Methods for evaluation of ride comfort as a function of vehicle reactions caused by railway alignment

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Abstract
In this report, methods for quantifying ride comfort in different alignment alternatives are analysed. There are many relations between quantities describing the track geometry and physical quantities describing the motions of the vehicle which may cause passenger discomfort. 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.

In the standards BS 6841 (1987), ISO 2631-1 (1985) and the draft ISO 2631-1.2 (1995) there are guidelines for assessment of the discomfort caused by motions. The standards did not consider the relative phases between different frequencies and directions, neither did they consider accelerations with frequencies lower than 0.5 Hz (which are crucial when evaluating alignments).

Comfort tests within railway vehicles have been conducted in many countries. Some of them are of little use when evaluating alignments, since only one-dimensional analyses have been conducted. In these cases, complex motions have been rated by test objects, but the relative importance of various jerks and accelerations to the perceived discomfort has not been investigated.

The $P_{CT}$ functions, originally suggested by British Rail, consider the relative influences of lateral acceleration, lateral jerk and roll velocity. Therefore, it is concluded that these functions are the most reasonable comfort functions when evaluating railway alignments.
Preface

This work has been carried out at the Division of Railway Technology, Department of Vehicle Engineering, Royal Institute of Technology (KTH), Stockholm and the Railway Systems group, Division of Transport Systems, Swedish National Road and Transport Research Institute (VTI), Linköping.

The work is part of the joint project concerning track/vehicle interaction commissioned by Adtranz Sweden, the Swedish National Rail Administration (Banverket), the Swedish Transport and Communications Research Board (KFB), the National Swedish Board for Technical and Industrial Development (NUTEK), the Swedish State Railways (SJ) and the Swedish National Road and Transport Research Institute (VTI).

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Linköping, May 1997

Cover photo: Curves and crossovers in the Western Main Line (left) and the Nynäs Line (right), Älvsjö.

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Appendix 1 Terminology

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Summary

In this report, methods for quantifying ride comfort in different alignment alternatives are analysed. There are many relations between quantities describing the track geometry and physical quantities describing the motions of the vehicle which may cause passenger discomfort. 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. However, it would also be preferable to know the relations between passenger discomfort and other physical quantities such as yaw velocity, duration of lateral acceleration, magnitudes of time derivatives of jerk, magnitude of roll acceleration, magnitude of vertical acceleration, etc.

The best solution would be to have a function where passenger discomfort is expressed in monetary units and where all the relevant physical quantities are expressed as independent variables. However, very little seems to have been done in the area of monetary assessment of comfort. The next best solution would be to have a non-monetary comfort function, where the relative importance of each relevant physical quantity is known. With such a comfort function, equally costly alignment alternatives may be compared.

In the standards BS 6841 (1987), ISO 2631-1 (1985) and the draft ISO 2631-1.2 (1995) there are guidelines for assessment of the discomfort caused by motions. The standards considered frequencies, directions and durations of the accelerations, and they also gave some guidance as to how to combine the effects of motions in different directions. The standards did not consider the relative phases between different frequencies and directions, neither did they consider accelerations with frequencies lower than 0.5 Hz (which are crucial when evaluating alignments). From the research literature, it is also clear that there are many controversies concerning the guidelines in the standards.

Comfort tests within railway vehicles have been conducted in many countries. Some of them are of little use when evaluating alignments, since they did not consider low frequency motions. Various other studies are also of little relevance since only one-dimensional analyses have been conducted. In these cases, complex motions have been rated by test objects, but the relative importance of various jerks and accelerations to the perceived discomfort has not been investigated. Some of the authors have noticed that the different jerks and accelerations covaried very greatly in their tests.

The $P_{CT}$ functions, originally suggested by British Rail, consider the relative influences of lateral acceleration, lateral jerk and roll velocity. Therefore, it is concluded that these functions are the most reasonable comfort functions when evaluating railway alignments.

Keywords: Passenger comfort, vibrations, shocks, accelerations, jerks, alignment, track geometry, radius, transition curves, cant, superelevation ramps, turnouts.
1. Introduction

An optimisation problem concerning the alignment of railways could be formulated in many different ways. One way might be to design, within a predefined terrain corridor, the alignment for a predefined design speed whereby passenger discomfort is minimised. Another approach might be to predefine passenger comfort and instead maximise the design speed.

Alignments and design speed affect not only the ride comfort. Safety aspects must be considered, as well as aspects concerning wear and maintenance of the rolling stock and of the track. But permissible train speeds on modern tracks, with continuously welded rails, heavy sleepers and high quality fastenings, are to a great extent related to ride comfort on horizontal curves. The special trains for high speed curving, built by Adtranz, FIAT and others, have active tilting to improve the ride comfort. The tilt mechanism itself does not affect safety or wear on rolling stock and track.

The type of comfort discussed in this report is a subset of the total ride quality. In focus are the undesired motions of the vehicle floor which are created when a coach runs on curves. Comfort in terms of comfortable seats, air quality, temperature, light and service is not in focus. In this report, these variables are regarded as constant, at least within the mentioned studies. They may, however, vary between studies and comparisons between different studies should be made with care.

There are different opinions as to whether or not the comfort disturbances analysed in this report are the most serious and the most important to consider. However, to take an example, a British study (Branton 1972) found that 800 out of 1500 passengers wanted "less vibrations". The second measure in ranking order was "more space", which 779 out of the 1500 passengers wanted.

In another study (West et al 1973) concerning the Mark I and Mark II coaches, "less vibrations" was the measure given the highest priority by the passengers. 51% of the passengers wanted "less vibrations", compared with 46% who wanted less noise and 39% who wanted improved seats. However, the ranking order of the measures was dependent on the duration of the journey.

In a third study (Myers & Marshall 1980) ride comfort, travel time, ticket cost, schedule and location of terminals were assumed to have an influence on the utility of the passengers. With stated preference techniques concerning the Edmonton-Calgary route, it was found that the ride comfort affected the utility and had a significant effect on traveller modal choice.

Hence it is concluded that the comfort disturbances, caused by the combination of alignment standard, vehicle characteristics and train speed, should not be neglected.
2. Principal methods of comfort rating

2.1 General considerations

The motions of the vehicle floor are of a complex nature and are usually characterised by different physical quantities such as amplitudes, directions, frequencies and durations of translational or rotational accelerations. The motions may also be characterised by the amplitudes of the displacement, the velocity or the jerk, instead of amplitude of the acceleration. However, it must be recognised that such descriptions must contain a large amount of data to describe the motion in full. A description with only a few physical quantities neglects many aspects of the motion.

Figure 2.1 indicates some, but not all, of the relations between characteristics of the track geometry and physical quantities. For example, in the figure the horizontal and the vertical accelerations caused by changes in the gradient of the cant are not indicated. Neither is the longitudinal jerk caused by the horizontal curvature (discussed in Megyeri 1978) indicated.
### Figure 2.1

*Some relations between characteristics in the track geometry and certain physical quantities of vehicle motions.*

In order to evaluate different alternatives in the track geometry (according to ride comfort), there are obviously many physical quantities which must be considered and it is necessary to know how railway passengers experience the discomfort caused by
different magnitudes of the physical quantities. The ratings should cover all the relevant ranges of the various physical quantities.

It should also be noted that many physical quantities influence the ride simultaneously, such as relatively high frequency vibrations (caused by track irregularities), and roll velocity (caused by a superelevation ramp), and low frequency lateral jerk (caused by the superelevation ramp and a corresponding transition curve). When passengers perceive the comfort to be low, this may not have a single cause in many situations. When comparing different alignment alternatives, and when considering relevant boundary conditions, an improvement in one variable in the track geometry must usually be made at the expense of another (Kufver 1997). Thus, an improvement in one physical quantity usually has to be made at the expense of another.

The left side of Figure 2.1 includes characteristics of the track geometry which are associated with the design geometry. However, the track irregularities describe the difference between the actual position of the track and the design geometry. When comparing different track geometry alternatives, it is interesting to evaluate whether or not a change in the design track geometry results in a significant change in ride comfort, if different relevant levels of track irregularities are superimposed on the geometry alternatives.

Therefore, the best solution would be to have an overall comfort function, which indicates the passenger ride comfort with regard to all the relevant physical quantities.
2.2 Monetary assessment of ride comfort

In a world of perfect knowledge, there would exist a comfort function expressed in monetary units. With such a comfort function, units of the physical quantities would be associated with costs and it would be possible to predict whether or not a curve realignment (with a larger curve radius and hence lower lateral acceleration) could be justified when the present value of comfort and construction costs are considered. It would also be possible to calculate whether or not a change in train speed is profitable, when time costs, comfort costs and other costs are included in the analysis.

This monetary approach to comfort rating seems to be very rare, but has been used by Mäkälä and Lampinen (1985) for monetary assessment of ride comfort in road vehicles.

2.3 Non-monetary assessment of ride comfort

If a monetary assessment of different vehicle motions is not available, a second-best solution might consist of an overall comfort function expressed in non-monetary units. With such a comfort function, equally costly track geometry alternatives could be evaluated. However, contrary to the monetary comfort function, it is not possible to judge whether or not a higher construction cost for a better track geometry can be justified economically to improve passenger comfort.

A comfort function in non-monetary units may be estimated in the following ways (Griffin 1990): In absolute methods (also called scaling methods), the test subjects indicate on graded scales the perceived discomfort when they are exposed to different levels of the physical quantities. Such methods may be used in laboratory studies as well as field studies. In relative methods, the test subjects in laboratories, in order to achieve the same level of comfort, regulate the level of one of the physical quantities when they are exposed to different levels of another physical quantity. In this way, equivalent comfort contours may be estimated. Comfort assessments with the absolute method may include large uncertainties if the ends of the scale (comfortable, uncomfortable) are not further defined (Alm 1989).

A special case of the absolute method is the pushbutton method, which has been used in various field tests in railway vehicles, see ASEA and SJ (1979), Harborough (1986a), Förstberg (1993) and Wichser & Boesch (1993). In the British Rail HST and APT tests, the test subjects were instructed to press the button when the perceived ride comfort was rated uncomfortable or very uncomfortable on a scale with the levels very comfortable, comfortable, acceptable, uncomfortable and very uncomfortable (Chapell 1984).

It should be noted that with the use of semantic labels such as "uncomfortable" at the ends as well as in the middle of the scale, some information is lost. Even though the test subjects may have the same opinion about the comfort rating on a graded scale, they may have different opinions of where on the scale to put the semantic labels. According to Griffin (1990), the use of semantic labels introduces extra large inter-subject variability and he compared the use of such labels with a situation where test subjects were asked whether or not a string is "long".
2.4 Discussion of the methods of comfort rating

If several types of physical quantities were included in a monetary comfort function, it would be possible to construct iso cost curves when two comfort variables are analysed simultaneously. In a similar way, if the physical quantities were included in a non-monetary comfort function, it would be possible to construct equivalent comfort curves. If the results from the two different methods of quantifying ride comfort are consistent, the following must apply:

If certain combinations of the physical quantities \( q_a \) and \( q_b \) (\( \{q_{a1}, q_{b1}\}, \{q_{a2}, q_{b2}\} \) etc.) are located on the same iso cost curve, they must also be located on the same equivalent comfort curve. Hence, the iso cost curves and the equivalent comfort curves must define the same pattern in the \( q_a-q_b \) plane. Also, a combination of the quantities \( q_a \) and \( q_b \), which according to the monetary comfort function is rated more favourable than another combination, must also be rated more favourable according to the non-monetary comfort function. Each equivalent comfort curve should therefore be associated with a certain comfort cost and the comfort cost function should increase.\(^1\) However, the requirement on consistency could not state whether the relation between discomfort and comfort cost is linear or non-linear.

\[\text{Figure 2.2 Possible relation between discomfort and comfort cost.}\]

It does not seem unlikely that, even though the equivalent comfort curves would be stable over time, the relation in Figure 2.2 might change, for example as a function of the real income of the passengers.

\(^1\) In some cases, motion may increase comfort. Sitting in a seat for a longer time may be more comfortable in a vibrating environment than in a still one (Richards 1980). Such low amplitude effects will not be analysed in this document.
The main difference between an ordinary non-monetary comfort rating and the pushbutton rating is that the latter defines only two\(^2\) alternatives (to press the button or not). ORE (1987) has concluded that the results from graded ratings and from pushbutton ratings are different. Graded ratings give a better resolution, and comparisons between the two methods are regarded as possible only if large comfort disturbances are frequent and if the threshold level for pressing the button is well defined.

Here, another important difference between the two methods is illustrated with an example:

Assume the non-monetary comfort rating of the physical quantities \(q_a\) and \(q_b\) is studied. There are three test subjects and they perceive and rate comfort the same way, according to the comfort function

\[
\text{discomfort} = q_a + 2 \cdot q_b
\]  

[2.1]

![Figure 2.3 Equivalent comfort curves in the \(q_a-q_b\) plane.](image)

If these three test subjects are exposed to different combinations of the quantities \(q_a\) and \(q_b\), they will make such graded ratings in an ordinary non-monetary comfort rating that a subsequent multiple regression analysis will result in the comfort function [2.1].

Assume the test subjects are to rate the combinations with pushbutton technique and that the threshold level for pressing the button, according to inter-subject variability, is the

\(^2\) In some comfort tests in Sweden, there were three alternatives. See Sections 4.7 and 4.8.
discomfort level 6, 7 and 8 respectively, for the three test subjects in question. The relative frequencies for pressing the buttons will be as in Figure 2.4.

![Relative frequencies for button pressings in the $q_a$-$q_b$ plane.](image)

**Figure 2.4**  
Relative frequencies for button pressings in the $q_a$-$q_b$ plane.

If all integer combinations of the quantities $q_a$ and $q_b$ in the intervals $1 \leq q_a \leq 5$ and $1 \leq q_b \leq 5$ are used in a test series, a multiple regression analysis will give the following result:

$$
\text{Relative frequency} = 0.36 \cdot q_a + 0.6 \cdot q_b - 0.72 \tag{2.2}
$$

Formula [2.2] indicates that the quantity $q_b$ is not twice as uncomfortable as the quantity $q_a$, but 1.67 times more uncomfortable. Such a difference exists since the instrument does not distinguish between different comfort levels in the upper right corner and the lower left. This effect may explain why Wichser and Boesch (1993) believe that pushbutton techniques alone cannot indicate the relative importance between different physical quantities. However, the opinion of Eickhoff (1996) is that tests where most of the observations fall in the diagonal zone, in which the instrument can distinguish between comfort levels, will give valid results.
3. Some general relations between vibrations and discomfort

In BS 6481 (1987), ISO 2631-1 (1985) and also in a recent draft of ISO 2631-1.2 (1995) there are guidelines for comparing the discomfort caused by different complex motions in the frequency range 0.5-80 Hz. A comprehensive overview of results from comfort research and guidelines was also published by Griffin (1990), but he noted that most studies concerned motions with frequencies of at least 1 Hz.

3.1 Amplitudes

A common approach in the comfort tests is that perceived sensation $\psi$ increases with vibration amplitude $\varphi$ according to Steven’s power law

$$\psi = k \cdot \varphi^n$$ [3.1]

where the magnitude of the constant $k$ depends on the choice of units (Griffin 1990). The magnitude of the exponent $n$ is approximately 1. The unity value of $n$ indicates that comfort ratings for vibrations should not be based on logarithmic values of vibration amplitudes (e.g. decibels).

In a study by Shoenberger (1975) the exponent $n$ has been estimated to be in the range 1.34-1.48 in the frequency interval 0.25-4.0 Hz. Several studies (Griffin 1990, Howarth & Griffin 1988a, Shoenberger & Harris 1971) indicate that the exponent $n$ is independent of (or varies very little with) the frequency if the r.m.s. acceleration is less than 3 m/s². According to Howarth (1987) the exponent $n$ has a unity value for vertical shocks.

In a study, Howarth and Griffin (1988a) estimated the exponent $n$ to have values in the range 0.68-1.99 depending on the frequency, which was varied within the range 4-63 Hz.

In the literature, there are also alternatives to equation [3.1]. If there exists a threshold level of the stimulus $\varphi_0$, the sensory magnitude should be equal to zero at this level of the stimulus (Marks 1974). The modified power relations read

$$\psi = k \cdot (\varphi - \varphi_0)^n$$ [3.2]

and

$$\psi = k \cdot (\varphi^n - \varphi_0^n)$$ [3.3]
3.2 Frequencies

The perceived discomfort depends not only on the vibration amplitude but also on the frequency. At frequencies below 1-2 Hz, the human body reacts like a rigid body and the forces on the body are proportional to the acceleration amplitude. Griffin (1990) took this as an implication that at low frequencies discomfort is proportional to acceleration amplitude. At higher frequencies, humans are assumed to be less sensitive since the body provides an attenuation of the vibrations.

BS 6841 (1987) presented frequency weighting factors for translational and rotational accelerations. These weights are constant in the frequency range from 0.5 Hz up to 1-2 Hz (for rotations and translations respectively). Above 1-2 Hz, the weights are generally smaller, which indicates that perceived discomfort is lower for a vibration with the same acceleration amplitude but with a higher frequency. (However, for vertical translational accelerations, the weights have their largest values in the frequency range 5-16 Hz.). The way in which the values of the weights vary at higher frequencies indicates that perceived discomfort was assumed to be proportional to amplitude in velocity.

The frequency weighting factors in ISO 2631-1 (1985) and the second draft of ISO 2631-1.2 (1995) have generally the same characteristics as those in BS 6841 (1987). There are slightly different values for the weights for vertical translational accelerations and the numerical values for the weights are not presented separately, but multiplied by the numerical factors for the band-pass filters.

Both BS 6841 (1987) and ISO 2631-1 (1985) stated that the standards incorporate experience and research results. However, it is possible to find contradictory results. Yonekawa and Miwa (1972) have presented equivalent comfort curves where magnitude of vertical as well as horizontal acceleration is related to frequency in the power of 0.8 (in the frequency range 0.05-1 Hz). This indicates that perceived comfort is related to magnitude in velocity rather than acceleration.

Payne, Brinkley and Sandover (1994) argued, as Griffin (1990), that discomfort is proportional to forces. With this assumption, they used a mass/spring/damper-model (a DRI model) to show that perceived discomfort of shocks is independent of frequency in the range 0.2-2 Hz. They also stated that "rate of onset" (jerk) is not considered by "serious workers in the field" (Payne, Brinkley and Sandover 1994 p 21).

Concerning rotational accelerations, Griffin (1990) stated that for frequencies above 1 Hz, the perceived discomfort is roughly proportional to rotational velocity. The different equivalent comfort curves he presented show rotational acceleration as a function of frequency in the power of approximately 0.9 and confirm his conclusion. The equivalent comfort curve from Irwin (1981), concerning yaw accelerations, also covers the frequency range 0.063-1.0 Hz and in that paper perceived discomfort is also proportional to rotational velocity (with some minor exceptions around 1 Hz). Comfort tests reported by Parsons and Griffin (1978) also showed that discomfort caused by pitch motions was proportional to rotational velocity (the equivalent comfort curve increased with frequency in the power of unity), but discomfort caused by roll accelerations was proportional to frequency in the power of approximately 0.85. For the
frequencies in the study (2, 4, 8 and 16 Hz) the test subjects were significantly more sensitive to roll accelerations than to pitch accelerations. These results to some extent contradicted the frequency weights in BS 6841 (1987).

Griffin (1990) also made a remark that perceived discomfort of rotational accelerations is influenced by the position of the axis of rotation. If this axis is located far from the test subject, the translational accelerations caused by the rotational motions will dominate. He suggest the axis of rotation to be located directly beneath the ischial tuberosities, but for low frequency motions it might be desirable to locate the axis of rotation through the mass centre of the test subject. However, most studies had located the axis of rotation through the surface which supports the test subject.

3.3 Duration

In ISO 2631-1 (1985) there is no time dependence between perceived discomfort and vibration if the duration is less than 4 minutes. For longer durations, there is a non-linear relation between perceived discomfort and duration. In ISO 2631-1 (1985) there is also an approximate relation: If the duration is less than 10 minutes, there is no time dependence and for longer durations the perceived discomfort is assumed to be proportional to the square root of the duration divided by 10 minutes. This means that the discomfort is assumed to be proportional to the square root of the time integrated acceleration in the power of two.

In BS 6841 (1987) the time dependence is weaker (when the duration exceeds 10 minutes) since the perceived discomfort is related to the fourth root of the time integrated acceleration in the power of four. This integral is called vibration dose value, VDV, by Griffin (1990). The draft of ISO 2631-1.2 (1995) stated there is no universal time dependence of vibrations on comfort, but mentioned dose values (in which discomfort is proportional to the fourth root of time).

Payne, Brinkley and Sandover (1994) reported methods where acceleration in the power of eight is integrated over time.

Some studies indicate more complex time dependences. Kjellberg and Wikström (1985) studied time dependences within the range 0.1-128 seconds. They found that the time dependence may vary with duration, at least for some frequencies. The time dependences were calculated as equivalent comfort curves in a diagram where log acceleration $a$ of a vibration of fixed duration was plotted against log duration $t$ of a vibration with fixed acceleration magnitude. The regression coefficient $c_1$ from linear fittings (according to [3.4]) was in the range 0.13-0.25 for durations up to 4 seconds and in the range 0.03-0.06 for durations of 4-128 seconds. (R.m.s. integration corresponds to a regression coefficient $c_1$ of 0.5, while VDV corresponds to a regression coefficient $c_1$ of 0.25.)

$$\log a_{\text{fixed duration}} = c_0 + c_1 \cdot \log t_{\text{fixed magnitude}}$$ [3.4]
In another study, Kjellberg, Wikström and Dimberg (1985) found the regression coefficient $c_1$ to be in the range 0.040-0.045 when durations from 15 seconds to 64 minutes were analysed.

Oborne (1977) has reported time dependences which are different for frequencies in the range 0-4 Hz respectively 8-16 Hz. Payne (1996) suggested that the influence of duration should be related to the number of cycles of the excitation (at single frequency and single amplitude vibrations) rather than time, but he also stated that procedures for evaluating the duration of random vibration have not been solved.

3.4 Multiple frequencies

In the case of broad-band vibration, ISO 2631-1 (1985) recommended that the r.m.s. acceleration be calculated for each one-third octave band and that these r.m.s. accelerations be evaluated separately concerning the weights and suggested limits at the centre frequency of the respective band. This rating method assumes that there is no interaction between the effects of the various one-third octave bands: Only the worst component (the one-third octave band with the highest r.m.s. acceleration compared with its limit) will affect the comfort rating. However, ISO 2631-1 (1985) mentioned the weighting method, where the weighted r.m.s. acceleration is calculated for the entire broad-band.

BS 6841 (1987) recommended that the weighted r.m.s. acceleration be used. If the crest factor$^3$ is higher than 6.0, BS 6841 (1987) believed the r.m.s. acceleration to underestimate the discomfort and suggested a root-mean-quad method, where the r.m.q. acceleration is calculated as the fourth root of the time integral of the frequency-weighted acceleration in the power of four.

The draft of ISO 2631-1.2 (1995) states that frequency-weighted r.m.s. acceleration shall always be evaluated.

3.5 Impulsive motions

ISO 2631/1 (1985) stated that continuous shock-type excitations may be evaluated by its standard procedures if the energy is contained within the frequency range 1-80 Hz and the crest factor is below 6. If the crest factor is higher, these procedures may underestimate the perceived discomfort generated by the motion.

Griffin and Whitham (1980a, 1980b) used an approach where the idea was to use the same procedures for continuous vibrations, shocks and combinations of these. They found that the r.m.q. values were more highly correlated to perceived discomfort than r.m.s. values were. Howarth and Griffin (1988b) studied perceived discomfort from

$^3$ The crest factor is defined as the frequency-weighted peak acceleration divided by the frequency-weighted r.m.s. acceleration (BS 6841 1987). Unfortunately, the phase shifts in filters may affect the magnitude of the weighted peak acceleration (BS 6841 1987). Another problem with the crest factor definition is that the magnitude of the weighted peak acceleration tends to increase with increased duration of the measurement (ISO 2631-1 1985; Griffin 1990).
intermittent building vibrations and also found that r.m.q. values were more highly correlated to perceived discomfort. A later study by Howarth and Griffin (1991) confirmed these results.

BS 6841 (1987) suggested the use of r.m.q. methods and $VDV$ methods at intermittent vibration and/or when the crest factor exceeds 6.

The draft of ISO 2631-1.2 (1995) suggested that shocks and high crest factor motions should be evaluated either with the running r.m.s. methods or with $VDV$ methods. Two running r.m.s. methods were described. Both methods give a new time history, instead of a single discomfort value.

Payne, Brinkley and Sandover (1994) criticised r.m.q./$VDV$ methods to be too similar to r.m.s. methods and stated that methods with acceleration in the power of eight give a higher correlation to perceived discomfort.

### 3.6 Multiple-axis motions

When (translational) vibrations occur in two or three directions simultaneously, ISO 2631-1 (1985) and the draft of ISO 2631-1.2 (1995) suggested a vector summation of the frequency-weighted r.m.s. accelerations.

Also BS 6841 (1987) suggested a vector summation (root-sum-of-squares, r.s.s.) for translational and rotational multiple-axis r.m.s. accelerations respectively. However, in the appendix where the effects of vibration and repeated shocks on health (contrary to comfort) are discussed, BS 6841 (1987) suggested a method using the fourth root of the sum of the fourth powers of the vertical $VDV$ and the horizontal $VDV$.

According to Griffin (1990) there has been few studies concerning multiple-axis motions. Several comfort tests indicated that r.s.s. summations give a better correlation to perceived discomfort than worst component, linear summation or r.s.q. summation (root-sum-of-quad) of the r.m.s. accelerations. Even fewer studies concerned effects of the phase when combining vibrations in different axes and there were no general conclusions concerning perceived discomfort.

### 3.7 Simultaneous translational and rotational accelerations

ISO 2631-1 (1985) offered no guidance concerning rotational accelerations. BS 6841 (1987) stated that the r.s.s. accelerations for translational and rotational accelerations should be reported separately. The draft of ISO 2631-1.2 (1995) suggested that translational and rotational accelerations may be combined in a r.s.s. value when calculating an overall vibration total value.

Parsons and Griffin (1978) investigated comfort ratings in simultaneous translational and rotational accelerations by exposing test subjects for rotational accelerations with different distances to the axis of rotation. Three methods for aggregating the accelerations were evaluated: The worst component (the acceleration with the highest r.m.s. acceleration after frequency weighting), r.s.s. acceleration (of root-mean-squares)
and linear summation of r.m.s. accelerations. The highest correlation to comfort ratings was found for the method with the worst component.


Griffin (1990) noted that it does not seem appropriate to use sometimes the r.s.s. summation and sometimes the worst component method when different frequencies and directions are aggregated into a single comfort value. He suggested that the three translational accelerations and the three rotational accelerations respectively should be aggregated separately with the r.s.s. method. If one of these two root-sums-of-squares dominates the other, it will be sufficient to use the larger value as a comfort value. If the two sums are similar, they should be summed with the r.s.s. method.

3.8 An overall comfort function

A general model for calculating a total rating of the perceived discomfort has been suggested by Parsons and Griffin (1983). The suggested procedures are the following: Firstly, the accelerations in each axis (three translations and three rotations) are frequency-weighted. Secondly, for each axis the accelerations at each frequency are integrated over time. Thirdly, the effects of the six vibration axes are combined. If there are several contact points, and if they vibrate differently, the effects of these vibrating input positions are combined. Finally, the duration is taken into account. This general model is consistent with BS 6841 (1987) except that the standard did not suggest that the effects of translational and rotational accelerations are combined and that the effects of different vibrating input positions are also combined.
4. Comfort tests in railway vehicles

4.1 General remarks about ‘field’ tests

In this chapter, certain comfort investigations and comfort tests in railway vehicles will be reviewed. The focus will mainly be on physical quantities which are related to alignment and cant.

Generally, less comfort research has been conducted in ‘field’ conditions than in laboratories. One reason is that it is more difficult to control the physical quantities in the ‘field’. It has also been noted that passengers find it difficult to extract and consider just one physical physical quantity from all the others (Oborne & Clarke 1973).

Various other important differences exist between laboratory tests and 'field' tests. In the laboratory, the test subjects may be occupied with other activities than the train users. They are not fare-paying customers, they are not expecting a comfortable journey, they are not exposed to the ordinary journey environment, and they are not looking forward to their arrival at a destination (Oborne 1977).

In this chapter, most of the investigations were made in special test runs. However, the comfort observations on the German Railways were made in commercial service. ORE B153 (1989a) has also validated the results concerning "vehicle ride comfort" \( N_{MV} \) and "mean passenger comfort" \( N_{VA} \) for seated passengers (see 4.6 below) in surveys carried out in commercial traffic.

4.2 Comfort observations on the German Railways (DR)

Schramm (1937) reported various observations from the German Railways in 1935, when 300 S-shaped superelevation ramps were introduced and compared with the normal linear superelevation ramps. The reason why these non-linear ramps were introduced is that their corresponding transition curves (the Helmert curve) in some cases require smaller re-alignments (than the traditional clothoid type of transition curve) when permissible train speeds are to be increased on existing lines (see Kufver 1997). The maximum speed was 120 km/h for fast multiple units and the derivative of cant was 28-35 mm/s on the linear ramps and a maximum 55-75 mm/s on the S-shaped ramps. According to Schramm, measurements of vibrations and accelerations showed that S-shaped ramps with a maximum derivative of cant of 70 mm/s were equally good compared with linear ramps with derivatives of cant of 28 mm/s. The passengers did not notice any differences between the two types of layout, but the experts ("Der Kenner") were able to notice the differences and considerd the S-shaped ramps superior.
4.3 Comfort tests on the Japanese Railways

In comfort tests conducted during 1961-1962 on the Japanese Railways, the dependent variables "sensibility", "feeling" and "judgement" were used (Urabe, Koyama & Iwase 1966). The test subjects rated these variables according the levels in Table 4.1.

<table>
<thead>
<tr>
<th>Sensibility:</th>
<th>0 score</th>
<th>no sensation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1 score</td>
<td>a little sensation</td>
</tr>
<tr>
<td></td>
<td>2 scores</td>
<td>clear sensation</td>
</tr>
<tr>
<td></td>
<td>3 scores</td>
<td>strong sensation</td>
</tr>
<tr>
<td>Feeling:</td>
<td>0 score</td>
<td>comfortable</td>
</tr>
<tr>
<td></td>
<td>1 score</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2 scores</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3 scores</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4 scores</td>
<td>uncomfortable</td>
</tr>
<tr>
<td>Judgement</td>
<td>0 score</td>
<td>acceptable train riding</td>
</tr>
<tr>
<td></td>
<td>1 score</td>
<td>unacceptable</td>
</tr>
</tbody>
</table>

Table 4.1  Levels used in Japanese comfort tests 1961-1962.
(Source: Urabe, Koyama & Iwase 1966.)

The magnitude and duration of lateral steady-state acceleration, magnitude of jerk and magnitude of peak acceleration were used as independent variables.

The authors reported an analysis of variance which, however, cannot be used in a research project concerning the optimisation of the alignment. An analysis of variance does not give the necessary partial derivatives which show the relative importance of each independent physical quantity in regard to perceived discomfort.

Comfort tests with tilting trains 1983 have been reported by Koyanagi (1985). Comfort statements "good" respectively "not good" were plotted in a roll velocity - roll acceleration plane. When the roll velocity was lower than 5 degrees/second (approximately 0.1 rad/second) and the roll acceleration was lower than 15 degrees/s² (approximately 0.3 rad/s²) there were no statements of "not good" from the test subjects. It was reported that the shape of the roll velocity signal on a transition curve ought to be sinusoidal. The simultaneously acting lateral jerk and lateral acceleration were not reported.

4.4 Comfort tests on railways in the USA


The tests have been conducted in different coaches in trains between Washington DC, New York and Boston and a PATCO urban rapid rail vehicle. The comfort ratings were made for carefully selected sections of the track which were announced by the experimenter. According to Pepler et al (1978), 80% of the test sections were straight and horizontal. The curved test sections were not analysed separately.
Paid test subjects rated comfort on a 5-point or 7-point discontinuous scale.

The motions of the vehicle floor were measured in six directions (three translational accelerations and three rotational velocities). The characteristics of band pass filters and weighting filters were not reported. Also acoustic noise and temperature were measured.

The researchers found that the noise level had a significant impact on the perceived discomfort and also reported certain relations between the comfort ratings and r.m.s. values of the measured motions, and in some cases the r.m.s. about the mean. (This latter r.m.s. value may have eliminated low frequency motions of the vehicle caused by the alignment.) In the analysis, one of the conclusions was that the motions in the six directions were correlated in the tests, and that studies where these cross-correlations are broken down or altered are needed, for future vehicle design.

4.5 Comfort tests on British Rail (BR)

BR have conducted comfort tests in 1949 and 1983-1984.

In the 1949 tests twelve members of the Track Committee in question and one other observer acted as test subjects (Loach and Maycock 1952). The test train consisted of a 2-6-2 tank locomotive, a vestibule coach with the test subjects and finally the District Engineers saloon. The train speed was varied in the range 30-45 miles per hour. The lateral acceleration in the coach was registered with a Hallade recorder and an Askania accelerometer, both placed approximately over the trailing bogie.

The tests were conducted on a track between Bettws-y-Coed and Blaenau festiniog in North Wales, with a normal speed restriction of 30 miles per hour. The first 14 curves were re-aligned and transition curves were arranged with a linear change of curvature and with corresponding superelevation ramps. The last 38 curves were not specially set out. Several of these curves did not include transition curves and the superelevation ramp was placed within the circular part and in some cases partly on the adjacent straight line.

The comfort rating was made as soon as an entering transition, a circular curve, and a run-out transition respectively was passed. Each curve section was announced from a loadspeaker. The rating was made on a seven-degree scale, Table 4.2.

---

4 At that time, the Hallade recorder was the standard instrument for measuring accelerations caused by track irregularities.
## Table 4.2 Sensation levels in the comfort tests 1949.
(Source: Loach and Maycock 1952.)

<table>
<thead>
<tr>
<th>Level</th>
<th>Sensation</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Nothing noticed</td>
</tr>
<tr>
<td>1</td>
<td>Just noticeable</td>
</tr>
<tr>
<td>2</td>
<td>Noticeable</td>
</tr>
<tr>
<td>3</td>
<td>Pronounced</td>
</tr>
<tr>
<td>4</td>
<td>Very pronounced but not at all uncomfortable</td>
</tr>
<tr>
<td>5</td>
<td>Strong and slightly uncomfortable</td>
</tr>
<tr>
<td>6</td>
<td>Rather uncomfortable</td>
</tr>
</tbody>
</table>

The lateral acceleration (expressed in inches of cant deficiency, as measured within the coach) and the corresponding comfort rating were plotted for circular curves 1-14 and 15-52 separately. On curves 1-14, the scatter of the results was fairly small. On average, at 3.5 inches of cant deficiency (89 mm cant deficiency or 0.58 m/s² lateral acceleration) the rating was 2, "noticeable", and at 6.6 inches of cant deficiency (168 mm cant deficiency or 1.10 m/s² lateral acceleration) the rating was 4, "very pronounced but not at all uncomfortable".

On curves 1-14, with re-aligned transition curves with linear change of curvature and cant, there was a correlation between the rate of gain or loss of cant deficiency and the comfort rating. For low values of the rates of change, a rate of loss was regarded as less uncomfortable than a rate of gain. For high values of rates of change of cant deficiency, such as 3-4 inches per second (76-102 mm cant deficiency per second or a 0.50-0.66 m/s³ lateral jerk), a rate of loss was rated equally uncomfortable to the rate of gain. However, these results were also affected by the rate of change of cant.

From tests with equilibrium speed on the curves, it was possible to study the comfort rating as a function of the rate of cant. At the equilibrium speed, no steady-state lateral acceleration is present on the curves. Rate of gain of cant and rate of loss of cant were plotted against the comfort rating. Again, on curves 15-52 without proper transition curves, there was a large scatter in the results. These curves had irregular lateral jerks even if there was no steady-state lateral acceleration in the middle part of the circle. On curves 1-14, there was also scatter in the results concerning rate of loss of cant, but for the rate of gain of cant there was a correlation to the comfort rating. The plot showed low sensation ratings also at high rates of gain of cant. A high rate of change of cant,
such as 2.5 inches per second (64 mm cant per second, or roll velocity 0.04 rad/s or 2.4 degrees/second) gave a sensation rating somewhere between 0.7 and 1.1 ("just noticeable").

During 1983-1984, comfort tests were conducted with an HST (High Speed Train) and an APT (Advanced Passenger Train). In the HST tests, Mk III coaches were used, one with BT10 bogies and one with BT17 bogies. The coach with BT17 bogies had defective dampers and therfore had worse dynamic behaviour than expected (Chapell 1984). The APT was a tilting train.

The HST tests included 64 test persons per day for four days and the APT tests included 64 test subjects per day for six days. The test subjects were employees of BR, but not only technical staff (Pollard 1984). In the APT tests, also a number of retired BR employees participated. The tests were conducted with pushbutton technique and the test subjects were instructed to press the button if "any aspects of the lateral ride" (Chapell 1984 p 5, Harborough 1986b p 1), were considered "uncomfortable" or "very uncomfortable" of the levels in Table 4.3. The test subjects were instructed to continuously press the button in "continuous sideways shaking" and "prolonged sideways forces" and to press the button once in "sudden sideways jerks" (Chapell 1984 pp 5-6).

Table 4.3 Levels used in the comfort tests during 1983-1984.
(Source: Harborough 1986a.)

<table>
<thead>
<tr>
<th>Comfort Level</th>
</tr>
</thead>
<tbody>
<tr>
<td>Very comfortable</td>
</tr>
<tr>
<td>Comfortable</td>
</tr>
<tr>
<td>Acceptable</td>
</tr>
<tr>
<td>Uncomfortable</td>
</tr>
<tr>
<td>Very uncomfortable</td>
</tr>
</tbody>
</table>

The comfort disturbances were found to occur at track irregularities and/or transition curves. These two cases were analysed separately. In the analysis of 277 circular curves with only small track irregularities, very few comfort disturbances were found. No seated passenger pressed the button if the lateral acceleration was less than 1.4 m/s². No standing passenger pressed the button if the lateral acceleration was less than 1.2 m/s² (Harborough 1986a).

The comfort disturbances from track irregularities were analysed in the following way: Within each ¼ mile, the largest vehicle reaction due to irregularities was evaluated if the peak acceleration exceeded 1.5 m/s² and if at least 3 seconds had passed since the previous comfort disturbance and/or the previous transition curve. The value 1.5 m/s² was chosen in order to include almost all irregularities which resulted in comfort disturbances and not to include too many irregularities which did not. The 3-second limit was chosen because of delays in the voting machine. For each comfort disturbance, peak lateral acceleration, peak-to-peak lateral acceleration and steady-state lateral acceleration were registered. Multiple regression analysis showed correlation between button pressings and both peak-to-peak acceleration and steady-state acceleration, but not for peak acceleration. The passenger and vehicle reactions from 504 track irregularities were used in the final multiple regression analysis. In Table 4.4, peak-to-
peak acceleration $\ddot{y}_p$ and (absolute value of) steady-state acceleration $\ddot{y}_m$ are expressed in per cent of gravitational acceleration $g$, and the two acceleration quantities should be evaluated in a sliding 2-second rectangular window.

<table>
<thead>
<tr>
<th></th>
<th>$k_0$</th>
<th>$k_1$</th>
<th>$k_2$</th>
<th>$r^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{DE}$ standing passengers</td>
<td>-37.0</td>
<td>1.63</td>
<td>2.65</td>
<td>0.66</td>
</tr>
<tr>
<td>$P_{DE}$ seated passengers</td>
<td>-21.7</td>
<td>0.83</td>
<td>1.28</td>
<td>0.66</td>
</tr>
</tbody>
</table>

**Table 4.4** Percentage of disturbed passengers $P_{DE}$ at track irregularities.

$P_{DE} = k_0 + k_1 \cdot \ddot{y}_p + k_2 \cdot \ddot{y}_m$  
(Source: Harborough 1986a.)

**Figure 4.1** Definitions of $\ddot{y}_p$ and $\ddot{y}_m$ in the $P_{DE}$ formulas.

(Source: CEN 1995. Modified by the author.)

In the evaluation of transition curves it was found that the comfort disturbances appeared in entry transitions and in reverse transitions. Transition curves where lateral acceleration decreased, i.e. run-off transitions, did not lead to any significant comfort disturbances, unless they also included large track irregularities. An explanation for this result may be that on circular curves, passengers expect the curve and hence the lateral acceleration soon to end. When this happens, it does not come as a surprise to the passengers. On a straight track, similar expectations (approaching a transition curve) do not exist and the direction of the next lateral jerk is unknown. This explanation is supported by the observation that passengers seem to be more disturbed when a circular curve is followed by another circular curve with higher lateral acceleration. In these cases, the passengers had the wrong expectation concerning the direction of the jerk on
the intermediate transition curve. Unfortunately, there were few examples of such compound curves, so BR was unable to analyse these observations further.

A number of transition curves were excluded from the statistical analysis; those with a temporary speed reduction and those with large track irregularities. The latter was excluded since it was impossible to distinguish the prime cause of the perceived discomfort. The statistical analysis thus comprised 1725 transition curves.

In the statistical analysis of comfort on transition curves, the effect of roll velocity was analysed separately. In some tests, the APT had the tilt system adjusted for 100% compensation of lateral acceleration. In these tests, there was almost no lateral acceleration or lateral jerk present in the coach (excluding certain effects of delays in the tilt system and effects of track irregularities.) The button pressings could then be regarded as caused only by the rolling.

Parameter estimation of certain non-linear functions gave the following expressions for the percentage of disturbed passengers:

\[
\text{Percentage disturbed among standing passengers} = 0.185 \cdot (\dot{\theta})^{2.283} \quad [4.1]
\]

\[
\text{Percentage disturbed among seated passengers} = 0.120 \cdot (\dot{\theta})^{1.626} \quad [4.2]
\]

where maximum absolute value of the roll velocity \(\dot{\theta}\) is expressed in degrees per second.

It was assumed to be possible to superimpose the discomfort of rolling on the discomfort caused by lateral acceleration and lateral jerk. With the help of the expressions [4.1] and [4.2] a multiple linear regression was conducted to quantify the perceived discomfort of lateral acceleration and lateral jerk. For standing passengers, the percentage of disturbed passengers on transition curves \(P_{CT}\) was then estimated as

\[
P_{CT} = \max(2.80 \cdot \ddot{y} + 2.03 \cdot \ddot{y} - 11.1, \ 0) + 0.185 \cdot (\dot{\theta})^{2.283} \quad [4.3]
\]

and for seated passengers

\[
P_{CT} = \max(0.88 \cdot \ddot{y} + 0.95 \cdot \ddot{y} - 5.9, \ 0) + 0.120 \cdot (\dot{\theta})^{1.626} \quad [4.4]
\]

where \(\ddot{y}\) = maximum absolute value of lateral acceleration of vehicle body, expressed in per cent of \(g\)

\(\dddot{y}\) = maximum absolute value of lateral jerk of vehicle body, expressed in per cent of \(g\) per second

\(\dot{\theta}\) = maximum absolute value of roll velocity of vehicle body, expressed in degrees per second
Figure 4.2  **Definitions of $\ddot{y}$ and $\dddot{y}$ in the $P_{CT}$ formulas.**  
(Source: CEN 1995. Modified by the author.)

Figure 4.3  **Definition of $\dot{\varphi}$ in the $P_{CT}$ formulas.**  
(Source: CEN 1995. Modified by the author.)
In comparisons of the results from the HST tests on the Midland Main Line (MML) and the HST tests on the more curved West Coast Main Line (WCML) certain minor differences were found which could possibly have been eliminated by the use of non-linear relationships between $P_{CT}$ and lateral acceleration and lateral jerk in equations [4.3] and [4.4].

In evaluation of the HST tests, it was possible to compare results from coaches with different bogies. The coach with BT17 bogies had a poorer ride than the coach with B10 bogies, and the r.m.s. values for vibrations were approximately 10% higher. Despite this fact, there were no significant differences in the comfort ratings of the test subjects (Harborough 1984, Harborough 1986a).

In a European prestandard from CEN (1995), the $P_{DE}$ and $P_{CT}$ functions were mentioned as non-mandatory evaluation functions. CEN stated that $P_{DE}$ evaluation should normally be conducted throughout the test run. The fact that BR excluded the transition curves in their $P_{DE}$ evaluation was not commented upon by CEN. CEN also stated that the formulas for $P_{CT}$ should be used for transition curves with a duration longer than 2 seconds. No procedure for shorter transitions was described. This 2-second rule could not be found in either the reports from BR or in ORE B153 Rp12 (1987), which also reported on the British HST and APT tests.

As mentioned in Sections 2.3-2.4, there is always additional uncertainty and inter-subject variability when labels such as "acceptable" and "uncomfortable" are used. The magnitude of lateral acceleration, lateral jerk and roll velocity which is acceptable depends on the expectations of the passengers. Hence, the absolute values of $P_{CT}$ must be interpreted with care, but in any case a larger value of $P_{CT}$ indicates more discomfort (Eickhoff 1997).

### 4.6 Comfort tests within ORE (ERRI) B153

The general objective of the B153 Committee was to standardise the evaluation of the exposure of individuals to whole-body vibrations in railway transport (ORE 1989b). The evaluation procedure should be compatible with the modified ISO Standard 2631. The ORE Committee suggested three comfort functions: Vehicle ride comfort $N_{MV}$, mean passenger comfort $N_{VA}$ for seated passengers and mean passenger comfort $N_{VD}$ for standing passengers. These comfort functions are based on linear as well as r.s.s. summation of band-pass filtered (1-80 Hz) and frequency weighted r.m.s. accelerations in vertical, lateral and longitudinal directions.

The evaluation procedure include calculation of r.m.s. accelerations every 5 seconds, and for a 5-minute period median values and 95% percentiles of the different r.m.s. accelerations should be used in the summation procedure. The sums $N_{MV}$, $N_{VA}$ and $N_{VD}$ represent passenger comfort during the 5-minute period in question.

Institut de Recherche des Transports in Lyon has reported a poor correlation between high-pass filtered lateral acceleration and passengers perceived discomfort. According to ORE (1983) the explanation might be that the high-pass filter eliminates the steady-state
lateral acceleration in curves, and since the original ISO 2631-1 Standard does not take into account accelerations in the frequency band 0-1 Hz, ORE (1987) decided to study how these accelerations influence ride comfort.

The French Railways (SNCF) conducted comfort tests with a CORAIL coach A10 rtu. The tests were conducted on three different track sections at test speeds of 80, 140 and 160 km/h respectively. The coach was in normal condition and the air conditioning was working. Six test subjects from the test department indicated in a report the perceived general comfort, the vibration comfort, the acoustic comfort and the ability to read and write for a preceding three-minute period. During the same period, the noise level, low frequency lateral accelerations in the floor and accelerations in the frequency range up to 200 Hz in the floor and in the seats were recorded. The analysis included 63 three-minute periods.

In a multiple regression analysis, it was found that the low frequency accelerations only had an influence on the acoustic comfort. However, the noise level had an influence on general comfort, vibration comfort, acoustic comfort and the ability to read.

The Polish Railways (PKP) conducted tests with a first class coach with BKW 200 bogies. The test speeds were 70 and 90 km/h. Eight employees from the Polish Railways Institute were test subjects. They were instructed to press one of five buttons at any time they perceived discomfort. The five buttons were used to indicate the following levels:

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Weak</td>
</tr>
<tr>
<td>2.</td>
<td>Relatively strong</td>
</tr>
<tr>
<td>3.</td>
<td>Strong</td>
</tr>
<tr>
<td>4.</td>
<td>Very strong but acceptable</td>
</tr>
<tr>
<td>5.</td>
<td>Not acceptable</td>
</tr>
</tbody>
</table>

**Table 4.5** Levels used in comfort tests on PKP. *(Source: ORE 1987.)*

During the tests, band-pass filtered (1-80 Hz) vertical and horizontal vibrations as well as low-pass filtered (0-1 Hz) lateral accelerations were recorded. No weighting filter was used. The band-pass filtered accelerations were expressed as r.m.s. acceleration and from the low-pass filtered lateral acceleration three quantities were calculated: Peak acceleration, steady-state acceleration and peak value of the jerk.

In a multiple regression analysis, 24 track sections with different levels of low frequency lateral acceleration were studied. The band-pass filtered vertical acceleration and the difference between peak acceleration and steady-state acceleration were found to be correlated with the button pressings.

In the analysis, ORE (1987) made the following conclusions: It is necessary to use more exact definitions of discomfort. It is also necessary to distinguish between momentary comfort and the aggregate ride comfort for a longer time period. Low frequency lateral acceleration has an influence on the momentary comfort. The effect of low frequency lateral acceleration on the aggregate ride comfort for longer periods must be studied further. The noise level affects the perceived ride comfort and the perceived ride comfort.
comfort could be improved significantly by a reduction of the low frequency acceleration within the coach.

### 4.7 Comfort tests on the Swedish State Railways (SJ)

Comfort tests were conducted in 1979 on an X15 train, an older EMU rebuilt with a tilt system, and in 1992 on an X2 tilting train. The tests were conducted with the pushbutton technique.

The tests in 1979 involved 20 test subjects (ASEA & SJ 1979). They were railway employees and were regarded as experienced passengers since in previous years they had travelled by long-distance train more than six times a year. The following instruction was used on the first day:

"Press the button each time you perceive discomfort. You should not register shaking, jerks, etc. which according to your opinion are normal, i.e. not disturbing during train journeys. The button should be pressed twice if you perceive very strong discomfort." (ASEA & SJ 1979 p 10.13, Forstberg 1993 p 4, translated from Swedish)

This instruction gave too few pressings of the button, so from the second day a new instruction was used:

"Press the button each time you perceive shaking, jerks, etc. which according to your opinion correspond to the limit of a good train ride. In very strong disturbances, the button should be pressed twice." (ASEA & SJ 1979 p 10.13, Forstberg 1993 p 5, translated from Swedish)

After each test, the test subjects were asked to give a summary of the perceived comfort in a questionnaire. In this questionnaire the following scales were used:

<table>
<thead>
<tr>
<th>I consider the tilting of the train was ...</th>
</tr>
</thead>
<tbody>
<tr>
<td>very uncomfortable</td>
</tr>
<tr>
<td>slightly uncomfortable</td>
</tr>
<tr>
<td>perceptible, but not uncomfortable</td>
</tr>
<tr>
<td>not perceptible</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>In summary, I consider the journey was ...</th>
</tr>
</thead>
<tbody>
<tr>
<td>uncomfortable</td>
</tr>
<tr>
<td>slightly uncomfortable</td>
</tr>
<tr>
<td>acceptable</td>
</tr>
<tr>
<td>comfortable</td>
</tr>
<tr>
<td>very uncomfortable</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>According to my experience, I consider this journey on the X15 tilting train, compared with an ordinary long-distance train, was</th>
</tr>
</thead>
<tbody>
<tr>
<td>less comfortable</td>
</tr>
<tr>
<td>approximately equally comfortable</td>
</tr>
<tr>
<td>more comfortable</td>
</tr>
</tbody>
</table>

**Figure 4.4**

*Final comfort rating in the X15 tests.*

(Source: ASEA & SJ 1979, translated from Swedish.)
The following diagrams illustrate the results from the tests in 1979:

Figure 4.5 Reaction probability as a function of lateral acceleration.  
(Source: ASEA & SJ 1979.)

Figure 4.6 Reaction probability as a function of lateral jerk.  
(Source: ASEA & SJ 1979.)
Figure 4.7 Reaction probability as a function of roll velocity.
(Source: ASEA & SJ 1979.)

It was noted that the variables lateral acceleration, lateral jerk and roll velocity covaried and had their maximal magnitudes approximately at the same time; at the tangent point between transition curve and circular curve. This "... makes it difficult to judge whether a passenger reaction is caused by a high level of acceleration or jerk" (ASEA & SJ 1979 p 10.20, translated from Swedish). No statistical evaluation was presented.

It has also been reported that the level of acoustic noise was unusually high in the X15 train (Nilstam 1996).

Also the tests during 1992 included 20 test subjects and a new instruction was used:

"The aim of this test is to evaluate how you as a passenger perceive the discomfort of shaking in the vehicle, caused by shocks from track irregularities... Press the button once each time you perceive shaking, jerks, etc. which according to your opinion exceed your requirements on good comfort. Press twice rapidly if you perceive very strong disturbances... Observe that the aim of the tests is not to evaluate the general level of vibrations but your reactions to shaking caused by large irregularities." (Förstberg 1993 Appendix 4.7 p 1, translated from Swedish)

The 20 test subjects were track engineers from the Swedish National Railway Administration (BV), employees at the mechanical engineering division of the Swedish State Railways (SJ), employees at the rolling stock unit of the passenger division of SJ and employees on the X2000-trains.
In the evaluation of the tests in 1992, it was concluded that there are probably relations between the button pressings and a number of physical quantities. These quantities were peak-to-peak lateral acceleration, lateral jerk and vertical acceleration. Relations between the button pressings and quasi-static lateral acceleration, tilt angle, time derivative of tilt angle, roll velocity of the vehicle body and roll acceleration of the vehicle body were found to be unlikely.

The evaluation was performed in the following way: In a group of five test subjects, one pressing was interpreted as indicating that 20% of the passengers were disturbed, two pressings 40% and so on. Five pressings were interpreted as indicating that 100% of the passengers were disturbed and the same was concluded when six to ten pressings were recorded. "This because all five persons were assumed to have perceived discomfort and only the magnitudes of the comfort disturbances changed." (Fürstberg 1993 p 13, translated from Swedish)

The achieved percentages were used for parameter estimations of exponential functions and logit functions. It was stated that an advantage of the logit functions is that predicted button pressings cannot exceed 100%. Only one-factor analysis was performed. Achieved correlations between vertical acceleration and button pressings might be caused by the covariation between vertical and lateral acceleration in the tests.

<table>
<thead>
<tr>
<th>Independent variable</th>
<th>a</th>
<th>b</th>
</tr>
</thead>
<tbody>
<tr>
<td>x=lateral acceleration (BP 0.2 - 1.5 Hz)</td>
<td>0.73</td>
<td>72.44</td>
</tr>
<tr>
<td>x=lateral acceleration (BP 0.2 - 0.6 Hz)</td>
<td>0.79</td>
<td>1375</td>
</tr>
<tr>
<td>x=lateral jerk (BP 0.2 - 0.6 Hz)</td>
<td>0.54</td>
<td>23.44</td>
</tr>
<tr>
<td>x=lateral jerk (LP 0.3 Hz)</td>
<td>0.64</td>
<td>4510</td>
</tr>
</tbody>
</table>

*Table 4.6* Coefficients for an exponential function, standing passengers.  
\((y=a \cdot b^x)\) (Source: Fürstberg 1994.)

<table>
<thead>
<tr>
<th>Independent variable</th>
<th>a</th>
<th>b</th>
</tr>
</thead>
<tbody>
<tr>
<td>x=lateral acceleration (BP 0.2 - 1.5 Hz)</td>
<td>0.76</td>
<td>39.80</td>
</tr>
<tr>
<td>x=lateral acceleration (BP 0.2 - 0.6 Hz)</td>
<td>1.47</td>
<td>129.4</td>
</tr>
<tr>
<td>x=lateral jerk (BP 0.2 - 0.6 Hz)</td>
<td>0.46</td>
<td>21.16</td>
</tr>
<tr>
<td>x=lateral jerk (LP 0.3 Hz)</td>
<td>0.58</td>
<td>4720</td>
</tr>
</tbody>
</table>

*Table 4.7* Coefficients for an exponential function, seated passengers.  
\((y=a \cdot b^x)\) (Source: Fürstberg 1994.)

<table>
<thead>
<tr>
<th>Independent variable</th>
<th>A</th>
<th>B</th>
</tr>
</thead>
<tbody>
<tr>
<td>x=lateral acceleration (BP 0.2 - 1.5 Hz)</td>
<td>-4.94</td>
<td>4.75</td>
</tr>
<tr>
<td>x=lateral acceleration (BP 0.2 - 0.6 Hz)</td>
<td>-4.42</td>
<td>7.94</td>
</tr>
<tr>
<td>x=lateral jerk (BP 0.2 - 0.6 Hz)</td>
<td>-5.09</td>
<td>3.25</td>
</tr>
<tr>
<td>x=lateral jerk (LP 0.3 Hz)</td>
<td>-5.43</td>
<td>10.94</td>
</tr>
</tbody>
</table>

*Table 4.8* Coefficients for a logit function, standing passengers.  
\((y=e^{A+BX}/(1+e^{A+BX}))\) (Source: Fürstberg 1994.)
Table 4.9  Coefficients for a logit function, seated passengers.

\[ y = \frac{e^{A+Bx}}{1 + e^{A+Bx}} \]  

(Source: Försberg 1994.)

In Tables 4.6-4.9 BP denotes band-pass filter and LP denotes low-pass filter, with filter gradients of 24 dB/octave for the 1.5 Hz filter and 12 dB/octave for the other filters.

4.8  Comfort test on the Stockholm underground (SL)

At the request of SL, SJ and KFB, The Department of Transportation and Logistics at Chalmers University of Technology has investigated comfort disturbances at the passage of a turnout curve on the Stockholm underground (Johansson 1991). The tests were conducted in 1990 with 40 test subjects and with a train consisting of C13 and C15 multiple units.

The tests were conducted with the pushbutton technique. The test subject was instructed to press the button once when his/her judgement was "It is good the journey is fast, but if it is as jerky as this, I want the train to go slower" and to press the button twice when his/her opinion was "It is good the journey is fast, but I refuse to be treated like this" (Johansson 1991 p 3, translated from Swedish).

During the tests, the button pressings, the accelerations in longitudinal, lateral and vertical direction, the noise level and the train speed were registered. The acceleration signals were low-pass filtered with a cut-off frequency of 0.5 Hz. (The gradient of the filter was not presented.) The jerk signals have been calculated as the change of acceleration in one second.

The dependent variable in the comfort function was defined as the number of button pressings multiplied by 100 and divided by the number of test subjects. Hence, the best comfort rating was zero and the worst 200. Contrary to the SJ tests, all button pressings are regarded in the comfort function.

Tests were conducted in a circular turnout curve with the radius 300 m, without transition curves. The train passed the turnout at speeds of 30, 40 and 50 km/h.

The relations which were tested were linear relations where the comfort rating was dependent on

- lateral acceleration
- lateral jerk
- lateral acceleration and lateral jerk
- lateral acceleration in the power of two
- lateral jerk in the power of two
- longitudinal train speed
The results from the entire group of 40 test subjects are presented in Tables 4.10 and 4.11. In the final report of the project, the results from four subgroups (commuters, car users, students and retired persons) were presented. With such a separation of the test subjects, negative coefficients $b_1$ or $b_2$ were achieved for all subgroups in the multiple regression analysis. This indicates that a higher level of lateral acceleration or lateral jerk results in improved comfort. Johansson (1991 p 23) stated that "it is unlikely that this is the actual case" (translated from Swedish) but did not discuss these results further.

<table>
<thead>
<tr>
<th>Independent variable</th>
<th>a</th>
<th>$b_1$</th>
<th>$b_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x=$lateral acceleration</td>
<td>9.0</td>
<td>101.7</td>
<td>-</td>
</tr>
<tr>
<td>$x=$lateral jerk</td>
<td>33.9</td>
<td>74.2</td>
<td>-</td>
</tr>
<tr>
<td>$x_1=$acceleration; $x_2=$jerk</td>
<td>26.0</td>
<td>25.5</td>
<td>57.9</td>
</tr>
<tr>
<td>$x=$squared acceleration</td>
<td>46.5</td>
<td>63.2</td>
<td>-</td>
</tr>
<tr>
<td>$x=$squared jerk</td>
<td>65.1</td>
<td>34.0</td>
<td>-</td>
</tr>
<tr>
<td>$x=$train speed</td>
<td>-40.6</td>
<td>2.9</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 4.10 Coefficients in a comfort function, standing passengers. 
($B=a + b_1x_1 + b_2x_2$) (Source: Johansson 1991.)

<table>
<thead>
<tr>
<th>Independent variable</th>
<th>a</th>
<th>$b_1$</th>
<th>$b_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x=$lateral acceleration</td>
<td>-3.2</td>
<td>63.3</td>
<td>-</td>
</tr>
<tr>
<td>$x=$lateral jerk</td>
<td>12.7</td>
<td>46.2</td>
<td>-</td>
</tr>
<tr>
<td>$x_1=$acceleration; $x_2=$jerk</td>
<td>-2.9</td>
<td>61.1</td>
<td>2.0</td>
</tr>
<tr>
<td>$x=$squared acceleration</td>
<td>16.1</td>
<td>47.0</td>
<td>-</td>
</tr>
<tr>
<td>$x=$squared jerk</td>
<td>25.0</td>
<td>36.6</td>
<td>-</td>
</tr>
<tr>
<td>$x=$train speed</td>
<td>-2.0</td>
<td>1.06</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 4.11 Coefficients in a comfort function, seated passengers. 
($B=a + b_1x_1 + b_2x_2$) (Source: Johansson 1991.)

4.9 Comfort tests in ERRI B207

The aim of the work in ERRI B207 (1996) was to verify the procedures suggested by CEN WG7 TC256 (which are similar to $N_{MV}$ and $N_{VA}$ suggested by ORE B153 and $P_{CT}$ and $P_{DE}$ suggested by BR) and to integrate the procedure of evaluation of passenger momentary comfort in curves ($P_{CT}$) into the 5-minute average passenger comfort (represented by $N_{MV}$ and $N_{VA}$).

The committee compared pushbutton techniques used by BR and the use of a discontinuous scale. The committee decided to use scales in their own tests and hence comparisons with $P_{CT}$ will be difficult. In a discussion concerning statistical evaluation, they stated that the linear regression does not automatically provide the best description of the relations between physical quantities and perceived discomfort. The estimated parameters are dependent on the model chosen by the analyst. They required the
achieved model to be reasonable and consistent with a physical model. Hence, negative parameters (indicating that a higher level of acceleration decreases the perceived discomfort) cannot be accepted.

Tests were conducted in Poland and Italy. The Polish tests were conducted using a first class coach with 25 AN bogies on a track between Dzialdowo and Rakowice on the Warsaw - Gdansk line. Curves were passed with lateral acceleration of 0.6-1.0 m/s² in the superelevated track plane. Eleven of the twelve test subjects were employees of the Polish Railways, a third of those were technical staff with fairly good knowledge of vibration problems. The test subjects were asked to make two types of comfort assessments; instantaneous comfort at specific locations which were announced by lamps and acoustic signals and average comfort every 5 minutes. They were instructed to evaluate comfort related to vibration, oscillation, lateral acceleration, impacts and various general vehicle motions and not to take environmental aspects (noise, temperature etc.) into account. Comfort was rated on a five-level scale from "very comfortable" to "very uncomfortable".

The Italian tests were conducted with a Pendolino ETR 460 tilting train on the former Florence - Rome line as well as on the new Florence - Rome high speed line. Curves were passed with lateral acceleration up to 1.8 m/s² in the superelevated track plane. All twelve test subjects were employees of the Italian Railways. Also these test subjects were asked to rate momentary comfort as well as average comfort on a five-level scale. They were asked to assess comfort related to the dynamic behaviour of the vehicle (vibration, oscillation, impacts and vehicle motions in general) and to ignore other aspects of comfort such as noise, temperature, lighting and pressure shocks in tunnels.

From the test results, the committee draw the following conclusions. In tilting trains or when travelling at high speed on heavily curved lines, the average passenger comfort should be estimated as

\[
\text{Discomfort} = 0.84 \cdot N_{VA} + 0.30 \cdot \dot{\theta} - 1 \tag{4.5}
\]

where the coach roll velocity \( \dot{\theta} \) is the maximum average during 0.1 seconds of the absolute values and is expressed in degrees per second. The formula assumes that steady-state lateral acceleration is less than 1 m/s². The committee noted that jerk showed covariation with \( N_{VA} \) in the Polish tests and generally showed a low variation in the Italian tests. It would have been better to include jerk explicitly in the formula instead of concealing it.

Concerning comfort on transition curves, the committee wanted to avoid the use of independent variables which are laborious to obtain. Therefore, the committee suggested the following formula

\[
\text{Discomfort} = 1.4 \cdot \ddot{Y} + 0.23 \cdot \dot{\theta} + 0.98 \tag{4.6}
\]

where the lateral acceleration \( \ddot{Y} \) in the coach is expressed in m/s² and roll velocity \( \dot{\theta} \) of the coach is expressed in degrees per second. Both physical quantities are calculated as the maximum average during 0.1 seconds of the absolute values.
5. Discussion

From Chapters 3 and 4 it is clear that we are far from the world of perfect knowledge where discomfort costs may be added into a total cost function and where the optimal alignment can be derived from minimisation of that function of total cost. At best, it is possible to extract the most relevant relations between physical quantities and perceived discomfort, in order to minimise discomfort when comparing different but equally costly alignment alternatives. It would also be possible, in individual cases, to quantify the relation between the construction cost for an alignment and the lowest possible associated level of passenger discomfort.

The most complete comfort assessment is the general model of Parsons and Griffin. In principle, the model makes it possible to consider motions generated by the track geometry as well as track irregularities, motions in different directions, impulsive motions (generated by singular misalignments) and motion patterns of different duration (shorter or longer journeys). However, there is a crucial lack of knowledge about the discomfort caused by accelerations of very low frequencies (lower than 0.5 Hz).

Griffin (1990) stated that there are indications that perceived discomfort is proportional to acceleration, but data presented by Yonekawa and Miwa (1972) indicate that equivalent comfort curves (plotted in an acceleration-frequency plane), in the frequency range of 0.05-1.0 Hz, are functions where acceleration depends on frequency in the power of approximately 0.8.

Figure 5.1  Lateral accelerations in two alignment alternatives.
The use of the general model of Parsons and Griffin may be illustrated with two alternative alignments, which generate the lateral acceleration in Figure 5.1. In one of the alternatives, the lateral acceleration is zero for two seconds, linearly increasing for four seconds, constant at 1 m/s² for four seconds, linearly decreasing for four seconds and finally zero for two seconds. In the other case, the time scale is changed. In order to obtain the same duration for each level of acceleration, the time history is repeated. Therefore, also the total duration is the same for the two alternatives. (It should be noted that the two curve alternatives generate the same change of direction and hence may be substitutes with the same position of the adjacent straight lines.)

Without a frequency weighting, the two acceleration patterns would be regarded as equally comfortable. With normal frequency weighting (according to Section 3.2) for frequencies above 0.5 Hz, and assuming constant weights for frequencies below 0.5 Hz, the passengers are assumed to be less sensitive to higher frequencies. Therefore, as a result of the frequency weightings, the first (low frequency) alternative would be regarded as more uncomfortable than the second alternative. However, the first alternative has the same level of lateral acceleration and a lower lateral jerk than the second alternative. The first alternative ought to be regarded as better and calculations of \( P_{CT} \) values for the two alternatives indicate that the first alternative is more comfortable.

Even though the general model of Parsons and Griffin considers many aspects of the accelerations, it does not appear to give reasonable results at very low frequencies, which are caused by the alignment. If the frequency weightings are modified according to the equivalent comfort curves of Yonekawa and Miwa (1972), the results from the general model will favour high frequency motions even more. To obtain consistency between the general model and the \( P_{CT} \) evaluation, it would be necessary to have equivalent comfort curves where acceleration is a function of frequency in the power of a negative value (at least at very low frequencies).

---

5 This discussion makes a distinction between the frequency weightings and the band-pass filters, see BS 6841 (1987). In many other documents, this distinction is lost.
In the general model of Parsons and Griffin, as well as in the comfort evaluations of BS 6841 (1987) and ISO 2631-1 (1985), it does not matter whether a time history of lateral acceleration is run forwards or backwards (Figure 5.2), since the phase information is disregarded. However, the results from the British Rail comfort tests 1983-1984 showed that passengers were not disturbed when the absolute value of a lateral acceleration decreased. Comfort disturbances occurred only when the absolute value of the acceleration increased. Hence, $P_{CT}$ should only be calculated for such transition curves where the absolute value of the lateral acceleration increases, and in this way the two alternatives in Figure 5.2 will be rated differently. Vogel (1936) had drawn similar conclusions concerning expected and unexpected jerks respectively, and Loach and Maycock (1952) noted that test subjects were less disturbed by rate of loss of cant deficiency than by rate or gain of cant deficiency.

The results from the British Rail comfort tests in 1983-1984 include certain relations of great importance, even though they are based on the pushbutton technique. The magnitudes of lateral acceleration, lateral jerk and roll velocity, which are dependent on the curve radius, transition length and gradient in the superelevation ramp (as well as vehicle speed and roll coefficient), are taken into account in the $P_{CT}$ value. But it must also be noted that certain other quantities, which ought to affect the evaluation of railway alignments, are not included in $P_{CT}$. 

---

**Figure 5.2**  
Lateral acceleration with the same discomfort according to the general model of Parsons and Griffin but with different $P_{CT}$ values.
One quantity which is not included in $P_{CT}$, but which may be important, is the duration of the lateral acceleration since it is related to the lengths of the circular curves. In the general model of Parsons and Griffin, as well as in the comfort evaluations of BS 6841 (1987) and ISO 2631-1 (1985), this is principally taken into consideration through the frequency weightings. But few studies, if any, concerned frequencies so low that they are relevant for evaluations of railway alignments.

The duration of the journey is not taken into account in the $P_{CT}$ value, while there are time dependences in ISO 2631-1 (1985) and in the general model of Parsons and Griffin (1983).

It must also be questioned whether or not it is reasonable to use the $P_{CT}$ evaluations in unconventional alignments, such as alignments with continuous derivative of curvature (transition curves with S-shaped curvature function). Only clothoids were evaluated in the British tests, and they have their maximum values of curvature, derivative of curvature and derivative of cant on the same longitudinal position of the line, at the tangent point between the transition curve and the circular curve. Transition curves with S-shaped curvature pattern have maximum curvature at one end of the transition curve, while maximum values of derivative of curvature and derivative of cant are achieved somewhere in the middle of the transition curve. According to hypotheses concerning expectations, it should be an advantage that these derivatives have their maximum values in the middle and not at the ends of the transition curves. It should also be noted that the durations, with high magnitudes of the derivatives of curvature and cant, are short for the unconventional transition curves, while the clothoids have a constant level of the derivatives. (See also the comfort observations from the German Railways above). Hence, a parameter estimation for $P_{CT}$ functions may be different for different types of transition curves.

According to the CEN standard, the formulas for $P_{CT}$ are only valid for transition curves with a duration of at least 2 seconds. Such a limitation was not given in the reports from BR, and CEN did not give any explanation for the limitation. But since the jerk in the $P_{CT}$ function is averaged during 1 second, evaluation of transition curves shorter than 1 second could justify another definition of the jerk used in $P_{CT}$.

Since $P_{DE}$ is evaluated within a 2-second window, it might be reasonable to ask whether transition curves shorter than 2 seconds should be evaluated with $P_{DE}$ rather than $P_{CT}$. Calculations of $P_{CT}$ and $P_{DE}$ have been made for a lateral acceleration which starts with zero and in 2 seconds increases to a constant value. (Such an acceleration pattern disregards the dynamic behaviour of the vehicle and the possible additional influence of rolling.) If it was reasonable to switch between $P_{CT}$ and $P_{DE}$ evaluation, the two functions ought to have the same value at the 2-second transition. The results in Table 5.1 do not support the use of $P_{DE}$ for short transition curves.
<table>
<thead>
<tr>
<th>Maximum lateral acceleration</th>
<th>$P_{CT}$ standing</th>
<th>$P_{DE}$ standing</th>
<th>$P_{CT}$ seated</th>
<th>$P_{DE}$ seated</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05 \cdot g</td>
<td>8%</td>
<td>0</td>
<td>1%</td>
<td>0</td>
</tr>
<tr>
<td>0.10 \cdot g</td>
<td>27%</td>
<td>16%</td>
<td>8%</td>
<td>4%</td>
</tr>
<tr>
<td>0.15 \cdot g</td>
<td>46%</td>
<td>42%</td>
<td>14%</td>
<td>17%</td>
</tr>
<tr>
<td>0.20 \cdot g</td>
<td>65%</td>
<td>69%</td>
<td>21%</td>
<td>30%</td>
</tr>
<tr>
<td>0.25 \cdot g</td>
<td>84%</td>
<td>95%</td>
<td>28%</td>
<td>43%</td>
</tr>
</tbody>
</table>

Table 5.1  
$P_{CT}$ and $P_{DE}$ in a 2-second linearly increasing acceleration pattern.

In the estimation of parameters in $P_{CT}$, BR disregarded transition curves with large track irregularities. Therefore, the passenger discomfort of large track irregularities in transition curves has still not been investigated.

The reported results from the Swedish comfort tests, described in Section 4.7, are less useful when evaluating alignment alternatives, mainly because only one-dimensional analysis was performed. From the tests in 1979, it was reported that it was difficult to distinguish between discomfort caused by lateral acceleration and discomfort caused by lateral jerks.

According to Nilstam (1996) the test subjects were exposed to an abnormally high acoustic noise level in the X15 tests. This may have affected the comfort rating of the lateral accelerations, the lateral jerks and the roll velocities.

Howarth and Griffin (1990) found a reduced sensitivity to vertical accelerations in the presence of high noise levels. In such cases, the discomfort caused by the noise might dominate the discomfort caused by accelerations.
Also in the SNCF tests within ORE B153 (1987) it was concluded that the acoustic noise has a major influence on the general comfort level as well as the vibrational comfort level.

In the Swedish comfort tests in 1992, the test subjects were informed that the purpose of the tests was to evaluate comfort in impulsive track irregularities. It is likely that at least the five track engineers (working with alignment calculations), and perhaps also the employees at the mechanical engineering division of SJ and at the rolling stock unit of the passenger division of the same company, were well aware of the difference between track irregularities and horizontal curves, and for this reason did not indicate types of discomfort such as a high value of quasi-static lateral acceleration and high roll velocities on transition curves. The conclusions from the 1992 tests, that quasi-static lateral acceleration, roll velocity and roll acceleration do not affect passenger comfort, are not well grounded.

In the comfort tests on the Stockholm underground, only one type of alignment was tested, a circular curve of 300 metres radius connected directly to a straight line (without intermediate transition curve). This alignment was passed at different speeds. The resulting values for lateral acceleration and lateral jerk therefore covaried in a way that made it impossible to distinguish their effects on passenger comfort. Such a problem has already been discussed by Vogel (1936) and provides the most likely explanation of why Johansson obtained incorrect signs for his parameters when he separated his test subjects into four subgroups.

Figure 5.3  **Principal relations between vibrations and discomfort at three different levels of acoustic noise.** (Source: Howarth & Griffin 1990.)
It also seems likely that a large inter-subject variation was present in the comfort tests on the Stockholm underground. The test subjects were asked to press the button twice when their opinion was "... I refuse to be treated like this" (Johansson 1993 p 3, translated from Swedish). It is hard to believe that the test subjects really meant that they would cancel their journeys or select another mode of transport just because the underground trains pass a type of turnout at 30-50 km/h which in several countries in Europe is used for train speeds of 50 km/h.

The pushbutton technique may give biased results because of low resolution of the measuring instrument (see Figure 2.4). On the other hand, the use of graded scales may make it difficult to judge whether comfort is sufficient or not. (It may seem unnecessary to rate and compare acceleration patterns which are in any case unacceptable for railway operations.) The solution to these problems may be to use the measuring instrument which the Japanese Railways used in their comfort tests in 1961-1962 (see Section 4.3).

The rotational motions of yaw and pitch have been disregarded in the comfort tests in railway vehicles reported in Chapter 4 (except Jacobson, Richards and Kuhlthau 1980 and Richards, Jacobson and Pepler 1980). The reason for this may be the fact that when such motions are caused by the alignment, the simultaneously created translations may dominate the rotational motions. It must also be noted that the direction and magnitude of translational accelerations caused by rotational velocities and rotational accelerations depend on the position of the test subject. Hence, different test subjects in the same test run are exposed to different translational accelerations.

Until now, the comfort tests within ERRI B207 have not provided useful formulas for the optimisation of alignments. The committee required that a statistical model be consistent with a reasonable physical model, but their suggested comfort function for evaluation of transition curves does not regard jerk. Such an oversimplified formula states that if no roll velocity is present, there is no benefit of lengthened transition curves (which is unlikely to be generally true). In fact, what is needed for a better base for optimising alignments is a more complex comfort function which regards more, not fewer, of the relations indicated in Figure 2.1.
6. Conclusions

The conclusion of this report is that the $P_{CT}$ functions are the most reasonable overall comfort functions when optimising railway alignments. The reason for this is that $P_{CT}$ includes the lateral acceleration, lateral jerk and roll velocity, which are the most basic physical quantities when calculating track geometry. Another reason for using the $P_{CT}$ evaluation is that $P_{CT}$ has been proposed by CEN as an international standard.

An alternative method for comfort rating which may be considered is the general model of Parsons and Griffin. However, there is a crucial lack of knowledge concerning the weightings of very low frequency accelerations.

On the other hand, when using $P_{CT}$ there is a lack of information concerning effects of the duration of the journey, the number of curve entries and large track irregularities, as well as the possible benefit of transition curves with S-shaped curvature functions. However, when optimising the alignment of single curves, the duration of the journey and the number of curve entries will be the same for all alignment alternatives. It must also be questioned whether or not large track irregularities should be taken into account when designing alignments.

Research activities aimed at establishing methods for optimising the alignment may be described as a chain linking different areas of knowledge. The first link describes a model for the geometric base for comparisons of alignment alternatives. The second link describes the relations between the track geometry and the vehicle reactions and the third link evaluates the vehicle reactions. If the third link is weak, because of uncertainties concerning the validity of comfort assessment with $P_{CT}$ functions, there is still a value in establishing the other links. These may enable the optimisation to be revised, with an improved comfort function.

The need for further research concerning discomfort in railway vehicles has been pointed out by Andersson (1988) and Parsons and Griffin (1993). One possible result from a project concerning optimisation of railway alignments is a more detailed description of the need for further knowledge of the relations between ride comfort, physical quantities and quantities concerning track geometry.
References


Appendix 1

Terminology

Alignment
The alignment is a mathematical description of the designed position of the centre line of the track. An alignment consists of elements which are connected at tangent points. An alignment can be divided into a horizontal alignment and a vertical alignment.

Cant
The difference between the heights of the right and left rails, at the same longitudinal position, is called cant and/or superelevation. The cant is positive if the outer rail is higher than the inner rail.

Cant deficiency
When the cant is lower than equilibrium cant, the difference is called cant deficiency.

Compound curve
A chain of elements consisting of at least two circular curves in the same direction, without any intermediate straight line.

Crest factor
The crest factor is defined as the frequency-weighted peak value divided by the frequency-weighted r.m.s. value of the same variable.

Curvature
The horizontal curvature is the derivative of the direction of the track. On a straight line, the curvature is zero and on a horizontal circular curve, the absolute value of curvature is the inverse of the radius. On a vertical curve, the vertical curvature is the derivative of arcus tangens of the slope.

Element
An element is a section of the track with specific mathematical characteristics. An element may be a straight line, a circular curve or a transition curve.

Equilibrium cant
The equilibrium cant is the amount of cant which eliminates the quasi-static lateral acceleration (in the track plane).

Gauge
The lateral distance between the inner edges of the two rails is called gauge. Standard gauge is 1435 mm.

Jerk
Time derivative of acceleration.

Misalignment
A misalignment is a track irregularity in the horizontal alignment.

Octave band
A frequency interval where the upper limit is twice the lower limit, \( f_{\text{low}} \leq f < f_{\text{high}} = 2 \cdot f_{\text{low}} \). Octave bands may be divided into one-third octave bands, \( f_{\text{low}} \leq f < f_{\text{high}} = \frac{3}{2} \cdot f_{\text{low}} \).
Appendix 1

R.m.q. Root-mean-quad, an average quantity calculated from a time history of a signal, \( a_{r_{mq}} = \sqrt{\frac{1}{T_2 - T_1} \int_{T_1}^{T_2} a^4(t) \, dt} \).

R.m.s. Root-mean-square, an average quantity calculated from a time history of a signal, \( a_{r_{ms}} = \sqrt{\frac{1}{T_2 - T_1} \int_{T_1}^{T_2} a^2(t) \, dt} \).

R.s.q. Root-sum-of-quad, a summation procedure for accelerations in different directions, \( a_{r_{sq}} = \sqrt{a_x^4 + a_y^4 + a_z^4} \).

R.s.s. Root-sum-of-squares, a summation procedure for accelerations in different directions. If the accelerations are of the same type (translational or rotational) and if the directions are orthogonal, the r.s.s. procedure is equivalent to vector summation, \( a_{r_{ss}} = \sqrt{a_x^2 + a_y^2 + a_z^2} \).

Roll angle Roll angle \( \varphi \) defines rotation around x axis. In some literature the roll angle is denoted \( r_x \).

Running r.m.s. A filtering procedure for smoothing a very irregular time history (shocks and high crest factor motions). A new time history is created, \( a_r(t_0) = \sqrt{\frac{1}{T} \int_{t_0}^{t_0+T} a^2(t) \, dt} \) or alternatively
\[
a_r(t_0) = \sqrt{\frac{1}{T} \int_{-\infty}^{t_0} a^2(t) e^{-\frac{t-t_0}{T}} \, dt}.
\]

Superelevation See cant.

Superelevation ramp A superelevation ramp is a section of the track where the cant changes gradually. (According to the standards of many railway companies, a superelevation ramp should normally coincide with a transition curve.)

Tangent point A tangent point is where two elements are connected to each other.
Appendix 1

Track irregularity

When the actual track position does not coincide with the designed position, track irregularities exist. The mismatch between actual position and designed position could involve horizontal alignment, gauge, vertical alignment and/or cant. The track irregularities are more frequently described by deviations in the first derivative (cant) or second derivative (alignment) of the position, rather than deviations in the position itself.

Track plane

The track plane is a function of longitudinal distance. The instantaneous direction of the track centre line defines the x-axis within the track plane, while the y-direction within the track plane is tangent to the designed positions of the top surfaces of the two rails.

Transition curve

A transition curve is a type of element where the curvature changes gradually.

\[ VDV = \sqrt[4]{\int_0^T a^4(t) \, dt} \]

Vertical curve

A vertical curve is a section of the track where the slope changes gradually. Normally, vertical curves consist of circular curves or parabolas.

x, y and z-directions

In ISO 2631-1 (1985, 1995) and BS 6841 (1987) the x, y and z-directions are related to individuals, with the x-direction back-to-chest, the y-direction right-to-left side and the z-direction buttocks-to-head. However, when evaluating passenger comfort in railway vehicles, the directions refer to the vehicle body; the x and y-directions are longitudinal and lateral directions parallel to the floor, while the z-direction is perpendicular to the floor.
Appendix 2

Notations

\( \dot{a} \) Acceleration (m/s\(^2\))
\( a \) Regression coefficient
\( A \) Regression coefficient (-)
\( b_i \) Regression coefficients
\( B \) Comfort function (-)
\( B \) Regression coefficient
\( c_i \) Regression coefficients (-)
\( f \) Frequency (Hz)
\( g \) Gravitational acceleration (m/s\(^2\))
\( k \) Constant in Steven's power law
\( k_i \) Regression coefficients
\( n \) Exponent in Steven's power law (-)
\( \text{N} \text{MV} \) Vehicle ride comfort (-)
\( \text{N} \text{VA} \) Mean passenger comfort for seated passengers (-)
\( \text{N} \text{VD} \) Mean passenger comfort for standing passengers (-)
\( P_{CT} \) Comfort in curve transitions (%)
\( P_{DE} \) Comfort on discrete events (%)
\( q_i \) Physical quantity
\( r \) Correlation coefficient (-)
\( t \) Time (s)
\( T \) Time (s)
\( VDV \) Vibration dose value (m/s\(^{1.75}\))
\( x_i \) Independent variable in comfort function
\( y \) Comfort function (%)
\( \ddot{y} \) Maximum absolute value of lateral acceleration (% of g)
\( \dddot{y}_m \) Absolute mean value of lateral acceleration (% of g)
\( \dddot{y}_p \) Maximum peak to peak value of lateral acceleration (% of g)
\( \dddot{Y} \) Maximum absolute value of lateral jerk (% of g per second)
\( \dot{Y} \) Maximum absolute value of lateral acceleration (m/s\(^2\))
\( \dot{\theta} \) Maximum absolute value of roll velocity (degrees per second)
\( \dot{\theta} \) Maximum absolute value of roll velocity (degrees per second)
\( \tau \) Duration (s)
\( \varphi \) Vibration amplitude in Stevens power law
\( \varphi_0 \) Threshold level in vibration amplitude
\( \psi \) Perceived sensation in Stevens power law (-)
\( (\cdot)_r \) Running r.m.s. of (\cdot)
\( (\cdot)_{rmq} \) Root-mean-quad of (\cdot)
\( (\cdot)_{rms} \) Root-mean-square of (\cdot)
\( (\cdot)_{rsq} \) Root-sum-of-quad of (\cdot), (\cdot), ...
\( (\cdot)_{rss} \) Root-sum-of-squares of (\cdot), (\cdot), ...

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**The Swedish National Road and Transport Research Institute (VTI)** has laboratories and know-how for advanced research commissions in transport and welfare economics, road safety, vehicles and the environment. It also has research capabilities for the construction, operation and maintenance of roads and railways.

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