Vehicle Dynamics and the Wheel/Rail Interface

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1. INTRODUCTION

The formation of Rolling Contact Fatigue (RCF) in rails is due to the combination of contact stress, tangential creep forces and creepage in the wheel/rail contact patch. Most of these parameters cannot be measured directly with current technology. However, the science of railway vehicle dynamics allows us to predict with confidence what the values of these parameters are for a wide range of different conditions. This gives a valuable insight into the influence of the many different factors that affect the incidence of RCF.

Most RCF damage in the UK is found in one of two areas, either on the high rail in curved track, or in switches and crossings. Vehicle dynamic simulations have concentrated mainly on the curved track situation, as the varying curvature and rapidly varying rail profiles at switch and crossing locations multiplies the number of variables to be considered.

Most of the results presented in this paper are from quasi-static simulations, which assume a constant curvature without geometric imperfections. The effect of the dynamic response to track irregularities is also discussed towards the end of the paper.

1.1 Introduction to Curving Behaviour

Railway vehicles operating in the UK use wheelsets comprising two wheels fixed to a common axle. Wheels tend to roll in the direction in which they are facing. In a curve the leading wheelset will tend to roll towards the outside of the curve, and the trailing wheelset will tend to roll towards the inside. as shown in Figure 1.

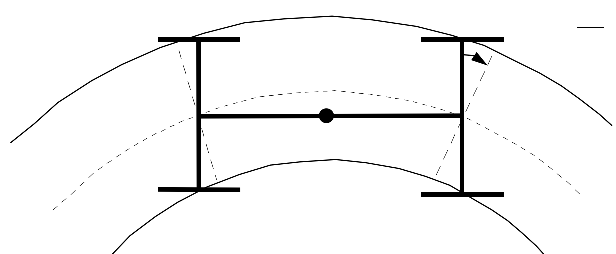

Figure 1 Vehicle on a curve

Because of the coning of the wheels, as the leading wheelset moves outwards, the radius of the outer wheel becomes greater than the inner wheel, as shown in Figure 2. As both wheels are rotating at the same speed, the larger radius wheel tries to roll further than the smaller radius wheel, thus steering the wheelset towards a radial alignment, when it will roll smoothly around the curve. The opposite process happens on the trailing wheelset as it moves inwards on the curve.

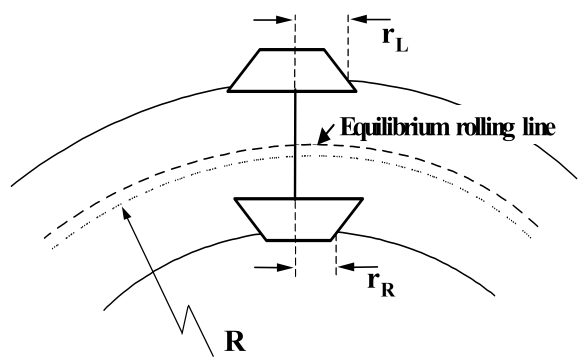

Figure 2 Rolling radius difference

The outer rail on the curve is longer than the inner rail, so that unconstrained wheelsets can curve freely by running along the equilibrium rolling line, where the rolling radius difference balances the difference in the lengths of the rails, shown again in Figure 2.

In practice, rotation of the wheelsets into radial alignment is resisted by the vehicle suspension. The stiffer the primary yaw suspension, the larger the forces which will be required to achieve the required rotation. These forces are generated by the leading wheelset moving out beyond the equilibrium rolling line to give an excess of rolling radius difference that gives rise to creepage (or microslip), and consequently creep force, to steer the wheelset relative to the rail. Similarly, the required steering forces at the trailing wheelset are generated by moving inwards from the equilibrium rolling line.

If the curve radius is smaller, or the bogie wheelbase is greater, the wheelset must rotate through a greater angle. Thus, larger steering forces must be generated, so the wheelsets must move further from the equilibrium rolling line. The forces that can be generated depend on the "effective conicity" of the wheelset on the rail. The larger the conicity, the greater the rolling radius difference for a given lateral shift. Conicity tends to increase with increasing wheel tread wear.

The steering forces are ultimately limited by one of two mechanisms. The first limit is the available adhesion. The second limit is the flange, which limits the lateral shift of the wheelset, preventing the wheelset from generating sufficient rolling radius difference.

Once the wheelset is unable to generate sufficient longitudinal forces to steer into the radial position, the wheelset will have an "angle of attack" to the track, and will run in flange contact. Because of the angle of attack, both of the tread contact points will be generating forces to push the wheelset into the flange, which must be resisted by the flange contact force. These forces are a major cause of wear.

As the equilibrium rolling line is closer to the outer rail than the inner, the leading wheelset will always reach flange contact before the trailing wheelset. Also, the lateral movements of the wheelsets tend to yaw the whole bogie or vehicle relative to the track, which increases the rotation required to achieve radial alignment at the leading end, and reduces it at the trailing. These factors ensure that the curving forces are always larger at the leading than at the trailing wheelset.

The curving diagram for a Mark 3 passenger coach on a 1000m radius curve, running at balancing speed (where the cant exactly balances the lateral acceleration) is shown in Figure 3. In this diagram the position and attitude of each wheelset is shown schematically relative to the flangeway clearance, and the size of the forces on the wheels indicated by the size of the arrows.

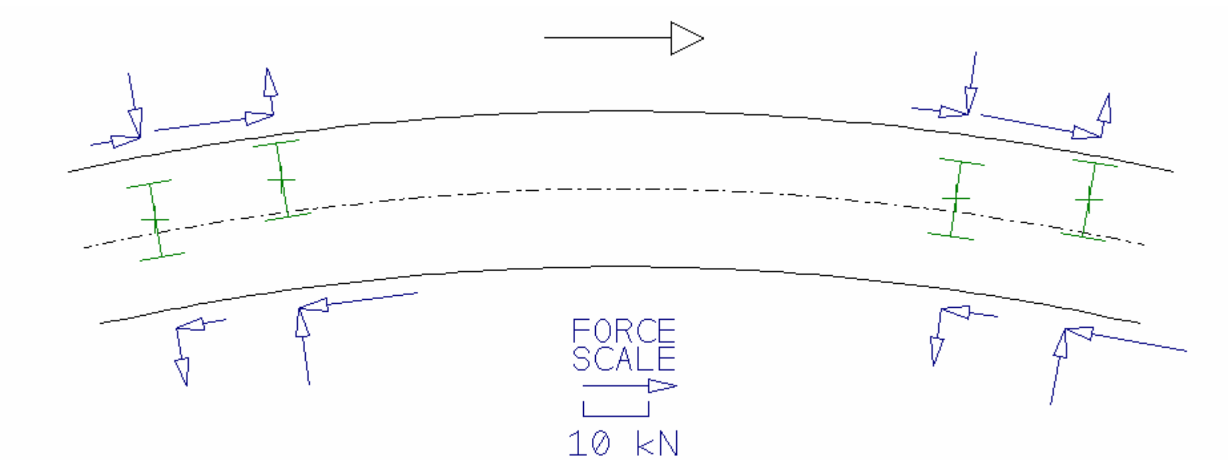

Figure 3 Curving diagram for Mark 3 coach on 1000m curve at balancing speed

In general, the resistance of the secondary suspension to bogie rotation is relatively low, so that the behaviour of the leading and trailing bogies is similar.

Curving at high speeds with cant deficiency means that the wheelsets must generate additional lateral forces to overcome the centrifugal forces in the curve. Where there is sufficient adhesion in larger radius curves, this can be achieved by the wheelsets "over-steering" to achieve a positive angle of attack. In practice, most of the lateral force tends to be carried by the trailing wheelset, which moves further outwards on the curve, improving the steering and reducing the forces on the leading wheelset.

2. COMPUTER MODELLING OF RAIL VEHICLE DYNAMICS

Using modern computer packages it is possible to carry out realistic simulations of the dynamic behaviour of railway vehicles running on real track. The theoretical basis of the mathematical modelling used is now mature and reliable and programs originally written by research institutes have been developed into powerful, validated and user-friendly packages.

Validation of the software and of the particular vehicle model is of great importance in ensuring that the results are truly representative of the real situation. Vampire, for example, has been extensively validated against a series of full scale experiments using load measuring wheels on locomotives, passenger and freight vehicles.

2.1 Modelling the vehicle

The first stage in setting up a computer model is to prepare a set of mathematical equations that represent the vehicle dynamics. These are called the equations of motion, and can be prepared automatically by the computer package through a user interface requiring the vehicle parameters to be described in graphical form, or by entering a set of co-ordinates describing all the important aspects of the suspension. The amount of detail used to prepare the model will vary according to the type of suspension and the required outcome of the modelling. Depending on the purpose of the simulation a wide range of outputs for example displacements, accelerations, forces at any point can be extracted.

The vehicle is represented by a network of bodies connected to each other by flexible elements. This is called a multibody system and the complexity of the system can be varied to suit the vehicle and the results required. The bodies are usually rigid but can be flexible with a given value of stiffness. Masses and moments of inertia need to be specified. Points on the bodies, or nodes, are defined as connection locations and dimensions are specified for these. Springs, dampers, links, joints, friction surfaces or wheel-rail contact elements can be selected from a library and connected between any of the nodes.

Track inputs to the model are usually made at each wheelset. Typical inputs are cross level, gauge and vertical and lateral alignment. These can be idealised discrete events such as dipped joints or switches or can be measured values from a real section of track. Additional forces may be specified such as wind loading or powered actuators.

There are several methods available to analyse the equations of motion. The first of these is an eigenvalue calculation on the matrices that represent the equations of motion. This will indicate the natural frequencies of the various modes of oscillation. The motion can be self-exciting and become unstable, and the eigenvalue analysis can indicate the critical speed at which this instability, or hunting, may occur.

Another alternative is a quasi-static analysis of the equations of motion for the particular case of a vehicle travelling on a curve of constant radius at a constant speed. This is known as 'steady state curving' and involves repeated evaluation of the equations of motion, while adjusting the position vector of the vehicle, until the forces are in equilibrium. The aim of this analysis is to predict the steady state attitude of all the bodies and the resulting wheel rail suspension forces.

The most powerful solution method is time stepping integration of the equations of motion. This is known as 'dynamic analysis' and describes the situation encountered when a vehicle enters a curve or transition or negotiates a series of curves of differing radii or a curve of changing radius. Switches or crossings or track defects can also be analysed. Nonlinearities which occur in the suspension, especially if displacements are large, and at the wheel-rail contact point due to creep and flange contact can be included using this method. Many numerical methods are available to perform the time stepping integration.

2.2 The track model

Additional elements are included below the rail to account for the track structure. This includes the vertical, lateral and roll stiffness and damping of the rails, pads, clips, sleepers and ballast.

2.3 Wheel rail contact

The computer packages all contain sophisticated routines for calculating the shape and size of the contact patch and the normal and tangential forces. Contact patches of varying shape and multiple contact points can be calculated as shown in figure 4.

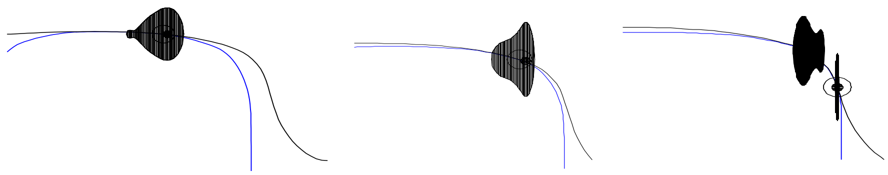

Figure 4 Contact patches [ref 2]

2.4 Computer packages

A number of vehicle dynamics analysis packages have been developed by research institutes and railway administrations around the world. Examples are: ADAMS/Rail, Vampire, Gensys, Nucars and Simpack. These have often grown out of the in house software tools that were developed to solve specific problems and are thus different in their operation and capability. Benchmarking of the main vehicle system dynamics packages has been carried out at MMU and is published in ‘Vehicle System Dynamics’ [ref 1]. In the post Hatfield work Vampire, ADAMS/Rail and Nucars have been used to carry out simulations of a range of vehicles running on new and worn wheels.

A typical ADAMS/Rail screen is shown in figure 5 and a typical Vampire screen in figure 6.

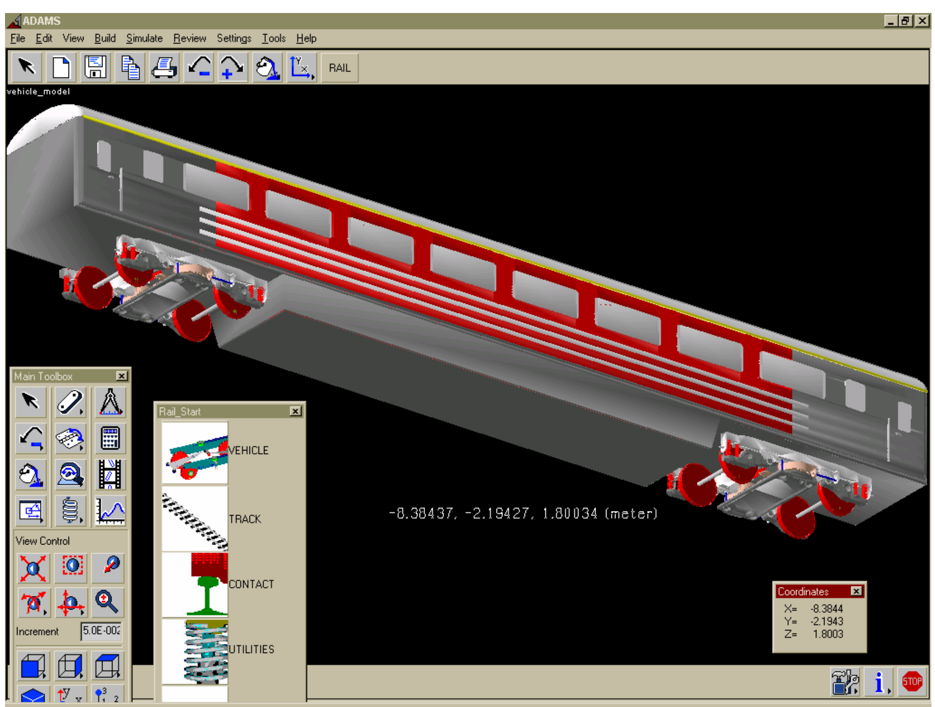

Figure 5 A typical ADAMS/Rail screen

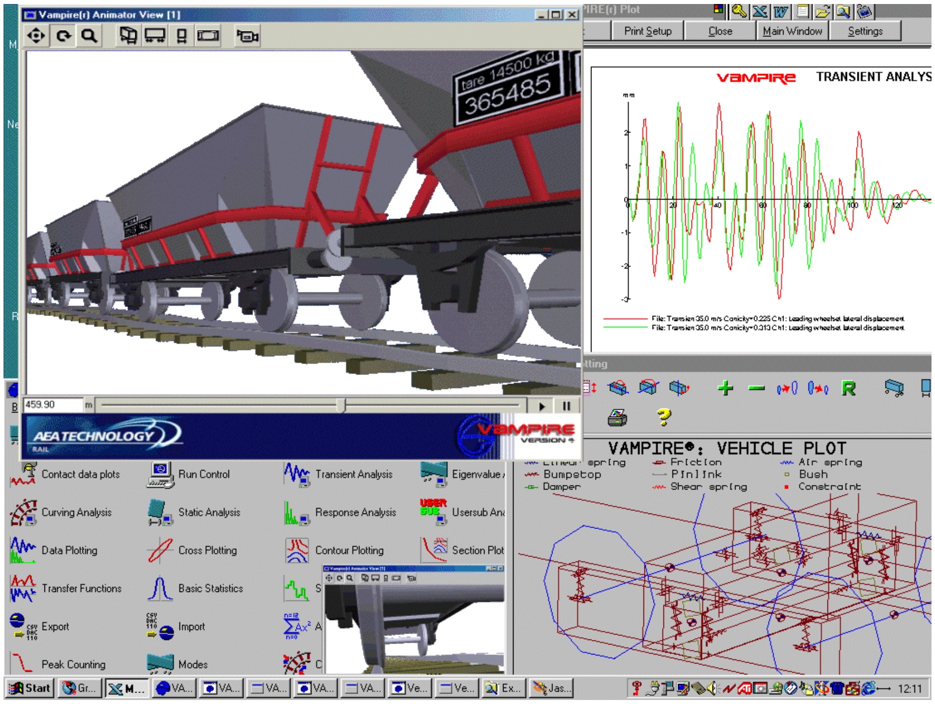

Figure 6 A typical Vampire screen

2.5 Example model – The class 43 HST Power Car

The primary suspension includes (per axle) 4 coil springs, 2 vertical dampers, 4 traction links with end bushes and 2 vertical bumpstops. The secondary suspension includes 4 coil springs, 1 traction centre equivalent bush, 2 vertical dampers, 2 lateral dampers, 2 yaw dampers, 2 lateral bumpstops and 2 vertical bumpstops. The wheel profiles were heavily worn as measured by TTCI on a typical vehicle using a miniprof device.

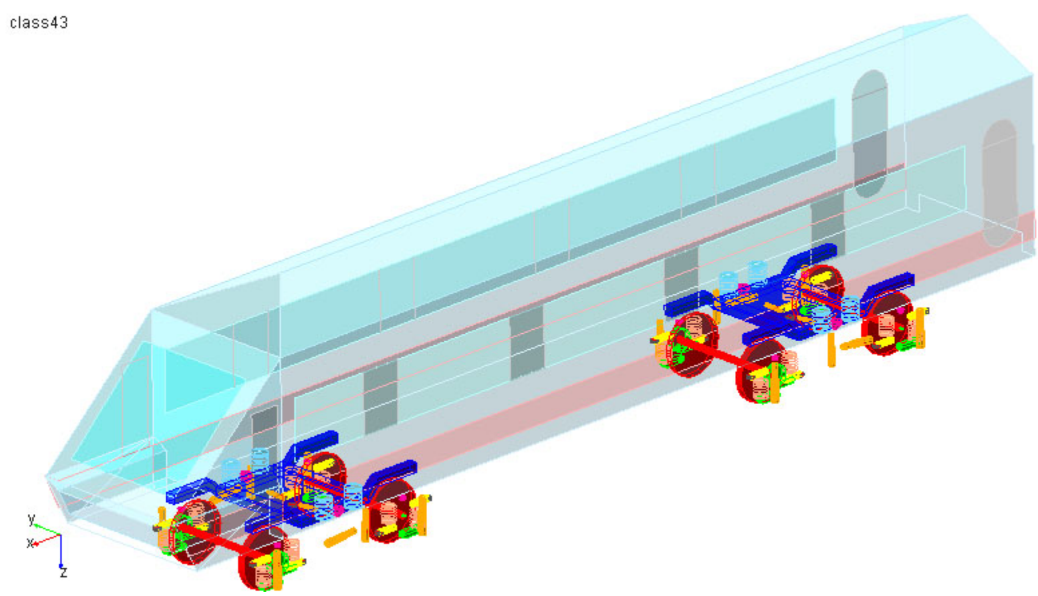

Figure 7 The class 43 model

2.6 Sample output

Figure 8 shows the simulation results for the contact patch position on the wheel and on the rail at the front left wheel from the ADAMS/Rail model of the class 43 power car as it runs into and around the curve at Aycliffe at 68.9 mph. This gives the exact position in lateral coordinate on both profiles during the simulation. In the transition, the wheel is oscillating from left to right on the rail, and then reaches a steady position in the curve.

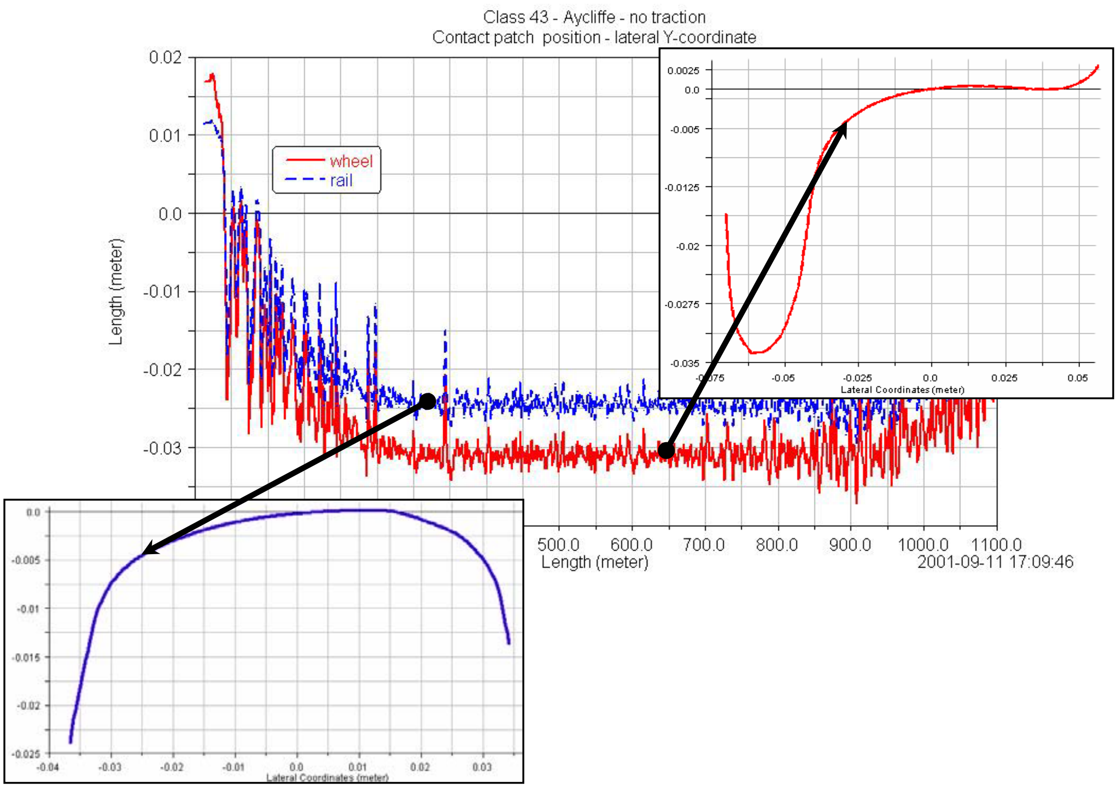

Figure 8 Lateral contact patch position

3. FACTORS AFFECTING ROLLING CONTACT FATIGUE IN CURVES

The development of rolling contact fatigue in rails depends on the interplay between crack growth, which is governed by the contact stress and the tangential force at the contact patch, and wear which depends on the tangential force (again) and the creepage at the contact patch. These parameters are dependent on a large number of inter-dependent factors, in particular…

- Curve Radius
- Vehicle Configuration – wheelbase, axleload, wheel diameter
- Suspension Design – in particular primary yaw stiffness
- Wheel Profiles – nominal profile and state of wear
- Rail Profiles – nominal profile and state of wear
- Wheel/rail Friction
- Cant Deficiency (depends on speed, radius and cant)
- Traction and Braking Forces
- Track Geometric Quality
- Wheel and rail material properties

The large number of variables makes the analysis of the big picture a massive undertaking. During the investigation of RCF for Railtrack, well over 2000 separate cases were simulated, and work is still continuing to fill in gaps in the jigsaw. The influence of these different factors is discussed in more detail in subsequent sections of this paper.

3.1 Initiation and Growth of RCF Cracks

The initiation and growth of RCF cracks depends on the contact stress and tangential force in the contact patch. These are shown on a shakedown diagram, by plotting the contact stress against traction coefficient (defined as the ratio of the tangential to normal forces). The tangential force is given by the vector sum of the longitudinal and lateral creep forces.

Figure 9 shows an example of a shakedown diagram for a 1500m radius curve. This shows the tread contact on the leading outer wheel for nine different vehicle types. Four cases are shown for each vehicle, with different combinations of wheel and rail profile. All the cases are calculated at 150mm cant deficiency, with a high coefficient of friction of 0.45.

The shakedown diagram also shows the shakedown limit for two different rail steels, grade 220 rail with a yield stress in shear of 213MN/m^2, and grade 350HT hardened rail with a yield stress of 449MN/m^2. The limits are based on results published by Bower and Johnson (Ref. 3). These limits are approximate, and represent a case with partial slip, point contact and longitudinal traction.

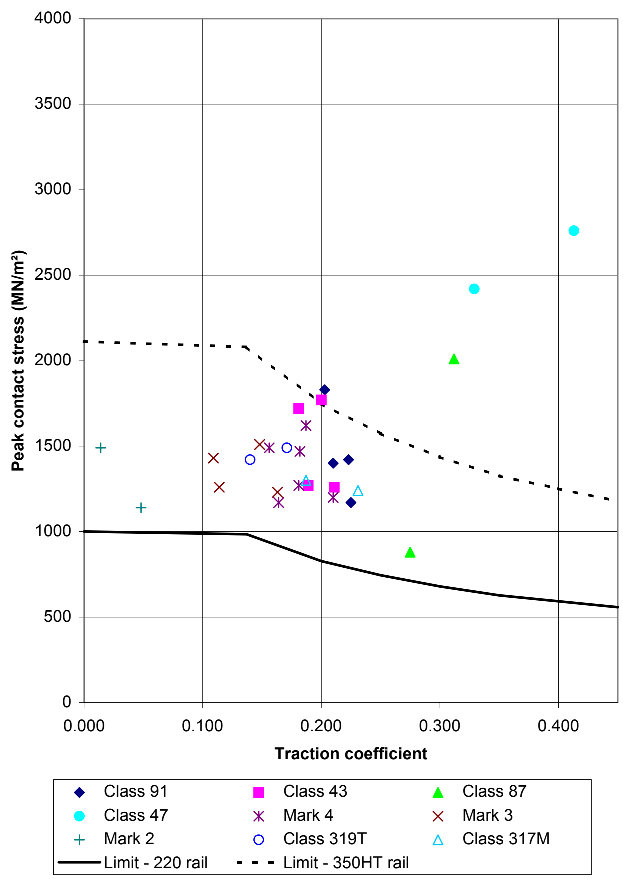

Figure 9 Shakedown diagram, 1500m radius

This diagram illustrates the range of conditions that can be obtained with different vehicles, with a selection of different wheel and rail profiles. It will be noted that every case is outside the shakedown limit for the standard rail. Some of the cases are outside the limit even for the hardened rail.

Thus, shakedown theory implies that cracking is inevitable on normal grade rails with any vehicle, even with perfectly smooth track, which is not consistent with real life experience. Furthermore, shakedown theory suggests that hardened rails should be far more resistant to cracking than normal grade rails, whereas experimental evidence (Ref. 4) indicates that in practice the opposite is true. These anomalies are due to influence of wear.

3.2 Wear of the Rail

Wear of the rail surface acts to prevent the development of RCF by wearing away the incipient cracks before they are able to grow.

Empirical studies both on a full-scale laboratory test rig and in the field (Ref.5) have shown that the wear of wheels and rails depends on the rate of dissipation of energy in the contact patch. This can be calculated from the tangential creep force multiplied by the creepage. This parameter is known as the wear number $T\gamma$.

The relationship between the wear number and the traction coefficient is shown in Figure 10. This was calculated using a Mark 4 coach for a range of radii and four different wheel and rail profile combinations at 150mm cant deficiency. Calculations were undertaken for three different friction coefficients, and a best fit line has been drawn for each case.

The relationship between the traction coefficient and wear is clearly non-linear, with the wear increasing rapidly as the traction coefficient approaches the limiting value represented by the friction coefficient.

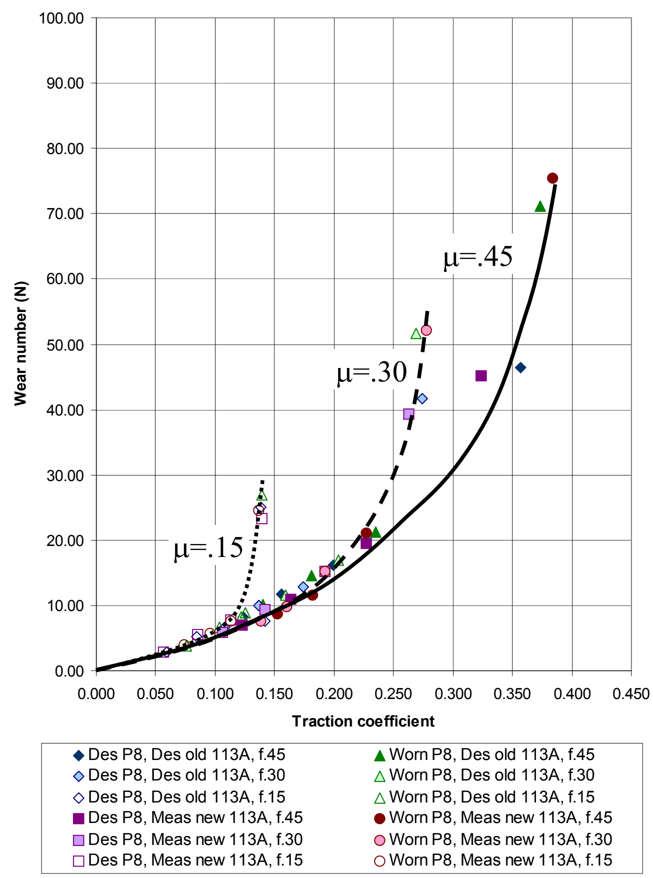

Figure 10 Relationship between wear and traction coefficient

3.3 Interpreting the Shakedown Diagram

The interplay between crack growth rates and wear means that predicting the risk of RCF development for a particular combination of contact stress and traction coefficient is not straightforward.

Cases that fall in the bottom left quarter of the shakedown diagram, below the shakedown limit, will be safe from RCF development. In principle, as conditions move upwards and towards the right the rates of crack initiation and growth will increase. However, as conditions move towards the right, the traction coefficient increases towards full slip and the wear rate increases rapidly. In these conditions, it is again believed that RCF is unlikely to develop. Thus there exists an area in the middle of the diagram where the greatest risk of RCF will be found.

Work continues to establish the boundary between the areas where crack growth dominates and the area where wear dominates. A tremendous amount of effort has been expended on vehicle dynamic simulation to establish the contact conditions prevailing with a wide range of vehicles in a wide range of conditions. The full value of this work can only be realised once we have a robust understanding of the relationship between crack growth and wear.

The case of hardened rail illustrates the significance of wear. Wear rates are much lower in hardened rail, and despite shakedown theory showing that hardened rail should have a lower risk of crack development, in practice it develops RCF more quickly than softer grades (Ref.4). This suggests that the wear mechanism has a very major role in the development of RCF.

3.4 The Influence of Curve Radius

Curve radius has a key influence on curving behaviour. Figure 11 shows the shakedown diagram for the same nine vehicle types and the same conditions as Figure 9, but calculated for a curve radius of 700m rather than 1500m. At this curve radius, many of the cases with lower conicity wheel/rail combinations run in flange contact. Where there are separate flange and tread contact points, the conditions at the point with the higher contact stress and traction coefficient are shown in the diagram. This is usually the flange contact point.

By comparison with the shakedown diagram for 1500m radius, all the points have moved to the right, indicating higher traction coefficients, and most have also moved upwards, indicating higher contact stress.

From shakedown theory, this condition would be considered substantially worse than the 1500m radius curve in Figure 9. However, in practice, RCF has been found to be significantly more prevalent at radii around 1500m, and less prevalent at radii around 700m. This would suggest that the conditions shown in Figure 9 are in fact more damaging than those shown in Figure 11. The likely explanation is that higher wear in the 700m radius case is more than enough to offset the increased shakedown risk.

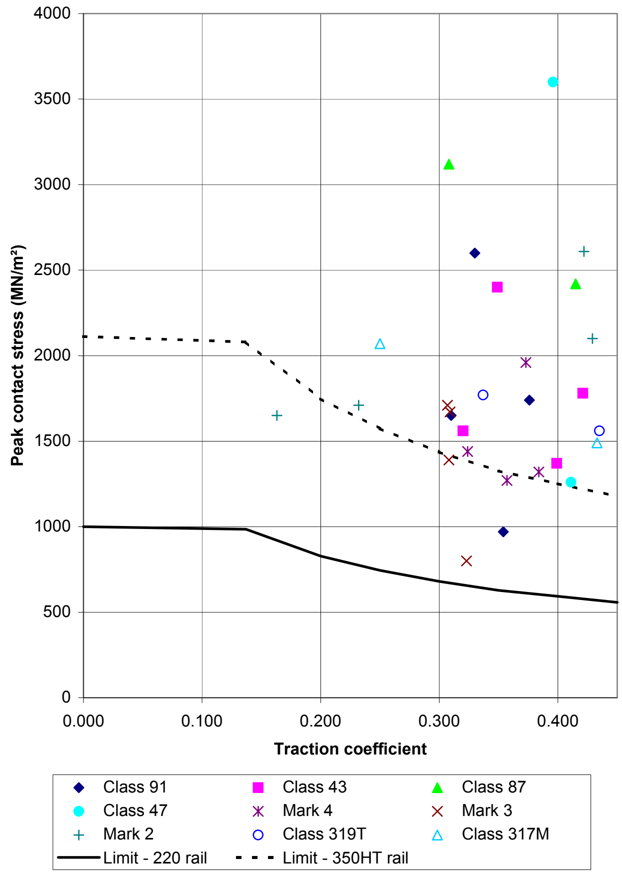

Figure 11 Shakedown Diagram, 700m radius

4. VEHICLE FACTORS AFFECTING ROLLING CONTACT FATIGUE

Two vehicle-related factors affect RCF. These are the configuration of the vehicle itself – in particular the wheelbase and the primary yaw suspension stiffness – and the wheel profile.

The wheel profile is not an independent variable, different vehicles use different wheel profiles for technical and economic reasons. For example, an experiment undertaken post-Hatfield to fit a P1 wheel profile on class 315 multiple units in place of the normal P8 has demonstrated that excessive flangewear is obtained with the older profile on a vehicle with a modern design of bogie.

4.1 Passenger Vehicles

Figure 12 shows a shakedown diagram for three different types of passenger vehicle in a 1500m radius curve at 150mm cant deficiency, 0.45 friction. Four wheel/rail profile combinations are shown for each vehicle. These three vehicles are representative of the three broad groupings into which passenger bogies can be split.

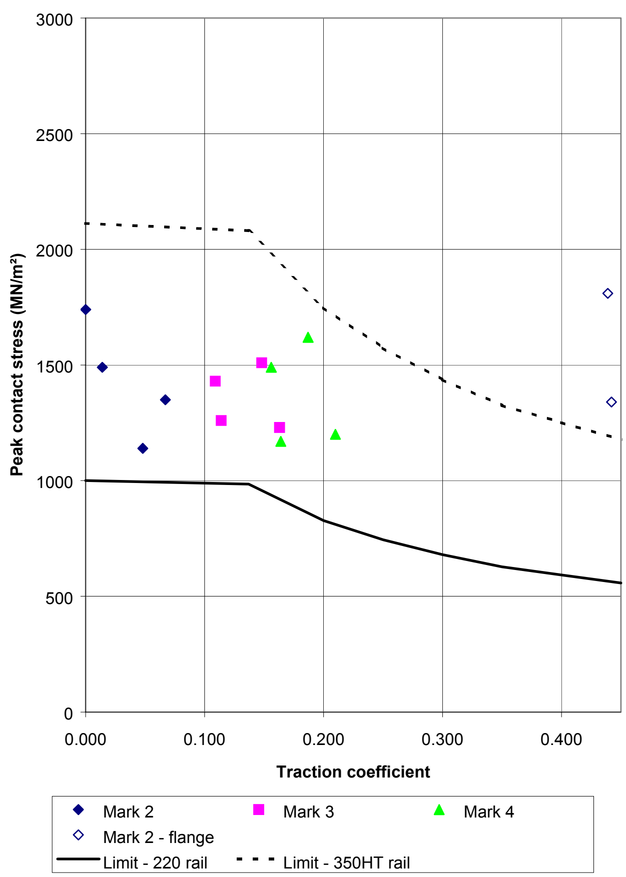

Figure 12 Shakedown Diagram - passenger coaches

The first vehicle is a Mark 2 coach with P11 wheel profiles. This vehicle has B4 bogies, and has a low primary yaw stiffness. Its behaviour is likely to be similar to other older designs of coaching or slam-door multiple unit stock.

The second vehicle is a Mark 3 coach, which has a more modern bogie design with P8 profiles. The BT10 bogie has a moderate primary yaw stiffness, sufficient to maintain stability up to 125 mile/hr, but still giving quite good curving behaviour. The behaviour of other passenger rolling stock with British-designed bogies, including much multiple unit stock, will be similar.

The third vehicle is a Mark 4 coach. This vehicle conforms to typical continental bogie design practice, with very high primary yaw stiffness. Other vehicles with similarly high primary yaw stiffness include Coradia and Juniper multiple units and Eurostar trains.

In the conditions shown, all three types of vehicle give similar contact stresses, but the traction coefficients vary in proportion to the primary yaw stiffness. Even on such a shallow radius curve, the Mark 2 coach with new wheel profiles is in flange contact, due to the low conicity of the P11 profile.

The Mark 2 coach probably has the lowest RCF risk, as the low traction coefficients at the tread contact are likely to be benign, while the very high traction coefficients at the flange contact could be expected to lead to wear. However, this type of bogie is not acceptable for speeds much in excess of 100mile/h, and the cost of maintaining the low conicity wheel profiles is relatively high.

The very stiff Mark 4 suspension will give higher crack growth and wear rates than the softer Mark 3 suspension. The effect on RCF will depend on whether the difference in wear outweighs the difference in crack growth, and is not readily quantifiable with the current level of knowledge.

4.2 Locomotives

Figure 13 shows a shakedown diagram for four different types of locomotive, for the same conditions as Figure 11. Either two or four wheel/rail combinations are shown.

Compared with the passenger coaches, the pattern of behaviour is less clear-cut, but some examples, particularly the older locomotives, can give very high contact stresses.

The class 47 is an old design of Co-Co locomotive, with P1 wheel profiles. The class 87 has a more modern bogie with P6 profiles. With design-case wheels both locomotives are in flange contact, giving quite low contact stresses on the tread, but with worn wheels, both locomotives give single-point contact with high contact stresses and traction coefficients.

The class 43 (HST Power Car) and class 91 are modern designs with P8 wheel profiles. The class 43 has a stiffer primary yaw suspension but a shorter wheelbase than the class 91, so that both have similar curving performance. They give slightly higher contact stresses and traction coefficients than the Mark 4 coach.

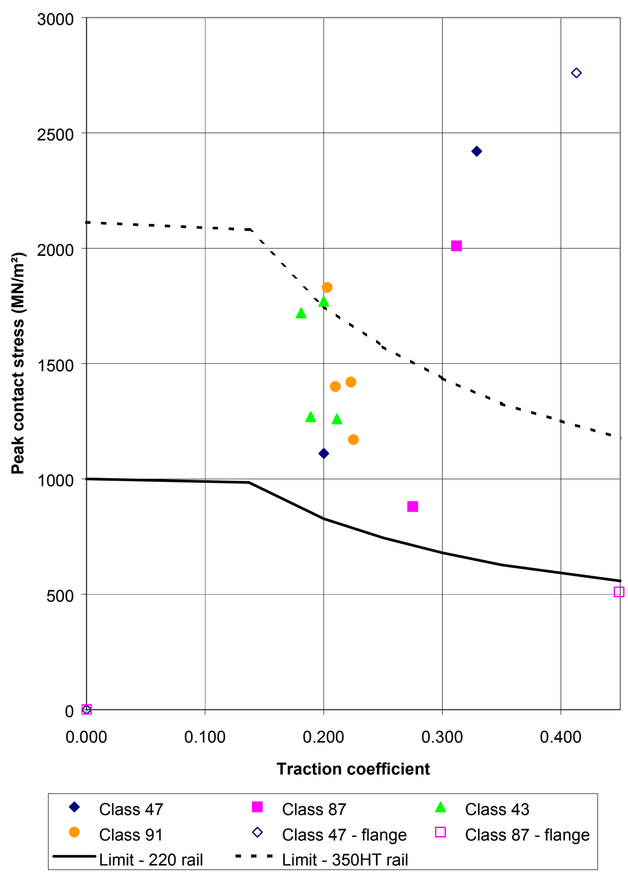

Figure 13 Shakedown Diagram - Locomotives

Locomotives are much less numerous than coaches and wagons, which may help to reduce their relative importance.

4.3 Freight vehicles

Freight vehicles are thought to be less damaging than passenger vehicles. This is based on the observation that on most UK four-track main lines, RCF damage is more common on Fast lines than Slow lines.

Figure 14 shows the shakedown diagram for a laden JRA 90t box wagon, with Y-series bogies with P10 wheel profiles. Again, four different wheel/rail combinations are shown, for a coefficient of friction of 0.45. The wagon is running at cant excess in this curve, as a consequence of its lower speed.

Simulations have also been carried out for a selection of other types of wagon, showing generally similar behaviour.

This wagon shows similar contact stresses to the passenger coaches, but higher traction coefficients. If it is correct that these wagons are less damaging than passenger coaches, then this would be as a result of the increased wear from the higher traction coefficients.

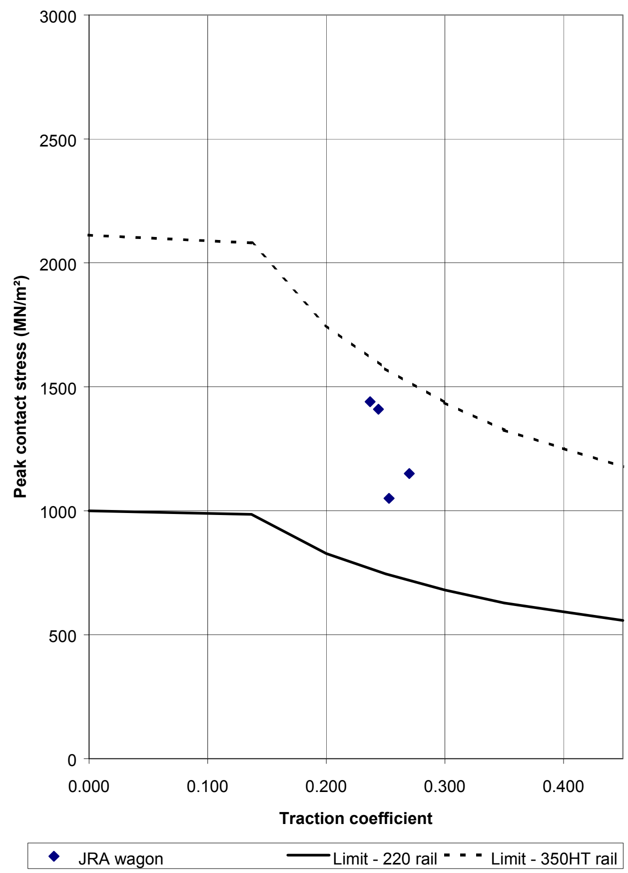

Figure 14 Shakedown Diagram - freight wagon

4.4 Traction and Braking

Traction and braking forces modify conditions in the contact patch. It is generally believed that braking forces are not a significant issue, as they act in the wrong direction to cause rail crack propagation. This is confirmed by the tendency for RCF to occur on the high rail in curves, where the steering forces are in the same direction as traction forces.

The class 91 locomotive is the most powerful high-speed locomotive in the UK. Representative traction forces for each speed have been added to the simulations for this locomotive with the results shown in Figure 15. This plot is with the 0.45 coefficient of friction, 150mm cant deficiency, and shows design-case and worn P8 profiles on the original design case BS113A rail.

The effect of the traction in every case is to increase the traction coefficient, but leaving the contact stress almost unchanged. The effect of this may increase the rate of crack growth, but will also increase wear rates, so that the net effect may be a reduction in the risk of RCF.

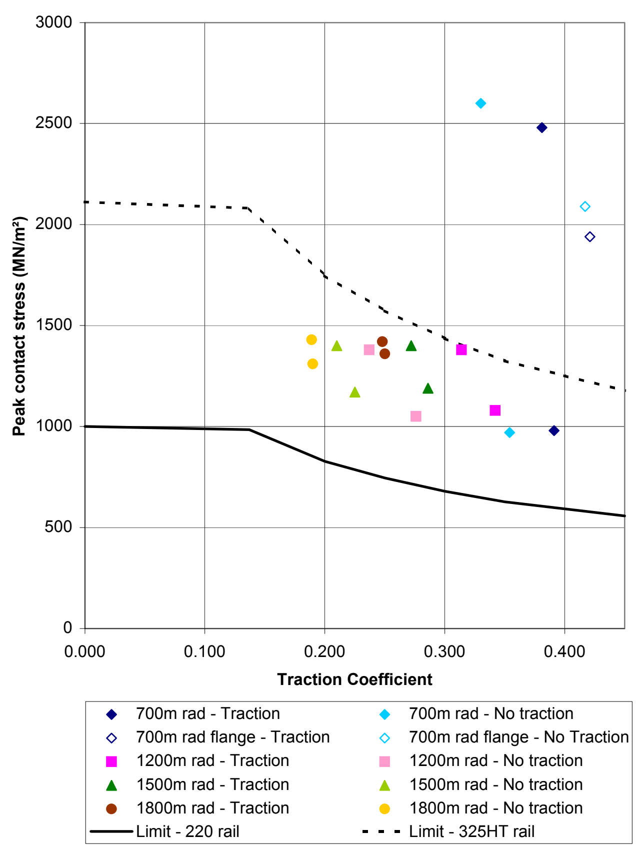

Figure 15 Shakedown diagram - effect of traction

5. WHEEL AND RAIL PROFILES

5.1 Design Wheel Profiles

Design case wheel profiles in the UK differ mainly in the effective conicity across the tread, and in the shape and thickness of the flange. The choice of wheel profile depends on the vehicle suspension and the type of routes on which it operates. Inappropriate wheel profiles may have severe technical or economic consequences.

The evidence suggests that lower conicity profiles are more benign from the point of view of RCF, but they can give severe flange wear if used in current designs of bogie. Conicity usually rises as a wheel wears, and maintaining a low conicity may require regular and costly re-profiling. The P8 profile is essentially no more than a part-worn P1, which was introduced to maintain a stable conicity range and reduce the amount of material that had to be removed at each re-profiling.

5.2 Wear of wheel profiles

The pattern of wear varies enormously from application to application. Poor curving vehicles on curvaceous routes will suffer mainly flange wear, whereas good curving vehicles on relatively straight routes will suffer mainly tread wear.

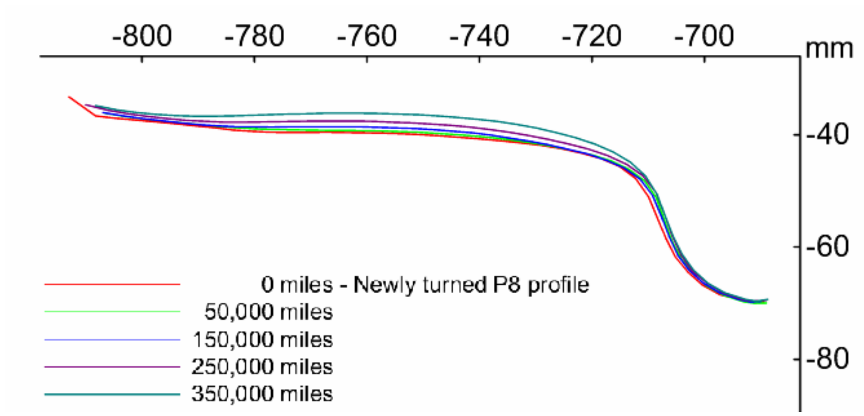

Figure 16 Measured P8 wheel profiles

Figure 16 shows a selection of wheel profiles measured in 1996 on Mark 4 coaches that had covered a range of different mileages since re-profiling. The wear can be seen to be predominantly on the tread, although some flange wear is also apparent.

Figure 17 shows the contact stress generated by these profiles when running on a design-case new BS113A rail. The simulations were run for five different curve radii, assuming 150mm cant deficiency and a friction coefficient of 0.45.

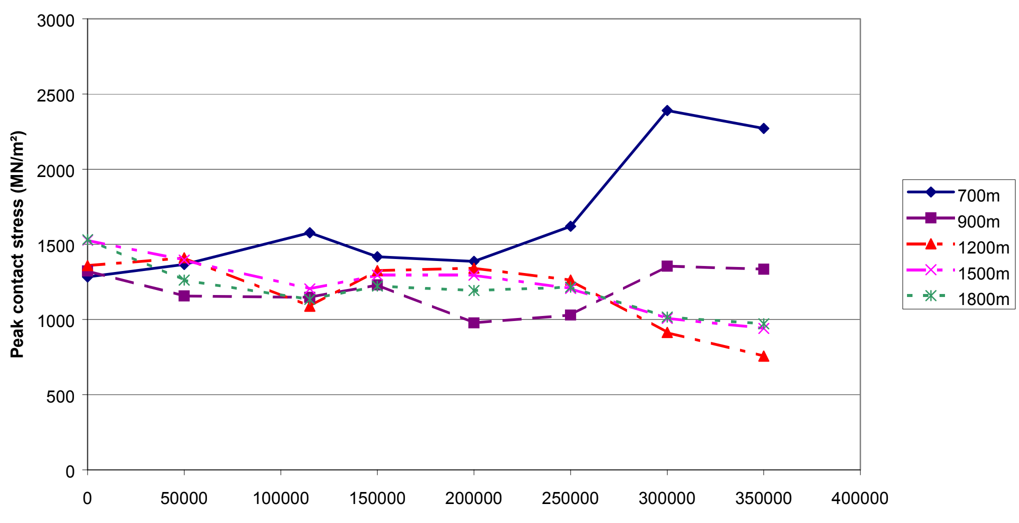

Figure 17 Variation of contact stress with wheel profile wear

At the larger curve radii, 1200m to 1800m, where RCF is most prevalent, it will be seen that the contact stress drops as the wear progresses. At tighter radii the contact stresses can start to rise quite rapidly, but these curves are less prone to RCF.

The results shown in Figure 17 are just a snapshot for one vehicle type in one set of conditions. Nevertheless, they demonstrate that wear of the wheel profile is not necessarily detrimental to the development of RCF, and indeed may be beneficial in some circumstances.

5.3 Design Case Rail Profiles

There have been two variations of the BS113A rail profile. The original BS113A profile tended to give excessive conicity, and hence poor vehicle ride, particularly when laid tight-to-gauge. Accordingly, a revised BS113A profile was introduced in 1988, with a higher crown, which tends to reduce conicity. The higher crown will also reduce the tendency to contact in the gauge corner area, and could be expected to reduce the incidence of RCF in the gauge corner.

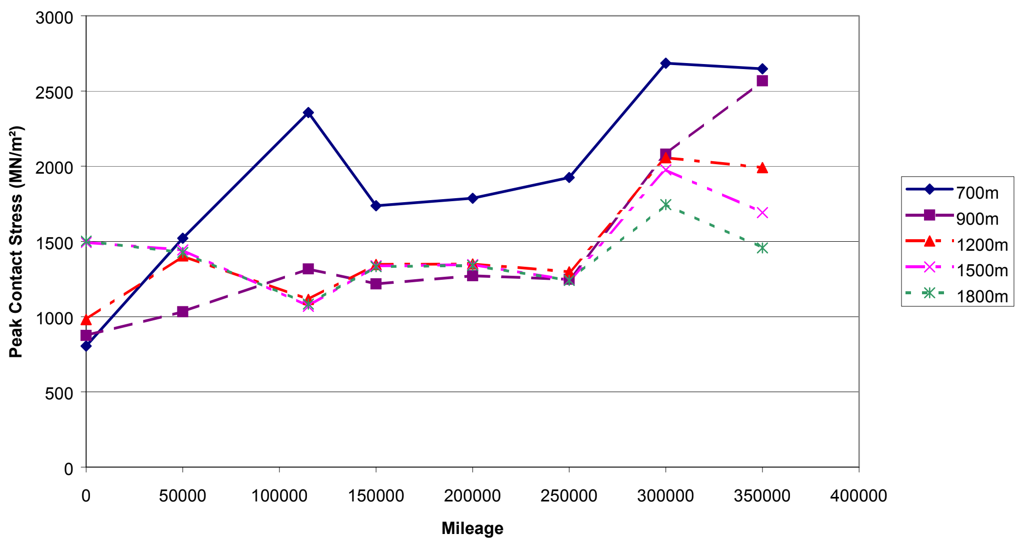

Figure 18 Variation of contact stress with wheel wear - old BS113A rail profile

By comparison with the high-crown BS113A shown in Figure 17, Figure 18 shows the same results for the original flatter BS113A rail. It can be seen that with the flatter rail head there is a generally increasing trend for contact stress with wheel wear, and on the 700m curve, high contact stresses are generated even with moderate wear. UIC60 rail currently being introduced in the UK has a similar rail head profile to the original BS113A rail, so the UIC60 rail may be more prone to RCF in the gauge corner than the current BS113A rail.

5.4 Wear of Rail Profiles

To investigate the effect of rail wear, simulations were undertaken for four different vehicle types on a 1500m radius curve, for a range of speeds, with and without traction. Four different rail profiles were used with different levels of wear as shown in Figure 19.

Figure 20 shows the resulting points plotted on a shakedown diagram. There is a very wide scatter in the results, and the state of

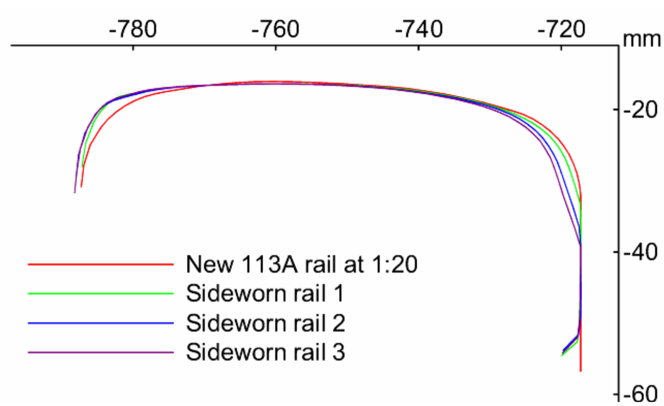

Figure 19 Worn rail profiles

wear of the rail has only a minor influence on the conditions. New rail tends to give slightly higher contact stresses than the worn rails, which all give broadly similar behaviour.

5.5 Grinding profiles

Railtrack have implemented a programme of rail grinding in curves as a control measure to mitigate RCF.

The target grinding profile is based on the standard BS113A rail-head profile with a slightly increased inclination of 3.4° rather than the normal 2.86° (1:20). This is intended to give relief of the gauge shoulder. The tolerances for the grinding process are +0mm/-0.6mm in the gauge shoulder to ensure that the as-ground profile has more gauge-shoulder relief than the target.

In terms of vehicle dynamic behaviour, the effect of relieving the gauge shoulder on the rail is to reduce the ability of the wheelset to steer in the curve. This results in flange contact at larger radii than hitherto, which could lead to increased flange wear. The Arup/TTCI report (Ref.6) recommends that grinding of rails on curves for relief of RCF should be accompanied by lubrication, which would mitigate the increased wear.

In practice, contact on the gauge shoulder usually occurs in tighter radius curves, and simulations confirm that the grinding profile does tend to reduce contact stresses on these curves. However, on the larger radius curves, where RCF has been most prevalent, the benefit of the ground profile is not so clear cut.

Figure 21 shows the shakedown diagram for a variety of vehicles and a selection of wheel profiles, on a 1200m radius lubricated curve. For simplicity only the tread contact is shown.

There is a clear trend that contact stresses are slightly increased by going from the current BS113A profile to the target grinding profile, and further increased by going to the as-ground profile. Conversely, traction coefficients tend to decrease. This evidence suggests that on moderate radius curves, the grinding profile is not achieving the desired object of reducing contact stresses.

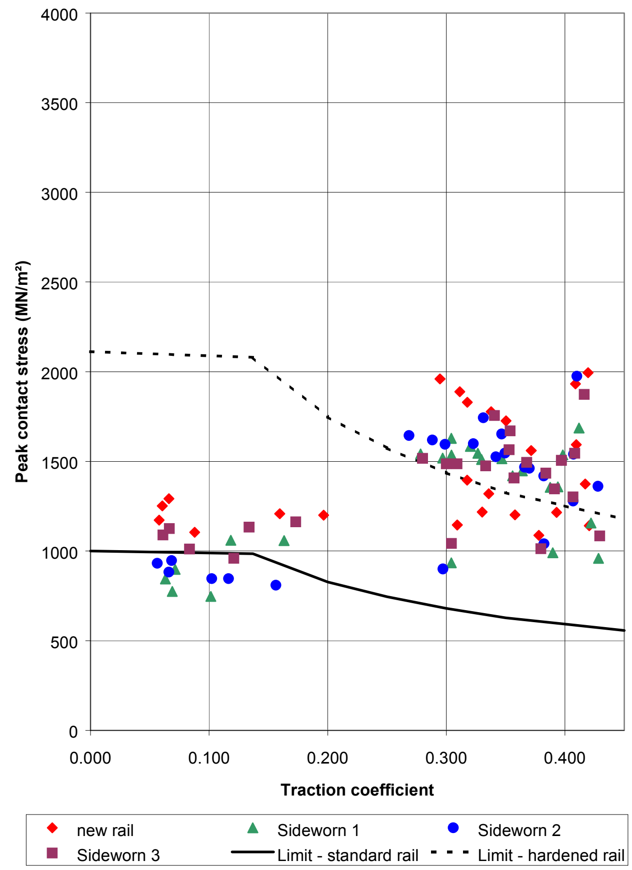

Figure 20 Shakedown Diagram - rail wear

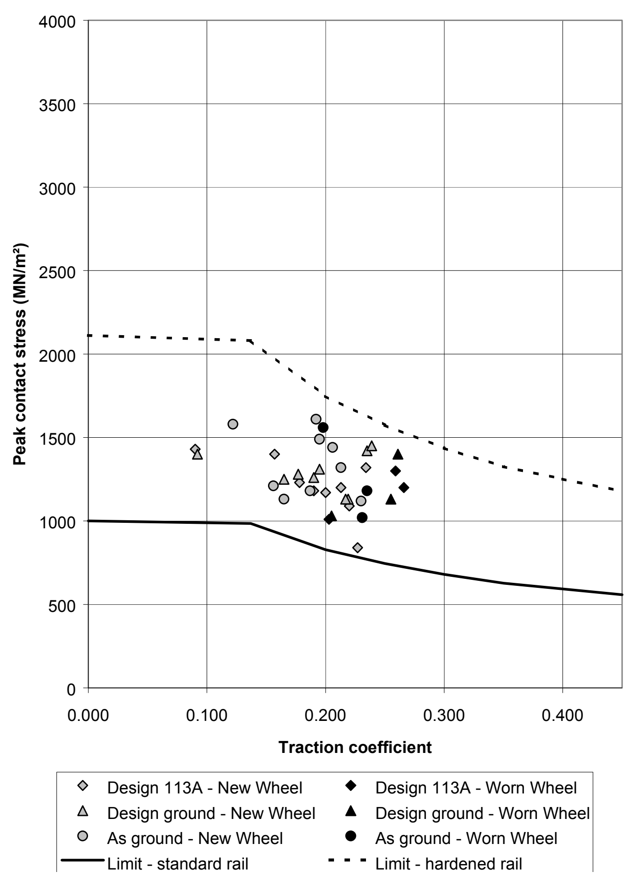

Figure 21 Shakedown diagram - ground rails

6. FRICTION AND LUBRICATION

Shakedown theory (Ref.3) indicates that partial slip at high friction is the worst case for crack initiation. Therefore, analysis has concentrated on a friction coefficient of 0.45, which corresponds to the highest friction level normally found in UK railway conditions. However, calculations have also been undertaken with a friction level of 0.15, characteristic of wet or damp rails (although lower friction may be found in the leaf fall season), and an intermediate value of 0.3.

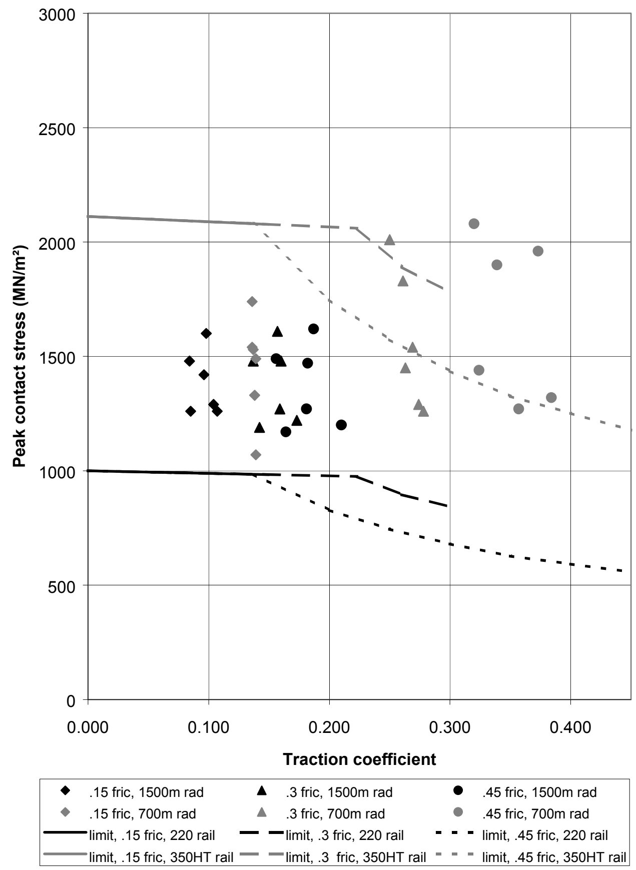

Figure 22 Shakedown Diagram – varying friction

Figure 22 shows the behaviour of a Mark 4 coach at maximum cant deficiency with these three friction levels at 1500m and 700m radii. For each condition, the behaviour with six different wheel/rail combinations is plotted.

At 1500m radius, the traction coefficient increases as the friction level increases, while the contact stresses remain similar. Even at the lowest friction level, the wheelset is still only in partial slip (full slip would imply a traction coefficient of 0.15 in this case).

At the tighter 700m radius, the wheels are much closer to full slip, and the traction coefficient changes much more markedly. Low friction is also affecting the steering behaviour, and hence wheelset position, so that the contact stress can also change – some cases show higher contact stress at higher friction.

The shakedown limit for 0.3 friction is higher than for 0.45 friction (the limiting line for 0.15 friction is hidden by the other two lines which extend further). Nevertheless even at the lower friction levels all of the points lie above the limit for conventional rail steel.

In principle, reducing the coefficient of friction should reduce the incidence of RCF. However, the only full-scale controlled trial of flange lubrication to reduce RCF (Ref.4) showed that it had no significant effect. It is also reported that a Japanese experiment with continuous water spraying resulted in premature rail failure due to rolling contact fatigue. This may be because the presence of fluid can promote crack propagation.

7. CANT DEFICIENCY

Increasing cant deficiency increases the vertical loads to be carried by the high-rail wheels, and reduces those on the low-rail wheels. It also requires greater lateral wheel/rail forces to be generated to resist the centrifugal forces. However, these forces are generated largely at the trailing wheelset of the bogie, where contact stresses and longitudinal steering forces are generally lower. At the leading wheelset, increasing cant deficiency tends to move the wheelset further towards flange contact, increasing contact stress with worn profiles, but reducing the tangential force.

Figure 23 shows the shakedown diagram for the leading outer wheel of a Mark 4 coach at cant deficiencies of 0mm, 110mm and 150mm, with curve radii of 1500m and 700m, and 0.45 friction coefficient.

There is a clear trend for each radius that as the cant deficiency increases, contact stress increases and the traction coefficient decreases. The change follows the slope of the limiting curves, which would suggest that increasing cant deficiency is neutral in terms of shakedown risk. However, the rate of wear will be reduced at higher cant deficiency, which could increase the overall incidence of RCF.

8. TRACK GAUGE

Simulations for the class 91 locomotive on a range of track gauges are shown in Figure 24. Two curve radii are shown, for 150mm cant deficiency and 0.3 friction coefficient.

The only significant effect of varying gauge is with the worn P8 profile on the 1500m radius curve, where the contact stress rises sharply at the tightest gauge. Although the stress is still lower than on the 700m radius curve, the 700m curve has a higher traction coefficient, and will therefore experience much higher wear, whereas the 1500m case is more likely to give rise to cracking.

It therefore appears that track gauge tighter than 1432mm may be a risk factor for RCF development on high speed curves.

9. THE INFLUENCE OF DYNAMIC EFFECTS

The analysis so far has been confined to investigations of the quasi-static behaviour of vehicles in curves. In practice, the dynamic response of vehicles also has a significant influence on the RCF behaviour.

Many cases of RCF on curves have clusters of cracks separated by sections of crack-free rail. Sometimes clusters are associated with welds, which can represent a discontinuity in the rolling line seen by the wheel. Even if a weld is perfectly aligned, a change in rail profile

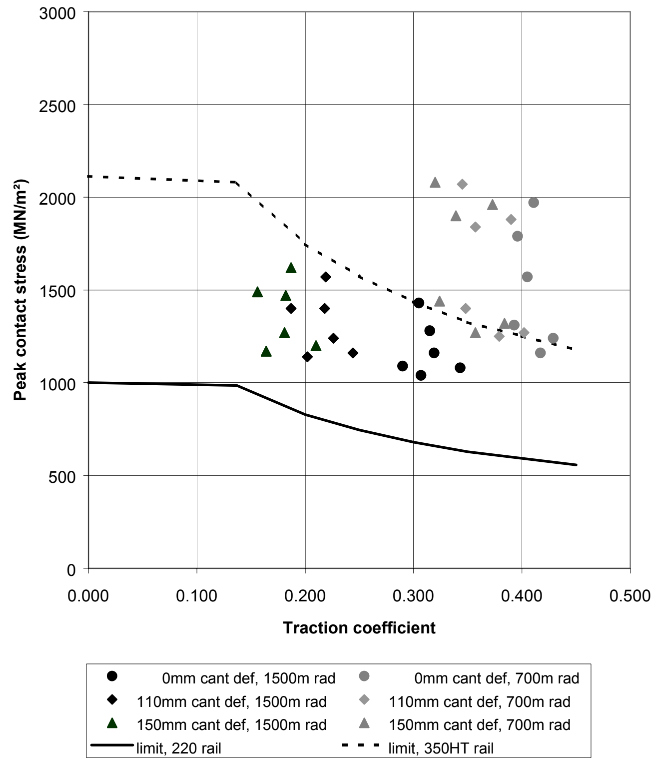

Figure 23 Shakedown Diagram – cant deficiency

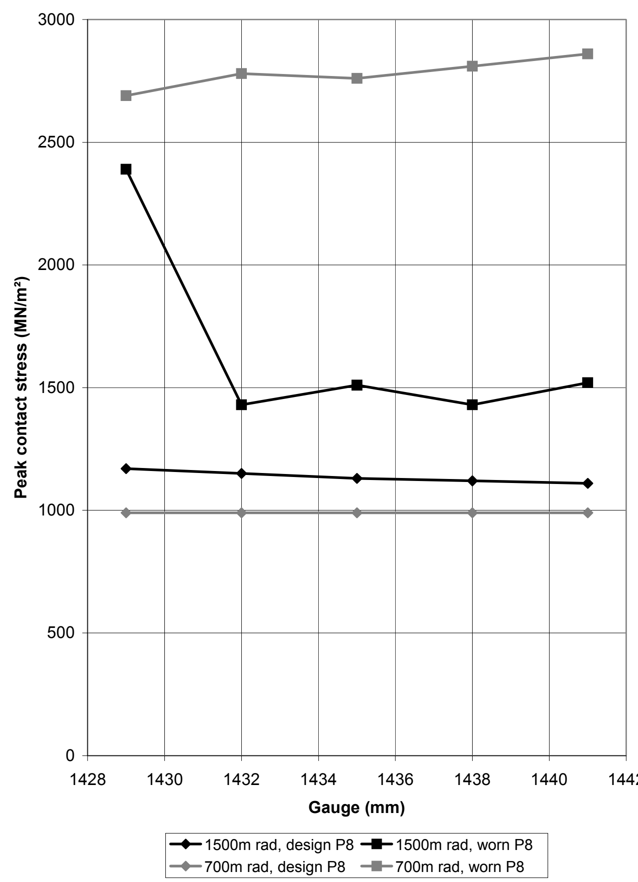

Figure 24 Effect of track gauge on contact stress

from worn to new can give a rapid change in the position of the rolling line which will cause sudden steering forces in the wheel/rail contact while the wheelset position adapts to the new conditions.

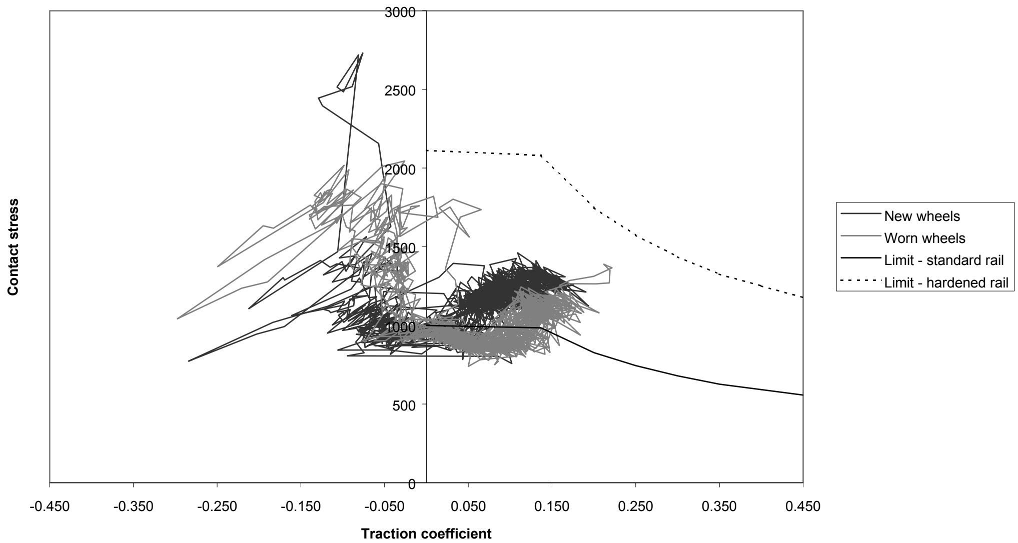

Figure 25 Dynamic Shakedown Diagram

Even in the absence of welds, the dynamic response of the vehicle, both to the design geometry of the transition and to random track irregularities, can cause wide variations in contact conditions. This can be seen in Figure 25, which shows the shakedown time history response of a class 91 locomotive running at 125 mile/h into the measured geometry of a 3600m radius curve at Biggleswade.

The clustering phenomenon is key to a better understanding of the varying influence of contact stress, traction coefficient and wear, because it enables us to identify adjacent sections of rail where many of the key variables can be eliminated. By simulating the variation along the track of these key parameters for a representative range of vehicles and wheel profiles, and correlating this with cracked and uncracked parts of the rail, it should be possible to obtain a far better understanding of the conditions that promote RCF in real life.

Limited preliminary work in this area was undertaken towards the end of Phase 2 of the Railtrack funded study into RCF. Further investigations await additional funding.

10. SWITCH AND CROSSING

Although this paper has been concerned primarily with plain curves there has also been a major problem with RCF in S&C.

There are some factors particular to S&C that may contribute to RCF. Firstly, most S&C is laid with vertical rail, which contacts closer to the gauge corner, usually giving higher contact stresses. Secondly, even when following a straight path through S&C, there are drastic changes in rail profile when running onto the switch blade and through the crossing. The changes in rolling radius difference associated with the changing rail profile give the longitudinal creep forces that are required to generate RCF. Finally, crossings can be damaged by the “false flange” that may develop on the field side of wheels that have experienced severe tread wear, which can give high contact stresses.

The vehicle dynamic influence on S&C is much more complicated to analyse than plain curves, as a result of the rapidly changing geometry. Once a full understanding of the key RCF influences has been obtained by study of the simpler plain line case, then the lessons can also be applied to S&C in the future.

11. FUTURE DEVELOPMENTS

The eventual aim of research into the understanding of RCF must be to find ways to eliminate, or at least manage the problem in the most cost-effective way.

Grinding is a key tool to manage RCF, primarily because it removes incipient cracks before they can grow to damaging lengths. It can also reduce vehicle dynamic effects by imposing a consistent rail profile through welds. However, grinding is costly and rail life will be limited by the loss of section, so other measures to tackle the problem at source may be more cost-effective in the long term. Some benefit may be gained by consideration of rail steels that have a higher natural wear rate and thus reduce the need for grinding as artificial wear.

Changes to wheel and rail profile are clearly worth considering, and can be applied to existing vehicles and rails. Unfortunately, changes to the wheel or rail profile that act to reduce contact stresses and /or traction coefficients also tend to reduce the steering ability in curves. Thus, improvements in RCF by profile change are likely to be at the cost of additional lubrication and/or additional maintenance costs for wheels and rails.

The current state of knowledge is in any case not sufficient to assess the influence of alternative wheel and rail profiles with confidence. The current grinding profile is likely to be effective at preventing gauge corner cracking in tighter curves, but its effectiveness against head checking in high-speed curves is far from proven.

For the longer term, on the vehicle side, train manufacturers should be encouraged to take more account of RCF issues in future designs. However, the current state of the art cannot tell us whether the current British practice of using relatively soft yaw suspension to allow radial steering is better or worse than much stiffer continental designs.

More radical changes for future vehicles could include the use of cross-braced or forced-steering bogies to greatly improve the steering in curves. The class 66 locomotive uses a cross-bracing system that can give very low curving forces, but a consequence is that the vehicles suffer from hollow treadwear and the development of false flanges, which can promote RCF in S&C. Another radical solution is to eliminate longitudinal steering forces altogether by the adoption of independently rotating wheels. In combination with forced steering, as in the Spanish Talgo system, or the Copenhagen S-Tåg, adequate curving performance can be obtained without generating any longitudinal creep forces. This configuration could give the ultimate low-RCF train.

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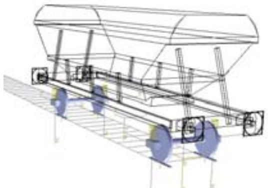
January, 2004
Simon Iwnicki
(RTU Manager)



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