 + \omega_3.M_3 \qquad \textit{Equation 9}$$

In practice, away from gauge face contact spin creep is relatively small; hence commonly only lateral and longitudinal creepage and force are used in this calculation of E_D. Units of E_D are Nm/m, which can be expressed as Newtons, or as Joules/m.

Small-scale and full-scale laboratory experiments have shown that, away from the high creepages seen in gauge face contact, wheel and rail wear is related to E_D in a complex way. Up to the saturation of friction, wear is proportional to E_D. Past this, there is evidence that wear is independent of E_D until thermal effects from very high dissipated energies cause a large increase in wear rate. Hence wheel and rail profile design should aim to reduce E_D by improving vehicle steering and reducing creepage and creepage forces - using vehicle dynamics simulation software.

Note that in straight track the calculated values of E_D will be relatively small (creepages and creepage forces are small) and vary little from wheel to wheel in a vehicle. E_D values will be larger in curved track, and will increase as track radius decreases. Further, E_D values will vary substantially from wheel to wheel, with the highest values tending to appear on the outer rail wheel of the leading wheelset in a bogie. Therefore calculations to assess the effect of newly developed wheel or rail profiles on wear need to focus mainly on these outer-rail leading-wheelset wheels, but the effect of the profiles on wear at other wheel positions should still be examined.

Chapters 4 and 7 of Reference 2 give further methods of predicting wear and RCF from creep, force, and stress calculations. Methods include wear maps, what is known as the “Archard” wear model, and methods based on the extent to which stresses and traction coefficients exceed the shakedown limits illustrated in Figure 11. The prediction method used will depend on the experience of the engineer, the available empirical data, and the types of output obtainable from the software used in the profile design process.

2.5 Rolling resistance

The energy dissipated (E_D, see equation 9) can also be expressed as Joules per metre rolled. The sum of energy dissipated over all (8) vehicle wheels can then be taken as a measure of rolling resistance. Vehicle dynamic simulations can be used to estimate the rolling resistance produced by newly developed wheel or rail profiles in straight track and track with a range of curves. If the track modelled in the simulation software is representative of a real stretch of track, then an estimate can be made of the effect of newly developed wheel or rail profiles on the energy dissipated over a whole route, and hence on fuel use.

For most passenger operations the fuel used to overcome rolling resistance is a small percentage of costs, and it is unlikely that a consideration of rolling resistance in wheel/rail profile development will be worthwhile.

In contrast, rolling resistance is of importance to freight operations. First, high rolling resistance requires more tractive effort and greater risk of locomotives stalling on curvaceous grades. This can be a particular problem for longer trains that do not use distributed locomotive power. Second, fuel costs are a higher percentage of operating costs on freight railways. Therefore for freight railways it may well be beneficial to consider the effect of wheel and rail profiles on overall vehicle rolling resistance both in curves and straight track.

2.6 Ease of application

Ease of application is unlikely to be an issue for new wheel profile designs. New wheels can simply be procured to the new wheel profile, which would need to be supplied to the manufacturer in digital format. Wheels already in service that are near to an old profile turning limit (for example on flange height or width) can be turned to the new profile assuming sufficient wheel material is left after turning to make the process cost-effective.

The situation with rails is different. The procurement of new rails to a new (and therefore non-standard) rail profile will require cooperation with the rail manufacturer who will need to produce new rolls for rail production. Production of new rolls and their installation in a production line is expensive and unlikely to be commercially attractive unless large quantities of rails are purchased.

With regard to the application of a new design profile to in-service rails, the key consideration is the amount of metal that needs to be removed from current rails to produce the new rail profile. An illustrative example of this is given in Figure 13 which shows a current in-service worn rail (in red) and a proposed new design profile (in blue). The figure shows that, in this case, large amounts of metal need to be removed from the shoulders of the in-service rail to produce the new profile. In practice, even more metal would need to be taken off, since in re-profiling it is normal practice to remove deformed and cracked material from the top of the rail.

Metal can be removed from in-service rails by milling and by grinding. Milling is the most efficient way to remove large amounts of metal, but the technique is still relatively new and requires very close control. Grinding is much more commonly undertaken, using commercially produced grinding trains – such as those produced by the grinding companies Loram, Speno, and Harsco. Careful control of grinding or milling to the desired shape is required in order to provide smooth radii to the rail profile and to avoid facets that could cause wheel damage.

The metal removal rate depends on considerations such as the number and type of stones on the grinding train, and the speed of the train. If the capabilities of the grinding train are known, then the number of grinding passes needed to transform the old profile to the new can be estimated. Therefore throughout the rail profile design process, it is important to consider the grinding passes needed to produce the emerging design profile to see if it is practically achievable. If its practicality appears questionable then it may be necessary to search for a profile that is non-optimal in terms of performance but can at least be applied in practice.

A further option to be considered is whether the desired rail profile can be applied in suitable stages, for example by rail grinding operations spread 6 months apart, so that the desired rail profile is eventually achieved. This type of gradual production of the final desired rail shape is increasingly being adopted, especially in freight track.

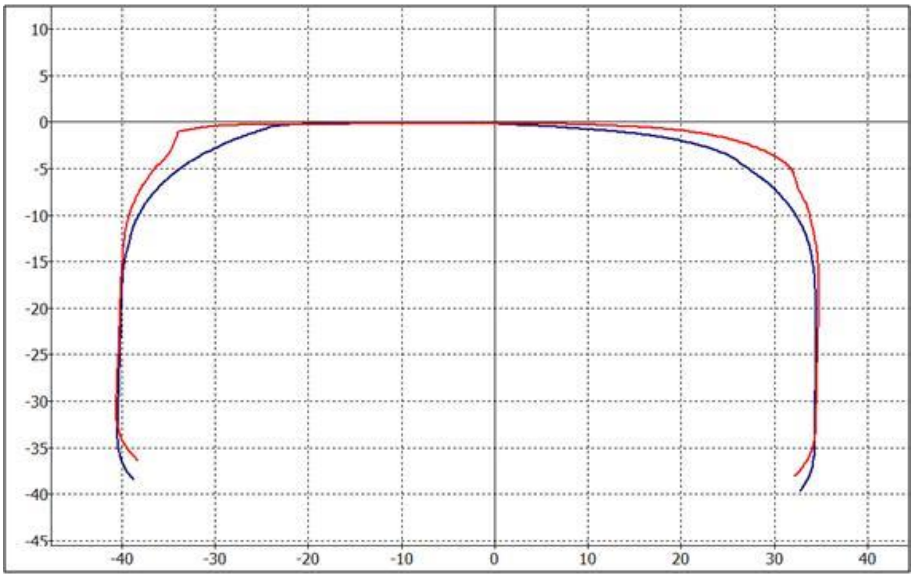

Figure 13: Example of metal removal needed to produce new design profile (blue) from current profile (red) (x-axis and y-axis dimensions are in mm, gauge face of rail is right-hand side in plot)

2.7 Other factors

Other factors may need to be considered when developing new wheel or rail profiles.

2.7.1 Collateral effects

As with ease of application, there should be few, if any, collateral effects with new wheel profile designs for vehicles running on existing track. The application of a new wheel profile on one fleet should have little effect on the performance of other fleets using the track.

However, if a new design rail profile is applied then it will affect the performance of all fleets that run over the track. If a new rail profile is designed to work with one (and likely the biggest) fleet running over the track, then the designer needs to undertake all necessary work to provide assurance that the profile will not adversely affect the safe (and possibly economic) performance of other fleets. This will likely require some or all of the considerations in Sections 2.1 to 2.5 to be revisited for these other fleets.

2.7.2 Track circuit and return current implications

Good electrical contact between wheel and rail is required for track circuit operation and for return current flow in electric traction. Under most conditions new wheel and rail profiles should have no significant effect on electrical contact. However, in some circumstances a new wheel (or rail) profile may force contact onto a running line on the rail (or wheel) that is rusted or otherwise contaminated. For example, a ground rail may have a narrow shiny running band with an otherwise rusty and contaminated head. A new wheel profile that forces contact off the narrow running band may cause track circuit or return current problems. The likelihood of this problem occurring needs to be considered early in the profile development exercise.

2.7.3 Adhesion and braking distance

As with electrical contact, it is unlikely that new wheel or rail profiles will adversely affect adhesion and hence braking distance. The change in wheel/rail contact area caused by new profiles might affect adhesion. However it is possible that forcing contact onto rusty and/or contaminated rail may change nominal adhesion (see Section 2.7.2 above).

It is also possible that a new wheel or rail profile will increase the probability that gauge face/flange lubricant could migrate to the rail head, reducing adhesion and increasing braking distance. This is unlikely, but the lack of any quantitative models of lubricant wheel/rail migration mean that it cannot be considered at the profile design stage. If such unwanted lubricant migration is considered to be a possibility, observations may need to be carried out at the profile trial stage. If the problem is real, the simple solutions of lowering the lubricant spreader bar height (wayside system) or altering the delivery nozzle angle (on-board system) may be feasible.

2.7.4 Set-up of turnouts and other special trackwork

Interlocking, and hence turnout control and operation, can be affected if switch and stock rail profiles are such that they allow significant relative vertical movement. Switch and stock rails are generally installed as pairs, and it is most unlikely that a new rail profile would be ground onto a stock rail without attention being given to the switch rail. Hence new rail profile designs ought not to influence turnout operation. However, if it is intended that the new profile should be applied through turnouts and other special trackwork, the effect on their operation should be considered.

Consideration should also be given to the transition between the inclined plain rail and the (generally) vertical rails in turnouts and other special trackwork, and how this section of rail will be treated if a new rail profile is considered.

2.7.5 Components and wear limits

In most cases when developing new wheel or rail profiles it ought not to be necessary to consider wheel and rail wear limits. However, in some circumstances, new wheel (or rail) profiles can cause problems with rails (or wheels) near their wear limits.

For example, if the rails are near their vertical wear limits and the new wheel profile forces contact to the edges (outer or inner) of the rails, it is possible that vertically split rail head defects will increase. This could occur as a consequence of the wheel load being concentrated on the section of a rail head that is both thin and unsupported by the web, leading to large shear stresses. In the same way, a new rail profile that forces contact towards the end-of-tread area of the wheel could cause vertical splits in the wheel if the wheel is near its wear limit. Also, the field side of rails need to clear the outer tread edge of hollow worn wheels.

At the beginning of profile development, therefore, the engineer should consider the wear state of the existing mating component. Towards the end of profile development, the engineer should consider whether existing wheel and rail wear limits remain applicable.

2.7.6 Uni-directional rolling stock

For rolling stock that travels mostly uni-directionally, wheels in trailing wheelsets could wear to a reduced flange angle. Then, if the direction of travel of the rolling stock is reversed, the reduced flange angle can encourage derailment by flange-climbing.

Also, the hollow wheel wear that arises in the trailing wheelsets of rolling stock that travels mostly uni-directionally, can cause hunting problems if the direction of travel of the Rollingstock is reversed.

3 IN-SERVICE ASSESSMENT OF WHEEL AND RAIL PROFILES

Wheel and rail profiles are assessed as a usual maintenance operation. In particular, close monitoring of wheel and rail profiles is required:

a. in order to gather data on the existing wheel and rail profiles when considering any change to the system; and

b. During trials of a change to the system.

There are two basic ways of assessing wheel and rail profiles in service; directly by measurement of profiles, and indirectly by visual inspection of wheel and rail surface condition. These are described briefly in Appendix A. For more detailed information, reference should be made to:

- For wheels: RISSB Code of Practice “Wheel Defects” (Reference 14);
- For rails: ARTC Procedure “Inspection of Rail Wear” (Reference 15) and UIC 712 “Rail Defects” (Reference 16); also:
- For wheels and rails, UIC publication “Atlas of Wheel and Rail Defects” (Reference 17).

4 CHANGE MANAGEMENT PROCESS FOR WHEEL/RAIL PROFILES

4.1 Approach to wheel and rail profile development

The development and implementation of new wheel and/or rail profiles needs to follow a structured approach:

- Stakeholder consultation.
- Definition of the objectives of profile development.
- Definition of performance requirements.
- Choice of the software and method that will be used in profile development.
- Collection of basic vehicle/track/operation data needed for the development process.
- Profile development, likely using an iterative approach with dynamic simulations.
- Conduct of a trial with analysis of results against success criteria.
- Stakeholder consultation and agreement.
- Implementation of the new profile(s).
- Documentation of all development work in a retrievable form, and preparation of a technical file.

These items are described in Sections 4.2 to 4.11 and illustrated in flowchart form in Figure 7.

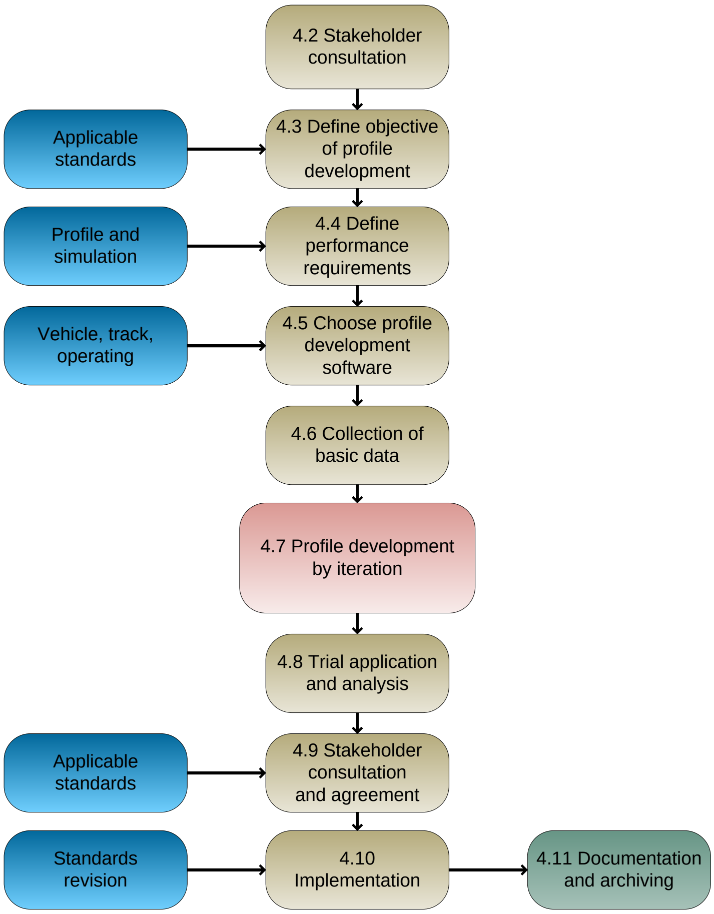

Figure 7: Flowchart of profile development process

4.2 Stakeholder consultation

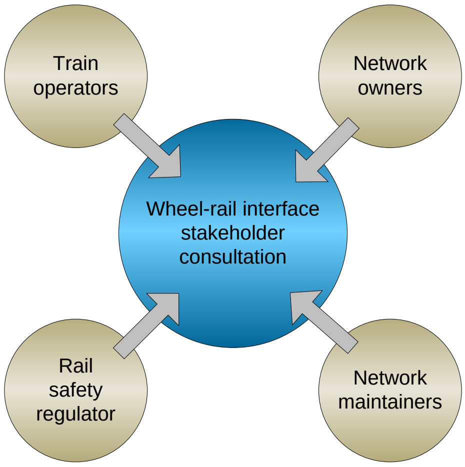

Figure 8: Wheel-rail interface stakeholder consultation

The wheel-rail interface is a system that affects the train operators and the network owner and maintainers. Therefore, the starting point for any proposed change that affects the wheel-rail interface should be to consult all the interested parties. Since there could be safety implications, the consultation should include the rail safety regulator.

The Wheel-Rail Interface Committee could provide a useful source of information and advice.

4.3 Definition of objectives of profile development

This is the most important task in wheel/rail profile development, since a clear understanding of the objectives of the development essentially defines all further work. Objectives need to be specific. For example, the goal may be to achieve longer wheel life, but the objective needs to specify how this will be achieved. In this example, the objective might be “to achieve longer wheel life by reducing spalls/shells arising from angled RCF cracks.” As other examples, the primary goals may be to reduce hunting in straight track, or reduce lateral forces in curved track, or to reduce lateral forces while also reducing RCF on the low rail. Quite likely goals will be mixed and require compromise and trade-offs, such as improving curving performance whilst also improving stability in straight track.

Whatever the goals of the development, it cannot be overstressed that the objectives should be clear and specific, beneficial, judged achievable from simple engineering considerations, agreed with stakeholders, and understood by all in the process.

Objectives will also depend on whether the wheel/rail profiles are for a new system, for new vehicles entering service on an existing system, or for modifying wheel or rail profiles on an existing system.

4.3.1 Wheel profiles and rail profiles for a new system

This is perhaps the most straightforward situation. In most cases it should be possible to use a standard pair of wheel/rail profiles that is already available, and this should be the preferred option. Examples of these are Australian, US and European wheel/rail profiles. If this is not possible, or if available profiles are felt unsuitable (for whatever reason), then objectives for new profiles need to be defined. These will depend on factors such as:

- Vehicle design, especially of the suspension.

- Track design, including curve radii, gauge, superelevation rules, design rail inclination, turnouts and other special trackwork.
- Type of service: high-speed passenger, metro, freight, etc.
- Range of operating speeds.
- Regulatory requirements that control safety, which might include ride quality (for example, lateral accelerations), track forces (for example, wheel and wheelset L/V limits), and minimum flange angle. If these parameters are not covered by regulatory bodies, they will still likely need addressing in any development.

Based on this list, development of wheel and/or rail profiles for a new system may require some or all of the following objectives:

- Minimum flange angle of at least 65º, and preferably at least 68º.
- Ride quality and track forces that meet all regulatory requirements.
- No vehicle instability at speeds up to 10% over maximum operating speed.
- Low predicted levels of wheel and rail wear and RCF. Ideally the objective should be to minimise wear and RCF, but in practice minimisation cannot be guaranteed. Hence low levels of wear and RCF should be the aim, based on the theories described in Section 2.4).

4.3.2 Wheel profiles for new vehicles on existing rail system(s)

In this case, it is likely that the existing rail profiles will be maintained and only new wheel profiles for the new vehicles need be considered. It is necessary to know what the actual rail profiles are for the existing system (or systems where the vehicles will operate on more than one track owners' system). However, the factors and suggested objectives given in Section 4.3.1 are still relevant.

4.3.3 Change of wheel profiles for existing vehicles on existing rail system(s)

This case is similar to that in Section 4.3.2, and again, the factors and suggested objectives given in Section 4.3.1 are still relevant.

4.3.4 Modified rail profiles for an existing system

This case is different to those in Sections 4.3.1 and 4.3.2 in that new rail profiles are likely only to be required at specific track locations and applied by grinding the rails. Different rail profiles might be needed in straight track, moderate curves, and severe curves. The factors and objectives described in Section 4.3.1 are still relevant, but the profile developer needs to ask the following questions before objectives are set:

- Will grinding the worn rail back to its design shape give acceptable performance?
- If not, are different profiles needed for straight track and curves track?
- What are the main objectives for a new profile for straight track? In practice, wear and RCF are generally minimal in straight track and preventing vehicle instability (hunting) is likely to be the key objective.
- What are the main objectives for a new profile (or new profiles) for curved track? In practice, wear and RCF and maximum flange angle are likely to be the key considerations.

- Are different profiles needed for track with moderate and severe curvature? Ideally this may be so, but consideration should be given to the logistics of having more than one rail profile for curved track – especially if the benefits offered by the second profile over the first are relatively small.
- Are different profiles required for the outer and inner rails? The use of different rail profiles on the outer and inner rails (so-called asymmetric profiles) can offer benefits in curving performance and wear and RCF.

4.4 Definition of performance requirements

Many performance requirements will arise from the consideration of objectives undertaken in Sections 4.3.1 to 4.3.4. These will relate to general profile requirements, such as low predicted levels of wheel and rail wear, and need to be clearly identified. However, it is also essential that all applicable standards and company documents are collected and their requirements understood as well. (This is also needed to ensure that the correct objectives are set in Section 4.2.) For Australia, Table 2 lists the regulatory documents and standards that need to be consulted.

The outcome will be a list of performance requirements, of which some will be mandatory and some desired. During the profile development process the developer will need to make trade-offs between requirements that point to conflicting profile changes. For example, achieving low wear and RCF may not be compatible with ensuring good curving performance and stability. In these trade-offs the mandatory requirements must be met and the desired requirements achieved as best possible.

Document	Relevant content
AS7508: Railway rolling stock track forces and stresses (Reference 11)	Lateral wheel-to-rail force limits Maximum bogie side L/V
AS7509: Railway rolling stock dynamic behaviour (Reference 7)	Maximum lateral acceleration Maximum vertical acceleration Minimum vertical wheel/rail force Maximum axle sum L/V
AS7514: Railway rolling stock –wheels (Reference 18)	Current Australian wheel profiles Wheel rim width
AS1085.1: Railway track material: Steel rails + Amendment 1 (References 19 & 20)	Current Australian rail profiles

Table 2: List of required documents for profile development

For development of new wheel profiles, suitable checks are required to ensure that the proposed wheel profile shape, in new and worn condition, is compatible with the rail geometry at turnouts and other special trackwork.

4.5 Choice of profile development method and software

There are two basic tasks in profile development, first the development of a profile either from new or by manipulation of a current profile and second, vehicle dynamic simulations using the new profile to predict performance.

4.5.1 Profile development

Historically profiles have been represented by a series of straight lines and circular arcs meeting at common tangents to prevent abrupt changes in slope; an example is shown in Figure 9. If the decision in profile development is to modify an existing design, then this can be done simply by changing the radii and intersection points of the circular arcs at the rail head or wheel tread – taking care to ensure that adjacent arcs meet at a common tangent. This type of modification can be done using common CAD software or mathematical software such as MATLAB®.

Once changes are completed the profile needs to be converted to a digital format that can be input to the vehicle dynamic simulation software. One simple way of doing this is to transcribe the radii and transition points to the ∗.mpt file format used by MiniProf software. All common simulation software accepts profiles in this format. Typically, profiles are measured with between 2 and 80 points per millimetre.

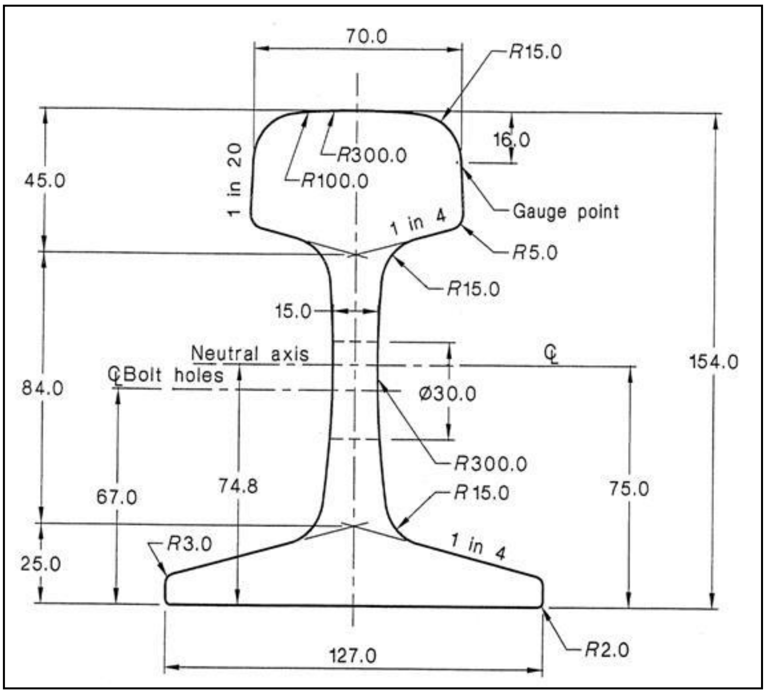

Figure 9: Australian 50 kg rail defined by straight lines and circular arcs (dimensions given in mm)

If the existing profile is in digital format (where x-y pairs define the profile), or if the new profile is to be designed in digital format then modification/design is less straightforward since no commercial software is available. If work is to be done with digital profiles the options are to use packages such as MSOffice EXCEL and MATLAB®.

4.5.2 Vehicle dynamic simulation

A number of commercial software packages are available for the prediction of railway vehicle dynamics, for example:

- VAMPIRE®
- NUCARS®
- GENSYS
- SIMPACK
- VI RAIL (formerly Adams/Rail)

- The VOCO series of software developed at INRETS (France).
- Universal Mechanism Loco, developed at Bryansk State Technical University, Russia.

Whichever software package is proposed to be used, it is important that it has been shown to have been validated.

These software packages all tend to have the same basic packages and mode of operation (Figure 10):

- The vehicle is modelled by a network of bodies connected by flexible elements. This is called a multibody system. The bodies are usually rigid but can be flexible (masses and moments of inertia need to be specified), and springs, dampers, links, joints, friction surfaces or wheel-rail contact elements can be selected. The complexity of the system can be varied to suit the vehicle and the results required, but the amount of detail used to prepare the model will vary according to the type of suspension and the required outcome of the modelling. If several fleets are affected by the profile development exercise then it may be necessary to have separate models for each type of vehicle.
- Track can be modelled in three ways.
  - First, using track with specific irregularities designed to examine specific types of vehicle dynamic behaviour such as ability to negotiate isolated track irregularities, and ability to negotiate cyclic track irregularities. This is required by RISSB standards; refer to Sections 8 and 9 in Australian Standard AS7509.
  - Second, track can be modelled as smooth track with the design curvature, cross-level and gauge, and with idealised discrete misalignments as required. Smooth track gives results that are easier to understand, but results are less relevant than those found with measured track. This type of track is not recommended, except in some cases if the effect of parameter changes on basic behaviour is to be assessed.
  - Third, as track with measured (or estimated) track geometry including vertical and lateral roughness (misalignment) and cross-level and gauge changes. Track with roughness gives outputs with greater variability but which are more representative of “real” behaviour. Note that Australian Standard AS7509 specifies that track data for dynamic behaviour modelling should have a similar frequency spectrum and amplitude of irregularities to the actual track that the modelled vehicles will operate on.
  - It may be possible to assemble a single track model that is representative of the route or network. Alternatively it may be necessary to assemble several track models to fully characterise the route/network.
- Relatively sophisticated routines for calculating the shape and size of the contact patch and the normal and tangential forces arising from wheel/rail contact.
- External forces, such as coupler forces and wind loading can be added if necessary.
- Time-stepping integration to calculate forces, displacements and accelerations as the vehicle model travels over the track model. Nonlinearities which occur at the wheel-rail contact point due to creep and flange contact and in the suspension, especially if displacements are large, can be included in the integration.
- Post-processing and presentation of outputs.

Slightly different outputs will be given depending on which software package is used, but effect will be small compared with the errors which are introduced through inappropriate choice of vehicle, track, and wheel/rail parameters. It needs to be stressed that results from these theoretical prediction methods are only as good as the parameter values put into them; this applies both to the vehicle model and to the track input excitation.

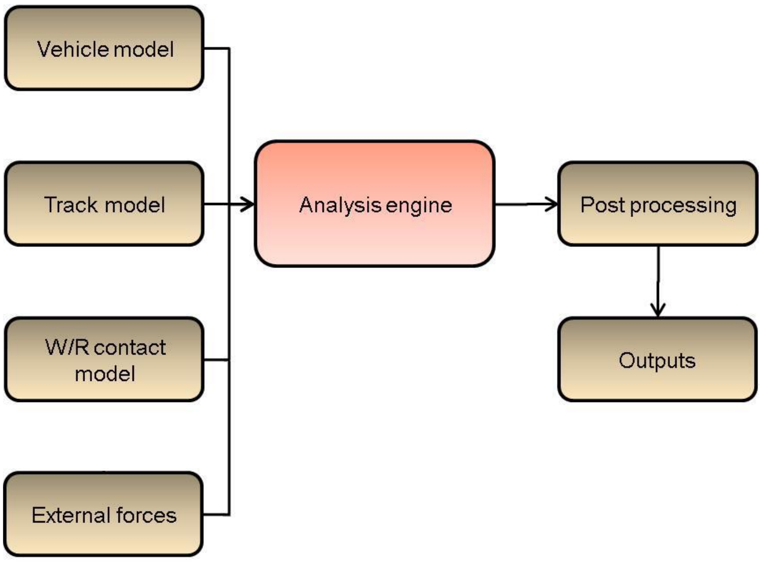

Figure 10: Schematic of software simulation

Much effort has gone in to try to make these software packages user-friendly, with simplified input and output routines and use of graphics to aid understanding. However, training is needed in use of the software, and considerable vehicle dynamics experience is needed to understand and interpret the outputs which can be complex. Understanding of natural frequencies and associated modes, frequency response, input and response PSDs can help to make sense of time histories. Caution needs to be exercised if the software user cannot understand why the predicted behaviour changes in a particular way when the inputs are changed.

Finally, before the profile development exercise is started, some effort needs to be put in to validate the assembled track and vehicle models. That is, trial simulations are needed to show that the models do in fact give predicted behaviour that is both internally and physically consistent and – if at all possible – generate results that compare well with measured or known vehicle behaviour. For example, if a new wheel profile is to be developed, simulations should be run with the old wheel profile and the outputs examined to confirm that the models agree with the current vehicle behaviour.

4.6 Collection of basic data

A large quantity of data needs to be collected, examined, and approved for use in a profile development exercise. The amount of data needed will depend on the extent of the simulations needed. For example, if the exercise is simply to change the wheel profile on one fleet of vehicles then only one vehicle model needs to be assembled. However, if the objective is to modify rail profiles in track over which a number of fleets run then vehicle models may be needed for all fleets. As a further example, if the track over which vehicles run is relatively uniform in quality, then the track model could be relatively short. But if the track is variable then it may be necessary to do simulations over several representative lengths of track.

The first task then is to decide on the extent of data needed. To some extent this will depend on considerations already addressed in Sections 4.2 to 4.4. Questions that need to be asked are:

- How many track models are likely to be needed to characterise the route or network?
- Can design track be used, or will track with measured geometry (roughness) be needed?
- How many different vehicles need to be considered in the exercise? Based on this, how many separate vehicle models are likely to be needed?
- Will any current wheel or rail profiles be unchanged in the exercise? For example, if the aim is to modify current wheel profiles, will current rail profiles be used throughout the development exercise?
- If rail profiles are to be changed, will this be at just one location or throughout the route or network? Will one new profile be developed for all track, or several profiles that will be applied in straight track and curved track?

The actual data needed for profile development will be defined by the manual for the simulation software used; but the following type of data is most likely to be needed:

- Vehicle parameter data (needed for each vehicle under consideration)
  - Masses and inertias (possibly in all three dimensions) of all major vehicle components, such as body shells, bolsters, wheelsets.
  - Positions of all force-generating connections such as springs and dampers.
  - Characteristics (in all appropriate dimensions) of all force-generating connections, such as spring stiffnesses and, damping rates and including non-linearities.
  - Major vehicle dimensions, such as wheelbase.
  - Extremes of loading.

It is possible that sufficient design details are available for collection of these vehicle details. However, unless parameter details are certain, physical testing should be used to validate key parameters. This is a requirement of the Australian Standard AS7509.

- Track parameter data
  - Track with isolated and cyclic track irregularities as required by Australian Standard AS7509 (Reference 7), Sections 8 and 9.
  - If a route is chosen and measured track geometry is unavailable, design alignment, including curvature, superelevation and gauge; plus a track geometry roughness file typical of the track under consideration.
  - If a route is chosen with measured track geometry, relevant samples of track containing straight track and a range of track curvatures.
  - Equivalent track mass, stiffness and damping.
  - Range of wheel/rail friction coefficients to be considered for flange and tread contact positions, typically 0.3 for the tread contact, but ranging from 0.3 up to 0.5 maximum, and typically 0.3 for the flange contact, but ranging from 0.1 (with flange or rail side lubrication) up to 0.5 maximum.
- Wheel/rail profile data
  - If a wheel profile is to be developed, representative measured rail profile pairs, with inclination.

- If a rail profile is to be developed, representative measured wheel profile pairs for each vehicle under consideration.

It is most important to ensure that there is a sufficient range of wheel and rail profiles to be representative. When measuring existing wheel/rail profiles it is useful to make notes (if information is available) of such as wheel kilometres since turning and rail age/accumulated tonnage, and the presence of any RCF. This information could be useful if profile development ends with simulation of wear of the new design profile (see Section 4.7), and it may also be useful when undertaking trials of the new profile. In both cases results for the new profile can be compared directly with performance of the old profile.

- Operating data
  - Operating speed profile and superelevation deficiency/excess profile.
  - Directional running of the vehicle including empty and loaded running for unit wagons.
  - Train force for curves and coupler yaw angles for determining lateral load inputs.
  - Traction and braking requirements and their effect on wheel/rail forces.

4.7 Profile development

There are two basic methods of developing a new wheel or rail profile. In the conventional method, which is an iterative technique, engineering judgement is used to progressively modify an existing profile until dynamic simulations show that it enables performance requirements to be met. This is described in Section 4.7.1. In the genetic algorithm method the software itself produces successive generations of profiles which, hopefully, eventually also lead to the desired performance requirements. This is described in Section 4.7.2.

Whatever approach is adopted, it needs to be stressed that wheel and rail profiles cannot be considered in isolation. They work together, and whether it is a new wheel profile or a new rail profile that is to be developed, the most important task is to focus on the wheel/rail interface.

4.7.1 Conventional profile development

Figure 11 shows a flowchart for a typical profile development exercise. In principle the approach is straightforward. However, the process does need considerable engineering judgement, and an ability to gauge the effect of profile modifications on likely dynamic performance, or rather the ability to look at simulation outputs and infer what profile modifications will likely give improved performance.

The first stage in profile development is to identify a pre-existing profile that is believed to be reasonably close to that which will finally be chosen, in the profile developer’s judgement. It is possible to start with a blank sheet of paper and start with a brand new profile, but this is likely to lead to many more iterations to develop a profile that meets requirements. Current Australian rail profiles are given in AS1085.1 (References 19 and 20).

Current Australian target rail profiles for grinding are given in Reference 21. Current Australian wheel profiles are given in AS7514 (Reference 18).

The second stage, which again requires good judgement, is to modify the chosen profile in ways that ought to lead to performance that better meets requirements. As examples, if stability at high speed is a primary requirement then reducing the slope of the wheel tread, which will reduce conicity, may be done. If good curving performance is required then increasing the radius in the wheel flange/tread transition will increase conicity in curves. If a new ground rail profile needs to be resistant to gauge-corner RCF then relief of the gauge corner may be needed.

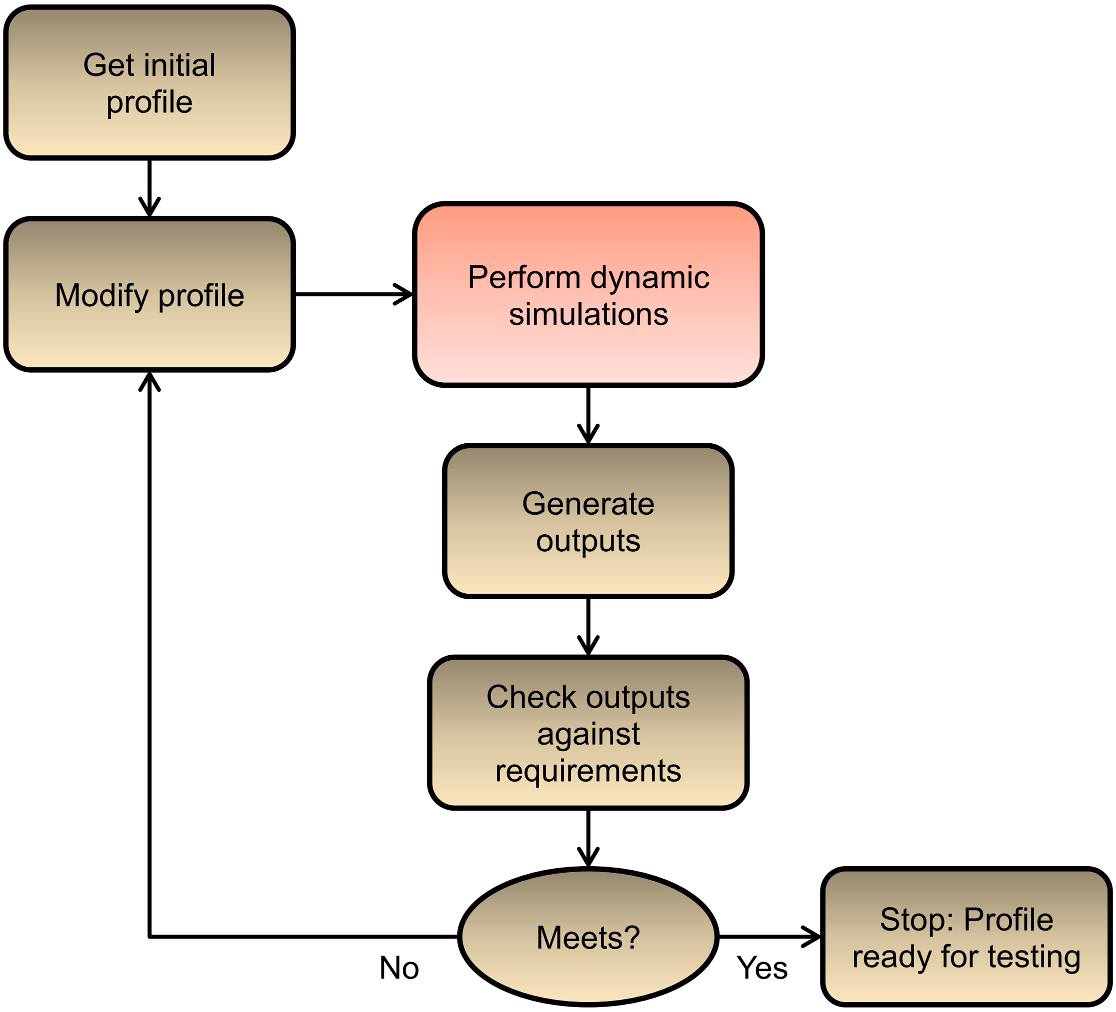

Figure 11: Flowchart of conventional profile development

The third stage is to perform dynamic simulations with the vehicle/track models using the newly modified profiles, and to generate suitable outputs. Note that if a new wheel profile is being produced, modelling needs to comply with the requirements of the Australian dynamic behaviour standard AS7509 if it is intended that no instrumented testing will be undertaken in the trial application (see Section 4.8).

The fourth stage is to compare the generated outputs with the defined requirements. If requirements are met then the process can stop, unless there are grounds for believing that further improvements are achievable. If requirements are not met, then further modifications are made to the profile and the simulation/checking process is repeated. This iteration goes on until the profile gives the required performance.

As described in Sections 4.3 and 4.4 profile development may well require trade-offs between conflicting requirements. It is emphasised that mandatory requirements must be met, with other desired requirements achieved as best possible.

The development should also consider sensitivity analysis of the wheel and rail profiles for acceptable tolerances and variations. As described in Section 2, these can include gauge variations, profile wear, effect of longitudinal train forces, curving speeds and bogie design.

Appendix B provides a check list of technical issues to be considered in development of wheel and rail profiles.

4.7.2 Emerging genetic algorithm method

Relatively recently a new technique has emerged for generating wheel and rail profiles. This is known as the genetic algorithm method, and its intention is to mimic evolutionary processes in nature. References 22 and 23 give details of the genetic approach.

As an example, considering rail profile development, the first approach is to develop a penalty index based on key performance parameters such as maximum contact stress, maximum lateral force on the track, maximum L/V, wear and ride index. Each parameter that makes up the penalty index can be weighted to reflect its importance to the profile designer, and a desired index is defined.

Then specific rail profiles are chosen as “parents” and their x, y coordinates represented by binary numbers (“genes”). These genes are then combined or ‘mated’ to produce offspring genes and then reconstructed into profiles that have random combinations of the properties of the parents. The offspring represent different profiles to the parent but share similar characteristics. Mutations are also made by randomly changing the genes to introduce occasional larger changes. Each of the offspring profiles is then evaluated by running a computer simulation of the behaviour of a vehicle running on rail with these profiles and calculating a penalty index that can be compared to the desired index. The process is then repeated with an algorithm included to choose the best profiles to act as parents for the next generation. In this way the profiles that have the best properties are more likely to be included but there is a chance for beneficial aspects of other profiles to survive. This process leads to a designed profile that meets or best approaches the desired penalty index, and it does not need the same level of engineering judgement as the conventional approach.

This genetic algorithm process looks attractive, but it has not yet been proved that it improves upon conventional profile development. Also, as a new technique it is not clear that commercially available software is available for general use. If this approach is to be used, it should be done only by developers with prior experience of its application.

For profiles developed by either route (Sections 4.7.1 and 4.7.2), a possible final step is calculation of how the profile varies over time as wear occurs. Methods are available to do this, but it is a specialist area requiring extensive knowledge and experience, and accurate information of both vehicle and track parameters – including measured track geometry. Calculations also depend sensitively over the assumptions required to spread calculated wear over the running surface of the wheel or rail. If the engineer undertaking profile development has sufficient experience, with accurate knowledge of both the vehicles and track that will see the profiles in question, this final step can be considered; but it is not a necessary requirement.

4.8 Trial application and analysis

Some degree of trial application of the final developed profile(s) is needed. This may range from simple measurements or observations (for example, visual observation to assess whether rail RCF has been reduced on trial ground rails) to extensive use of instrumentation to characterise vehicle performance.

The trials should:

- Ensure that the shape of the wheel or rail profile under trial has been implemented to accurately represent the profile shape intended.
- Record how the new wheel or rail profile changes its shape as it wears. This information could be used to revise the initial new profile. Thought is also required as to how to restore the worn profile back to its new shape.
- Be long-term trials that continue monitoring of the wheel and/or rail profile shape throughout its profile life, and not just a trial of the profile in its new condition.
- Consider wheel/rail squeal since calculation of squeal at design stage is difficult and little off-the-shelf software is available. (In practice, risk of squeal should be reduced by ensuring that – as far as possible - creep forces are not saturated on the inner rail in curves.)

4.8.1 Wheel profile changes

Australian Standard AS7509 states that changes to the wheel profile constitute rolling stock modifications within the meaning of the standards. The dynamic behaviour of new or modified passenger rolling stock should be evaluated for any areas of dynamic performance within those standards that could be affected by the change to wheel profiles.

However, in lieu of physical testing these standards allow use of computer simulation programmes that cater for wheel/rail interaction and rolling stock/suspension linearities. The software packages described in Section 4.5.2 allow such simulation. It is therefore the responsibility of the profile developer, in discussion with the profile user, to determine whether, and to what extent, testing is required in the trial. In this determination, the following factors need to be considered:

- Is there confidence that the simulations give an accurate assessment of actual dynamic behaviour?
- Is the new profile predicted to have minimal effect on safety-related performance?
- Is the new profile reasonably similar to profiles on wheels in service on other vehicles?
- Are the new profile wheels to be used on passenger or freight operations?
- At what speed will the new profile wheels operate?
- How well are the rail profile conditions known?
- How will the shape of the wheel profile change with wear until wheel reprofiling is undertaken?

It is clear from these questions that engineering judgement is needed in the decision of what sort of trial application – including timescale - is needed; but the decision needs to be taken conservatively. That is, if there is any doubt then instrumented testing should be undertaken. Whatever is decided, the decision-making process needs to be documented with the reasons for the final agreed decision clearly stated. Following this, a method statement for the trial needs to be prepared and agreed, with an approximate timescale and clear success criteria against which the results of the trial can be compared. (If instrumentation is used, many of the success criteria will be found in AS7509.

If the decision is only to use visual observations, then the trial timescale should be set appropriately. For example, if the objective of the new wheel profile is to reduce flange wear or wheel RCF then the trial timescale should be set to at least the time when wear/RCF normally becomes visible. Consideration should be given to using control wheels in the trial.

If the decision is taken to use instrumentation in the trial, then sufficient instrumentation and test tracks will be needed to meet the requirements of AS7509. Instrumentation might be based on such as vehicle-mounted accelerometers and displacement transducers, or it might be possible to use instrumented wheelsets if available. However, the application of the new profile to an existing instrumented wheelset (plus the possible consequent requirement for re-calibration) or the acquisition of a new instrumented wheelset may be prohibitively expensive. Finally, even if extensive instrumentation is used in the trial, the need for visual observations over a longer period may still be necessary, for example to demonstrate a reduction in wear/RCF.

4.8.2 Rail profile changes

Changes to the rail profile are not explicitly covered by the dynamic behaviour standards in AS7509, but, since rail profiles can have a similar effect to wheel profiles on dynamic behaviour, it is advisable to consider a rail profile trial in the light of these rolling stock references and to ask the same questions regarding the extent of a trial. Changes to the rail profile include a revision of the gauge or permitted gauge limits.

However, there is a key difference in that while trial wheels with a new profile will travel over all of a route, trial rail profiles will be applied only to one or two track sections. There may therefore be a reduced need for simulations and also for instrumentation. For example, if the new rail profile is intended only to be ground onto rail in straight track (to reduce hunting), there may be no need to consider extensive simulations or trials in curved track or transitions.

As with wheel profile changes (Section 4.8.1) decisions on a trial need to be arrived at conservatively, with the final agreed decision clearly stated. Following this, a method statement needs to be agreed with approximate timescale and success criteria. It is suggested that the trial should include a check that the intended rail profiles have been applied correctly, and that a check is made on the position and size of the wheel-rail contact bands (for example with spray paint).

The final stage of the trial is to analyse the data collected and determine if the new profile(s) have met the objectives of the development. If work has been done correctly and monitored throughout the development the trial ought to conclude with success. That is, the trial should not go ahead if results from the development imply that objectives will not be met. However, if the trial is unsuccessful, work needs to be done to understand why. No work on further profile modifications should go ahead until this understanding is reached. Work to understand failure should examine:

- The vehicle and track parameter inputs to the simulations.
- Possible errors in development of the vehicle and track models.
- Possible errors in undertaking the simulations and assessing the outputs.
- Whether some wheel profiles were not considered in the simulations.
- Rail profiles not applied correctly.
- Rail rotation under load not properly allowed for.

4.9 Stakeholder consultation and agreement

Prior to implementation of any proposed change that affects the wheel-rail interface, all the interested parties should be consulted again (see Section 4.2) and their agreement obtained to the proposed changes.

4.10 Implementation

Following a successful trial, implementation will be straightforward but will vary depending on the type of implementation.

For a new wheel profile, it is likely to be most cost-effective to apply the new profile as and when wheels are either purchased or re-turned on a maintenance basis. The benefits accruing from the new profile would need to be very great to justify re-profiling wheels that had not been already programmed for turning.

For a new rail profile intended to be applied by grinding, implementation can proceed as and when individual sections of track are scheduled for grinding maintenance. Again, benefits from the new profile are unlikely to be so great as to justify grinding rail that otherwise did not require grinding. The only exception to this would be if the new ground rail profile removed a clear threat to safety (derailment).

Implementation of a new profile for rail intended to be purchased is more of a problem. Because of the high cost of setting up production for a new rolled-rail profile, production is likely to be uneconomic unless large amounts of rail are to be purchased. During the profile development exercise, the developer should work with intended rail producers to establish likely costs of new rail.

Where a new wheel or rail profile is implemented, measurements of the new wheel or rail profile are recommended to ensure that the profile shape implemented accurately represents the profile shape intended.

Finally, wheel profiles currently allowed on Australian vehicles are specified in AS7514 (Reference 18). Rail profiles are specified in AS1085.1 (References 19 and 20) for new rail and Reference 21 for ground rail. Before implementation, the developer needs to work with the relevant standard organisations to add the new profiles to the current standards. This may involve development and approval of a technical case that demonstrates the conditions under which the new profiles can be safely used. This technical case can be based on the documentation and technical file development described in Section 4.11.

4.11 Documentation and archiving

It is important to ensure that the profile development process is documented fully with all relevant information archived for future examination as needed, because wheel and rail profiles have such a significant effect on vehicle performance, wheel/rail condition, and especially safety. It is good practice to document briefly all work undertaken including a technical file containing all data gathered and generated during the exercise.

The technical part of the file is the repository of all the detailed information related to the work, including intermediate and final simulation results. It will likely be mostly in electronic format, but may include documents/data in paper format if these cannot be scanned (for example for copyright reasons). The electronic component of the file should be backed up on a regular and defined basis. In the event of any problems with the developed profile(s) after implementation the technical information should allow any independent expert to understand what was done and where possible errors may have arisen. It will also act as a template for future profile development exercises

The documentation should ideally include the following information:

- The names of the organisations commissioning and undertaking the profile development.
- Summary of the vehicles/tracks to which the new profile(s) will apply, and the associated operating conditions.
- The objectives of the exercise.
- The relevant standards or company documents that need to be considered and, if necessary, complied with.
- Any other documents (such as published papers) relevant to the study.
- The requirements that the finished new profiles need to meet.
- All communications and minutes of meetings associated with the study.
- Summary of the software chosen for profile development: profile manipulation software and vehicle dynamics simulation software.
- Summary of the basic data acquired for the study. This need only be general, since actual data will be included in the technical file.
- An outline of the development process undertaken, with general results from intermediate profiles if profile iteration is undertaken.
- Details of all profile modifications/manipulations undertaken.
- Reasons for decisions taken with respect to profile iterations and final choice of trial profile(s).
- Simulation results for the final chosen profile(s). Basically the results should show that the profile(s) chosen meet the requirements set at the start of the study. Also:
  - A full list of all vehicle, track, and operating parameters used in the vehicle dynamic simulations.
  - All input files used in the dynamic simulations, including all profile iterations.
  - All output files arising from the simulations, including all profile iterations.
  - All files (for example MSOffice EXCEL) generated in the analysis of the simulations.
- The trial profile(s) either in a form such as that shown in Figure 9 or in digital format, such as a series of x-y pairs.
- Details of any trials undertaken, results, findings and conclusions.
- If applicable, details of any profile re-work needed based on the trials undertaken.
- If applicable, the final profile(s) chosen for implementation, in the form of Figure 9 or in digital format.
- Details of any implementation.

The easiest, and most efficient, way to produce this document is to begin it at the start of the exercise and add to it as and when work is done. In this way it is a living document, added to as needed and while work is fresh in the mind.

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A APPENDIX A – IN-SERVICE ASSESSMENT OF WHEEL AND RAIL PROFILES

There are two basic ways of assessing wheel and rail profiles in service; directly by measurement of profiles, and indirectly by visual inspection of wheel and rail surface condition. The first of these (measurement) is described in Section A.1, the second (visual inspection) is described in Section A.2.

A.1 Profile measurement

Profile measurement can be broken down into three levels of measurement of increasing sophistication: manual with gauges, manual with digital profile measuring machines, automated using such as laser-based systems and undertaken usually at speed.

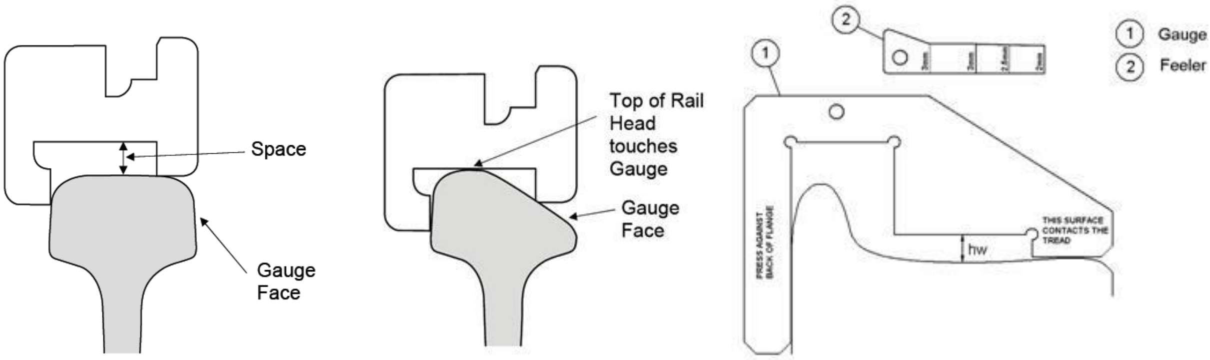
For any wheel or rail profile measurement it is important that the measured result has an appropriate reference. For example:

- Wheel profile measurements should be aligned with the vertical and lateral axes of the wheelset; and
- Rail profile measurements should be aligned with the track axis and referenced to track gauge.

A.1.1 Manual measurement with gauges

In this, the most common form of wheel and rail profile measurement, profiles are assessed using simple steel (or aluminium) gauges applied by hand to the wheel or rail. These gauges can be common within the railway industry, or may be developed by each rail network or operator to suit its own specific requirements. They frequently give a “go/no-go” profile measurement, but can be arranged to give an absolute profile measurement to enable trending.

Figure A1 illustrates two gauges in use in Australia. Figure A1a is an example of a go/no go gauge used to decide when wear on the rail gauge face is approaching condemning limits. Figure A1b is an example of a gauge that can be used for partial trending of hollow wear. Other wheel gauges are given in Reference 14.



a) Checking rail head width (Reference 15)

b) Checking wheel hollow wear (Reference 14)

Figure A1: Examples of rail profile and wheel profile gauges

The key function of a go/no go gauge is to help ensure continued safe operation. For example, the gauge of Figure A1a protects the structural integrity of the rail and also protects against loss of track gauge through side wear. The gauge of Figure A1b protects against unstable vehicle dynamics that can lead to overly high lateral force (L in equation 1), and also protects against derailment at turnouts where large values of hollow can cause the wheel to split the switch and stock rails in trailing turnout movements, or could cause a hollow worn wheel in flange contact to climb onto the end of a switch blade at facing turnouts.

The key requirements of a go/no go gauge are firstly that it represents a limit that should not be passed in routine operation and secondly that it has an associated action that needs to be taken if the limit is passed. For example, the rail head width gauge in Figure A1a defines a limit on rail side wear, and the associated action is that a rail failing the gauge is noted as being in unsatisfactory condition – with a requirement for more accurate measurement using callipers or a profile meter.

The limits and associated actions of go/no go gauges tend to be well-established but conservative, based on historic use and engineering judgement, and have been subsequently backed up by analysis.

A.1.2 Manual digital profile measurement

Manual profile measurement can be undertaken using commercial measuring equipment such as MiniProf (Reference 24) and WheelMate (Reference 25) that give profiles in digital form. Figure A2 shows examples of profiles manually measured using MiniProf equipment. Such profiles can be analysed to give the same information as go/no go gauges, and so can be used in their place. They can also be used to determine how profiles change over time (trending).

However, manual digital profile measurement is relatively time-consuming, requires sensitive and expensive measuring equipment, and requires skilled operators. For these reasons manual digital measurement tends to be used less for routine profile assessment and more for specific cases where accurate profile measurements are needed – for example in derailment investigations, or rail grinding trials, or where accurate in-service profiles are needed for vehicle dynamic simulations.

Figure A2: Illustration of manually measured wheel profile (red) and rail profile (blue)(x-axis and y-axis dimensions are in mm)

A.1.3 Automated profile measurement

Several suppliers are now able to offer systems for the automated measurement of wheel and rail profiles using laser-based systems. Rail measurement systems were the first to be developed. These typically use laser/camera systems on platforms attached to the underside of vehicles, with subsequent analysis to show the measured profile overlaid on the design profile.

Profiles can be taken at almost any interval along the track. An example of measurement and output is shown in Figure A3. Similar track- based systems are now available for the measurement and analysis of wheel profiles; usually one profile is taken per wheel. To make best use of these automated systems, which produce large amounts of data, vehicles need to be equipped with automatic equipment identification tags.

For both wheels and rails these automated systems allow rapid measurement of profiles and equally rapid analysis to give wear parameters such as flange height and thickness, rail head wear, and rail side wear. They are therefore a direct substitute for go/no go gauges, with the benefits of better resolution, automation of record-keeping, improved safety from reduced man/train interaction, and reduction in manpower.

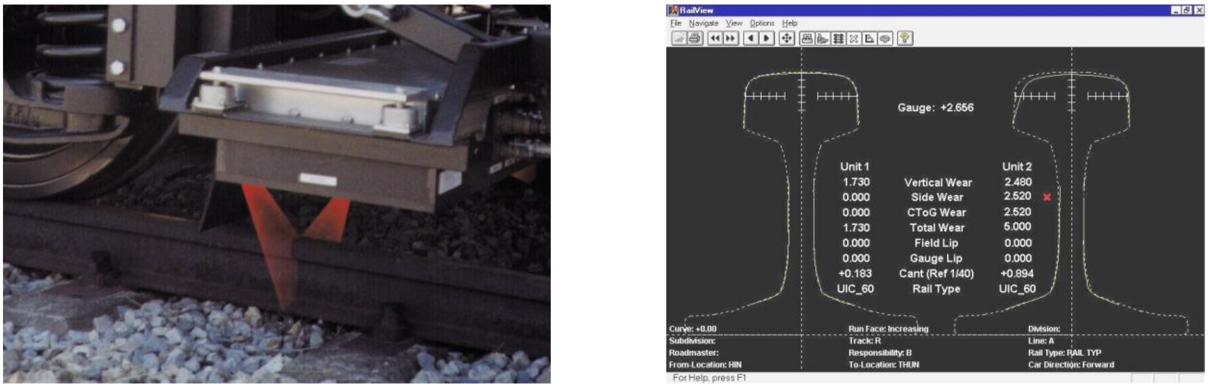


Figure A3: Illustration of automated profile measurement and output (pictures courtesy of KLD Labs, New York)

Because of the resolution achievable these automated measurements also enable trending of profile data such as wear. Such trending gives many benefits including:

- Detection of emerging profile problems, such as hollow wear.
- Identification of lubrication inefficiency through accelerated wear rates.
- Prediction of rail renewal and wheel turning or replacement.

An example of this type of trending is shown in Figure A4 which shows trending data from a North American freight railway. In this figure the goal is to predict when rail renewal, and hence capital expenditure, is needed.

Figure A4: Example of trending of wear data

The automated measurement of wheel and rail profiles raises the possibility of near-real-time programmed calculation of wheel/rail interaction to identify where wheel or rail profiles require remediation. For example, considering ride quality (Section 2.2) automated calculation of conicity using current wheel and rail profiles would indicate those vehicles at risk of giving poor ride quality, or conversely identify stretches of track likely to promote poor ride quality in the vehicles passing over.

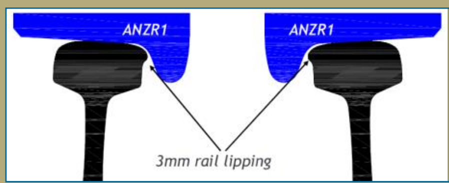
Automated wheel/rail interaction analysis is an emerging technology but is undergoing trial in North America (Reference 26). The principles of the scheme are:

- The technique is applied to essentially captive vehicles on a given line.
- Wheels from representative wheelsets are measured (using such as MiniProf or WheelMate machines), to give a number (N) of pairs of digital wheel profiles.
- These N wheel pairs are processed to represent N wheelsets, using the measured back-to-back spacings. Since wheelsets can be oriented in two way, this leads to 2N wheelsets that are then stored electronically.
- When a pair of rails is profiled by an automated (laser-based) system, these are also processed into a rail pair – including measured track gauge.

Case Study:

Incorrect positioning of ground rail profile:

Some rails had worn such that material flow from the rail head had caused significant 'lipping' on the gauge face of the rails.



The gauge face of the rail was used as a reference point for subsequent grinding of the rails to a new profile. The rail lip was incorrectly identified as the gauge face point resulting in the new rail profile being incorrectly positioned on the rails such the effective gauge was tight, with little flangeway clearance for the wheels.

The lesson learned is that it might be necessary to check track gauge at a position deeper than the usual 16mm below top of rail to ensure the gauge is being referenced correctly.

- The software then takes each of the 2N wheelsets in turn, matches it to the processed rail pair, shifts the wheelset laterally left and right, and produces the information described in Section 0; namely contact stress, flange angle, conicity. Other parameters, such as the extent of two-point contact between wheel and rail, can also be calculated.

- The software then compares these calculated results with limits and identifies where limits are exceeded.

The output either gives assurance that wheel/rail interaction between the reference wheelsets and the measured rails is acceptable, or identifies what profile changes are needed. Current use of this technology is to identify the most appropriate rail profile to be applied by grinding.

A.2 Visual inspection

A.2.1 Rail inspection

A subjective assessment of how wheel and rail profiles interact on a system can be gained by visual examination of the wheel and rail surfaces, and at turnouts and other special trackwork. The width of the contact band on the rail, and its position across the rail head, provides information on the wheel/rail contact patch.

Figure A5: RCF on the gauge corner of the outer rail in a curve

Figure A5 shows the outer rail in a curve with angled cracks on the gauge corner. These cracks are typical of RCF that tends to form when contact stresses are high (that is, when the wheel and rail profiles are non-conformal, Section 2.4) and when in-plane curving forces are high (Section 2.3). Such damage might indicate wheel and rail profiles that need to be made more conformal, or possibly – as an alternative approach – that the rail gauge corner needs to be relieved by grinding to reduce contact at the gauge corner.

Figure A6 shows the inner rail on a curve, with flow and deformation at the field side. This is symptomatic of the passage of wheels with hollow tread, where the wheel flange at the end-of-tread bears heavily on the rail, or where insufficient side relief has been applied in grinding. Hollow tread wheels also tend to give higher lateral force which, taken with the field-side contact, can cause the rail to roll outward if rail fastenings are weak.

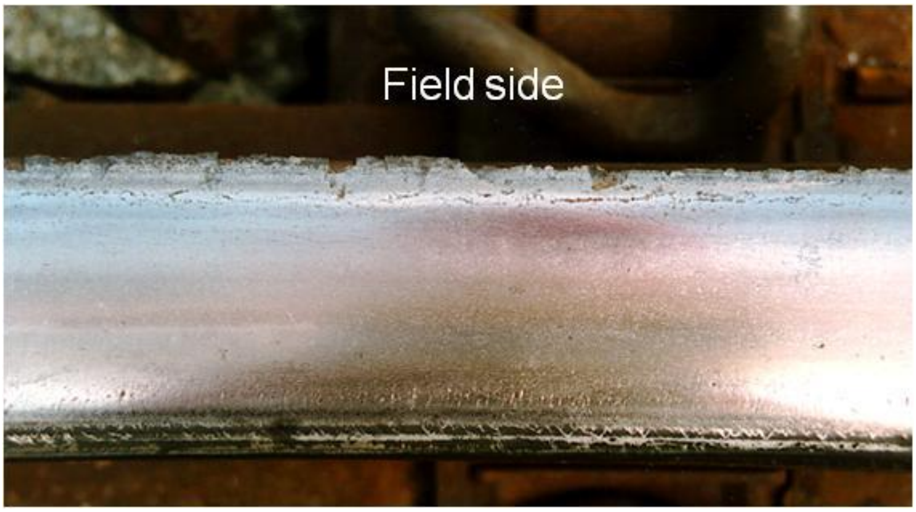


Figure A6: Metal flow and crushing on the field side of the inner rail in a curve

Figure A7 shows heavy deformation at a crossing where the wheel transfers from the wing rail to the nose, and vice-versa. This type of deformation is also an indication of wheels with hollow tread. Wing rail damage occurs when the wheel false flange bears directly on the wing rail as the wheel transfers to/from the nose. The nose damage tends to occur in facing moves when the wheel, supported on the wing rail by the false flange, suddenly drops onto the nose. Gouges on a stock rail where the wheel transfers from the stock to the switch rail are also an indication of hollow tread wheels. In this case the gouge occurs when the false flange passes over stock rail.

Figure A7: Heavy deformation at crossing nose and wing rail

A.2.2 Wheel inspection

As one example of damage in wheels, Figure A8 shows a wheel with various types of damage on the tread, identified by the letters A to D, where A is at the outer edge of the tread and D is in the flange root area of the wheel.

A: The shiny band at the outer edge of the tread (A) is caused by a misaligned brake block rubbing against the wheel. It indicates that the rail is unable to contact the wheel in this area, and the rubbing could eventually lead to a groove in the wheel. (With correctly-aligned brake blocks, this part of the wheel is likely to show surface corrosion indicating that it is not in contact with the rail.) As the wheel and rail wears to a more conformal shape this band will likely reduce in width. Its existence may indicate that the wheel and rail profiles are not best matched. However, it depends on whether gauge widening is applied in curves and on the wheel width used. Generally, 140mm wide wheels do not contact the rail in this area.

B: The angled cracks at B are typical of RCF and form when the wheel is on the inner rail in a curve. They indicate that curving forces are high (Section 2.3). Forces can be reduced by top of rail friction modifier and by modifications to bogie suspension stiffness, but wheel/rail profile re-design can also be beneficial. The cracks at B tend to move to a more transverse orientation as they approach the tread centre (C).

C: RCF is still the primary cause of the cracks at the centre of the tread C, but the change in orientation may be caused by in-plane forces from traction and braking or from thermal stresses caused by the brake block (if tread braking is used). Note that large cracks in this area, with associated deep shells, can indicate thermal fatigue from high heat input during tread braking.

D: The short cracks and spalls in the flange root area at area D are also a consequence of RCF; but in this case the damage occurs when the wheel is on the outer rail in a curve. As with the cracks at B, forces can be reduced by improved lubrication, but it is once again possible that modifications to wheel or rail profiles will be beneficial.

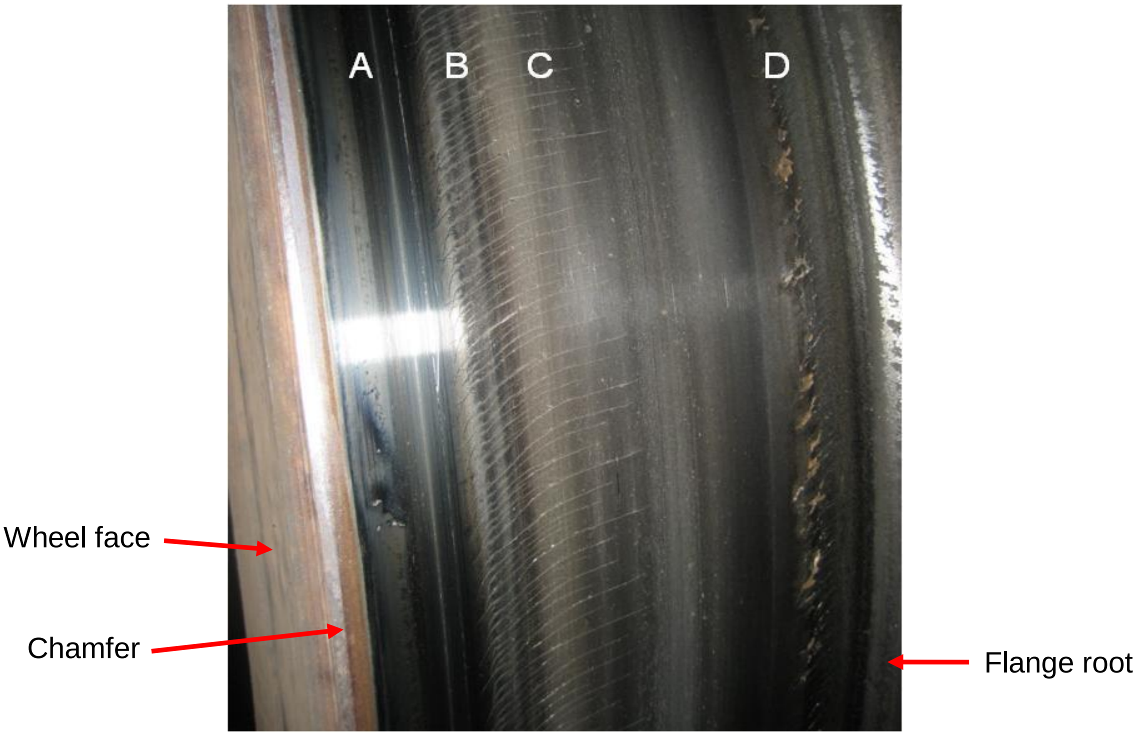


Figure A8: Variety of damage on a wheel

The Wheel Defects Code (Reference 14) provides more information on wheel inspection and defects.

B APPENDIX B – CHECK LIST OF TECHNICAL ISSUES

As an aid, this Appendix provides a summary list of technical issues to consider in development of a new wheel or rail profile. These are listed in alphabetical order, since the priority of each issue may depend on the application being considered.

This list is for guidance only and may not be comprehensive. Space is left for users to add their own issues.

Technical issues	For user
Adhesion, for traction and braking	
AS standards AS1085.1, AS7508, AS7509, AS7514	
Collateral effects – new wheel profiles on existing rail profiles or other system rail profiles	
Collateral effects – New rail profiles under existing wheel profiles or other operators' wheel profiles	
Conicity	
Contact, conformal or non-conformal	
Contact patch, position, size and shape	
Contact stress	
Curving performance	
Derailment safety, wheel L/V or wheelset L/V	
Ease of application, rail profile – material to be removed	
Ease of application, wheel profile – flange height and thickness	
Flange angle	
Flange height	
Flange thickness	
Flangeway clearance	
Friction modifiers	
Gauge, effect of gauge variations	
Gauge corner contact/relief, rail and wheel.	
Implementation	
Lubrication, of wheel flange or rail side.	
Lubrication, migration of lubricant with new wheel or rail profile	
Material and material properties, rails	
Material and material properties, wheels	
Rail profile, straight track, inner and outer rails of curves, large and small radius curves	

Rail profile, turnouts, whether to change rail profile in turnouts and other special trackwork, locking of turnouts

Rail profile, transition from inclined rails in plain line to vertical rails at turnouts and other special trackwork

Rail wear

Ride quality

Rolling contact fatigue (RCF), rails

Rolling contact fatigue (RCF), wheels

Rolling resistance (energy dissipated in wheel/rail contact patch)

Safety

Stability (i.e. resistance to hunting)

Track circuit operation (contact position, change of position, size)

Testing or trials of proposed wheel and/or rail profiles

Tolerances and variations - gauge, profile wear, longitudinal train forces, curving speeds, bogie design

Uni-directional rolling stock

Vehicle dynamic simulation

Wear of wheels and rails due to traction and braking

Wear limits, and inspection of wear, rail.

Wear limits, and inspection of wear, wheel.

Wheel wear

Other *(user to add)*

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