Dynamic vehicle response versus virtual transitions

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Abstract

The idea of transition curves was first presented by Pressel in 1854, and first used in the construction of the Brenner Railway 1864-1867. Nowadays, the use of transition curves in horizontal curves is common practice, both between a circular curve and an adjacent straight, and between circular curves with different radii. The transition curves increase the time span for changing lateral acceleration perceived by the passengers, hence reducing lateral jerk.

However, there are still curves (designed or existing) without transition curves. One of the reasons for omitting the transition curves may be that there is no change of cant between two elements with different curvature and lateral jerk is based on the assumptions of virtual transitions.

The concept of virtual transition is as follows. As long as both bogies of a vehicle are on a straight track, the lateral acceleration is zero. When both bogies are on a circular curve, the vehicle is subject to full lateral acceleration. In between these two extremes, when only one bogie has entered the circular curve, lateral acceleration is assumed to increase linearly (with chainage). Hence, the lateral jerk becomes (the change of) lateral acceleration multiplied with train speed and divided by bogie king pin spacing.

Dynamic vehicle response, when entering horizontal curves where transition curves are omitted, has been quantified in computer experiments with software for the dynamic interaction between vehicles and track. The simulations show that vehicle response does not correspond to the above assumptions for virtual transitions. The simulations also indicate that the wheel/rail forces have unfavourable peak values where the radius is changed without transition curves.
Keywords: transition curve, virtual transition, vehicle dynamics, lateral jerk.

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1 Background

The idea of transition curves was first presented by Pressel 1854, and first used in the construction of the Brenner Railway 1864-1867 [1]. Nowadays, the use of transition curves in horizontal curves is common practice, both between a circular curve and an adjacent straight, and between circular curves with different radii. The transition curves increase the time span for changing lateral acceleration perceived by the passengers, hence reducing lateral jerk. Transition curves are also the normal places for arranging the superelevation ramps, if cant is to be changed between two track elements with different curvature. The most common type of transition curve is the clothoid, where curvature changes linearly with chainage.

However, there are still curves (designed or existing) without transition curves. The reasons for omitting the transition curves may be:

1. There is only a small change in lateral acceleration (curve with large radius connected to straight, or two circular curves with close radii). Since the change of lateral acceleration is small, no attention is paid to the time span for changing it.
2. There is no space for transition curves. This may be the situation at the close proximity to turnouts.
3. There is no change of cant between the elements with different curvature and lateral jerk is not considered relevant.
4. There is no change of cant between the elements with different curvature and lateral jerk is based on the assumptions of virtual transitions.

The concept of virtual transition is as follows. As long as both bogies of a vehicle are on the straight track, the lateral acceleration is zero. When both bogies are on a circular curve, the vehicle is subject to full lateral acceleration. In between these two extremes, when only one bogie has entered the circular curve, lateral acceleration is assumed to increase linearly (with chainage and time). Hence, the lateral jerk becomes (the change of) lateral acceleration multiplied with train speed and divided by bogie king pin spacing. The concept is used in a similar manner between circular curves with different radii.

National track standards vary considerably. Earlier German track standards [2] permitted an instantaneous change of cant deficiency of 106 mm if train speeds are 100 km/h or less. In the speed interval 105-160 km/h, the limit was 82 mm, and at speeds above 160 km/h, the limit was 47 mm. For *main tracks*, current German track standards [3] permit an instantaneous change of cant deficiency of 40 mm if train speeds are 200 km/h or less. At speeds above 200 km/h, the limit is 20 mm. For *sidings*, the requirements in [3] are the same as in [2], for permissible train speeds less than 100 km/h. For permissible train speeds in the interval 100-200 km/h, permitted instantaneous change of cant deficiency is interpolated between the values mentioned above, and for permissible train speeds above 200 km/h, the limit is reduced even further.

Swedish track standards [4] accept an instantaneous change of cant deficiency of 25 mm on *main tracks*. The preferred maximum of instantaneous

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change of cant deficiency is as low as 9 mm. The limit has been set very low in order to ensure that the alignment has potential for increased permissible speeds for conventional non-tilting trains as well as enhanced speeds for tilting trains (a total of 65%). For *sidings*, the limit is 100 mm if permissible train speed is 100 km/h or less. In the interval 105-160 km/h, the limit is 80 mm, and in the interval 165-200 km/h, the limit is 60 mm.

German standards for tramways BOStrab [5] and British track standards [6] make use of the concept of virtual transitions. BOStrab [5] has a limit of 0.67 m/s^3 for lateral jerk, corresponding to a rate of change of cant deficiency of 100 mm/s. The British track standards [6] specify that the bogie king pin spacing shall be assumed to be 12.2 m. The limit for rate of change of cant deficiency (proportional to lateral jerk) in [6] is 70 mm/s on plain line and up to 95 mm/s on switches & crossings. However, according to [7], permissible speed on Gv24 turnouts (radius 1650 m) is 112 km/h. Hence, at the tip of the switch, rate of change of cant deficiency over the virtual transitions may become 229 mm/s.

Recent simulations of dynamic vehicle response on turnout curves resulted in lateral jerk values that differed considerable from those calculated according to the assumptions of virtual transitions [8].

A literature review showed that the concept of virtual transitions was published as early as 1936 [9]. The vehicle kinematics was illustrated together with the instantaneous centre of rotation. It was shown that the distance from the vehicle to the instantaneous centre of rotation was inversely proportional to the relation between the distance the front bogie has travelled along the circular curve and the bogie king pin spacing. However, lateral acceleration is proportional to the inverse of the instantaneous centre of rotation *only* if the centre is a fixed point in the space or moving at constant speed. In the case of entering a circular curve, the centre is accelerating. Hence, the lateral jerk will not be constant during the curve entry.

Railtrack's Track design handbook [7] states that the vehicle gradually acquires angular (yaw) velocity after the first bogie has passed a tangent point, and that the change continues until the second bogie reaches the tangent point, after which the vehicle moves around the curve with uniform angular velocity. The statement is correct from kinematic point of view. However, the lateral jerk (and the rate of change of cant deficiency) is not related to *yaw acceleration*. Lateral jerk is the rate of change of lateral acceleration. If the second bogie is not subjected to changes in lateral acceleration, the lateral jerk (of other parts of the vehicle) may be calculated as *yaw jerk* (rate of change of yaw acceleration) multiplied with longitudinal distance to the second bogie. It should also be noted that yaw acceleration does not vary with speed if a vehicle enters a curve with a certain amount of cant deficiency.

2 Simulations of dynamic vehicle response

As in earlier studies, [10], [11], [12], [13], it has been concluded that different alignment alternatives should be evaluated through simulations of the dynamic vehicle response. Simulations are less expensive than full-scale tests and also

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make it easier to control background variables such as vehicle speed, friction conditions, track stiffness and track irregularities.

The vehicle response has been simulated with the multibody code GENSYS. Each vehicle was modelled with seven rigid bodies, each of which has six degrees of freedom (translational and angular displacement in three directions each). The track was modelled as a lumped mass connected to each wheelset, each body having one degree of freedom: lateral displacement. Springs and dampers in the models are non-linear and the simulations are conducted in the time domain.

Vehicle response have been quantified for three types of vehicles: A Eurofima coach (a conventional passenger coach with rather stiff primary suspension, used by several railway companies in Europe), an SJ UA2 coach and an SJ X2 power car, see Figures 1a-1b. The two Swedish vehicles are used in the X2000 trainsets and have relative soft primary suspensions. At higher speeds, the UA2 coaches tilt on curves. At speeds lower than 80 km/h, the tilt system is inactive. In the present study, the tilt system is switched off at all speeds.

The track model has UIC60 rails inclined at 1:30 (according to Swedish standard) and the wheel models have UIC/ORE S1002 wheel profiles. In the present study, no track irregularities are superimposed on the design alignment and cant, and the track model contains no stiffness variations.

Figure 1: A Eurofima coach (left) and an X2000 trainset (right).

A description of the GENSYS software, a summary of various validation studies and a selection of vehicle data are published in Kufver [12].

3 Variables

Dynamic vehicle response is quantified according to international standards from CEN [14], [15], and UIC [16]. An analysis of the differences between evaluating standard vehicles on different track designs and evaluating different vehicle designs on standard track alignments is presented in Kufver [12]. The most important difference is that CEN and UIC recommend extensive tests on rather long (in certain cases at a minimum 500 m) track sections with constant cant and curvature, while the analysis of different track geometries is often focused on

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track sections where cant and curvature vary (comparing different lengths of transition curves, comparing different types of transition curves, etc.).

In the present study, the following variables have been evaluated:

1. Max. vertical wheel/rail force (*Max Q*), 30 Hz low-pass filtered (kN).
2. Min. vertical wheel/rail force (*Min Q*), 30 Hz low-pass filtered (kN).
3. Max. guiding force (*Max Y*), 30 Hz low-pass filtered (kN).
4. Max. track shift force (*Max* ΣY), 30 Hz low-pass filtered (kN).
5. Max. wheel/climbing ratio (*Max Y/Q*), 30 Hz low-pass filtered (-).
6. Max. lateral acceleration, 2 Hz low-pass filtered (m/s^2).
7. Max. lat. acceleration, 2 Hz low-pass filtered and averaged during 1 s (m/s^2).
8. Max. lat. acceleration, 0.5 Hz low-pass filtered (m/s^2).
9. Max. lat. jerk, based on Variable 7 and a time differential of 0.1 s (m/s^3).
10. Max. lat. jerk, based on Variable 7 and a time differential of 1.0 s (m/s^3).
11. Max. lat. jerk, 0.3 Hz low-pass filtered (m/s^3).
12. Max. roll velocity, 2 Hz low-pass filtered (rad/s).
13. Max. roll velocity, Variable 12 averaged during 1 s (rad/s).
14. Max. friction work in the wheel/rail contact patch (Nm/m).

Variables 1, 3-5, 7, 10 and 13 are suggested by CEN [15] and UIC [16]. Variables 7, 10 and 13 are suggested by CEN [14]. Variables 8 and 11 were used in earlier Swedish evaluation procedures.

Variable 2 is included since wheel unloading have been associated with derailments, and Variables 6, 9 and 12 are included too, since they register more transient motions than Variables 7, 10 and 13. Variable 14 is believed to be correlated to rail wear.

4 Lateral jerk on entrance of a circular curve

The vehicle response has been calculated for the following alignment cases. The vehicle starts on a straight and runs with constant speed into a circular curve with 100 mm of cant deficiency. Hence, the radius is varied according to Table 1.

Resulting jerk values for the three different types of vehicles are shown in Figures 2-4. The jerk variables are filtered according to Chapter 3 and are calculated for each vehicle body above first and rear bogie, and in the middle of the vehicle body, respectively. The solid curves (w) represent the worst position in the vehicle and the dotted curves (m) represent the centre of the coach. The thick straight line is the lateral jerk according to the concept of virtual transitions, calculated with the actual bogie king pin spacing for each vehicle.

The simplification according to virtual transitions seems to give a reasonably correct estimation of the lateral jerk in the *centre of the vehicle body* at a train speed of 40 km/h. However, the gradient of the simplified function is entirely wrong, and at slightly higher speeds it corresponds rather to the *worst position in the vehicle body*. At even higher speeds, the simplified function greatly overestimates the lateral jerk.

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Table 1: Curve radius as a function of train speed in the simulations.

Speed (km/h)	40	50	60	70	80	90	100	130	160	190	220
Radius (m)	189	295	425	579	756	956	1180	1995	3021	4260	5712

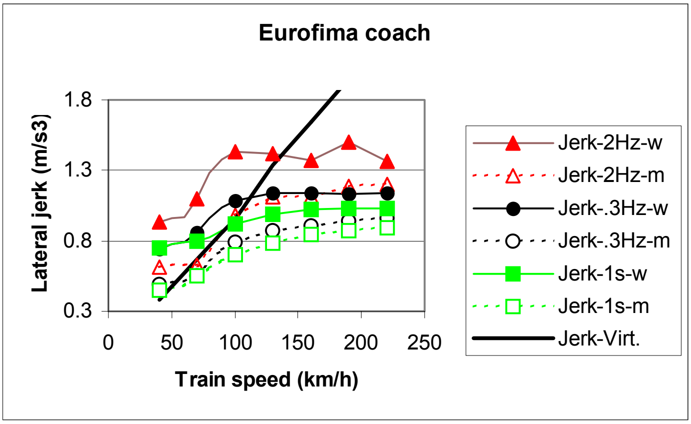

Figure 2: Lateral jerk in the vehicle body of the Eurofima coach when entering curves with an instantaneous change of cant deficiency of 100 mm.

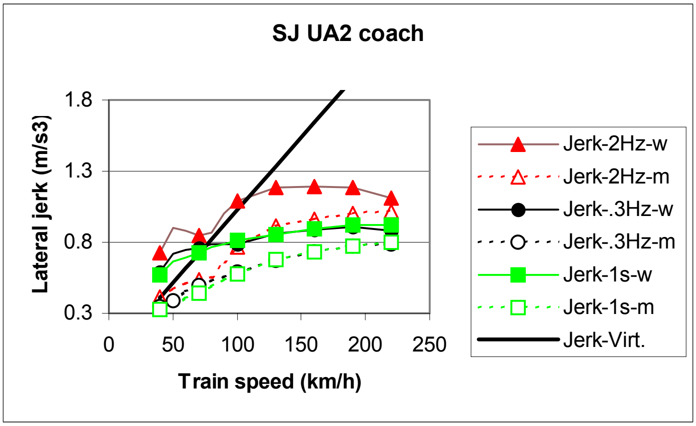

Figure 3: Lateral jerk in the vehicle body of the SJ UA2 coach when entering curves with an instantaneous change of cant deficiency of 100 mm.

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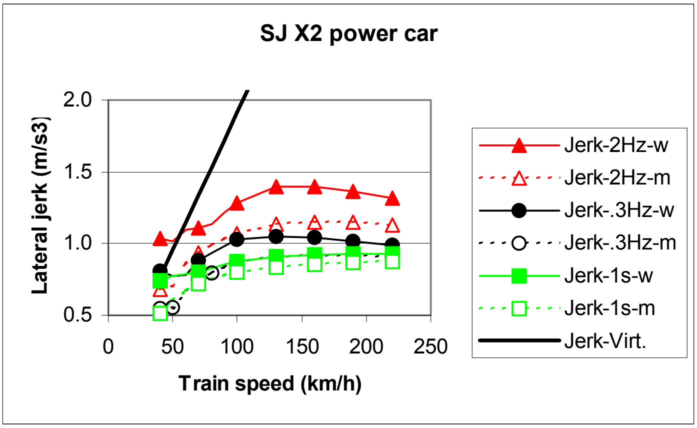

Figure 4: Lateral jerk in the vehicle body of the SJ X2 power car when entering curves with an instantaneous change of cant deficiency of 100 mm.

5 Other variables on entrance of a circular curve

The concept of virtual transitions is used for a simplified calculation of lateral jerk where transition curves are omitted. However, the simulations in this study also quantified other traditional evaluation variables.

The lateral wheel/rail-forces were of particular interest. When wheelsets run on a circular curve under steady state conditions, they will move laterally (in relation to track centre line) and take an angle of attack (angle between the wheel direction and the direction of the rail). The displacements depend on factors such as curve radius, suspension between wheelset and bogie, wheel and rail profiles, friction conditions etc. When entering the curve, there will be a transient change from the steady state position on the straight to a new steady state position on the curve.

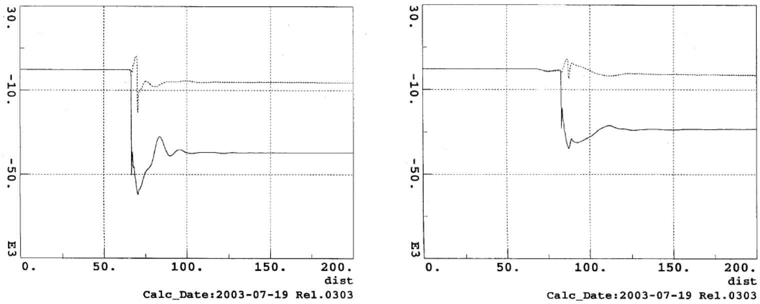

Figure 5: Lateral wheel/rail forces, plotted against chainage, for the two external wheels in the first bogie when entering a curve with a 295 m radius. SJ X2 power car (left) and SJ UA2 coach (right).

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Table 2: Dynamic vehicle response. A selection for the Eurofima coach.

Variable	Radius	189	295	425	579	756	956	1180
	Speed	40	50	60	70	80	90	100
1	Max Q	74.0	72.9	70.4	69.2	70.2	71.4	72.4
2	Min Q	37.7	37.9	39.3	39.0	37.4	36.2	36.3
3	Y	42.4	34.0	29.5	25.5	21.9	19.4	17.4
4	ΣY	26.9	20.0	14.5	14.5	17.6	19.5	20.5
5	Y/Q	0.64	0.57	0.50	0.44	0.38	0.34	0.30
6	Acc	0.95	0.98	0.97	1.07	1.20	1.30	1.37
7	Acc	0.85	0.93	0.89	0.90	0.92	0.95	0.98
8	Acc	0.90	0.96	0.92	0.92	0.92	0.95	0.98
9	Jerk	0.94	0.96	0.97	1.10	1.28	1.38	1.43
10	Jerk	0.75	0.78	0.79	0.75	0.82	0.88	0.92
11	Jerk	0.75	0.78	0.79	0.86	0.96	1.03	1.08
12	Roll	0.026	0.027	0.028	0.028	0.028	0.028	0.028
13	Roll	0.015	0.016	0.016	0.015	0.015	0.017	0.019

The simulations show that the transient change increase the dynamic lateral wheel/rail forces and the friction work in the wheel/ rail interface compared to the steady state values on the circular curve.

Figure 5 shows the lateral wheel/rail forces (in kN) for two wheels of the SJ X2 power car and the SJ UA2 coach, when entering a small radius curve without transition curves. For each curve entry, there is an unfavourable peak load. Table 2 shows a selection of the vehicle response for the Eurofima coach.

6 Discussion and conclusions

The concept of virtual transition seems to be based on faulty assumptions on kinematics.

Computer simulations of vehicle dynamics (for three different types vehicles) result in entirely different patterns for the values of lateral jerk (low-pass filtered with three different methods) than those proposed by the concept of virtual transitions. In fact, none of the evaluation variables increase linearly with speed.

Furthermore, lateral wheel/rail forces show a rather unfavourable pattern where real transition curves are missing.

Hence, an important practical conclusion is that transition curves should be used wherever possible.

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