Design of a high speed track

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With the increase of train speeds, requirements to the track geometry become more and more important in the design of a new high speed line, since safety should be guaranteed and a high level of comfort is desired. Contrary to safety, which can be indicated by the wheel-rail forces in the track plane, the level of passenger ride comfort is determined by the motions of the passenger within the vehicle. Current geometry standards are based on a quasi-static consideration of a vehicle in a horizontal curve and do not take into account the characteristics of the rolling stock, track and substructure. A three-dimensional simulation program was built in order to simulate the dynamic behaviour of a high-speed train as a result of the track alignment and to determine the level of passenger ride comfort by calculating the accelerations of the coach.

When evaluating track alignment and the influence of the rolling stock, these simulations show that the dynamic behaviour of the vehicle is not only determined by the amount of non-compensated lateral accelerations or cant deficiency / excess, but also by the geometry of superelevation ramps and the type of transition curves.

Key words: **design, alignment, computer modelling, passenger comfort, transition curve, cant**

1 Introduction

In order to offer an ecological alternative to short-distance air travel and to meet the growing demand for mobility, The Netherlands will be linked to the European high-speed rail network. Currently, high-speed trains are using existing track. As existing lines reach their capacity and are not designed for speeds over 200 km/h, it was decided for the HSL-Zuid to build a whole new track, running from Amsterdam/Schiphol via Rotterdam to Antwerp (Belgium).

Especially on high-speed lines, travellers should be offered attractive journey times and a high level of comfort. Furthermore, the construction of a high-speed line requires social acceptance, which can only be obtained if the line is integrated within the landscape and 'bundled' with other infrastructure. These factors, combined with the fact that because of the high speeds the curves should have a larger radius and therefore cover a long(er) length, result in the paradoxical situation that the horizontal alignment of the HSL-Zuid has hardly any straight parts.

Taking the vertical alignment in consideration, a continuous variation in the position in relation to ground level can be distinguished (elevated, on an embankment, semi-sunken or in a tunnel). This means great variation in (combinations of) horizontal and vertical curves, which leads to the question what restrictions are to be set to the curvature of the alignment of a high-speed line seen from the point of view of comfort. Research on this topic has been carried out at Delft University of Technology by order of the High-Speed Line Project Organisation. In this organisation, NS-Railinfrabeheer, Holland Railconsult and DHV work together under the responsibility of the Ministry of Transport, Public Works and water Management.

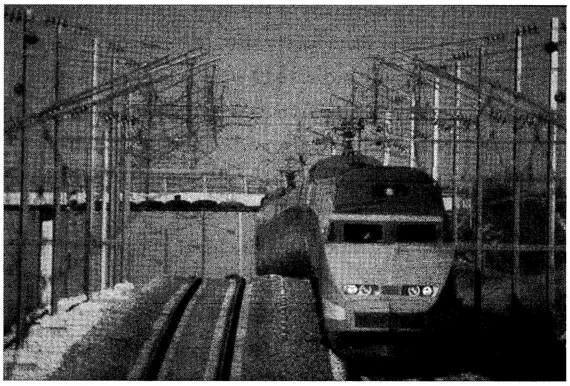


Fig. 1. TGV in a combined horizontal and vertical curve.

To answer the question better understanding of comfort is needed. The next paragraph will give insight in the relation between track alignment and the experience of comfort by passengers in the train. A method to predict the level of comfort is discussed. The third paragraph deals with the way new lines are designed and it compares standards currently used in The Netherlands to European standards. In paragraph four the computer model and calculation method are briefly explained. In the following paragraphs results of calculations on several cases are shown. Comparison is made between the quasi-static and dynamic effect of elevation ramps for several types of transition curves. The final paragraph draws some conclusions on the basis of this research.

2 Comfort

Comfort has both physiological and psychological components. As people are different, comfort-experience is a very subjective phenomenon. In the past, a lot of research is done on the way the human body experiences comfort and the determining factors [1]. As discomfort can occur in a lot of situations, this article only deals with passenger riding comfort, defined as the feeling of a person who is exposed to vehicle movements and which is influenced by dynamic, ambient, spatial and human factors.

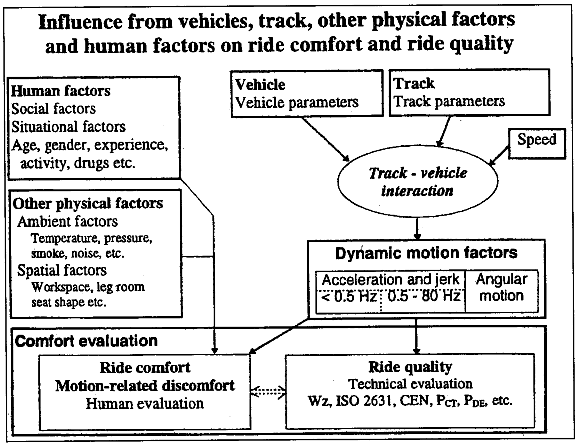


Fig. 2. Relation between track geometry and passenger comfort, taken from Förstberg [1].

Normally, only the dynamical movements (vibrations) of the vehicle are considered, since they can be quantified objectively and reduction of these vibrations is an important issue within railway engineering. These vibrations are usually described in terms of accelerations when considering the influence on the human body. In the standard ISO 2631 [2] there are guidelines for assessment of discomfort caused by motions. It considers frequencies, directions and duration of the accelerations and gives a method for combining the motions in different directions. Furthermore, in ISO 2631 distinction is made between the effect of vibrations on health, comfort and perception and the effect on motion sickness.

For vibration evaluation, ISO 2631 prescribes the use of the weighted root-mean-square (r.m.s.) acceleration, defined as the following equation or its equivalent in the frequency domain:

$$a_{\mathrm{w}} = \left[\frac{1}{T}\int_{0}^{T} a_{\mathrm{w}}^{2}(t)\,dt\right]^{\frac{1}{2}} \qquad [\mathrm{m/s^2}]$$

where a_{w} is the weighted acceleration (translational [$\mathrm{m/s^2}$] or rotational [$\mathrm{rad/s^2}$]) as a function of time and T is the duration of exposure [sec]. As a human being experiences vibrations in x, y and z-direction differently, frequency-weighting curves are given (see figure 3).

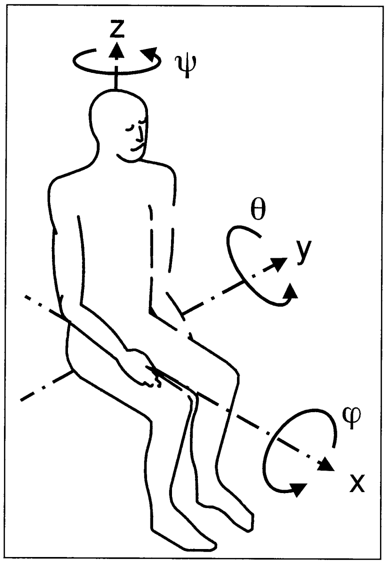


Fig. 3. Directions of motion for a human being.

The vibration total value of weighted r.m.s. acceleration a_v, determined from vibration in orthogonal coordinates, is calculated as follows:

$$a_v = (k_x^2 \cdot a_{wx}^2 + k_y^2 \cdot a_{wy}^2 + k_z^2 \cdot a_{wz}^2)^{\frac{1}{2}} \qquad [\mathrm{m/s^2}]$$

where a_{wx}, a_{wy} and a_{wz} are the weighted r.m.s. accelerations with respect to the orthogonal axes x,y and z and k_x, k_y and k_z are multiplying factors (which should be taken as 1 in the case of comfort).

As acceptable values of vibration magnitude for comfort depend on many factors which vary with each application, ISO 2631 only gives indications of likely reactions to various magnitudes of overall vibration total values in public transport:

Table 1. Human reaction on vibration magnitude.

a_v [m/s^2]	Reaction
< 0,315	Not uncomfortable
0,315 to 0,63	A little uncomfortable
0,5 to 1	Fairly uncomfortable
0,8 to 1,6	Uncomfortable
1,25 to 2,5	Very uncomfortable
> 2	Extremely uncomfortable

For evaluation of railway alignment design, two types of comfort experiences normally are distinguished, which are caused by motions. The first is a direct comfort disturbance caused by strong lateral acceleration or sudden movements of the vehicle due to low-frequency motions such as jerks and jolts. The other is discomfort due to prolonged low-frequency linear and angular motions, which results in dizziness, headache or nausea. In this paper, only the relation between alignment design and the first will be discussed. Therefore, the most important physical quantities to consider are lateral acceleration, lateral jerk and roll velocity, since they are dependent on track quantities such as curve radii, lengths of transition curves and cant gradients on superelevation ramps [1].

3 Geometry

For alignment considerations a railway vehicle traditionally is modelled as a mass point, which moves along the centreline of the track. If it is running through a curve, it is exposed to a lateral acceleration, which is proportional with the quadrate of the velocity and indirectly proportional with the curve's radius. This counts for both horizontal and vertical curves.

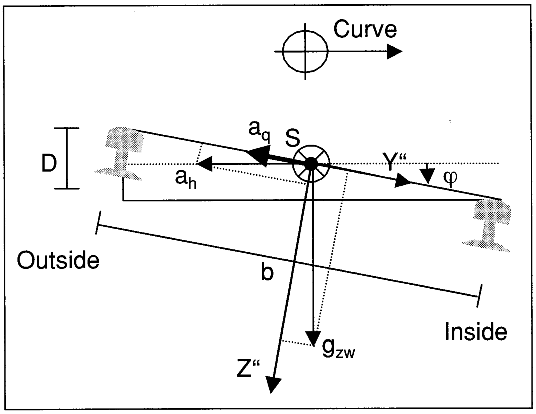


Fig. 4. Accelerations of a mass-point in a horizontal curve with cant.

Normally, the horizontal acceleration is compensated by the effect of cant *D*; by raising the outer rail the component of the horizontal acceleration in the track plane is compensated by the component of gravity (see figure 4). However, as trains run through the curve at different speeds, non-compensated centrifugal or centripetal accelerations occur.

Traditionally, track alignment designs were composed from straights and circular curves only. If a straight connects with a circular curve, both cant and (non-compensated) acceleration must 'jump' from zero to a fixed value and accordingly do the forces. Remedy was found in connecting the two elements with a transition curve, which has a reducing radius along its length to end with the correct curvature of the circular curve. Transition curves were firstly introduced by Wilhelm Pressel in 1864. Normally, transition curves are combined with superelevation ramps with increasing cant along its length.

Current standards for track alignment design (OVS, CEN [6]) give restricting values for the parameters mentioned to ensure safety and comfort. However, to describe the vehicle as a mass point is not correct, as the passenger is seated within the vehicle at a certain height above track plane. It can rather be described as a rigid body with reasonable measures and masses. Such a rigid body has 6 degrees of freedom (DOF). If longitudinal movement x, pitching around y and hunting around z are not taken into consideration, 3 DOF remain:

- Vertical displacement z
- Lateral displacement y
- Rotation around x : φ

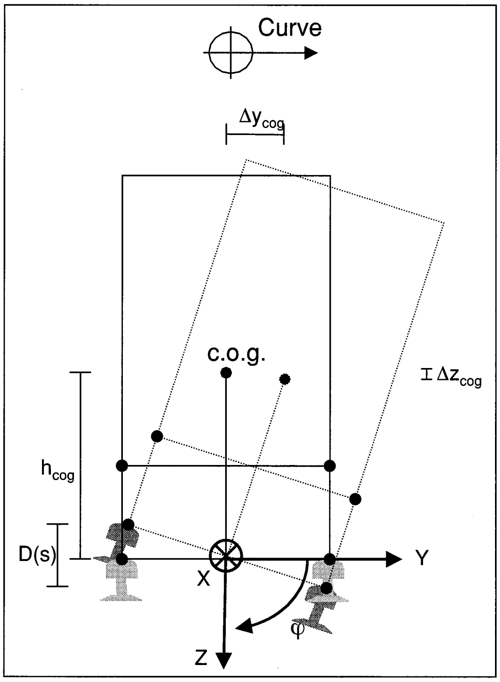


Fig. 5. Railway vehicle in a horizontal curve.

At straights and circular curves railway vehicles theoretically maintain movements without accelerations. In transitions the situation is different due to the need of changing the steady motion. Assuming the vehicle as a rigid body without length and the passenger to be in the centre of gravity (c.o.g.), the movement of the c.o.g. in transitions should be considered.

From the figures 5 and 6 it can be seen that the horizontal and vertical movement of the c.o.g. depends on both the height of the c.o.g. above the centreline of the track, as the amount of cant and the way it is applied.

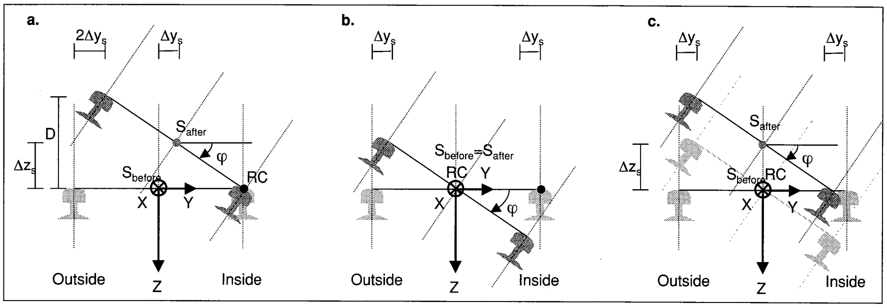


Fig. 6. Different ways to apply cant.

Theoretically speaking, cant could best be applied by rotating the track around the c.o.g. of the vehicle; the 'outswinging curve' ([5], see case 3 and figure 7), which has been tested at several locations on the Austrian network.

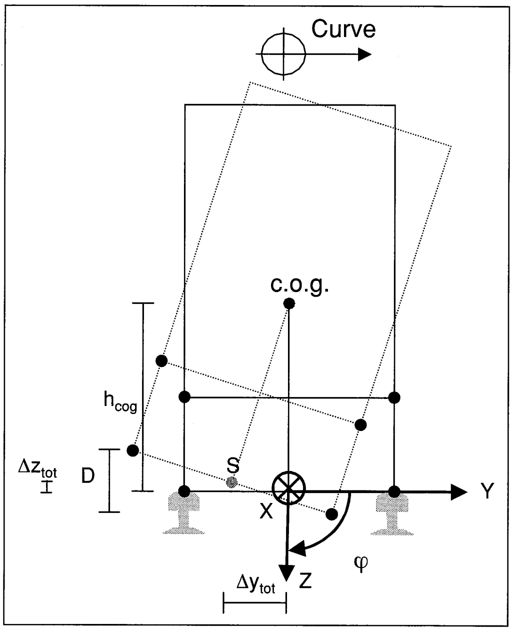


Fig. 7. Principle of the "outswinging curve".

To deal with the fact that the passenger is seated at a certain level above track plane, the European standard (CEN, Annex A) gives some additional values for the accelerations and change of accelerations at passenger level.

4 Modelling

When quasi-static behaviour of the vehicle is considered, the next step is to take vehicle characteristics (i.e. dimensions, primary and secondary suspension, masses) into account. In order to be able to analyse the dynamic behaviour of the High Speed Train as a result of track geometry, a 3D-simulation model was developed: SIMHSL.
SIMHSL uses a calculation method that is frequency based. The interaction between vehicle and track is described by response functions, giving the relation between the input (wheel displacement and non-compensated centrifugal forces) and output (displacement and rotation of the car body).
If the design of horizontal and vertical track alignment is known, they can be represented by their curvature – defined as the inverse radius – as a function of the distance. With these functions the position of the centreline of the track in space is formulated. To describe the track plane, the angle of cant is also needed as a function of the distance. Using train speed and train dimensions, variation in height (as a result of cant) and non-compensated centrifugal forces (as a result of curvature) on all bodies can be calculated as a function of time. By using the Fourier Transform (FFT) the inputsignal can be transferred from time domain to frequency domain:

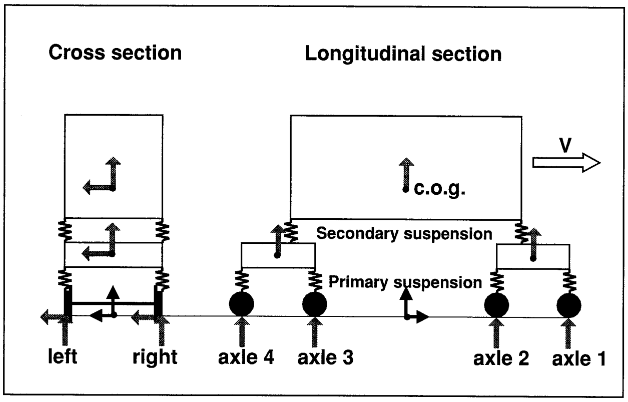
$$X(\omega) = \frac{1}{2\pi}\int_{-\infty}^{\infty} x(t)\cdot e^{-j\omega t}dt$$

For the High Speed Train a simplified 3D-model of an ICE was used (figure 8), consisting of 3 masses (the car body and two bogies) and the primary and secondary suspension (x, y and z direction).

In general, the formula for a multiple mass spring system (mss) which is excited by external forces is given by:

$$M\cdot\underline{\ddot{q}} + C\cdot\underline{\dot{q}} + K\cdot\underline{q} = F$$

where F is the external force and M, C and K are matrixes for mass, spring and damping of the system, which can be determined from internal equations of force (2nd Newton Law). Vector q summons the possible translations and rotations of the system.



Parameters

	masses		
m1	mass coach	36296	kg
m2	mass bogie	3625	kg
	primary suspension		
kx1	stiffness x-direction	45000000	N/m
ky1	stiffness y-direction	4000000	N/m
kz1	stiffness z-direction	800000	N/m
cx1	damping x-direction	52000	Ns/m
cy1	damping y-direction	9300	Ns/m
cz1	damping z-direction	10000	Ns/m
	secondary suspension		
kx2	stiffness x-direction	100000	N/m
ky2	stiffness y-direction	100000	N/m
kz2	stiffness z-direction	150000	N/m
cx2	damping x-direction	840	Ns/m
cy2	damping y-direction	840	Ns/m
cz2	damping z-direction	12000	Ns/m
	Rotational stiffness		
Ixx1	coach x-axis	97547	kgm^2
Iyy1	coach y-axis	1720704	kgm^2
Izz1	coach z-axis	1692411	kgm^2
Izz2	bogie x-axis (estimated)	*2000*	kgm^2
Iyy2	bogie y-axis (estimated)	*2000*	kgm^2
Izz2	bogie z-axis	4000	kgm^2

Fig. 8. Vehicle model of an ICE.

In case of SIMHSL this leads to 18 degrees of freedom (3 bodies with 6 DOF each). However, only vertical displacement, lateral displacement and rotation around the longitudinal axis as a result of cant are taken into account.

By assuming that input and output are excited at the same frequency

$$F = \hat{q}_0 \cdot e^{-j\omega t}$$

$$\underline{q} = \hat{q}_1 \cdot e^{-j\omega t}$$

the response function can be derived:

$$H(\omega) = \frac{F}{(-\omega^2 \cdot M - j\omega \cdot C + K)} = \frac{\hat{q}_1}{\hat{q}_0}$$

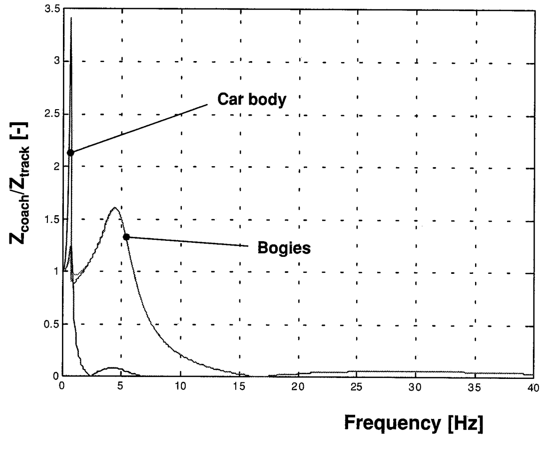


Fig. 9. Response function (Z_{coach}/Z_{track}) for an ICE running at 300 km/h.

Figure 9 shows the response function for the relation between vertical displacement of the car body (+bogies) and the track for an ICE, running at 300 km/h. The eigenfrequency (vertical direction) of the car body is about 0.6 Hz. As speed, wavelength and frequency are related by $v = \lambda \cdot f$, it is clear that vibrations (resulting from track geometry and track irregularities) with a wavelength of 80–160 m have a negative effect on the level of comfort a passenger experiences within an ICE running at 200–300 km/h.

By multiplying the spectrum of the inputsignal with the response function the spectrum of the output is derived, which gives the translation or rotation of the car body as a function of time after reverse Fourier Transform. Accelerations now can be calculated by taking the second derivative of the signal.

The following four cases give the results of some possible applications of SIMHSL.

5 Case 1: Horizontal curve

In order to determine the effect of vehicle characteristics, the case of a horizontal curve with adjacent transition curves is considered.

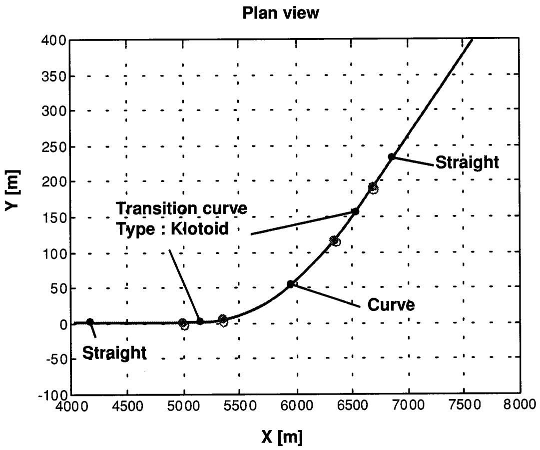
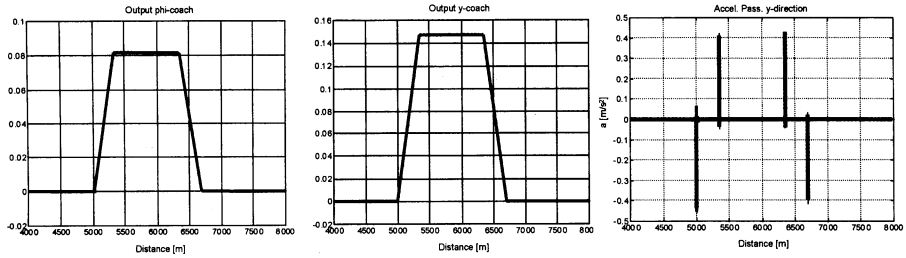


Fig. 10. Plan view of a horizontal curve.

For the cant the theoretical cant is taken, thus the passenger won't be subjected to (non-compensated) lateral accelerations. Clothoids are used for both transition curves and superelevation ramps, giving a linear increase or decrease of curvature and cant. In this first case lateral car body displacement and car body rotation around the longitudinal axis are considered as well as lateral accelerations.

Vehicle A



Vehicle B

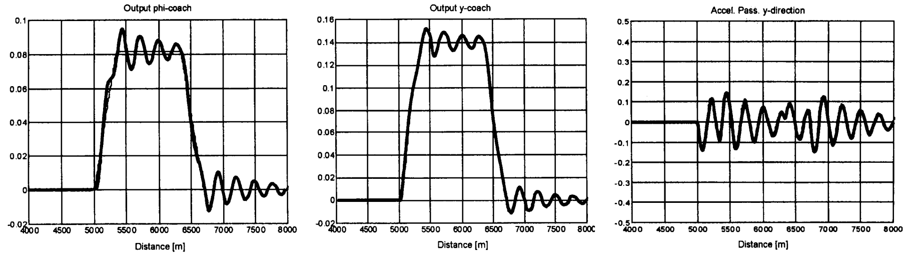


Fig. 11. Results of case 1 calculated by SIMHSL.

Comparison is made between two vehicles:

A. an imaginary vehicle without length, suspension system and mass (giving the results for the quasi-static behaviour);
B. The ICE for evaluation of the dynamic behaviour of high speed trains.

If the vehicle is considered as a rigid body (vehicle A) then its motions are determined by the track geometry, as can be seen in figure 11. Both the car body rotation and the lateral displacement can be calculated from the angle of cant. The passenger will only experience accelerations (and jerks) at the beginning and end of the transition curves.

Using model B more practical results are obtained. Basically, vehicle behaviour is also determined by the track geometry. As theoretical cant was applied, no non-compensated lateral force occurs and therefore the rotations of track and car body are equal. However, the moment of inertia causes the car body to start a rotational vibration which is attenuated only very slowly. Therefore, passengers will experience lateral, vertical and rotational accelerations.

6 Case 2: Transition curves

Experience shows that the deterioration of track geometry almost regularly starts at the beginning and end of transition curves. When taking into consideration that the locomotive or traction unit has very stiff primary and secondary suspension, from figure 11 (vehicle A) it is clear that – in the case of clothoids – high forces occur at these points. In reality the corners at the beginning and end of the transition curve are smoothed which reduces the impact but doesn't avoid it!

For this reason several types of transition curves and elevation ramps have been developed [5]. If curvature is described by

$$\kappa = \kappa_1 + (\kappa_2 - \kappa_1) \cdot \delta(\chi)$$

where κ_1 and κ_2 are the curvature at the beginning and end of the transition and

$$\chi = \frac{s}{l}$$

the following ramps can be distinguished:

- Linear ramp (clothoid)

$$\delta(\chi) = \chi$$

- S-shaped ramp (two parabolas)

$$\delta(\chi) = \left\{ \begin{array}{cc} 2\chi^2 & 0 \le \chi \le \frac{1}{2} \\ 1 - 2 \cdot (1-\chi)^2 & \frac{1}{2} \le \chi \le 1 \end{array} \right\}$$

- Bloss ramp (Cubic parabola)

$$\delta(\chi) = \chi^2 \cdot (3 - 2\chi)$$

- JNR-ramp (sinusoidal ramp)

$$\delta(\chi) = \sin^2\left(\frac{\pi}{2} \cdot \chi\right)$$

- Alternative ramp

$$\delta(\chi) = \chi - \frac{1}{2\pi} \cdot \sin(2\pi\chi)$$

- New ramp

$$\delta(\chi) = \frac{1}{2}\left\{1 + \tanh \cdot \frac{c \cdot (2\chi - 1)}{[4\chi \cdot (1-\chi)]^v}\right\}$$

(c and v being design parameters)

From quasi-static theory, both the 'alternative' and 'new' ramp should be preferred, as they show no 'jump' in the first and second derivative [4]. Using SIMHSL, the dynamic effect of these transitions and superelevation ramps can be investigated.

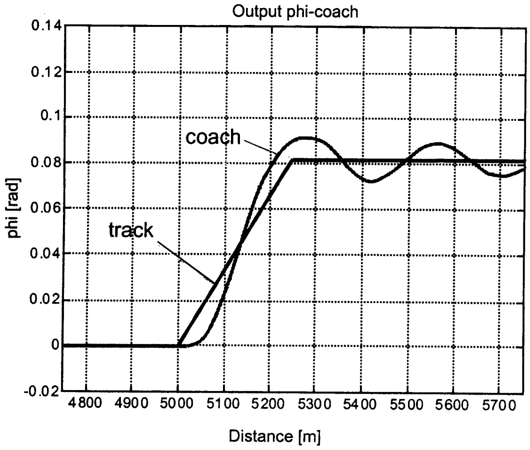


Fig. 12. Dynamic behaviour vehicle (roll angle coach) in case of a clothoid.

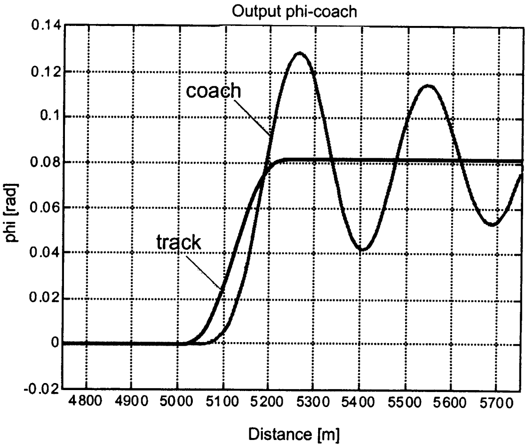


Fig. 13. Dynamic behaviour vehicle (roll angle coach) in case of an 'alternative' ramp.

Although the transition length is the same, it is clear from figure 12 and 13 that the dynamic motions of the car body lead to greater accelerations in case of the 'alternative ramp'.
As both the beginning and end of the ramp are 'smoothed', the slope in the middle is steeper, therefore causing greater dynamic motions.
From this point of view, the clothoid would be the optimal ramp as it minimises dynamic motions.

7 Case 3: The 'Outswinging curve'

As mentioned before, the 'outswinging curve' would be the optimal transition. As can be seen in figure 14, both the dynamic effects and the lateral displacement of the coach are minimal.

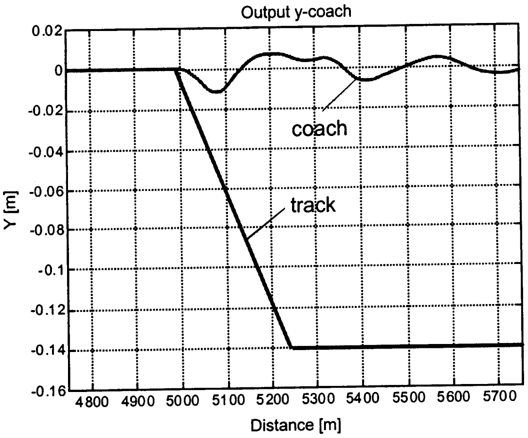


Fig. 14. Lateral displacement of the coach in a 'outswinging curve'.

8 Case 4: HSL-Zuid

Both to determine the influence of combined horizontal and vertical curvature and speed (cant deficiency or cant excess), as to investigate the level of comfort at the HSL-Zuid, part of the alignment of the HSL-Zuid has been investigated [3].

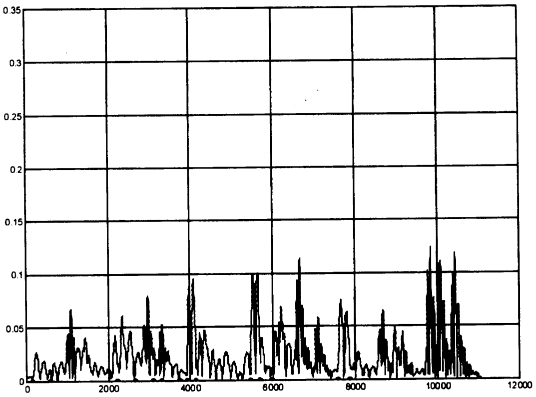


Fig. 15. ISO-weighted car body accelerations OTB5 – ICE at 300 km/h.

Although a high level of comfort is obtained ($a_v < 0.315$ m/s^2), the results made clear that both jumps in the (horizontal and vertical) curvature, as the gradient of cant on superelevation ramps influence the dynamic motions of the vehicle.

Two remarks should be made:

1. Research has only been done on the influence of track alignment and the vehicle suspension system on passenger ride comfort. The influence of track characteristics (i.e. irregularities) and substructure dynamics are not taken into account.
2. In this case, direct comfort disturbance was considered. When evaluating motion sickness, other weighting factors should be applied.

9 Conclusions

Current track geometry standards are based on simple (quasi-static) consideration and don't take vehicle-characteristics into account. SIMHSL is a tool to evaluate track alignment design of high-speed lines on basis of comfort.

Especially when designing high-speed lines, dynamic motions have to be considered.
Both curvature (leading to centrifugal forces) and cant (leading to lateral and vertical displacement and rotation of the coach) should be applied gradually in order to minimise dynamic motions.

Although vertical accelerations normally are small, the use of vertical transition curves will increase the level of comfort. For this reason a Spline-function seems to have great advantages.

Finally, railway alignment design should be derived by rotating the track around the c.o.g. reducing vehicle oscillations. An 'outswinging curve' can avoid free lateral accelerations (and jerk) to a great extent.

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