Track Strength And Track Geometry Design Requirements If Heavy Freight Traffic Has To Coexist With Semi High-speed Passenger Traffic

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1. Preamble:

It is well known that the combination of heavy freight traffic and semi high-speed passenger traffic poses challenges in the design, construction and maintenance of track, because of the conflicting requirements. The main concern of the track engineers in the field is that the track deterioration caused predominantly by freight trains becomes impediment to passenger trains. A lengthy preamble will be like carrying coals to Newcastle. It would be meaningful to plunge straight into analysis of the technicality behind track fatigue and other related issues, so as to acquire clear understanding and to find solutions.

2. Track Deterioration Due To On-loading Caused By The Wheels

2.1 Track maintenance is meant to compensate for the deterioration caused by the dynamic wheel loads. Therefore, both Designer and Maintainer should have a thorough insight into the on-loading caused by wheels, i.e. the mechanism of transmission of additional dynamic load on rail / track.

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2.2 Advanced railway technology treats Track and Vehicle as an integrated system. The momentary wheel load Q will be varying instant to instant during the run, with respect to QN which is the static wheel load (also called nominal wheel load). Off-loading (i.e., Q < QN) and on-loading (i.e., Q > QN) occur in cyclic manner. Off-loading may contribute to derailment, whereas on-loading causes track deterioration. Successive cycles of on-loading and off-loading cause fatigue of vehicle components – a concern for rolling stock engineers.

2.3 There are two types of on-loading, which can occur singly or jointly:

- Dynamic Overload, due to track geometry variation and/or due to the fluctuating response of the vehicle's suspension system.
- Impact Overload, due to (i) surface defects on rail (corrugation, rail scab, defective weld, etc.) or (ii) surface defects on wheel (wheel-flat, tyre scab, etc.)

Dynamic Overload, combined with static wheel load, decides the maximum bending stresses in rail and fatigue of rail-metal, bearing pressure transmitted by the sleeper on ballast, formation pressure at the bottom of ballast, etc. The effect of Dynamic Overload penetrates down to various layers of the supporting structure, though it diminishes gradually. This is illustrated in Fig.1, where Wheel Load Q (100kN) may be taken to include Dynamic Overload.

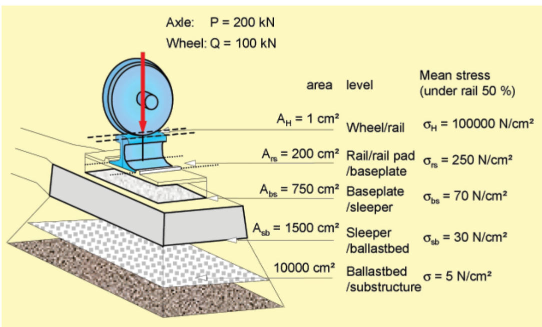


Fig.1

In contrast, Impact Overload, combined with static wheel load, creates intensive local effect on the rail, such as, rail-wheel contact stress and the consequent shear stresses causing fatigue of rail steel. Plastic flow of rail steel may also take place due to yield shear stress exceeded. This impact is transmitted also as striking force at the rail-seat of the concrete sleeper, which is moderated by the rail-pad. The effect of Impact Overload should not extend beyond rail pad in a well-designed track. Powdering of concrete sleeper or ballast, arising due to defect in rail-surface, should be considered as symptoms of bad maintenance.

These two types of wheel on-loading will be further discussed.

3. Dynamic Wheel Overload

3.1 As the vehicle speeds up, the wheel load is augmented by dynamic effect as initiated by track geometry variations. From the vehicle side, the dynamic wheel-overload is contributed by two components, namely, (i) non-suspended masses and (ii) suspended masses. The former component may sometimes attain much bigger magnitude than the latter, even though non-suspended mass is by itself far less than suspended mass. This is due to the acceleration of high values associated with the oscillation of non-suspended mass compared to that of suspended mass. Both the components of dynamic wheel-overload increase with increasing speed. The extent of defects in track geometry and the efficacy of suspension system in the bogie are the vital factors, deciding the dynamic wheel-overload.

3.2 The proportion of dynamic wheel-overload to static wheel load, attributable to non-suspended mass, is the highest in the case of locomotive, due to the heavy traction motors and heavy axles. This proportion is much lesser in the case of electrical multiple unit (EMU), since the traction motors are lighter and are distributed on more number of axles and the bogie suspension is designed to be more sophisticated. In some EMUs of latest design, the traction motor itself is shifted to the vehicle under-frame, with a torque shaft driving one of the wheels in the bogie. These features enable EMU to be run much faster than the locomotive without increasing dynamic wheel-overload. Passenger Coach (loco hauled) may be considered very close to EMU, though axle mounted traction motors are absent. As regards freight wagon, non-suspended masses are far lesser compared to locomotive (due to the absence of traction motor, etc.), but the bogie suspension system is surely inferior. As the combined effect of both these features, dynamic wheel-overload arising in a wagon is of higher magnitude and therefore wagons are rendered suitable only for slow speed.

- EMU or Passenger Coach: Lighter axle load - Medium non-suspended mass - Superior suspension under good maintenance: Hence, high speed is permitted.
- Locomotive: Heavy axle load - Heavy non-suspended mass - Good suspension under good maintenance: Hence, medium speed is permitted.
- Wagon: Heavy axle load - Less of non-suspended mass - Simplified design of suspension under somewhat inferior maintenance: Hence, low speed is permitted.

The above arguments lead to the logical conclusion that it would be wise to have dedicated high-speed track and dedicated freight haulage track. Unfortunately, this has not been possible in India; in fact, even in advanced railways. Hence compromise is inevitable in the design and operation of track meant for mixed traffic.

3.3 Contents of paragraph 3.2 may be summarised as under, in a crude way:

3.4 This subject has been dealt more professionally by Dr. A. Prud'homme (Erstwhile Head of the Track Designing and Research Department, SNCF) in his paper 'The Track', published in 1970, while explaining the rationale for the design of the first TGV track. Even after 46 years, his findings continue to be relevant, more so in this seminar. He has related the effect of fluctuating wheel load (caused by suspended and non-suspended masses) to the track fatigue. Track Fatigue Factor (TFF) is taken as Q^3_N (1 + 3σ2), where Q_N is the static wheel load and is the standard relative deviation of dynamic wheel-overload, which was found nearly equal to (½) (Q_A/Q_N) where Q_A is the maximum dynamic wheel-overload, instantly occurring. takes into account the efficacy of suspension system, effects of track geometry and proportions of suspended and non-suspended weights to axle load. Value of may be determined by field investigation using Measuring Wheel technique, instead of taking it as (½) (Q_A/Q_N).

3.5 This expression TFF = (½)(Q_A/Q_N) was used by Prud'Homme to compute relative track fatigue caused by various vehicles on Standard Gauge track of SNCF, as follows:

- EMU with 16 t axle load at 300 km/h: TFF = 710
- Locomotive with 20.6 t axle load at 200 km/h: TFF = 1375
- Freight wagon with 20 t axle load at 70 km/h: TFF = 1120

These data cannot be taken to be entirely valid for IR track, because of the varying values of the contributing parameters. However the basic principle holds well that maximum design speed of a vehicle has to be decided in accordance with the gross effect of axle load, suspension characteristics and permissible track tolerances on the track fatigue it causes. In the case of mixed traffic line, the relative fatigue values attributable to different types of vehicles/trains can be estimated, by multiplying TFF by the number of respective axles.

3.6 The ORE Committee (D-161) have also concluded in 1988 (much later to Prud'Homme's paper) that :-

➢ Rail fatigue, fatigue of other components and track geometry deterioration varies as the 3rd power of wheel load

➢ Rail surface defect varies as power 3.5 of wheel load.

Therefore, Prud'Homme's finding related to track fatigue (or track deterioration) continues to be valid for the current design situations.

3.7 Though track engineers qualitatively know that freight wheel load causes more damage than passenger wheel load, the contents of the foregoing paragraphs will enable a better quantitative understanding. It would therefore appear, if the Track Engineer caters to the needs of freight traffic, the same may suffice for passenger traffic. But this is not entirely true, because track tolerances for passenger traffic are tighter for achieving ride comfort. This is the real challenge. The deterioration caused by heavy axle load should not affect the requisite track geometry for passenger traffic.

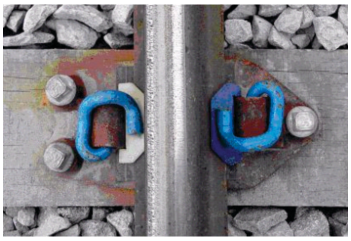
3.8 Here are the solutions to resolve the harmful effects of Dynamic Over-loading:

Principal aim should be to install and maintain the best possible track geometry, by way of creating structurally sound track structure, resting on good ballast and sub-ballast, supported on stable formation. Dynamic Over-loading is predominantly attributable to surfacing defects in vertical mode. But the track engineer should not ignore misalignment in horizontal mode, because it leads to lurches which in turn result in wheel over-loading. To minimise lurches, rolling stock engineer is required to closely monitor and rectify the wheel tyre profile, apart from ensuring proper functioning of suspension system and the track engineer is required to attend to alignment defects, especially in curves and turnouts.

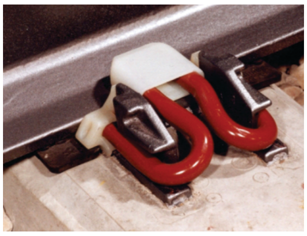
3.9 The Author feels that certain technological gaps have appeared between advanced railways of the world and Indian Railways. By bridging these gaps, laying and maintenance of track under mixed traffic conditions will be rendered more efficient. These are briefly indicated here.

3.9.1 Rail is the principal component of track and the improvement needed is dealt topic 4.

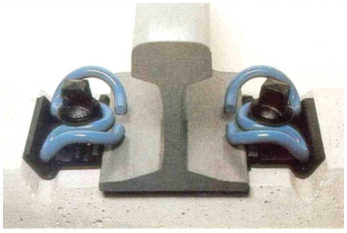
3.9.2 As regards fasteners, we are hanging on to ERC of 1970s design, though different versions of ERC have been successively introduced. But these clips do not sustain designated toe load, due to various reasons on Indian Railways. ERC is based on Pandrol 401, which has completely vanished in all other countries. Several types of fasteners, such as, Pandrol e-Clips, Pandrol FAST and Vossloh SKL Clamps are extensively used in tracks of different kinds in advanced countries – tramways, metro rail track, high-speed track, freight traffic track, slab tracks of various designs in open and tunnels. (Fig.2) Role of efficient fastener is to absorb vibrations, prolong retentivity of ballast packing, maintain alignment, control rail gap in case of fracture, etc. Excellent EN Specifications, namely, EN-13481 (7 Pts.) & EN-13146 (7 Pts.), have been evolved for performance requirement, manufacture and testing of elastic fasteners, embodying the benefits of investigations and experience from European railways. It is high time to turn towards such modern fasteners. It can be saistated, if someone develops a new fastener and if it passes through the tests successfully as prescribed in EN specification, the fastener is ready to be used in track.



Pandrol E-Clips



Pandrol FAST Clips



Vossloh SKL Clamps

Fig.2

3.9.3 As regards concrete sleepers, IR Specification T-39-85 has hit obsolescence. The sleeper is designed to resist prescribed moment of resistance as per IRS. Its relation to the Axle load, Dynamic augment and other parameters of stress control remains non-transparent. In contrast, rational design procedure, manufacturing process and quality control have been stipulated in Specifications

UIC-713-R and EN-13230-5Pts. These specifications enable optimal design for different service conditions. For dedicated freight corridor, these design principles are very important.

3.9.4 As regards upkeep of track geometry, the practice of trolley inspection should give way to scientific assessment and recording of track parameters. Presently, all Track Recording Cars are controlled by RDSO. It is learnt that, due to various kinds of bottleneck, TRC running schedule could not be strictly adhered to. Time is ripe to introduce self-propelled track recording cars and to keep them under the control of zonal railways. If Accelerometer cars can be operated by zonal railways, why not TRC? Some European companies have marketed self-propelled Track & OHE measuring cars, which are PC based, with reliable mechanism giving admirably accurate results.

4. Impact Wheel-overload

4.1 The largest impact loads in rail-wheel interaction arise from irregularities on wheel, such as wheel flat or from the wheel impinging on defective rail surface, such as, rail scab, depression at welded joint, etc. Advanced instrumentation is needed to investigate into this subject. ORE D-161 Committee and British Railway have done good work in this direction.

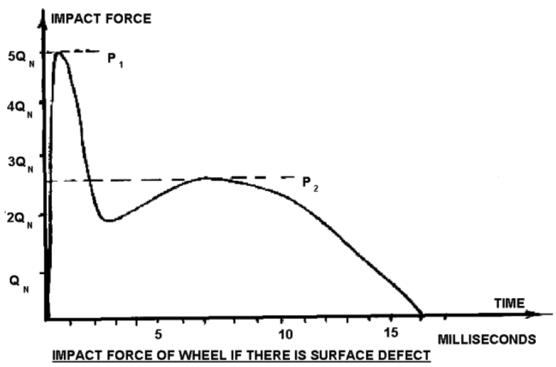


Fig.3

4.2 The impact mechanics will be explained now in a simplified form. Referring Fig.3, as a wheel passes over a rail surface defect, within 1 millisecond of the start of the impact, a force designated as P_1 of a very high magnitude equal to 4 to 6 times Q_N hits the rail. This impact force reduces soon; but it again attains a second peak value, at the end of 6-8 milliseconds from the start of the impact. The second peak force designated as P_2 is also of high order, may be equal to 2 to 3 times Q_N. The P_1 force decides the instantaneous contact stress generated in rail. P_1 increases with the speed of the wheel, whereas P_2 is more or less steady irrespective of the speed. The Amplification Factors Fa_1 and Fa_2, that is, (P_1/QN) or (P_2/Q_N) depend on the severity of wheel flat or rail defect. It is the P_1 force that must be considered as the instantaneous wheel load, for computing the correct values of contact stresses actually generated.

4.3 When rail-wheel contact pressure comes into action, the rail steel tends to heave out, similar to what happens if you step on a clayey ground. The maximum shear stress occurring at 4-6 mm below the contact surface is given by the formula

$$\text{(expressed in N/mm}^2\text{)} = 412\sqrt{(Q/r)}$$

where Q is the instantaneous wheel load (i.e. $Q = P_1$) in kN and r is the wheel radius in mm. It is important to note from this formula that contact stress can be reduced by increasing the wheel diameter. In fact, there are other complicated contact stress formulae giving more precise results. For the simplicity of explaining, easier formula has been selected; but it gives reasonably accurate result.

4.4 It is obvious that should not exceed the maximum permissible shear stress appropriate to the grade of rail steel. If ϵ is the UTS of rail in MPa (N/mm^2), should not exceed 0.38ϵ, according to material science. Applying this in the formula given in paragraph 4.3, we get the upper limit of Q, (that is, P_1 value) as $[8.507 \times 10^{-7} \times r \times \epsilon^2]$ expressed in kN.

4.5 This expression gives direct relation between permissible impact wheel load P_1 and wheel diameter depending on the UTS of rail steel. Taking two grades of rail steels with UTS of 900MPa and of 1175 Mpa, the following values emerge:

Vehicle	With UTS 900 MPa	With UTS 1175 Mpa
Electric Locomotive (2r=840mm)	$P_1 \leq$ 289.4 kN = 29.5 t	$P_1 \leq$ 493.3 kN = 50.3 t
Diesel Locomotive (2r=914mm)	$P_1 \leq$ 314.9 kN = 32.1 t	$P_1 \leq$ 536.7 kN = 54.7 t
Wagon (2r=780mm)	$P_1 \leq$ 268.7 kN = 27.4 t	$P_1 \leq$ 458.1 kN = 46.7 t

4.6 Having thus worked out the value of P1, value of Static Axle Load will have to be determined, which requires estimated value of Amplification Factor Fa1.

$Q_N = P_1/Fa_1$; Hence, Axle load = $2Q_N = 2P_1/Fa_1$

In the absence of research work carried out by RDSO regarding Fa_1, we may work out the values of Axle Loads assuming Fa_1 = 5, 4 and 3, successively. The results are tabulated:

Vehicle	F_{a1} assumed	Permissible Axle Load	
		For rail with UTS 900 MPa	For rail with UTS 1175 MPa
Elec. Loco	5	(2x29.5/5) = **11.8 t**	(2x50.3/5) = **20.1 t**
	4	(2x29.5/4) = **14.8 t**	(2x50.3/4) = **25.2 t**
	3	(2x29.5/3) = **19.7 t**	(2x50.3/3) = **33.5 t**
Dsl. Loco	5	(2x32.1/5) = **12.8 t**	(2x54.7/5) = **21.9 t**
	4	(2x32.1/4) = **16.1 t**	(2x54.7/4) = **27.4 t**
	3	(2x32.1/3) = **21.4 t**	(2x54.7/3) = **36.5 t**
Wagon	5	(2x27.4/5) = **11.0 t**	(2X46.7/5) = **18.7 t**
	4	(2x27.4/4) = **13.7 t**	(2X46.7/4) = **23.4 t**
	3	(2x27.4/3) = **18.3 t**	(2X46.7/3) = **31.1 t**

4.7 Now the tabulated results will be discussed. Research works on advanced railways have shown that the value of Amplification Factor Fa1 is expected to be 4-6. Hence the highlighted rows (with F_{a1}=5) are of practical importance to Indian Railways. Thus many track engineers will be surprised to find that rail of UTS 900 MPa is grossly inadequate to withstand the axle loads of Locomotive or Wagon, which are in the order of 20t. The initial design of track for heavy freight operation should comprise of rails of Grade R350HT or R350LHT (UTS 1175 MPa) complying with the Specification EN 13674-1. Only by doing this, frequent rail fractures, as happening now, can be avoided and the safety of both freight and passenger trains can be guaranteed. Last but not the least - Maintenance burden on permanent way staff and their mental agony can also be resolved.

4.8 As a locomotive starts or when brakes are applied on loco or wagons, rail scab is created. This is also a phenomenon of shear failure of rail steel. These rail scabs worsen the intensity of stresses under further wheel impacts and initiate rail fracture.

4.9 Question may arise – why not convert the rails of UTS 900 MPa as head hardened rail by heat-treatment process. Consequent to the investigation into the derailment near Hatfield in UK on 17.10.2000 killing 4 passengers, the technology of head hardening has disappeared at least one decade back. The author has seen two abandoned Head Hardening Plants in China. As such there is no current UIC or EN Specification for head hardening. Instead, the whole rail is hardened by special heat treatment as per EN 13674-1. Rails of Grade R350HT or R350LHT come under the "Fully Heat-treated" category as indicated by "HT". Therefore, instead of going in for head hardened rail, which may lead to accidents, it would be wise to adopt HT rails. UTS of these HT rails increases by about 10-15% more than1175 MPa in the region of head surface due to the heat treatment and this provides perfect solution to combat Rolling Contact Fatigue (RCF). Further these rails are more corrosion resistant too.

4.10 Here are the solutions to resolve the harmful effects of Impact Over-loading:

4.10.1 Methods to generate effective withstanding of the impact intensity:

- Use of rails of EN Grade R350HT or R350LHT (UTS 1175 Mpa)
- Welding of rails to superior quality, complying to EN-14587-1 for Flash-butt Welding and EN-14730-1 for Alumino-thermic welding

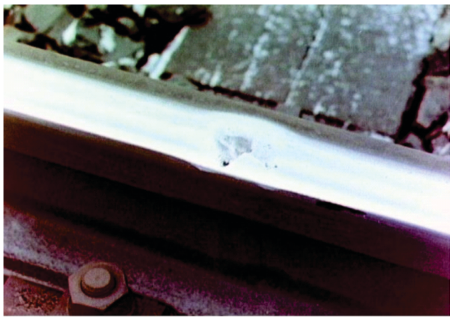
- Use of appropriate rail pads of requisite elasticity, such as EVA, HDPE, rubber, etc. depending on the relative mix of freight and passenger wheel loads, so that the effect of impact stops with rail pad

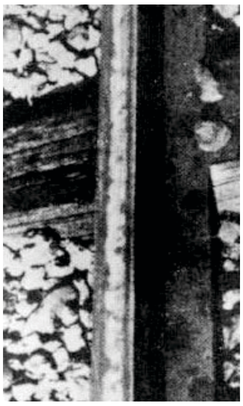
4.10.2 Methods to restrict the occurrence of Impact, (i.e., by keeping F_{a1} < 3-4):

- Rail Grinding Machine should be deployed periodically to eliminate surface defects on rail head and gauge face
- Wheel flats should be limited to smaller dimensions than those at present level
- Wheel tyre profiles should be maintained to strict tolerances. Specifically, false flange occurrence should not be allowed, since it causes continuous spalling of metal on rail head, ultimately leading to many types of fatigue failure.

Shear stress related rail defects and failures are illustrated here.


Isolated wheel burn


Local depression of rail head


Shelling, initial stage


Shelling, advanced stage

Gauge corner shelling

Continuous spalling due to false flange

Fig.4

5. Design Of Horizontal Curves In Mixed Traffic Lines

5.1 All the provisions in Chapter IV "Curved Track and Realignment of Curves" of IR Permanent Way Manual are applicable for the design of Mixed Traffic Lines. Specifications UIC-703-R and EN-13803-1 deal with track alignment design and the provisions therein are not at variance with respect to IRPWM. The clauses of IRPWM to determine Cant, Cant Deficiency, Cant Excess, Transition curve length based on 3 criteria (namely, time-rate of cant variation, time-rate of cant deficiency variation, geometric gradient of cant) should be entirely followed.

5.2 However, for mixed traffic section, the stipulation in Paragraph 406(b) is not specific for deciding the equilibrium speed to calculate actual Super Elevation to be provided. In fact, the stipulation requires Chief Engineer's intervention in deciding the most probable speeds of freight and passenger trains in each sub-section. But this is not being implemented in practice. Lecture notes of Training institutions suggest that 75% of passenger train's speed may be taken as equilibrium speed. This over-simplification resulting in lack of accuracy has entered into actual field design too. If the design equilibrium speed is more than the actual speed of freight trains, flattening of inner rail occurs. If the design equilibrium speed is less, side wear of outer rail occurs. Some UIC guidelines are given in paragraph 5.3.

5.3 The standard formula for super elevation is $C = GV^2/127R$. In this, $[V^2/R]$ represents the centrifugal force and this formula therefore shows that Cant is proportional to centrifugal force. Suppose N number of trains are running at different speeds, then $\epsilon[V^2/R]/ N$ will logically represent the average centrifugal force. For a given curve, R is constant. Hence only the average of V2 needs to be computed for applying in the formula $C = GV^2/127R$.

Suppose N^1 trains travel at speed V^1, N^2 trains travel at speed V^2, ...etc.,

Then the weighted average of V^2 will be $(N_1V_1^2 + N_2V_2^2 + N_3V_3^2 + ...) / (N_1 + N_2 + N_3 + ..)$

This will therefore be V_E^2, where V_E represents equilibrium speed.

This principle of computing V_E^2 is used in UIC method, with further refinement, by taking into account the weights of various trains, since rail damage caused by any one train is proportional to its gross weight.

Thus $V_E^2 = \{W_1N_1V_1^2 + W_2N_2V_2^2 + W_3N_3V_3^2 + ..\} / \{W_1N_1 + W_2N_2 + W_3N_3 + ..\}$

(Instead of actual weights of trains, proportional weights may be substituted in this formula.)

This methodology appears to be logical and may be applied in conjunction with the provisions in paragraph 406(b) of P. Way Manual. It will be necessary to divide the section into appropriate sub-sections. In sections of continuous long gradient, UP and DOWN sections should be treated as sub-sections for applying these formulae. Computer Simulation of Train Operation can be advantageously used to get the values of V^1, V^2, etc.

5.4 For mixed train operation, it is important to limit Cant Excess as 75mm, vide Paragraph 406(3). After deciding VE as explained above, Cant Excess is calculated as:

Cant corresponding to V_E – Cant corresponding to the actual speed of freight train in the curve under consideration. (This is an improvement over the present stipulation in IRPWM.)

6. Issue Of High Centre Of Gravity In Double Stack Container Transport

6.1 For normal BG freight stock, the overall size of Maximum Moving Dimension is (B)3300mm x (H)4265mm. Height of CG above rail level is nearly 1800mm.

6.2 For the Western Dedicated Freight Corridor, designed for double stack container traffic, the size of Maximum Moving Dimension is 3660mm x 7100mm. For the Eastern Freight Corridor, the same is 3600mm x 5100mm. The height of CG above rail level for Western Corridor is stated to be 2600mm.

6.2 The Author has made some calculations, in line with the practice of (erstwhile) Japanese National Railways (JNR), to compare the stability of Double Stack Container Stock with that of normal BG stock.

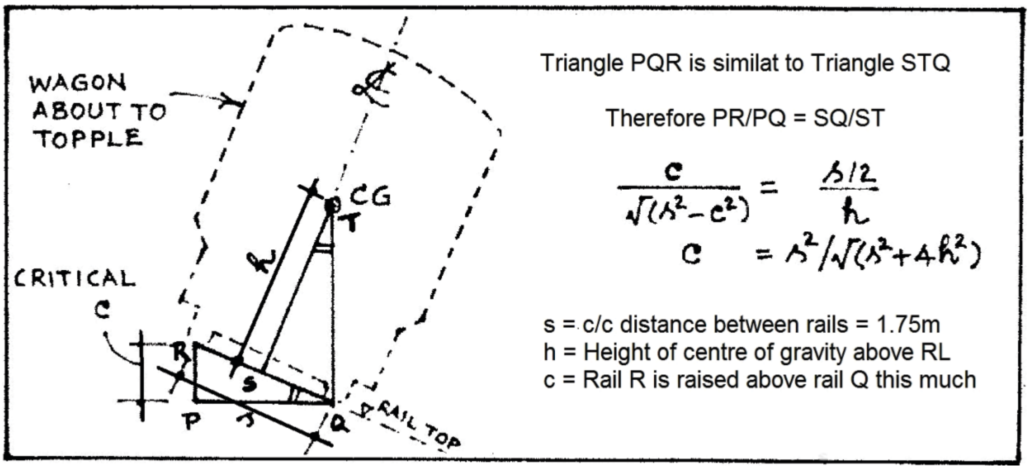


Fig. 5

In Fig.5, the rolling stock is tilted by raising rail R above the other rail Q. Tilting is continued till the wagon reaches the critical condition of toppling, i.e., the vertical from CG falls at the centre of lower rail Q. At this instance, vertical difference between the rails is c.

The computation in Fig.5 shows that the critical value of

$c = s^2/\sqrt{(s^2 + 4h^2)}$, where s=1.75m
For normal BG stock with h = 1.8m, c = 0.765m = 765mm
For double stack wagon with h = 2.6m, c = 0.558m = 558mm

JNR had prescribed that maximum cant in a curve should not exceed c/3, so as to ensure a factor of safety of 3 against toppling of a slow moving vehicle, considering any unfavourable wind / earthquake force, lateral sliding of merchandise carried, abrupt coupling forces, etc.

Therefore, maximum permissible cant can be 255 mm for normal BG stock and 186 mm for Double stack container wagon.

Maximum cant permissible as per IRPWM is 165mm. This is acceptable for both kinds of rolling stock. However, the following precautions should be taken in the case of double stack container wagon:

- The merchandise carried should be tied/held to prevent sliding under wagon-lurch.
- Loaded container should not be stacked above empty container by mistake.

6.3 That is not all. On-loading and Off-loading of wheels (namely as $Q+\Delta Q$ and $Q-\Delta Q$) will be more pronounced in the case of double stack wagon, compared to normal wagon.

If F is the lurching force acting horizontally at CG of wagon, the ΔQ = Fh/s. In other words, for the same lurch-force, ΔQ will be proportional to h. Therefore, ΔQ in the case of double stack wagon will be 1.44 times (=2.6/1.8) that in the case of normal wagon.

6.4 Track engineers should be thus informed that variation of vertical forces on track will be 44% more in the case of double stack wagons than that for normal BG wagon. This will increase the track loading on one hand and increase derailment possibility due to off-loading on the other hand.

7. Minimum Distance To Be Provided Between Adjacent Grade-change Points In Freight Operation Line

7.1 A few months back, in the Western Dedicated Freight Corridor Project, the Design Consultant suggested that a minimum distance of 750m (one freight train length) may be provided between adjacent grade-change points, as a good practice. This will ensure that the train will not travel over more than two grades at any instance, so that the travel is jerk-free and the coupling forces do not fluctuate violently within the train length. The Project Authority did not accept this suggestion, since this will increase the project cost, apart from exceeding the scope of agreement with the constructing company. Coincident with this decision making, the Author gave a technical note to the Chief Project Manager / Vadodara responding to his informal talk. This note concluded that the Design Consultant's suggestion need not be implemented even from technical consideration. This Analysis will be useful to all track engineers. Also this fills in a void in para 419 'Vertical Curve' of the IRPWM.

7.2 Double stack container trains are operated in USA, Australia and China. Here is the photo of such a train travelling over a hilly terrain in USA:

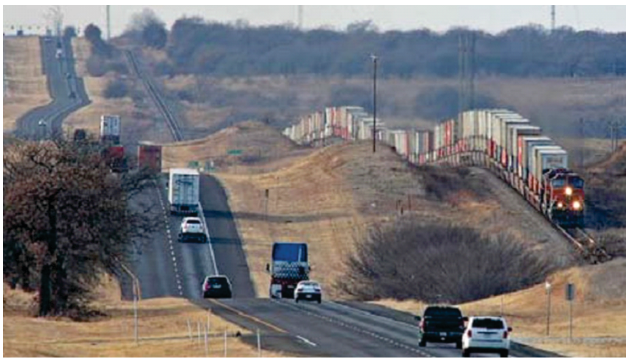


Fig.6

The photo from USA in Fig.6 provides evidence that a train can safely travel over multiple gradients within its length. Chapter 6 of 'Railway Track Design' by AREMA (American Railway Engineering and Maintenance-of-Way Association) does not directly stipulate the minimum distance between grade-change points; but it gives some rational stipulations for vertical curves separately for passenger and freight lines, from which this minimum distance can be derived. IRPWM may need similar clauses.

7.3 Let us take the case of a high-speed passenger train running at 300 km/h, because it is easy to visualize the effect of grade change on the riding of the train. Assume that there is a crest point where a rising grade of 1:100 is followed by a falling grade of 1:100. A vertical curve of radius R (m) will have to be provided for the track profile. Let the speed of train be v (m/sec). A passenger travelling in the train will be subjected to a vertical acceleration of v^2/R (m/sec^2) in upward direction. That is, he will experience a weight loss of $(W/g).v^2/R$, where W is his weight. If however the train passes over a similar 'valley profile', he will experience a weight addition of $(W/g).v^2/R$. This reduction or addition of weight causes passenger discomfort. For freight train, this situation may lead to vertical forces acting at couplers, sometimes leading to train parting. In case sensitive goods such as chemicals or costly goods such as cars are carried, the vertical acceleration may cause damages.

7.4 IRPWM has prescribed the following values of radius of vertical curve:

Route Classification	R
Group A	4000m
Group B	3000m
Groups C, D & E	2500m

The rationale for stipulating the value of R is not given in IRPWM. UIC, EN and AREMA documents have prescribed the limiting value of acceleration v^2/R instead of R.

7.5 The recommended value of acceleration adopted for passenger train over European Railways is 2% of g = 0.02x9.81 m/sec^2 = 0.1962 m/sec^2 (0.64 ft/sec^2). Exceptionally, this limit of 0.02g is increased up to 0.06g on SNCF. AREMA has prescribed 0.6 ft/sec^2 (1.86% of g) for passenger trains and 0.1 ft/sec^2 for freight trains. The reason for prescribing significantly lesser value of acceleration for freight train (which will result in far greater value of R) may be due to the limitation of coupler performance. Further, adoption of 0.1 ft/sec^2 (0.0305 m/sec^2) will reduce the proportional off-loading of wheel load, eliminating derailment possibility. Therefore, it would be apt to adopt the limiting value as (i.e., 0.1 ft/sec^2) for freight lines on IR. Freight train derailing at valley location due to bunching of wagons is not uncommon and therefore a large radius for vertical curve will be a desirable feature.

7.6 AREMA formulae for the radius R of vertical curve and its length L, when converted to metric units, are as follows: (Refer Fig.7)

$R = L/D$

$L = 0.07716\ V^2D / A$; (can be derived from the basic formula, i.e., Acceleration = v^2/R)

Where D = algebraic difference of grade in decimal, V = train speed in km/h,

A = vertical acceleration = 0.0305 m/sec^2 for freight, 0.183 m/sec^2 for passenger or transit

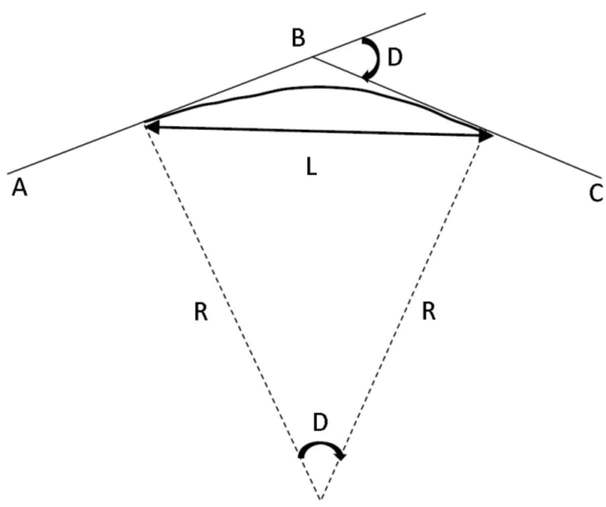


Fig. 7

7.7 Taking the example of a freight line with ruling grade 1:200 (0.5%), let us assume that a grade of +0.5% is followed by a grade -0.5% (crest profile).

D = 0.005 - (-0.005) = 0.01; A=0.0305

L = 0.07716 x 1002 x 0.01 / 0.0305 = 253 m. We may provide 250m.

R = L/D = 250/0.01 = 25000 m, which is far greater than the range of 2500-4000m prescribed in IRPWM.

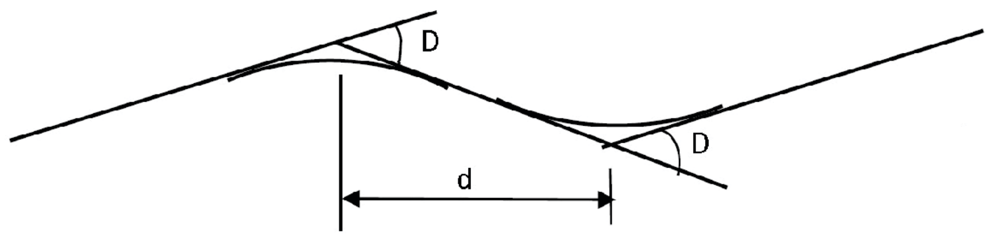


Fig. 8

7.8 On this basis, let us calculate the minimum distance between adjacent grade-change points. Assume, as shown in Fig.8, the most unfavourable situation of one crest followed by the next valley, each of D=0.01. If there is no straight between the vertical curves, the distance d between crest and valley will be L/2 + L/2 = L = 250m. It is desirable to provide a straight profile between the tangent points, equal to 2 second travel length, which is 2x(100/3.6) = 55.5m or say, 60m. Provision of such a non-curved profile enables stabilizing of wagon oscillations as it enters from + grade to - grade. Thus the distance d between crest and valley will be 250+60 = 310m.

7.9 Had we adopted R=3000m as per IRPWM, the distance between crest and trough would have been calculated as (0.01x3000) + 60 = 90m. The rational approach as per AREMA and other international practice, has yielded a value of 310m, instead of mere 90m as per IRPWM or 750m (one train length) as per the Consultant's suggestion referred in para 7.1. The optimised solution will enhance safety and smooth travel of the long train.

7.10 In case the minimum distance between adjacent grade-change points cannot be provided as 310m, due to site restriction or due to cost consideration, this distance can be recalculated, in site-specific manner, taking into account the actual values of gradients.

7.11 The following restrictions in locating the grade-change point should however be observed:

- This should not infringe with turnout zone
- This should not interfere with the transition length of horizontal curve.

7.12 It is thus summarised that the radius of vertical curve should not be prescribed arbitrarily; but based on limiting the vertical acceleration, appropriate for passenger or freight traffic. Minimum distance between adjacent grade-change points should also be decided, following the above principle, on site-specific basis, considering the speed and gradients of the vertical profile.

8. Conclusion

Unlike a Report compiled by a Committee assigned with specific investigation, it is not possible to summarise the findings or recommendations with serial numbers. This Paper has hovered over many topics, packing technical information in each page. The objective is to seed interest in the minds of the track engineers and to achieve clear understanding. Also, opportunity has been availed to bring to the kind attention of this Forum and Decision Makers that a wide gap has been allowed to grow between IR and other advanced railways of the world. We should continuously watch for the advancements abroad and straightaway implement the good practices instead of inventing wheel every time. The Author will feel contended, if the thought process initiated in this Paper is taken forward to bring in essential advancement in track technology and alleviate the challenges faced by the field engineers in maintaining track carrying heavy freight traffic along with semi-high speed passenger traffic.

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