Ride Comfort Evaluation Using Revised ISO Standard 2631

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Ride comfort is based on the subjective human perception. Numerous experimental investigations in ergonomics have shown that the human perception of vibrations depends on the acceleration, \( K = K(a) \), where \( K \) is a non-dimensional perception measure and \( a \) is the absolute value of the acceleration in horizontal (\( x, y \)) or vertical (\( z \)) direction, respectively. In ergonomics the vibration tests have been performed using deterministic, especially harmonic, excitations. In vehicles, however, the human body is usually exposed to random vibrations.

As response to stochastic excitation, the rms-value of the resulting Gaussian acceleration process \( a(t) \) can be found. For the consideration of the human perception a frequency weighting is required resulting in a weighted acceleration process \( \tilde{a}(t) \). The eigendynamics of humans represented by the frequency weighting are found by comparison with the results for deterministic excitation given in ISO Standard 2631 (1974) using a shape filter. Then, the perception is described immediately with respect to the rms-value or variance of the frequency weighted accelerations. The revised standard, ISO 2631 (2004) uses a slightly modified perception measure. Moreover, the exposure time is no longer included in the standard. Instead, it is assumed that the perception is proportional to the energy involved.

In this paper three models are considered to study the ride comfort: a quarter-car, a half-car and a three-wheeler. All models are subject to random excitation by the unevenness of the road resulting in linear, planar and spatial vibrations. The evaluation of the human perception is based on both, the vertical acceleration of the mechanical models considered and the frequency weighted acceleration obtained by the shape filter. The overall dynamical system is treated by the covariance analysis leading to an algebraic Lyapunov matrix equation, see Popp and Schiehlen (2010). Thus, a numerically efficient investigation of the ride comfort evaluation based on the original and the revised ISO standard 2631 is achieved.

The original ISO standard 2631 is well qualified for the comprehension of the assessment problem and will be used at first. Then, the more recently published and revised standard is discussed. The last results show a remarkable change in the ride comfort evaluation due to the revision of the standard 2631. This means that for future studies of ride comfort the use of the revised standard is highly recommended. In this paper results based on all three different models are discussed in detail, see also Taenzler (2009).

References


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ABSTRACT

Passengers traveling in vehicles running on uneven roads and tracks are subject to whole-body vibrations. Ride comfort is based on the subjective human perception of mechanical vertical vibrations due to randomly uneven guideways. Thus, for the ride comfort evaluation the global system consisting of the guideway, the vehicle suspension, the vehicle body and the passenger has to be modelled and dynamically analyzed. The state equations of the vehicle-guideway system and the human perception dynamics via ISO standards are presented as well as the covariance analysis of the resulting randomly excited global system. The ride comfort analysis is discussed for two vehicle models: a half-car and a full car. The results show a remarkable change in the ride comfort evaluation due to the revised standard ISO 2631.

1 INTRODUCTION

Ride comfort evaluation is one of the major problems in vehicle dynamics for a long time. Jacklin (1936) published an early paper on human reaction to vibration in the SAE Journal. Jacklin (1948) presented some vehicle vibration limits to fit passengers which also were considered in the SAE Ride and Vibration Data Manual (1965). For example, in the frequency range of 6-20 Hz the peak value of the acceleration is recommended to be less than 0.33 m/s². Dieckmann (1957) reported on the effect of mechanical vibrations on humans in automobiles. Mitschke (1972) devoted a whole chapter in his book to random vibrations and ride comfort evaluation. In particular, he used the spectral density analysis for computation and the national standard VDI 2057 (1963) for assessment of the ride comfort.

The general guide ISO 2631 (1974) for human exposure to whole-body vibrations was developed and agreed upon as an international standard. Wong (1978) used this standard for the determination of ride characteristics with two degree-of-freedom models excited by random guideway unevenness and treated by spectral density analysis.

Based on the state-space representation of vehicles Mueller et al. (1980) introduced the covariance analysis for randomly excited vehicle systems. The application to ride comfort evaluation was also summarized by Popp and Schiehlen (1993) in a book on vehicle dynamics. Moreover, numerous papers on ride comfort problems have been published in the journal Vehicle System Dynamics, e.g. by Scheibe and Smith (2009).

More recently the international standard ISO 2631 (1974) has been revised. In this paper the original international standard from 1974 and the revised international standard from 2004 are compared with respect to vehicle applications. For the computation the efficient covariance analysis is used.

2 STATE EQUATIONS OF VEHICLE SYSTEMS

For the modeling it is advisable to decompose the global vehicle system in subsystems, e.g. the components shown in Figure 1 with interfaces for forces and motions clearly defined, see Popp and Schiehlen (2010). Then, the subsystems may be modeled separately and composed by modular assembly to the mathematical model of the overall system. The modular concept allows different modeling approaches for the individual subsystems, it offers the flexibility for the larger number of design variations required today.

The dynamics of all the subsystems can be described consistently by state equations. The state space representation is widely used in control and system theory, and it proves to be most appropriate for vehicle dynamics, too. References of the state representation for dynamical systems are found in Hinrichsen and Pritchard (2005), for control problems in Föllinger (2004) and for vibration engineering in Mueller and Schiehlen (1985).

The equations of motion of a vehicle read according to Popp and Schiehlen (2010)

\[ M(y,\dot{y})\ddot{y}(t) + k(y,\dot{y},t) = q(y,\dot{y},t) \]

or in linearized form appropriate for vertical vehicle vibrations

\[ My(t) + (D + G)y(t) + (K + N)y(t) = h(t) , \]

where the \( f \times 1 \)-vector \( y(t) \) summarizes the generalized coordinates and the \( f \times 1 \)-vector \( h(t) \) includes the generalized coordinates.
The suspension systems of a vehicle are described in the most general case by linear differential equations of the first order, see Popp an Schiehlen (2010)

$$c_f \ddot{f}(t) + c_f f(t) = c_s(t) + c_s \dot{t} + c_u \dot{t} , \quad f(0) = f_0 \, .$$  \hspace{1cm} (7)

Considering that a vehicle has several suspension devices, one gets from (7) state equations of the form

$$\begin{align*}
x_T(t) &= A_T x_T(t) + B_T u_T(t) , \\
x_T(0) &= x_{T0} ,
\end{align*}$$

where $x_T$ is a $n_T \times 1$-state vector of force variables, $A_T$ is the corresponding $n_T \times n_T$-system matrix and $u_T$ is the $r_T \times 1$-input vector of the relative motions of the suspension elements and the control variables if there is an active suspension.

Randomly uneven guideways require to be modelled by stochastic processes. Starting with a white noise excitation it follows a colored noise process via a shape filter as described in Popp and Schiehlen (2010) resulting in

$$\begin{align*}
x_W(t) &= C_W R x_R(t) , \\
x_R(t) &= A_R x_R(t) + B_R w(t) , \\
x_R(0) &= 0 , \\
w(t) &\sim (0,Q_W) ,
\end{align*}$$

where the shape filter is characterized by the time-invariant matrices $A_R$, $B_R$ and $C_W R$. The $n_R \times 1$-vector $x_R$ represents the random process generated by the $r_R \times 1$-vector process of white noise $w(t)$ with an $r_R \times r_R$-intensity matrix $Q_W$.

For the assembly of the complete system the input and output variable of the subsystems have to be connected properly to each other. The generalized forces acting on the vehicle are summarized in the $r_F \times 1$-vector $u_F$. Further, forces are generated by the suspension system, and they serve as state variables featuring the $n_T \times 1$-vector $x_T$ of the suspension devices. Therefore, it yields

$$\begin{align*}
u_F(t) &= C_{FT} x_T(t) ,
\end{align*}$$

with the $r_F \times n_T$-coupling matrix $C_{FT}$. On the other hand, the suspension system is controlled by the $r_F \times 1$-input vector $u_F$. This input depends on the relative motions of the vehicle and the guideway as well as the control variables of the control device. In particular, the controller feeds back the measured motions of the vehicle and the guideway. With the assumption of a simple PD-controller, it yields

$$u_T(t) = C_{TF} x_T(t) + C_{TW} x_W(t) ,$$

where the $r_T \times n_T$-coupling matrix $C_{TF}$ and the $r_T \times n_R$-matrix $C_{TW}$, respectively, occur.
Thus, vertical vibrations on a randomly uneven rigid road result in state equations of the global system based on (4), (8), (9), (10), (11) and (12) as

\[
\begin{bmatrix}
\dot{x}_F \\
\dot{x}_T \\
\dot{x}_R
\end{bmatrix} =
\begin{bmatrix}
A_F & B_F C_{FT} & 0 \\
B_T C_{TF} & A_T & B_T C_{TR} C_{WR} \\
0 & 0 & A_R
\end{bmatrix}
\begin{bmatrix}
x_F \\
x_T \\
x_R
\end{bmatrix} +
\begin{bmatrix}
0 \\
0 \\
B_R
\end{bmatrix} w(t),
\]

\[
\dot{x}(t) = A(t) x(t) + B(t) w(t)
\]  

(13)

where \( x \) means the \( n \times 1 \)-state vector, \( w \) the random \( r \times 1 \)-excitation vector, \( A \) the \( n \times n \)-system matrix and \( B \) the \( n \times r \)-input matrix. In many cases the matrices \( A \) and \( B \) are time-invariant so that the computations are more simple.

A good approximation of the colored noise process via the shape filter (9) and (10) is a white velocity noise excitation characterized by the corresponding excitation vector, \( \dot{w}(t) \), and input matrix. In many cases the matrices \( A \) and \( B \) are time-invariant so that the computations are more simple.

\[
x_W(t) \equiv \dot{w}(t)
\]  

(14)

3.1 Deterministic Excitation

For a rough qualitative assessment of ride comfort during deterministic excitation it yields

\[
K \sim a_{\text{max}},
\]

(17)
i.e. the maximum acceleration value is a first perception measure. As a thumb rule \( a_{\text{max}} \leq 0.5 \text{ m/s}^2 \) results in an acceptable ride comfort. More refined relations between \( K \) and \( a \) are given in international standards that are based on extensive ergonomical investigations. The ISO International Standard ISO 2631 (1974) is better qualified for the understanding of the assessment problem. Then, the revised standard published more recently is discussed. For harmonic excitations the aforementioned standards give a precise relation that suits well for vehicle dynamics. Starting from the harmonic acceleration

\[
a(t) = A \sin \omega t, \quad \omega = 2\pi f,
\]

(18)

with amplitude \( A \) [m/s\(^2\)] and frequency \( f \) [Hz] the root mean square (rms) value \( a_{\text{rms}} \) of the acceleration can be determined,

\[
a_{\text{rms}} = \sqrt{\frac{1}{T} \int_0^T a^2(t) \, dt} = \frac{A}{\sqrt{2}} \text{[m/s}^2\text{]},
\]

(19)

where the unit m/s\(^2\) has to be used. Then, the perception measure \( K \) can be found from Figure 2. The perception measure

\[\begin{array}{c}
\text{FIGURE 2. FREQUENCY RESPONSE OF PERCEPTION MEASURE: ISO 2631 (1974)}\\
\end{array}\]

required for road vehicles is about

\[
2 < K < 10.
\]

(20)
Then, a comfortable ride can be expected for exposure times of 30 minutes up to 8 hours, i.e. humans feel comfortable with $K = 2$ for eight hours and $K = 10$ for half an hour.

### 3.2 Random Excitation

Experimental results for a stochastic excitation of test persons are limited. However, they confirm the interpretation of the subjective human perception as response of a vibration system to stochastic excitation, too, and the possibility to describe the perception using methods of linear system theory. The rms-value of a Gaussian process $a(t)$ is equivalent to its standard deviation $\sigma_a$,

$$a_{\text{rms}} = \sigma_a = \sqrt{\int_0^\infty \Phi_a(\omega) \, d\omega},$$

(21)

where $\Phi_a(\omega)$ is the single sided power spectral density (PSD) of the process under consideration. The standard deviation is a scalar parameter characterizing globally the stochastic process, however, it is frequency independent. Thus, the weighting of frequencies and the calculation of the rms-value have to be interchanged what is allowed for linear stochastic systems. This results in the variance $\sigma^2_{\bar{a}}$ of the weighted process $\bar{a}(t)$,

$$\sigma^2_{\bar{a}} = \int_0^\infty \alpha^2 |F(\omega)|^2 \Phi_a(\omega) \, d\omega.$$  

(22)

Here, $\alpha$ is a dimensional factor, $F(\omega)$ is the frequency response function of a weighting filter and $\Phi_a(\omega)$ is the single sided PSD of the mechanical acceleration process $a(t)$. The still unknown quantities of the frequency weighting in (22) are found by comparison with the results for deterministic excitation. For vertical excitation the dimensional factor reads $\alpha = 20 \text{s}^2/\text{m}$ and the frequency response function is given by

$$|F(\omega)| = \frac{1}{20} K(\alpha_{\text{rms}}, f), \quad f = \omega/2\pi,$$

(23)

where the perception measure $K$ according to Figure 2 is used. The frequency response function (23) can be well approximated by a linear weighting filter or shape filter, respectively.

The shape filter can be described equivalently by differential equations,

$$\bar{a}(t) = \alpha \dot{\mathbf{h}}^T \mathbf{v}(t),$$

$$\mathbf{v}(t) = \mathbf{F}\mathbf{v}(t) + \mathbf{g}a(t).$$

(24)

Here, $\mathbf{v}$ is the $s \times 1$-state vector of the shape filter that is excited by the acceleration $a(t)$. The scalar product of the $s \times 1$-vectors $\mathbf{h}$ and $\mathbf{v}(t)$ results in the frequency weighted scalar acceleration $\bar{a}(t)$. The coefficients of the shape filter have to be determined in such a way that the given frequency response function (23) is approximated sufficiently well. Thus, the computation of the perception measure $K$ results simply in

$$K = \sigma^2_{\bar{a}},$$

(25)

what includes the frequency weighting automatically.

### 3.3 Revised Standard

The revised standards ISO 2631 (2004) uses a slightly changed frequency responses of the perception measure, see Figure 3. Moreover, the exposure time is no longer included in the standard. Instead, it is assumed that the perception is proportional to the energy involved. Then, it yields for the variances of the frequency-weighted acceleration $\sigma^2_{\bar{a}}$ and the energy-equivalent acceleration $\sigma_{\bar{e}}$ during the exposure time $T_e$ the relation

$$\sigma^2_{\bar{a}} T_0 = \sigma^2_{\bar{e}} T_e$$

(26)

where the assessment period $T_0$ is usually 8 hours and $T_e < T_0$ is the exposure time which may be composed by several segments. The perception is described immediately with respect to the root mean square of the frequency weighted accelerations. The shape filter of the human perception can be easily adapted to the revised perception measure as shown by Rill (2007).

![Figure 3. Frequency Response: Original and Revised Standard ISO 2631](image)

### 4 RANDOM VIBRATION ANALYSIS

The investigation of random vibrations may be performed in the frequency domain using the spectral density analysis or
in the time domain applying the covariance analysis. Assuming that \( x(t) = 0 \) is an asymptotically stable equilibrium position, then from \( E\{w(t)\} = 0 \) it follows \( E\{x(t)\} = 0 \). i.e. the mean value of the state vector vanishes in the steady state on a horizontal guideway. Thus, as essential goal it remains to calculate the characteristic variance \( \sigma_f^2 \) of the frequency-weighted acceleration \( \pi(t) \).

The input of the shape filter is the acceleration \( a(t) \) at some seat position, a quantity composed of the states and excitation variables of the entire vehicle system, see Popp and Schiehlen (2010),

\[
a(t) = c^T x(t) + d^T w(t) ,
\]

where \( c \) is a weighting vector of the states and \( d \) a weighting vector of the excitations. Thus, the state equations (13) extended by the shape filter (24) read

\[
\begin{bmatrix}
\dot{v} \\
\dot{\pi}
\end{bmatrix} =
\begin{bmatrix}
A & 0 \\
gc^T & F
\end{bmatrix}
\begin{bmatrix}
v \\
\pi
\end{bmatrix} +
\begin{bmatrix}
B \\
gd^T
\end{bmatrix} w ,
\]

\[
\dot{x} = A x + B w .
\]

The extended system (28) has exactly the same structure as the vehicle system (13), so that system (13) is considered subsequently without loss of generality.

4.1 Spectral Density Analysis

The scalar spectral density of the scalar acceleration \( a(t) \), given by (27) reads considering the state equations (13) as

\[
S_a(\omega) = (c^T F_{x}(\omega)B + d^T) Q_w(B^T F_{x}^T(-\omega) c + d)
\]

(29)

where the \( n \times n \)-frequency response matrix \( F_{x}(\omega) = (i\omega E - A)^{-1} \) has been used. Based on the spectral density (29), the frequency weighting required due to (22) can be performed easily. It remains

\[
S_\pi(\omega) = \alpha^2 |F(\omega)|^2 S_a .
\]

(30)

The variance or square rms-value, respectively, follows from the spectral density by integration over an infinite interval,

\[
\sigma_f^2 = \int_{-\infty}^{\infty} S_\pi(\omega) d\omega .
\]

(31)

The numerical computation of the spectral density (31) is generally not difficult. However, one has to deal with complex matrices, many approximation points and a very large integration interval. This can lead to numerical errors and considerable computation times.

Special care requires the spectral analysis of two-axle vehicles. The excitation at different axles is given by an excitation function with time delay,

\[
B_w(t) = 2 \sum_{i=1}^{2} B_i \zeta_i(t) ,
\]

\[
\zeta_i(t) = \zeta(i - t_1) , \quad 0 = t_1 < t_2 , \quad t_2 = \frac{l_2}{v} ,
\]

(32)

where \( l_2 \) denotes the distance between the front axle (FA), and the rear axle (RA) and \( v = \text{const} \) is the vehicle speed. The spectral density matrix of a two-axle vehicle under random excitation reads

\[
S_i(\omega) = (i\omega E - A)^{-1} \left( B_1 S_{\pi} B_1^T + B_2 S_{\pi} B_2^T + e^{i\omega(t_2 - t_1)} B_2 S_{\pi} B_1^T + e^{i\omega(t_2 - t_1)} B_1 S_{\pi} B_2^T \right) (-i\omega E - A)^{-1} .
\]

(33)

In frequency domain, the time delay \( (t_2 - t_1) \) results in additional terms, weighted by the exponential function.

4.2 Covariance Analysis

In contrast to spectral density analysis, the covariance analysis yields directly the variances that are required for the assessment of the vehicle performance. The covariance matrix of the entire vehicle system follows from an algebraic equation, the so-called Lyapunov matrix equation, and integrations are not required. An essential prerequisite of the covariance analysis is a white noise input process. This can always be achieved by modeling the random vehicle excitation by means of a shape filter. The Lyapunov matrix equation corresponding to the state equations (13) of the entire vehicle system reads

\[
AP + P A^T + B Q_w B^T = 0 ,
\]

(34)

where \( P = E\{ xx^T \} \) denotes the symmetric \( n \times n \)-covariance matrix and \( Q_w \) is the \( r \times r \)-intensity matrix of the white noise input. An extensive derivation of (34) can be found in Mueller and Schiehlen (1985).

The covariance analysis has to be applied to the extended system (28). The covariance matrix of the extended state vector \( \tilde{x} \) reads

\[
P_\tilde{x} = \begin{bmatrix}
P_x & P_{\pi x} \\
P_{\pi x} & P_{\pi \pi}
\end{bmatrix} ,
\]

(35)
where $P_{xx} = P_{yy}^{-1}$ holds. The $s \times 1$-vector process $\mathbf{v}(t)$ yields according to (24) immediately the frequency-weighted acceleration and its variance using an appropriate vector $\mathbf{h}$ as

$$
\sigma^2 = \alpha^2 \mathbf{h}^T P_{yy}^{-1} \mathbf{h}.
$$

This is the relation corresponding to (31), and represents immediately the perception measure.

In the case of a two-axle vehicle the reduced state equations (15) are used. Then, the extended intensity matrix reads

$$
\mathbf{A} P_s + P_s \mathbf{A}^T + \mathbf{Q} = 0,
$$

$$
\mathbf{Q} = q \begin{bmatrix} \mathbf{b}_1 \mathbf{b}_1^T + \mathbf{b}_2 \mathbf{b}_2^T + e^{t_2} \mathbf{b}_1 \mathbf{b}_1^T + \mathbf{b}_2 \mathbf{b}_2^T e^{t_2} \end{bmatrix}.
$$

where $\mathbf{b}_1$ and $\mathbf{b}_2$ are subvectors of the excitation matrix $\mathbf{B}$ and $t_2$ characterizes the time delay between front and rear axle, too.

From $P_s$, the variances $\sigma^2(x_p(t))$ at any position $x_p$ with longitudinal coordinate $x_p = \mathbf{a}^T \mathbf{y}$ can immediately be calculated by

$$
\sigma^2 = \begin{bmatrix} \mathbf{a}^T \mathbf{p_s} \mathbf{0} \end{bmatrix}.
$$

6 Conclusion

The international standard ISO 2631 on human exposure to whole-body vibrations characterizes the frequency response of the perception of passengers travelling in ground vehicles. For modelling and analysis of ride comfort evaluations the global dynamics of the vehicle system including the guideway unevenness and the whole-body vibrations have to be considered. It turns out that the revised standard ISO 2631 from 2004 results in higher rms-values of the frequency-weighted vertical accelerations than the original standard ISO 2631 from 1974. Thus, for vehicle suspension design and optimization the evaluation of ride comfort based on the revised standard is strongly recommended.

REFERENCES


FIGURE 4. PLANAR HALF CAR MODEL WITH 4 DEGREES OF FREEDOM

FIGURE 5. SPATIAL FULL CAR MODEL WITH 7 DEGREES OF FREEDOM
FIGURE 6. NORMALIZED VERTICAL BODY ACCELERATION FREQUENCY WEIGHTED BY THE ORIGINAL ISO STANDARD 2631 (1974)

FIGURE 8. NORMALIZED VERTICAL BODY ACCELERATION FREQUENCY WEIGHTED BY THE ORIGINAL ISO STANDARD 2631 (1974)