

# **Tracking and Curving by Railway Vehicle : Issues in Heavy Haul**

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## **1.0 Abstract**

Damages to track and vehicle are greatly influenced by the tracking/ curving characteristics of the vehicle. Mitigating these damages, and thus the maintenance requirements, has become more important with increased axle load of vehicle, which has the potential of causing disproportionately higher damages. Conicity of wheel-rail and angle-of-attack of wheelset are important parameters affecting tracking/ curving ability. These, in turn, get affected by radius of curve, design of vehicle suspension, frictional characteristics at the rail-wheel contact, wear of rail and different components of vehicle, and several other parameters. This paper analyses the mechanism of tracking/curving, generation of tangential forces and their influence on wear and Rolling Contact Fatigue defect generation in rail and Wheel Climb derailment. Elements of design as well as maintenance of rail-wheel interface and vehicle suspension which would mitigate the problem have also been discussed.

## **2.0 Introduction**

Development of railway vehicle with improved tracking ability has been an important field in railway engineering. Tracking ability of rolling stock affects the stability of run, in turn affecting safety as well as maintenance requirements of both rolling stock and track. Increased vulnerability to derailment and excessive wear and damages to track/ vehicle in curves have been known to railway engineers for very long. However, understanding the rail-wheel interaction had been difficult due to the complexities involved. Recent advancements in the fields of Numerical Techniques and Field Measurements have contributed a great deal in developing a better understanding of the subject.

Introduction of higher axle loads on different Railways resulted in a large increase in damages in the form of wear and Rolling Contact Fatigue. Thus improved design of vehicle and better maintenance practices became unavoidable. On Indian Railways also, three-piece bogies (Casnub) were introduced for running heavier loads. Although these bogies were superior in respect of negotiation of track unevenness, their curving performance was far from satisfactory, which resulted in severe wear in curves and a large number of derailments as well. There have been several improvements made in the design and maintenance practices for overcoming the problem. These have yielded positive results but problems such as severe track damage in routes with sharp curves still remains a challenge.

### 3.0 Self-Centering Motion of Wheel

Wheel treads are machined at an angle  $\gamma$  to the rotational axis of the axle, or to a more complex profile. As a result, when the wheelset is displaced laterally with respect to the track, a rolling radius differential is generated between the two wheels of the wheelset. If displacement of the wheelset is in the correct sense, it will induce a self-centering action of the wheelset.

Klingel mathematically worked out that the self-centering motion is represented by a sine wave, with wave length on a straight track as

$$\lambda = 2\pi (l^*r_0/\gamma)^{1/2}$$

Where,  $2l^*$  = Dynamic track gauge,  
 $r_0$  = Dynamic wheel radius

The wave length of oscillation gets modified (increased) when wheelsets get coupled through a rigid wheel base, e.g. in a bogie. Track damage in the form of wear and rolling contact fatigue patterns on the rail at wave lengths equal to the kinematic wavelength of the wheelset is a common observation in the field.

In a curve, the outer and inner wheels have to travel different lengths in order of the wheelset maintaining a perfectly radial position. This is possible under the conditions of pure rolling if the wheelset shifts towards the outer rail by an amount 'y' given by

$$y = r_0^*l/R_c^* \gamma$$

Where,  $R_c$  is the radius of curve

### 4.0 Hunting

The self-centering motion of a wheelset normally decays in a damped manner as the wheelset corrects its motion to lateral track disturbances. However, when a wheelset hunts, the self-centering action of the wheels becomes unstable through the action of the inertia of the wheelset, in combination with (or without) other car components. The wheelsets over-react to the steering forces and oscillate violently on the track in a sustained self-centering action. In such a situation, the wheelset or bogie continuously oscillates from one rail to the other rail while traversing down tangent track. This phenomena is also observed in curves, but is much less likely to occur. Bogie hunting can be caused by low warp restraint, reduced shear stiffness, worn wheel profiles, and/or other worn bogie components.

As the kinematic frequency of wheelsets can be increased by coupling them, they are suitably constrained in a bogie to prevent hunting at practical running speeds.

## 5.0 Creep and Tangential Forces

**5.1 Longitudinal Creepage:** Let us consider a wheelset deflected laterally from a pure rolling position by a distance 'y' ( Figure 5.1-1). This is referred to as the “Initial State”. On straight track , the pure rolling position is the centerline of the track. On rolling forward with a velocity,  $v$ , the deflected wheelset will want to roll to a “Preferred State”. If the wheelset is constrained to remain in a similar attitude to the track, as it was in the “Initial State,” creepage takes place as the wheels roll. In the case illustrated, the wheel of larger rolling diameter slips back, with the smaller rolling diameter wheel slipping forward. Longitudinal creepage is defined as the ratio of the creep velocity to the forward velocity of wheelset.

Slip forces,  $F_s$  are generated on the wheelset, which react against the constraining forces at the journals. Forces opposite to  $F_s$  are act on the rail. The amount of creepage and the creep force generated are directly proportional to the displacement  $y$  and the cone angle  $\gamma$ .

The above description is of a quasi-static form for the sake of simplicity. A similar model may be drawn in a dynamic sense with the inertia of the wheelset and suspension design adding to the constraining forces.

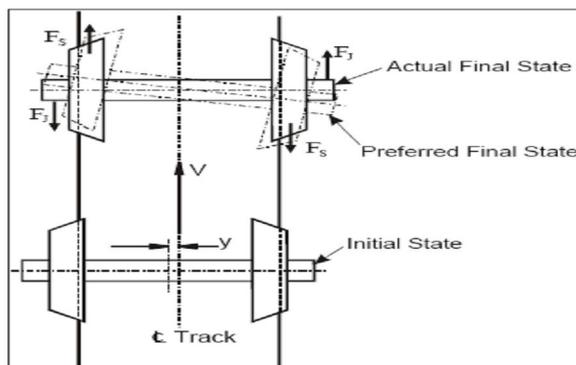


Figure 5.1-1

The effects of longitudinal creepage are very commonly observed in running track. Excessive longitudinal creepage, combined with high-contact stresses, is often seen in material flow of the rail producing head checks and subsequent shelling.

**5.2 Lateral Creepage:** Consider a wheelset placed at an angle of attack ( angle of yaw),  $\alpha$ , on the track (Figure 5.2-1). As the wheelset rolls forward, the preferred final state of the wheelset is shown by the dotted line. If the wheel is constrained by the vehicle suspension or a flange force to be oriented to the track in a similar position as the initial state, the wheelset must have slipped laterally. This is lateral creepage and the associated forces are lateral creep force.

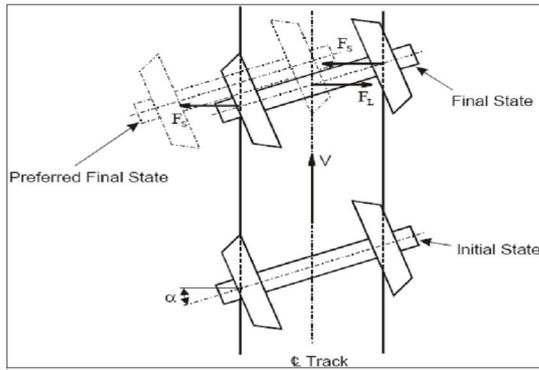


Figure 5.2-1

In case the wheelset is not fully constrained in the lateral direction, the creepage gets reduced as discussed below

The lateral velocity  $V_t$  of a wheel due to its rotational velocity is given by

$$V_t = -\omega r_0 \sin(\alpha)$$

where  $r_0$  is rolling radius ,  
 $\alpha$  is wheelset angle of attack.

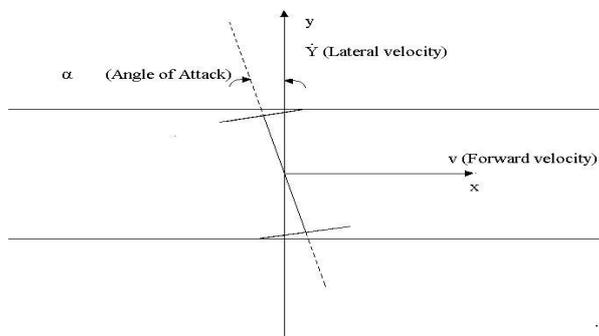


Figure 5.2-2

Figure 5.2-2 shows a plan view of a wheelset with an Angle of Attack,  $\alpha$  relative to the track. This angle contributes to the lateral creepage through a component of the wheelset's rotational velocity. If the wheelset has a lateral velocity,  $\dot{y}$ , in addition to the component of lateral velocity due to its rotation, the net lateral velocity of the wheelset at the contact zone, for small angle of attack ( $\alpha = \sin \alpha$ ), is given by

$$V_y = \dot{y} - \omega r_0 \alpha$$

Lateral creepage is defined as the wheel-rail relative lateral velocity divided by the forward velocity

$$\gamma_y = (\alpha - \dot{y}/V)$$

The term  $(\alpha - \dot{y}/V)$  is commonly known as the 'effective angle of attack' and is a function of the wheelset lateral velocity. Thus, if the wheelset is moving towards flange

contact with a positive angle of attack, the lateral velocity tends to reduce the effective angle of attack.

The force associated with lateral creepage are proportional to the effective angle of attack. The constant of proportionality is dependent, inter alia, on axle load and contact geometry. Further, Spin creepage also affects lateral creep force. Direction of lateral creep force depends on the resultant of the contribution of both lateral and spin creepages. However the effect of spin creepage is usually small.

The effects of lateral creepage are very commonly observed in running track. When steering cannot be achieved by means of creep forces, flange contact is made and a lateral flange force is developed. High lateral creepage results in lateral material flows in the rail crown in sharp curves or at large lateral track discontinuities as shown in Fig 5.2-3.

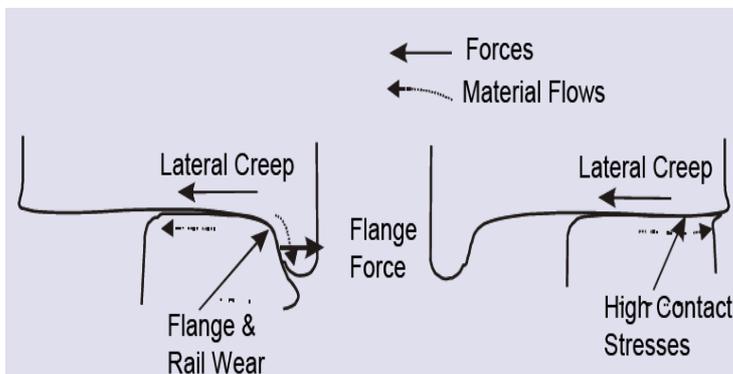


Figure 5.2-3

### 5.3 Other Forces

Lateral Flange force has a Frictional force associated, which is generated as a result of the rolling of the wheel flange about the tread contact point. The Frictional force has a vertical component which causes a reduction in vertical load over the contact patch and may result in wheel climb and subsequent derailment.

Other forces, like spin creep and the gravitational forces, also act on the wheelset but are often of lesser magnitude than those described above and do not necessarily play a significant role. Spin creep occurs when different parts of the contact roll on different radii relative to the axis of the axle. Hence, a rotational “scrubbing” action occurs at high contact angles. This has been associated, together with high contact stresses, with the formation of head checks. The couple associated with spin is considered to have a minimal influence on rail/wheel forces.

Gravitational force is generated on the wheelset when the lateral components of the normal reaction to the contact patch are unequal. This force occurs when the wheelset is

deflected laterally and profiled wheels, like worn wheel profile wheel used on the Indian Railways, are used.

## 6.0 Curving

**6.1 Curving and Tracking Ability:** Direction of Longitudinal creepage affects steering of wheelset in a curve. Consider a longitudinally constrained single wheelset negotiating a curve, Fig 6.1-1. Direction of wheel Longitudinal creepage would depend on the lateral position of the wheelset with respect to the pure rolling position as shown.

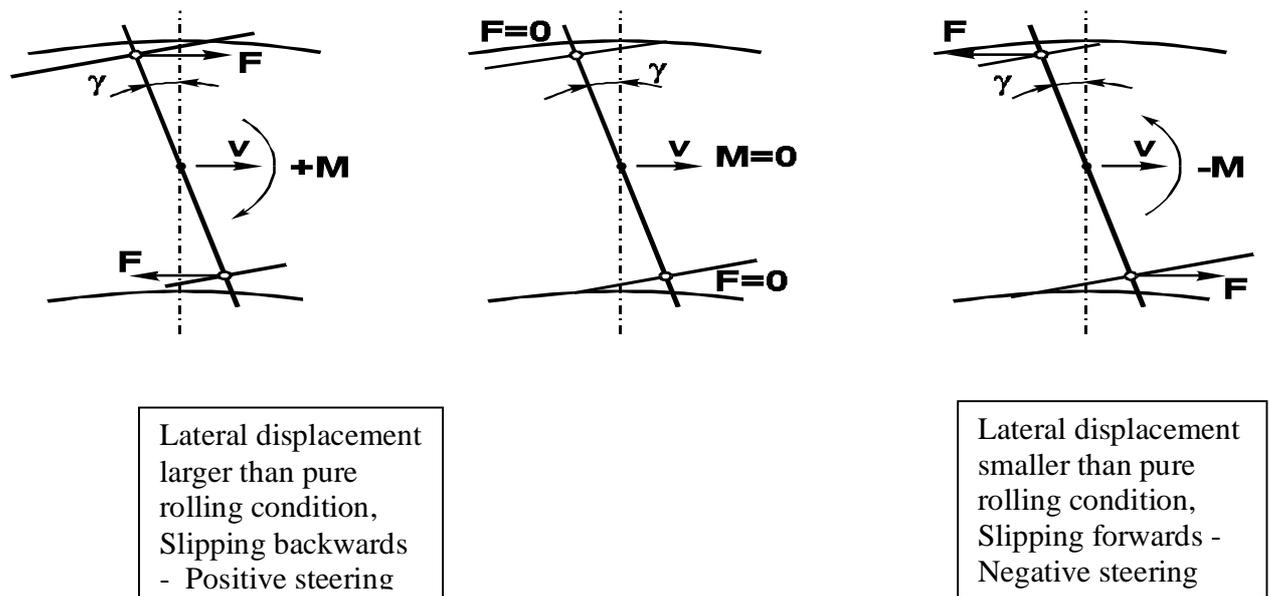


Figure 6.1-1

As discussed earlier, wheelsets require coupling in the horizontal plane so that wheelset stability is obtained at any practical vehicle speed. The manner in which this is done influences the ability of the coupled wheelsets to negotiate curved track.

When two wheelsets are moved relative to one another in opposing yaw senses, resistance to this motion is depends on the bending stiffness. If two wheelsets are deflected laterally relative to one another in opposite senses while retaining parallelism between axle centerlines, they are said to have moved in a shear and the resistance depends on shear stiffness (Fig 6.1-2).

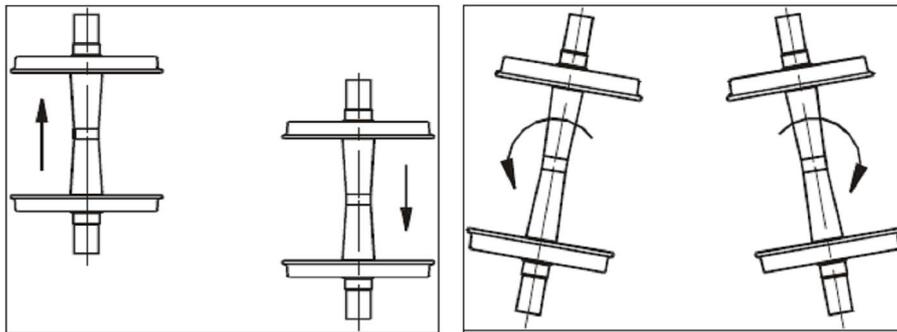


Figure 6.1-2

Bogies with varying bending and shear stiffness are in practical use. A combination of stiffness is required to be chosen to give a stable run. Theoretically, it has been established that selecting a minimum value of either of the stiffness requires a relatively higher value for the other.

Let us consider the case of a conventional bogie with a high bending stiffness. This implies that both wheelsets remain essentially parallel to one another and hence may not attain a radial position in a curve. Curving may take place with or without flange contact, as discussed below.

Fig 6.1-3 shows the case of curving without flange contact. The “clockwise” moments resulting from longitudinal creep are in balance with the “anti-clockwise” moments resulting from lateral creep. Hence the bogie is kept in equilibrium.

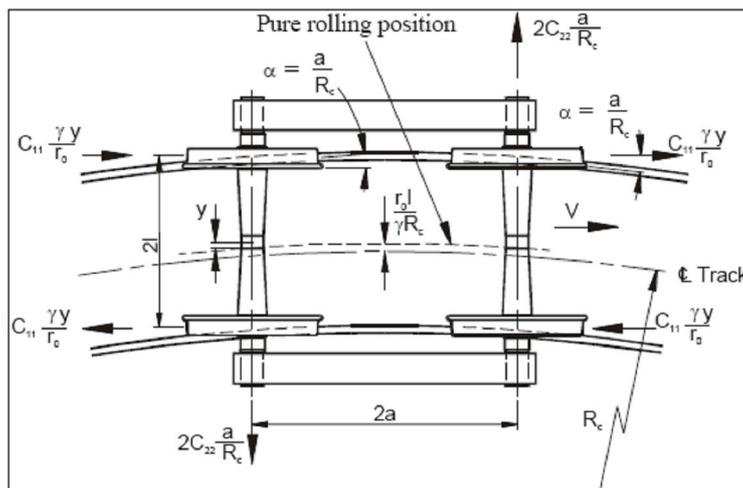


Figure 6.1-3

Figure 6.1-4 shows the lateral displacement under flange free curving for a bogie of 1.83 m wheel base, for wheelset conicity 0.05 and 0.2 on standard gauge for

different curve radius. Typically, bogies on standard gauge, with wheelbases of approx. 1.8 m may negotiate curves of 1500- 2000 m radius without flange contact. Under these conditions, and with an accurately aligned bogie, the lateral and longitudinal creepage is low. Minimal rail and track damage is experienced in such situations. It has been observed on heavy haul routes that side and crown wear is minimal, with a degree of material flow to the field side of both high and low leg after about 200 GMT.

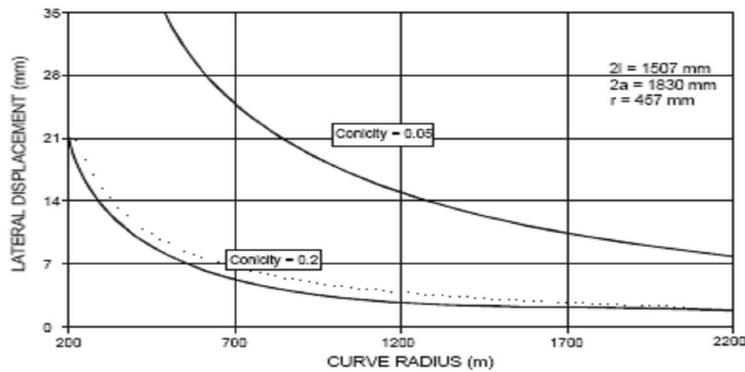


Figure 6.1-4

**6.2 Negotiation of Sharp Curves:** There is a limit on the ability of the bogie to negotiate sharp curves without flange contact. Where the “anti-clockwise” moments on the bogie due to the lateral creepage resulting from the angle of attack of the wheels are larger than the “clockwise” moments that can be generated by the longitudinal creepage, the latter must be supplemented by a flange force. This limit is a function of track gauge, bogie wheelbase, wheelset conicity, rail-wheel contact patch characteristics, gauge clearance, and bogie rotational resistance. In sharper curves, below a radius of 1500 m, with low wheelset conicities and reduced gauge clearance, or at large track discontinuities or in mis-aligned bogies, flange contact is made as shown in Figure 6.2-1.

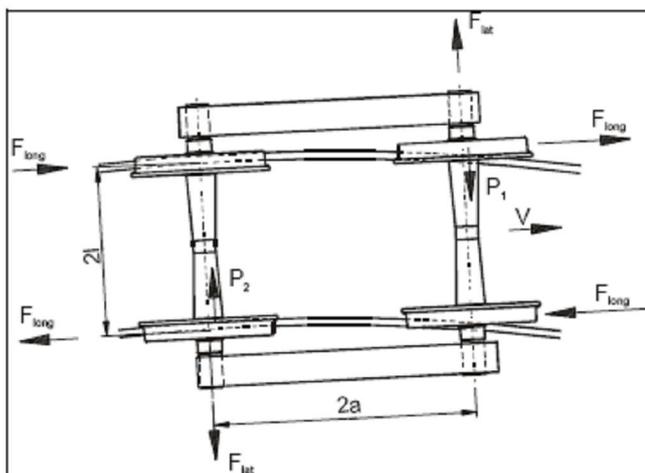


Figure 6.2-1

Under these conditions, high gauge corner wear is experienced and excessive material flow to the field side of the low leg. The most cost effective and quick remedy in such situations is to apply lubricant to the flange and/or gauge corner of the high rail.

High conicity profiles should be generated to encourage flange free curving. However, as the same wheelset are required to be run on the tangent as well as the curved track, high value of conicity provided to the wheelset would have to be limited in order to avoid the problem of hunting. High conicity can still be achieved in curves by means of asymmetric grinding of the rail (while adopting a different profile in tangent track), as shown in Fig 6.2-2.

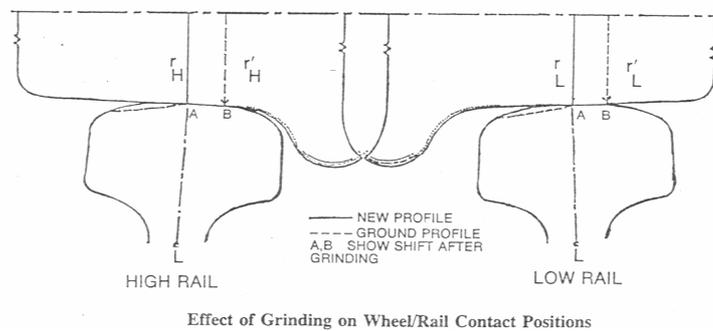


Figure 6.2-2

Asymmetric rail profile are often not long lasting, needing repeated and frequent attention under heavy axle load conditions as the effect of the lateral creepage wears the rail crown down and negates the rail grinding action. These profiles can result in concentration of high stresses near the gauge corner of high rail, resulting in gauge corner fatigue defect, which would require close monitoring.

Super elevation in curves: Excessive cant will cause the bogie to steer out of the curve to counter the resulting inward force. This will increase the angle of attack and lateral creepage and the resulting flange force. Therefore, speed on curve has to be optimized so that the vertical forces on the left and the right leg of the curve are almost equal and unbalanced lateral forces are small. This will also prevent one side of the track to be overloaded under heavy haul traffic.

## 7.0 Heavy Haul Bogies

The conventional three-piece bogie has been the standard freight car bogie for many years. It has been predominantly used for heavy haul operation. It has the advantage of having low manufacturing and maintenance costs. Its load redistribution over uneven track is satisfactory. However, its disadvantages are several - poor curving performance (resulting large angle of attack) that results in higher wear between the wheel flange and the rail gauge

corner, may not have adequate vehicle stability at higher speeds, and may result in derailments due to excessive lateral forces and a high curving resistance.

In conventional three-piece bogies, the inter-axle shear and bending stiffness is obtained from the lateral and longitudinal stiffness of the primary suspension, which acts between the axles via the bogie frame. As the lateral and longitudinal suspension stiffness act in series, a reduction in bending stiffness would also reduce the shear stiffness, thus limiting the maximum allowable operating speed.

A suspension arrangement which ensures a virtually pure rolling motion of the wheelset in a curve and adequate hunting stability on straight track is needed for optimal performance. For optimized curving performance, yaw constraints lower than that acceptable in conventional bogies are required. This necessitates the inclusion of direct linkages between the wheelsets so that an inter-axle shear stiffness independent of the inter-axle bending stiffness can be provided. Such designs are found in the so called self-steering and forced-steering bogie designs. The main advantages of steering bogies are reduced flange wear, improved lateral to vertical wheel/rail force ratios, lower curving resistance, and better derailment and hunting stability.

## 7.1 Maintenance Issues and Improvements in Three-Piece Bogie

**7.1.1 Friction Damping of Vertical Oscillations:** The friction damper typically consists of cast wedge, wedged between the side frame and the bolster, supported by a snubber spring (Fig. 7.1.1-1). This suspension arrangement is thus designed to provide friction forces between the vertical surface of the wedge and the side frame wear plate, and to keep the side frame and bolster square relative to each other. The latter helps to control bogie hunting.

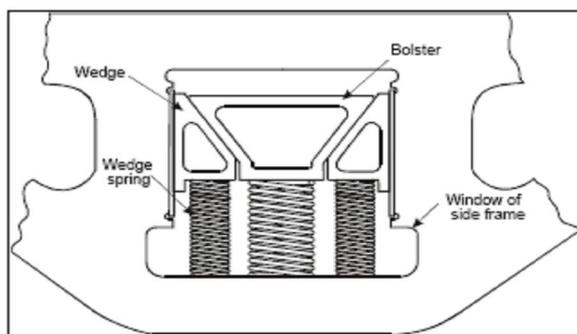


Figure 7.1.1-1

Under certain conditions, such as high operating speeds or adverse track conditions, a high wear rate between rubbing surfaces is experienced. The resulting friction wedge rise between the bolster and the wear liners in the side frame pocket causes a reduction in the frictional damping force and a change in the vertical, lateral and warp stiffness. This can lead to unacceptable running dynamics. In addition, loss of vertical suspension due to wedge binding as a result of wedge rotation due to side frame wear results

in high impact forces being transmitted between the wheel and the rail, damaging vehicle and track.

In order to overcome the above problem, some railroads have implemented resilient friction elements. These urethane elastomeric wear surfaces significantly reduce wear. Hence, the available friction damping and the warp stiffness is maintained for longer periods of service, eliminating regular bolster slope rework.

**7.1.2 Bearing Adapters:** It assist in preventing the three-piece bogie frame from losenging. Insufficient losenging constraint can be a cause of severe hunting at low operating speeds. Under these conditions, the bearing adapters generally experience wear that further reduces the losenging constraint. Therefore, in conventional 3-piece bogies, it is very important to limit wear in the friction wedge area, as well as on the bearing adapters. Bogies have been provided with a stiff rubber shear pad above the bearing adapters to overcome the problem.

**7.1.3 Shear Stiffness Enhancers:** Spring plank, frame bracing and friction wedges all enhance the shear stiffness of the three-piece bogie while permitting a soft lateral flexi coil stiffness. In the absence of high lateral friction wedge forces, rocker and pendulum designs can be used to further soften the lateral ride quality of the vehicle.

The swing motion bogie (Fig7.1.3-1) improves lateral ride by providing the equivalent of swing hangers. The bolster is supported on springs through a spring plank, which is carried on longitudinal knife edges at the bottom of the bolster opening in the side frames. The side frames in turn have knife edges in the pedestal areas that engage with the bearing adapters. Hence, the side frames function as swing links.

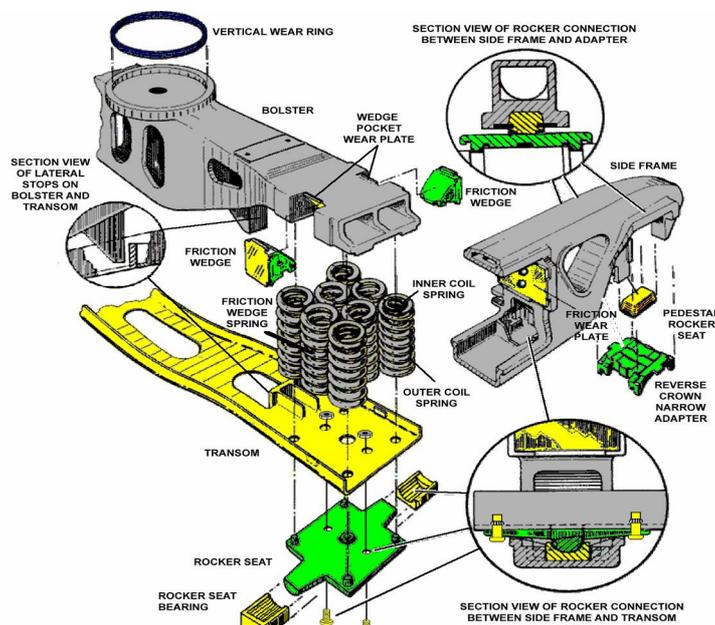


Figure 7.1.3-1

Swing motion bogie also has an increased warping stiffness due to its transom connection, which helps it reduce angle of attack in curves. The lateral stiffness of the truck is reduced, and the carbody can swing (displace) laterally by about 2.5 inches, far more than the 1 inch that a standard 3-piece truck provides. This increases the critical velocity for the onset of truck hunting.

One of the more successful methods of controlling bogie hunting has been bogie side frame cross-bracing as shown in Figure 7.1.3-2. In this design, diagonal cross-braces are added between the side frames and resilient pads are installed between the side frame and the axle bearing adapters.

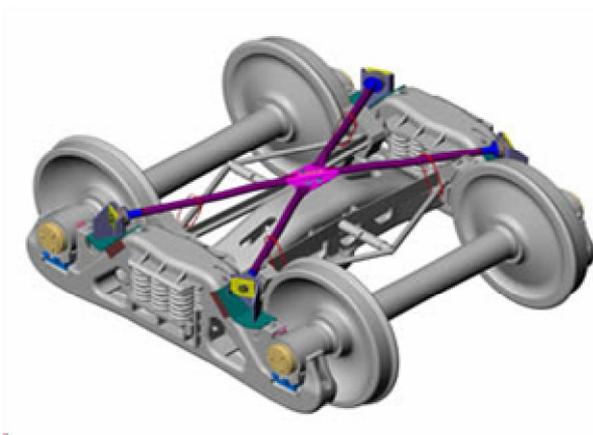


Figure 7.1.3-2

This bogie has the desired high shear stiffness while maintaining a low yaw stiffness, as shown in Fig7.1.3-3. Such bogies would be ideally suited for use on routes with very high curve density. However, it is more complicated than the standard 3-piece truck in terms of the connections between various elements, and the cost of manufacture as well as maintenance would be higher. Such bogies have been tried in North America, but with limited success only.

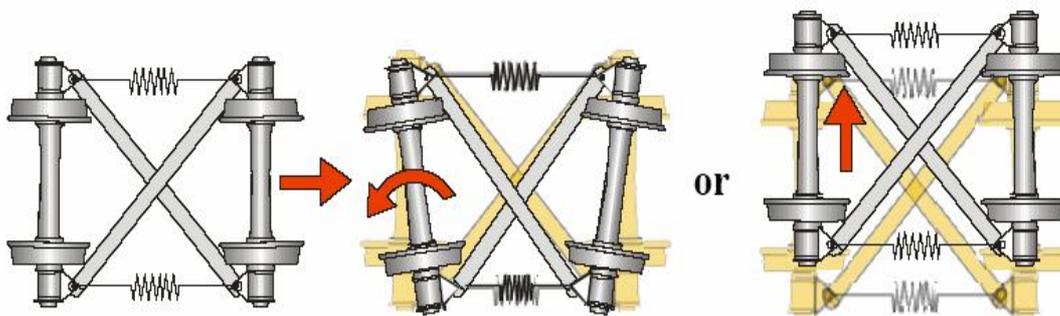


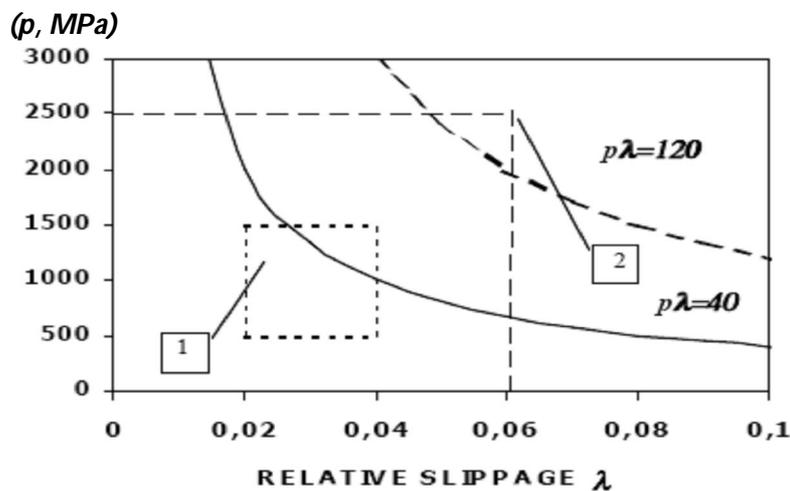
Figure 7.1.3-3

**7.1.4 Rotational Resistance:** The rotational resistance between bogie and vehicle body is important to stabilize the bogie and thus to prevent it from hunting. It is generated through friction in the centre plate, but can also be assisted or replaced by hydraulic damping.

Constant contact side bearer arrangement provides an improved rotational resistance, resulting into improved hunting and rolling performance. However, its adverse effect on curving performance must be taken into consideration. Hydraulic damping for rotational motion is expensive and, hence, provided in premium bogies only.

## 8.0 Wear

Rail and wheel wear is generally assumed to be proportional to the energy dissipated in overcoming wheel and rail rolling resistance. The wear is determined by the relative slippage  $\lambda$  and stresses  $p$  at contact patches, as shown in Fig 8-1. In turn, the relative slippage and stresses depend on dynamic parameters of wheel/rail interaction. Further, wear is highly dependent on the third body properties, which are strongly influenced by lubrication, environment conditions (humidity, rain and snow), and presence of sand. Based on laboratory wear simulated tests in unlubricated conditions, three major wear modes have been identified as (1) mild, (2) severe and (3) catastrophic. These modes are characterized by different wear rates, worn surfaces, and wear debris form and size.



**Figure 8-1**

Rail gauge face and wheel flange wear takes place mainly when a bogie is negotiating a curve, though it may occur for short lengths on tangent track also, particularly if cars are hunting. In sharp curves under dry conditions, conventional three-piece bogie negotiation leads to the catastrophic wear mode of high intensity. This results from severe flange contact due poor curving performance.

## 9.0 Rolling Contact Fatigue (RCF) Damage

RCF damage to rail have been observed in running track and studied in simulations. Results of a comprehensive parametric study performed using TTCI's NUCAR simulation software ( by John Tuna and Curtis Urban, TTCI, Pueblo), using different freight suspensions are discussed below.

Figure 9.0-1 shows the RCF damage function that has been used in this study to calculate a continuous output of rail surface damage, in order to be consistent with other similar studies. Average rail surface damage over the body of the curve has been computed and used to make comparison between different simulations. Longitudinal and Lateral tangential forces ( $T_x$  ,  $T_y$ ) and creepages ( $\gamma_x$  ,  $\gamma_y$  ) have been combined to give a continuous output of  $T\gamma$ ,

$$T\gamma = T_x\gamma_x + T_y\gamma_y$$

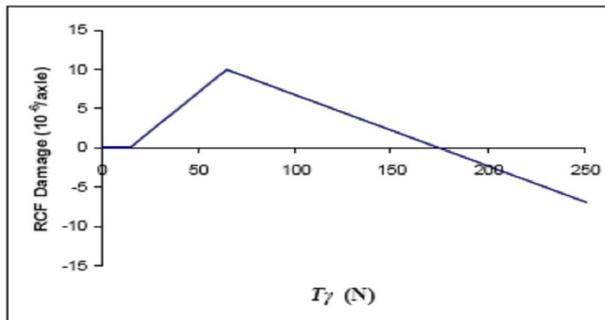


Figure 9.0-1

**9.1 RCF damage and radius of curve:** Assessment done for the enhanced 3-piece bogie predicts the damage depends on the distribution of curves on the route. RCF damage would be maximum for the route with curves in the 600m to 1,200m radius range, while only wear would be produced for curves sharper than 500m radius.

**9.2 RCF damage and type of vehicle:** In general, the vehicle with the enhanced three-piece bogies is predicted to produce the most RCF damage. The bogie is a three-piece bogie (bolster and two side frames) with no primary suspension. The reason being that this type of bogie can produce longitudinal and lateral creep forces resulting into appreciable RCF damage on large radius curves (and even on straight track).

The Y25 bogie has a solid frame and a primary suspension. The axles are relatively free to move longitudinally until the clearances ( $\pm 2$ mm ) are taken up in the axle guides. Axles can align themselves radially in curves of radius 360m or more (standard gauge). Hence the results show very little RCF damage, except on small radius curves.

**9.3 Conicity:** Conicities considered (as measured for typical wheel-rail profile) were 12, 23, and 35 percent. Reducing conicity on large radius curve resulted in reduced RCF damage –

the explanation being generation of smaller longitudinal creep forces for the same lateral displacement of the wheelset. However, reducing conicity on small radius curves resulted into large RCF damage- the explanation being generation of smaller longitudinal creep force, which does not allow closer radial alignment of the axles and results in large lateral creep forces that increase RCF damage.

Increasing conicity improves the ability of wheelsets to steer in all, except the smallest, radius curves. However, for large radius curves, any small offset in wheelset lateral position will result in longitudinal creep forces and RCF damage. Such small offsets could be caused by irregularities in track geometry. Conicity clearly has a significant effect on RCF damage. Conicity is affected by the profile of both the wheels and rails. Assessment of RCF damage requires knowledge of the variation of wheel profile over the vehicle fleet and the distribution of rail profiles on the route.

## 10.0 Wear Management and Lubrication

There exists an '**Optimal Wear Rate**' at which fatigue and wear are in balance. The optimal wear rate is obtained when the surface material wears just enough to prevent small fatigue cracks from developing in the rail and propagating to become detail fractures. The optimal wear rate depends on differences in traffic type and density, axle load, rail metallurgy, and track curvature. On the average, an optimal wear rate is estimated as about 0.02 mm/MGT. For providing the optimal wear rate, it is necessary to introduce friction management technology and wear monitoring facilities.

The coefficient of friction as it relates to railway operation is categorised into three general categories: (1) low friction — 0.2 or less, (2) intermediate friction — 0.2 to 0.4, and high friction — 0.4 or greater. Based on the experience gained on different heavy haul operation, and laboratory/ simulation studies, following have been identified as the objectives of friction management, in respect of maintenance of the coefficient of friction:

- at wheel flange/rail gauge interface at the low level,
- at the wheel tread/top of rail interface of freight cars at the intermediate level, and
- at the wheel tread/top of rail interface of locomotives at high level.

Friction management is a concept that allows, by careful selection of specific materials (friction modifiers), the development of layers which obtain desired wheel/rail frictional characteristics.

## 11.0 Wheel Climb Derailment and Angle of Attack

**11.1 Nadal Single Wheel L/V Criteria:** Following is the Nadal equation for the condition of incipient derailment by the mechanism of wheel climb

$$L/V = (\tan\beta - \mu) / (1 + \mu * \tan\beta)$$

Where  $L$  = lateral force on wheel  
 $V$  = vertical force on wheel  
 $\beta$  = Flange slope  
 $\mu$  = friction co-efficient between rail and wheel

Nadal equation has been used on different Railways, including Indian Railways, as one of the criteria for certification of speed of rolling stock.

The results of a single axle wheel climbing rail obtained from simulations using NUCARS (TTCI's simulation software) and flange climb tests conducted using the Association of American Railroads Track-Loading Vehicle is shown in Fig 11.1-1. The graph indicates that for large wheelset angle of attack (about 10 mrad and more), derailments occurred at Nadal's value. However, for smaller and negative angles of attack, the  $L/V$  ratio required for derailment increased considerably, Nadal's criterion being conservative in such situations.

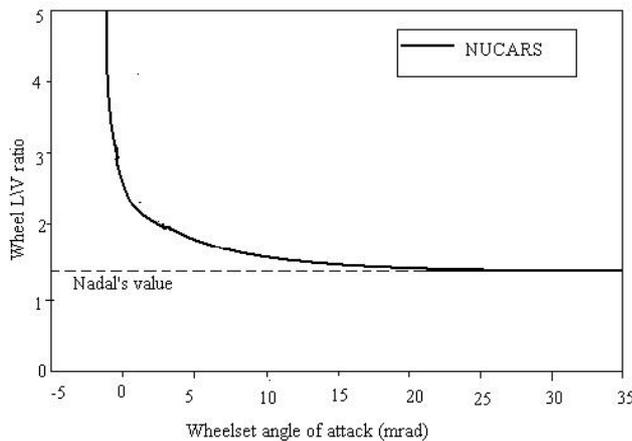


Figure 11.1-1

Nadal assumed that wheel was initially in two-point contact with flange point leading the tread point. He concluded that wheel material at flange contact point was moving downwards relative to the rail material, due to the wheel rolling about the tread contact. He considered the total frictional force generated at the flange point contact acting vertically upwards, assisting wheel climb. However, in reality, direction of the frictional force would actually depend on the position of the flange point vis-à-vis the tread point and would be near vertical only for a large angle of attack. Hence for smaller angles of attack, criteria as per Nadal equation would be conservative.

Another important factor, not considered by Nadal, was the influence of the other wheel of the axle on the subject wheel. The wheelset will have a large lateral velocity under the influence of the derailing lateral forces. At small angle of attack, say upto 5 mrad, the Effective Angle of Attack,  $(\alpha - \hat{y}/V)$ , (discussed earlier in section 5.2) would be negative. This would result in generation of a lateral creep force opposing the derailment on the non-

flanging wheel. Magnitude of this lateral force would depend upon the Effective Angle of Attack. Therefore, criteria as per Nadal equation is conservative at small Angles of Attack.

**11.2 Wheel climb duration limit:** Nadal equation assumes flange-climbing derailment is instantaneous, once the L/V ratio limit has been exceeded. However, based on research and considerable experience in on-track testing of freight vehicles, a 50 msec time duration was adopted by the AAR for the chapter XI certification testing of new freight vehicles. The chapter XI criterion states “ The individual wheel L/V should not exceed 1.0 on any wheels measured . The instantaneous sum of absolute L/V’s on any axle shall not exceed 1.5”, and “ (Those values) not to exceed indicated value for a period greater than 50 msec per exceedence.”

**11.3 Proposed TTCI Wheel climb distance criteria:** The TTCI flange climb criterion has been developed for North American freight cars using AARIB wheel profile with a 75° flange angle at speeds below 80 km/h in curving. This criterion encompasses two limits , the single wheel L/V limit and L/V distance limit. The distance limit is the maximum distance that the single wheel L/V limit can be exceeded without risk of flange climb derailment. It is possibly the first time that the wheelset angle of attack has explicitly been included in the flange climb criterion. Figure 11.3-1 shows the simulation results of the L/V distance limits under different wheelset angle of attack. The test and simulation results showed that the distance limit is a function of wheelset angle of attack.

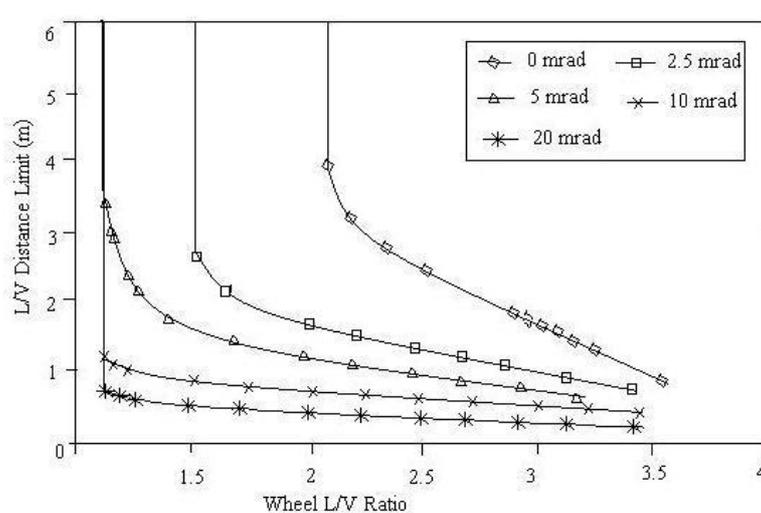


Figure 11.3-1

## 12.0 Conclusion

Inability of wheelset to radially align in sharp curves results in increased damage in the form of wear or RCF defect. High axle load bogies require to be provided with improved features which result in reduced yaw stiffness while maintaining other parameters within appropriate limits and, hence, radial alignment and better curving. Although such bogies would be costlier, they are likely to result in overall economy considering reduced

maintenance cost of track. This must be associated with provision of adequate conicity by suitable profiling of wheel (by turning) and rail (by grinding) and an appropriate friction management regime. Indian Railways presently operate the same freight vehicles even on routes with a large density of sharp curves, as on any other route. Rail profile management by rail grinding is still in the planning stage and friction management regimes are not being effectively implemented. As a result, problem of severe damage to track is prevalent on routes with high density of sharp curves having high density of freight traffic.

### **13.0 Reference**

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