# RAIL WHEEL INTERACTION: NEGOTIATION OF DIVERGENCE IMPOSED BY CURVES AND TURNOUTS – HIGHER AXLE LOADS AND HIGHER SPEEDS

By *VIVEK KUMAR GUPTA , GM/RITES/MUMBAI*

**Synopsis**

Rail wheel interaction is a complex phenomenon and is fascinating the imagination of Railway Engineers and travelling public since inception of Railway. It requires more insights when complexities of diversion in form of curves or turnouts is superimposed. Not much basic work has been done in India in recent past on the issue resulting in ambiguous understanding of the concepts even to Railway P Way Engineers. This paper is meant to discuss basics of rail wheel interaction mechanism in context of negotiation of curves and turnouts by railway vehicles. The issue assumes greater importance in context of introduction of higher axle loads and requirement of negotiation of curves and turnouts at higher speeds.

## 1.0 INTRODUCTION

Railway vehicle by its peculiarity of construction has in determinant behaviour of sprung and unsprung support loads and solution becomes more complex by taking into consideration inelastic base medium consisting of rail/sleepers/formation. The complex mechanism throws unparalleled challenges to engineers for correct modelling of pattern of behaviour and understanding their interactive relationship. The number of variables further increases when behaviour of this interaction in negotiation of divergence imposed by curves or turnouts is attempted to be studied. The affect on rail wheel interaction caused by curves and turnouts associated with higher axle loads and higher speeds require study of parameters of stability and maintainability of track as well as rolling stock design point of view. Study of phenomenon of development of RCF( rolling contact fatigue) on divergence imposed by curves and turnouts is most relevant in present context from point of view of additional requirement of maintenance of track/rolling stock.

## 2.0 MECHANISM AND PARAMETERS OF RAILWAY VEHICLE NEGOTIATIONG A CURVE

Railway vehicles use wheel sets 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 wheel set will tend to roll towards the outside of the curve, and the trailing wheel set will tend to roll towards the inside.

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 both wheels are rotating at the same speed, the larger radius wheel tries to roll further than the smaller radius wheel, thus steering the wheel set towards a radial alignment, when it will roll smoothly around the curve. The opposite process happens on the trailing wheel set as it moves inwards on the curve.

The outer rail on the curve is longer than the Inner rail, so that unconstrained wheel sets 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 in Fig 1.

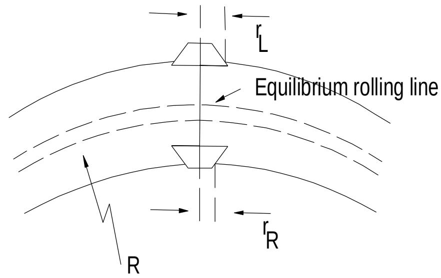


Figure 1 Rolling radius difference

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.

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.

## 2.1 FACTORS AFFECTING RCF (ROLLING CONTACT FATIGUE) ON 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

There are a range of conditions that can be obtained with different vehicles, with a selection of different wheel and rail profiles. Cracking is considered to be inevitable on normal grade rails with any vehicle, even with perfectly smooth track, which is not consistent with real life experience. Head hardened rails are supposed to be far more resistant to cracking than normal grade rails whereas there are experimental evidences that in practice the opposite is true. These anomalies are due to influence of wear.

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 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 ( ratio of tangential to normal force) is is non-linear,
with the wear increasing rapidly as the traction coefficient approaches the limiting value represented by the friction coefficient.

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.

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.

## 2.2 **THEORY OF CONING**

On a level track, as soon as the axle moves towards one rail, the diameter of the wheel tread over the rail increases, while it decreases over the other rail. This prevents the further movement and axle retreats back to its original position (i.e., with equal diameters on both rails and equal pressure on both rails).

On curved path, it is seen that due to rigidity of the wheel base either of the wheel must slip by an amount equal to the difference of length or the axle must slightly move outwards to provide a tread of longer diameter over the outer rail and smaller diameter over the inner rail.If the tread diameter on both the rails is same, the amount of slip will be given by

$$\text{Slip} = \theta\,(R_1 - R_2)\ \text{, where}$$

$$R_1 = R + \frac{G}{2} \quad \text{and}$$

$$R_2 = R - \frac{G}{2}$$

θ =Angle at centre in radians

$$\text{Slip} = \theta \times G$$

For B.G. Track G= 1.676 meters

and $$\text{Slip} = \frac{2Л\theta^{o} \times 1.676}{360}$$

where θ°= angle at centre degrees

Slip=0·029 (approx. for 1° of central angle)

Therefore, the slip is about 0·029 m per degree of deflection angle.

Conning of wheels on curves is not of much use as the leading axle if due to centrifugal force movers towards the outer rail the rear axle (or trailing axle) will move towards the inner rail and the full advantage of coning wheels cannot ne availed.

## 2.3 <u>**PROVISION OF CANT AND LOAD TRANSFER MECHANISM**</u>

When a Railway vehicle passes over a curve, the element of centrifugal force comes in play. The centre of gravity of vehicle is O and F and W are the centrifugal force and weight respectively, U is the resultant of these two in the Fig 2/3.

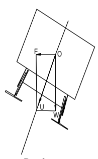


Figure 2

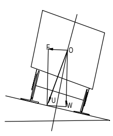


Figure 3

The resultant U is central for a combination of cant of track and speed of vehicle. At this equilibrium, the load carried by each wheel is the same, the springs are supposed to be equally compressed and passenger does not tend to lean in either direction.

When the cant provided in track is less for a particular speed of vehicle, the resultant U inclines towards the higher rail and more weight is carried by outer wheels and outer springs are compressed more than inner and passenger tend to lean outwards under resultant force.

The centrifugal force acting on a vehicle is given by equation

$$F = \frac{W v^2}{g R}$$

Where F is centrifugal force , W is weight of vehicle, v is speed , R is radius of curve and g acceleration due to gravity.

If E is the cant and G the distance between rail centres, the inward force due to cant is WE/G.

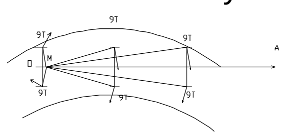
Therefore, for equilibrium conditions,

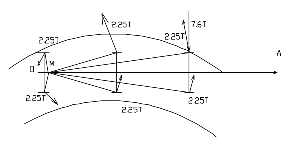
$$\frac{WE}{G} = \frac{W v^2}{gR}$$

Substituting values, equilibrium cant is worked out by

$$E_{eq} = \frac{G v^2}{127 R} \text{ m}$$

When a Railway vehicle enters a curve, the outer leading wheel flange strikes the outer rail and a considerable force is developed between the two even at low speeds. This causes the vehicle to rotate about some point M known as centre of friction and slipping take place under each wheel as indicated by small arrows in Fig 4. The slipping introduces frictional forces indicated by small arrows in Fig 5 .



For a representative movement of 6 axle locomotive , the flange force is 7.6 T and wheel load is 10 T . If the same vehicle is made to negotiate a curve of cant deficiency of 125 mm, the load is transferred from the inner to outer wheels and flange force increase to about 9.5 T . It is reasonable to conclude that no amount of cant can abolish the flange forces and magnitude of this force depends on design of vehicle only. Special attention is therefore required for the transfer of weight from outer wheel to inner wheel under conditions of cant deficiency and from outer rail to inner rail under conditions of cant excess. Under cant deficiency, a large flange force can exist without risk of derailment . However, under unfavourable conditions, an excess of cant may contribute to derailment by reducing the weight on outer leading wheel to such an extant that it can mount the rail. Certain types of goods vehicle when running empty at low speeds round heavily canted curves have been known to derail due to this reason.

## 2.4 EFFECT OF PLAY IN NEGOTIATION OF CURVE

Play is the gap between gauge face of rail and wheel flange. The instantaneous value of play depends on track gauge and thickness of flange. With standard track gauge of 1673 mm on PSC track, the available play is 16 mm (1673-1600-2*(28.5)) for new wheel set and 41 mm for worn-out wheel set. It is therefore important to know the total play between the gauge face of the rail and wheel flange and not merely standard value of gauge and standard flange thickness.

When a vehicle negotiate a curve , the combined effect of centrifugal force and play need to be studied for visualising the behaviour from rail wheel interaction point of view. Under the effect of centrifugal force (Wv2/gR), the critical leading wheel tend to move outwardly thereby reducing the play at the critical contact point. Simultaneous slip at corresponding wheels result in higher frictional forces thereby increasing the lateral force at non critical wheel sets. Reductions in play also reduce angle of attack thereby reducing angularity of attack of critical leading wheel. Since magnitude of lateral force is proportional to square of speed , higher speeds and consequent cant deficiency result in increasing the vertical load with increased lateral force maintaining the proportion undisturbed to any great extent. The positive impact of reduced angularity due to reduced play result in simultaneous achievement of increased vehicle stability. At lower speeds, however, the play may increase due to increased centripetal force due to cant excess which may give rise to higher positive angularity of leading critical wheel.

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 wheel sets “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.

## 3.0 MECHANISM AND PARAMETERS FOR RAILWAY VEHICLES NEGOTIATING A TURNOUT

Turnouts are essential adjuncts of Railway system providing basic facility to rolling stock to pass from line to another. These are to be given special emphasis in their design and maintenance from considerations of safety, comfort and maintainability. Improvement in design of turnouts can reduce the differential of speeds over mainline and turnout sides thus improving the operations and average speed of movement.

Switches and Crossings provide a solution to continuity of rolling and guidance of axles in relation to the normal continuity of the running line. The existence of discontinuities in running and guiding (edges) and the incidences of partially unbalanced lateral acceleration are clearly apparent while negotiating a turnout.

## 3.1 GEOMETRIC CONSIDERATIONS OF TURNOUT ASSEMBLY

Geometrical considerations of turnout assembly consisting of switch, lead and crossings as three components must evolve a design fulfilling requirements of safety, comfort and maintainability.

From the installation aspect, turnouts are to be designed for two categories;

(a) Those situated in big yards with facilities of quick " fanning out" to allow prompt clearance of trains from stations even at the expense of slower speeds ( say 40 KMPH) permitting wide crossing angle.

(b) Those in which speed ( 75 KMPH or more) is important and differential in speeds has to be less ,thereby necessitating use of narrower crossing angle.

51st TSC( march 1975) laid a roadmap for development of both category turnouts but no significant progress for category (b) turnouts has been made in our country so far.

## 3.2 EFFECT OF VEHICLES-DECIDING MIMIMUM RADIUS

After deciding category of turnout, another geometrical consideration is negotiation of worst vehicle required to travel through the turnout giving rise to limiting value of radius of turnout curve. The passage of six-axled locomotive with little lateral play and powerful side springs result in lateral sliding of axles while negotiating sharp radius curve of turnout. These intermittent lateral sliding movements may pose a serious threat to the integrity of switch blade and lead to partial opening.

Taking into consideration worst vehicle, loading gauge and switch assembly construction, the limiting value of ridius of turnout curve has been specified on IR in Schedule Of Dimensions (SOD) as 218 M. This limitation is primarily from the consideration of lateral sliding of worst vehicle.

## 3.3 KINEMATIC AND DYNAMIC CONSIDERATIONS OF TURNOUT ASSEMBLY

The design of turnouts from kinematic and dynamic considerations require study of speeds in relation with radius of turnout, lateral accelerations and lateral forces while negotiation of vehicle over a turnout assembly.

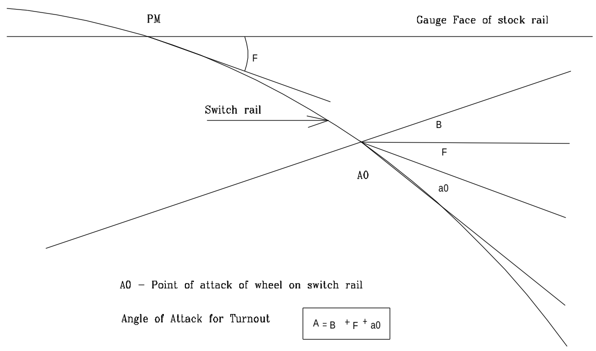
Guidance on a turnout is to be studied for the worst vehicle by determining the locus of flanges over its path of travel. The contact of wheel tyre at rail flange is at a distance z below the rolling plane and the locus obtained gives the running edge of the component of turnout in this same plane at dimension z.The axles occupy various positions in the track and tyre profile vary from new to wornout. The two planes are, therefore, different and difficult to reconcile. This is precisely the reason that for the running edges of items of switches, a recource is made to the single plane of dimension z ( 13-14mm in our country) called the machining plane.

### 3.3.1 LATERAL FORCES AND ACCELERATIONS

The assessment of lateral forces and accelerations takes into considerations the maximum speed permitted on the diverging track. If this speed is studied to passenger vehicles, comfort is the limiting criterion than that of safety.
The lack of compensation of lateral acceleration by cant is considered in calculating the speed limit to be imposed on turnout. The sudden lateral acceleration due to negotiation of curve without transition together with presence of an entry angle at the toe of switch have an effect on impact.
It is therefore important to define the lateral percussive effects permissible from the points of view of comfort and to analyse the composition of these thrusts for the various arrangements of the turnouts.

### 3.3.2 ANGLE OF ATTACK (σ )

For the study of rail wheel interaction on turnouts, angle of attack assumes great significance. Angle of attack can be segregated into three distinct components of (1) Crabbing angle ; (2) Switch entry angle ; and (3) Differential entry angle. ( Fig 4)




1. Crabbing angle (ψ):

This is the angle of the axle relative to through track resulting from the oscillation . This is a fraction of total possible crabbing angle permitted due to play. This angle is basically formed due to sinusoidal movement of axle and is directly related to play between the rail gauge face and wheel set.
On sharp curve, the value of crabbing angle is likely to be lesser due to component of centrifugal force directed outwards towards outer rail coming into affect. The play itself reduces during movement of vehicle over a curve. This affect is more preponderant in sharper curves.

2. Switch entry angle (φ):

This angle is a fixed component depending directly on the construction of switch assembly. The angle is independent of the laying geometry and depends entirely on fabrication methodology.

3. Differential entry angle (α) :

This angle is the difference in entry angle due to point of attack of the axle being different than the mathematical intersection point . This angle is a function of play in the track and radius of switch.

Of the three angles defined above, only SEA can be controlled during fabrication of switches. Other two angles depend upon rail wheel interaction and largely influenced by play in track and general geometrical condition of layout. SEA is therefore only controlable factor out of the three components of angle of attack and usually taken as a design parameter in place of angle of attack in design of turnouts. The value of SEA for 1 in 12,60kg/52 kg PSC layouts to RDSO drawing no T- 4218/4733 is $0^0$-20'-0". As per deliberations of item no 695 of 51st TSC , the permitted speeds by advanced world railways for turnouts with SEA of 20' ( or nearabout this value) are

| *Railway* | *Switch entry angle* | *Speed permitted in kmph* | *Remarks* |
|---|---|---|---|
| *SNCF* | *25'* | *70* | *With 1 in 12 turnout* |
| *DB* | *18' 34"* | *50* | *With 1 in 9 turnout* |
| *BR* | *21'* | *80* | *With 1 in 20 turnout* |

As per article 3-125 of the interim report -2 of research conducted by ORE under Question D-72, angle of attack is a function of crabbing angle, switch entry angle and differential entry angle at speeds greater than 100 KMPH ( on turnout side). On lower speeds associated with sharp curves, switch entry angle is less important.

### 3.3.3 VIRTUAL SWITCH ENTRY ANGLE

Owing to the pecularity of availability of only SEA as a predefined and measurable angle out of three components of angle of attack, same has assumed great importance in discussions of turnout design. Varied combinations of turnouts with different SEA have been designed all over the world and sufficient experience gained for deciding the band of SEA for suitability of turnouts for various locations. The term VSEA ( virtual switch entry angle) is used in reference to SEA for turnouts taking off from curves in contraflexure. For turnouts taking off in similar flexure, no net increase or decrease in SEA under dynamic conditions of wheel movement is considered and effect is reflected in increased value of differential entry angle since actual point of cantact is further away from the mathematical point of intersection. This is due to the fact that vehicle is already negotiating curved path before entering in the divergence imposed by switch toe and it is only by amount of designed SEA that the wheel flange is further diverted away.

### 3.3.4 THEORETICAL VALUE OF ANGLE OF ATTACK

It is mathimatically possible to determine maximum theoretical value of angle of attack by adding up three components viz. Crabbing angle,SEA, and differential entry angle.

$$\sigma = \psi + \varphi + \alpha$$

Question D-72 of ORE deals with determination of maximum angle of attack for a SNCF coach under Appendix 2A of their study. The mathematical equation derived contains following variables

(a) Wave length of oscillation of wheel
(b) Play in track used to determine maximum possible amplitude of oscillation
(c) Displacement of axle in relation to the through track ( at point of attack)
(d) Entry angle of switch
(e) Radius of divergence

For a typical case of SNCF turnout having SEA of $0^0$-18'-0", the maximum angle of attack is calculated as $0^0$-35.5'. The component of SEA is about half of total angle of attack for the case under study.
For turnouts situated on curves in similar flexure, value of angle of attack is not likely to be significantly different from average angle of attack since increase in differential entry angle gets compensated by reduced crabbing angle, SEA being constant.

### 3.3.5 **THEORETICAL VALUE OF LATERAL ACCELERATION**

As per validated theoretical analysis, the lateral acceleration has been worked out as

$$\ddot{Y}\,max = \frac{V^2}{R_1}\; 0.0873 + 1,159\, V \left\{ \left( -\frac{\Psi o}{2} - \phi\left(1 - \frac{e^{-0,0272v}}{2}\right) \right) + \underline{\alpha^{o}} \right\}$$

V- Speed of Coach.

R1- Radius of divergence of the switch.

-Crabbing angle of the axle relative to the through track resulting from the oscillation.

(Fraction of the total crabbing angle permitted by the play in the track).

Ø Entry angle of the switch

-α Increase of the entry angle between the mathematical intersection point and the point of attack of the axle on the switch blade.

This formula is derived from the theoretical mathematical tools and validated by field trials on range of turnouts by ORE and reported in Question No. D-72.

### 3.4 SWITCH ENTRY ANGLE AND PERMISSIBLE SPEED ON TURNOUTS

As per item 695 of 51st TSC ( March 1975)
*The trials carried out by RDSO on 1 in 12 turnout with curved switch entry angle of 27' -35" have indicated speed potential of 40 kmph. Trials on 1 in 16 turnout, with curved switch entry angle of 24'- 27" have indicated a speed potential of 50 kmph.*
*Based on the world practices, ORE recommendations and trials carried out by RDSO mentioned above, it is proposed to provide a switch entry angle of 0°- 20' – 0" for a speed potential of about 60 kmph. In a 1 in 12 turnout, however, the maximum permissible speed on the lead curve with a cant deficiency of 75 mm is 50 km/h. For the time being, therefore, the maximum permissible speed on a 1 in 12 turnout with curved switch with an entry angle of 0° 20°- 0° can be taken as 50 km/h. As and when the cant deficiency is increased to 100 mm, it will be possible to increase the speed further to 58 km/h. i.e, about 60 km/h.*

Above discussion of TSC is the basis for defining safe speeds with regard to SEA for 1 in 12 turnouts. For 1 in 8.5 turnouts having SEA of 47'-27" ( or 46'-59" for improved design), the permissible speeds from SEA consideration is extrapolated to be 30 KMPH.( permitted speed of 40 KMPH for SEA of 40'-30")

### 3.5 ADOPTED DESIGNS OF TURNOUTS AND REFERENCE PARAMETERS ON IR

Standard crossing angles 1 in 8.5/12/16/20

Type of switch Intersecting Type Overriding Curved Switch

Switch Entry Angles $0^o$ -47'-35" for 1 in 8.5 and $0^0$-20'-0" for 1 in 12

Lateral accelerations Upto (0.30-O.35) g

Lateral Forces Upto 9.5 T with 18.8 T axle load vehicle

The location specific permissible speed requirement as noted in 51st TSC is

| *Speed* | *Location* |
|---|---|
| *25 km/h* | *For Goods yards* |
| *50 km/h* | *For passenger yards* |
| *75 km/h* | *For turnouts on the outskirts of big yards* |
| *100 km/h* | *At junctions between single lines and double* |

Above parameters laid a roadmap for development of turnout designs in India. Use of Fan Shaped PSC layouts with or without thick web swithches meet the speed criterion of 25/50 KMPH specified for Goods and Passenger yard movements. RDSO has yet to work for turnouts of 75/100 KMPH.

## 3.6 CRITERION FOR PERMISSIBLE SPEEDS ON TURNOUTS

Subitem 5.3 of item no 735 ( 55th TSC) lays criterions for determining permissible speed on a turnout as

(a) The lateral force exerted at the switch ( which is limited to about 9.5T on BG for vehicles with 18.8 T axle load- the force of this magnitude being measured on 1 in 8.5 turnout with straight switch at the present permissible speed of 10 KMPH)

(b) Cant deficiency on lead curve ( limited to 75 mm)

### 3.6.1 LATERAL FORCES AND PERMISSBLE SPEEDS

The components of switch assembly are subjected to lateral thrust and must be strong enough to withstand designated forces. RDSO have carried out field trials for measurement of lateral guiding force at speeds upto 50 KMPH for 1 in 8.5 and 1 in 12 turnouts with curved switches for WDM2 locomotive.

| *Speed (km/h)* | *1 in 12 Curved switch (switch entry angle 27' 35")* | | *1 in 8.5 Curved switch (Switch entry angle 48' 27")* |
|---|---|---|---|
| *10* | | *5.0* | |
| *20* | | *5.0* | |
| *30* | | *5.5* | |
| *40* | | *8.0* | |
| *50* | | *10.5* | |

### 3.6.2 CANT DEFICIENCY AND PERMISSIBLE SPEEDS

The unbalanced component of lateral acceleration impose restriction on maximum permissible speeds on turnouts on comfort criterion. Since, turnouts are uncanted, entire centrifugal acceleration is to be absorbed by vehicle for which maximum value of cant deficiency has been predefined on Indian Railways. As per para 406 of IRPWM maximum value of cant deficiency is given as 75 mm for BG. The calculated speeds on 1 in 8.5 and 1 in 12 turnouts with 75 mm cant deficiency works out to be 35 and 50 KMPH respectively. Annexure A of item 695 of 51st TSC observed that as and when cant deficiency limit is increased to 100mm, higher speeds upto 60 KMPH for 1 in 12 turnouts would be possible.

### 3.6.3 PERMISSIBLE SPEEDS ON TURNOUTS

Taking into considerations existing design parameters, the permissible speeds on 1 in 8.5 and 1 in 12 PSC turnouts on IR can be worked out from SEA , lateral force and cant deficiency criterions and lowest speed permitted . The details are

| Criterion | Speed in KMPH for 1 in 8.5 turnout | Speed in KMPH for 1 in 12 turnout |
|---|---|---|
| SEA | 30 | 60 |
| Lateral Force | 30 | 50 |
| Cant Deficiency | 36 | 50 |
| NET SPEED | 30 | 50 |

### 3.6.4 TSC's RECOMMENDATIONS OF PERMISSIBLE SPEEDS ON EXISTING LAYOUTS

Based on field trials by RDSO, permissible speeds for various types of existing turnouts were deliberated in 51st TSC in March75. The permissible speeds on then-existing turnouts were recommended as

| *Turnout* | *switch entry angle* | *Permissible Speed (km/h)* | *Remarks* |
|---|---|---|---|
| *1 in 8½ with straight switch* | *1° 34' 27"* | *10* | *On the basis of lateral forces on switch.* |
| *1 in 8½ with curved switch* | *0° 47' 27"* | *25* | *On the basis of lateral force on turn in curve.* |
| | | *30* | *On the basis of lateral force on switch in case there is no turn in curve as in case of cross-over.* |
| *1 in 12 with straight switch* | *1° 8' 0"* | *16* | *On the basis of lateral force on switch.* |
| *1 in 12 with curved switch* | *0° 27' 35"* | *40* | *On the basis of lateral force on switch.* |

It is permissible to have speed of 30 KMPH on 1 in 8.5 curved switch turnout of cross over. Also speed of 40 KMPH has been adopted for 1 in 12 curved switch turnout with a SEA of 00-27’35” as against existing SEA of 00-20’-0”

## 4.0 CONSIDERATIONS FOR COMBINED EFFECT OF CURVE AND TURNOUT

It is not desirable to have turnouts located on curves, especially in similar flexure. However, same is a necessary evil and better to be studied and tackled instead of avoiding. The resultant radius of turnout curve reduces unfavourably for the turnouts taking off from inside of curve and favourably for turnouts taking off from outside or in ccintraflexure. Since the effect is unfavourable only for turnouts taking off from inside in similar flexure, the discussion is restricted to this situation only.

When a turnout takes off in similar flexure, the resultant degree of curvature of turnout curve increases to the extent of simple mathematical summation of degree of main line curve. Important parameters for considerations are

(a) Resultant radius of turnout curve not to reduce below 218 m for BG

As per SOD, the resultant radius of lead curve can not be reduced below 218m ( 8 degree) . This stipulation prohibits laying of any 1 in 8.5 turnout in similar flexure. It also imposes restriction on laying of 1 in 12 turnout in any curve sharper than 4 degree. This provision is the most important field provision with regard to turnouts located on curves.

(b) Superelevation considerations on mainline and turnout for permissible speeds

The provision of laying of turnout in same plane mandates equal cant on mainline and turnout side. This effects the speed potential on mainline taking limiting value of cant excess on turnout side with slower speed and limiting value of cant deficiency on mainline for higher speeds.

(c) Angle of attack ( or SEA) consideration for imposing change of direction

For speeds on turnout side still ranging upto 50/30 KMPH in our country, angle of attack (or SEA) decided as 00-20’-0” for 1 in 12 and 00-47’-35” for 1 in 8.5 does not have any bearing on permissible speeds of turnouts permitted as per radius considerations. Also as discussed in para 3.3.4 above, the net increase in angle of attack due to laying of turnout is not likely to be more merely because of its laying on curve.

(d) Lateral accelerations not to exceed permissible value

Lateral accelerations while entering into a curved turnout is to be kept within specified limits of (0.30-0.35) g . By theoretical analysis and field trials, it has been established by RDSO that these limits are not reachable even for the worst combination of type of turnout ( 1 in 8.5) and vehicle. Infact, RDSO even abandoned the trials of lateral accelerations. It can, therefore, be safely concluded that lateral acceleration is not the limiting criterion for curved turnouts and limiting the permitted radius would suffice .

(e) Lateral force not to exceed permissible value

Higher value of lateral thrust for the curved turnouts is to be controlled for ensuring structural stability of switch . RDSO has prescribed limiting value of 9.5T of lateral force. Mesaurements of lateral forces in instrumented trials conducted at RaeBareilly yard of NR in 1970’s indicate that these forces are maximum for 6 axled locomotives and increase proportionately with degree of lead curve, SEA and speed.
For speeds on turnout side still ranging upto 50/30 KMPH in our country, lateral forces for designed 1 in 12 and 1 in 8.5 turnouts with SEA of 00-20’-0” for 1 in 12 and 00-47’-35” do not exceed for the worst combination of curved turnout and vehicle. Also the limits of 9.5T were decided in 1975 for wooden layout to drg no. RDSO T-1917 and may be considered for upward revision for PSC Fanshaped strengthened/thick web switch turnout.

Based on above discussion, it is very safely concluded that for PSC fanshaped/thick web turnouts of 1 in 12 and 1 in 8.5 and upto a speed potential of 50/30 KMPH respectively, only criterion to be reckoned even for turnouts laid in similar flexure is the radius of lead curve . No slow down limits are required to be imposed for these turnouts since they are already very much on conservative side.

## 5.0 **WAY FORWARD**

In the quest of moving ahead and keeping pace with changing requirements of heavier axle loads and higher speeds , it is high time that critical review of speeds on turnouts is undertaken. With adoption of improved design of turnouts, it is very much feasible and practicable to improve speeds as envisaged way back in 1975. Railway Board had approved standarisation of turnouts of four designs as

| *Speed* | *Location* | *Remarks* |
|---|---|---|
| *25 km/h* | *For Goods yards* | *1 in 8½ turnout with curves switch to existing meet this requirement t.* |
| *50 km/h* | *For passenger yards (for 52 kg)* | *1 in 12 turnout with curved switch to drawing no. RDSO/T 1917. Meet this requirement* |
| *75 km/h* | *For turnouts on the outskirts of big yards* | *New designs are require to be evolved by RDSO* |
| *100 km/h* | *At junctions between single lines and double* | *-do-* |

Further work for standardisation of appropriate design of turnouts suitable for proposed speeds have to be undertaken for implementation in field. The suitability of higher axle load operation and its affect on maintainability of track and rolling stock is to be given due cognizance while adopting new designs.