Comfortableness Evaluation of an Autovehicle Equipped with Colloidal Suspensions*

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Abstract
Common autovehicle suspensions employ hydro-pneumatic absorbers placed in parallel with compression springs that provide the necessary restoring force. Since the spring can be omitted, compact and lighter design can be achieved by using the recently proposed colloidal suspensions. In this work, frontal and rear colloidal suspensions were designed to replace the classical suspensions, and tests of an autovehicle traveling on normal road with an asphalt step were performed. From the impulse response of tested autovehicle one evaluates its comfortableness, both based on the \(K\) factor method and based on the equivalent acceleration recommended by the ISO 2631 standard. Such testing method allows comfortableness evaluation without using an expensive test rig on which the autovehicle is placed over four actuators and excited to simulate the real road conditions. Results obtained are firstly validated in the case of classical suspensions consisted of oil dampers mounted in parallel with compression springs. Then, colloidal dampers with and without attached compression springs were evaluated. Relationship between the travel speed of the autovehicle and the level of vibration perception, as well as the influence on the sickness, concentration and health was obtained for various values of the tire inflation pressure. Ride-comfort decreases at augmentation of the travel speed and the tire inflation pressure. Although the colloidal suspension was found to provide inferior comfortableness than the classical suspension, results obtained are encouraging, since better performances are to be expected by optimal design of the colloidal spring.

Key words: Damping, Damper, Automobile, Comfortability in Riding

1. Introduction
A review of the literature on the possibility of colloidal damper usage for autovehicle suspension reveals that the French researchers proposed firstly a hybrid hydraulic damper; this absorber employed a colloidal mixture of silica gel and water encapsulated into elastic tanks, and such several damping elements were placed inside the cylinder of a traditional oil damper (1), (2). Reports on the principle, structure and damping characteristics of such hybrid damper were presented, but its usefulness for autovehicle suspension system remained to be clarified (1), (2). In order to achieve structural simplification, compact and light design, as well as ecological benefits, Japanese researchers proposed an alternative solution in which the colloid was directly placed and sealed inside of the working cylinder (3)-(7). Then, in order to extend the life of this oil-free damper to accommodate real applications, such as
vehicle suspensions, silica gel was introduced into a rigid tank that was separated from the main cylinder by a filter permeable to water \(^{(8),(9)}\). Replacement of the filter by an elastic rubber membrane or a diaphragm was also proposed in order to improve the dynamic performances of colloidal dampers. During endurance tests, by using a thermographical method, temperature distribution on the external surface of the colloidal damper was recorded versus the working time, and the temperature inside of the cylinder, as well as the absorber’s generated heat were evaluated \(^{(10),(11)}\). Although the conversion mechanism of mechanical energy was not fully understood, such thermographical tests on colloidal dampers showed quite stable damping characteristics against temperature variation and surprisingly reduced heating. Thus, while the traditional hydro-pneumatic absorbers transform almost integrally the dissipated energy into heat, apparently only a small amount of the dissipated energy was converted into heat by the novel colloidal absorber \(^{(10),(11)}\).

Firstly, based on bench excitation tests, damping and elastic characteristics of colloidal dampers destined to vehicle suspensions were estimated, and then, based on travel tests on real roads, the vehicle’s ride-comfort and the transmissibility of vibration from the rough pavement to the drive-shaft were evaluated \(^{(12)-(15)}\). In comparison with classical automobile suspensions that employ for instance oil dampers mounted in parallel with compression helical springs, the latter providing for the necessary restoring force, in the case of a colloidal damper, since the liquid naturally exudes from the hydrophobized nanoporous silica matrix at decompression, the restoring force is intrinsically achieved, and the compression helical spring can be omitted \(^{(13)-(15)}\).

Recently, vehicle manufactures showed interest in evaluating the static and dynamic performances of colloidal dampers, as well as the comfortableness of vehicles equipped with colloidal suspensions, in order to decide on the feasibility of such novel concept of automobile suspension \(^{(16)}\).

One traditional method to evaluate comfortableness is to place the concerned vehicle over four expensive actuators and to excite it by using a complex controlling system able to simulate the real road conditions \(^{(17),(18)}\). Then, from the measured acceleration data at specified points on the vehicle one estimates its comfortableness by using a quite complex method, as recommended by the ISO 2631 standard \(^{(19)}\). As a relatively inexpensive alternative, in this work, tests of an automobile traveling on normal road with an asphalt step were performed. From the response at impulse excitation of the tested automobile one evaluates its comfortableness, both based on the simplified \(K\) factor method \(^{(20)}\) and based on the more elaborate method recommended by the ISO 2631 standard \(^{(19)}\). Frontal and rear colloidal suspensions, i.e., colloidal dampers with and without compression helical springs mounted in parallel, are designed to replace the classical suspensions, i.e., oil dampers with attached springs. Actual suspensions are used to obtain a database of reference, necessary both to validate the results obtained and to critically evaluate the performances of the proposed colloidal suspensions. One clarifies the influence of the vehicle travel speed and tire inflation pressure on comfortableness, expressed as an equivalent acceleration \(^{(19),(20)}\). In order to improve the ride-comfort of the vehicle when equipped with colloidal suspensions, some measures are proposed based on the understanding of the vehicle’s dynamic behavior.

### 2. Evaluation Methods of the Ride-Comfort

Ride-comfort of the concerned vehicle is expressed as an equivalent acceleration, calculated in this work based both on the on the simplified \(K\) factor method \(^{(20)}\) and based on the more elaborate method recommended by the ISO 2631 standard \(^{(19)}\).

#### 2.1. Evaluation of the Vehicle Ride-Comfort based on the \(K\) Factor Method

In this method an equivalent acceleration \(a_e\) of the seat surface is calculated as \(^{(20)}\):
\[ a_K = 20 \sqrt{\sum_i [(G \cdot a_i)^2]} , \] (1)

where \( G \) is the vibration transfer function of the human body in vertical direction, called also frequency weighting or filter (see Fig. 1) applied to the acceleration data measured in the vertical direction \( a_z \) at the seat surface. Table 1 illustrates the correlation between the equivalent acceleration \( a_K \) and the human perception of vibration. One observes that the equivalent acceleration \( a_K \) decreases as the vehicle’s ride-comfort increases.

### 2.2. Evaluation of the Vehicle Ride-Comfort based on the ISO 2631 Standard

In this method an equivalent (weighted and composite) acceleration \( a_w \) is calculated as
\[ a_w = \sqrt{\sum_i W_x^2 a_x^2 + W_y^2 a_y^2 + W_z^2 a_z^2 + \frac{4a_d^2}{25} + \frac{a_b^2}{25} + \frac{a_s^2}{4} + \frac{a_f^2}{16} + \frac{4a_n^2}{25}} , \] (2)

where \( W_x, W_y, W_z \) are the frequency weighting functions or filters defined by the ISO 2631 standard to be applied to the acceleration data (see Fig. 1). In Eq. (2) \( a \) represents the acceleration along the axes of a 3D Cartesian system of coordinates (subscripts \( x, y \) and \( z \)) measured at the feet surface (subscript \( F \)), at the seat surface (subscript \( S \)) and at the seat-back surface (subscript \( B \)), respectively. Table 2 shows the correlation between the equivalent acceleration \( a_w \) and the human perception of vibration. In the same way as observed for the \( K \) factor method, the equivalent acceleration \( a_w \) decreases as the ride-comfort increases.

<table>
<thead>
<tr>
<th>Equivalent acc., ( a_K ) [m/s²]</th>
<th>Perception level</th>
</tr>
</thead>
<tbody>
<tr>
<td>( a_K &lt; 0.1 )</td>
<td>Imperceptible</td>
</tr>
<tr>
<td>( 0.1 \leq a_K &lt; 0.4 )</td>
<td>Perceptible</td>
</tr>
<tr>
<td>( 0.4 \leq a_K &lt; 1.6 )</td>
<td>Agreeable sensation</td>
</tr>
<tr>
<td>( 1.6 \leq a_K &lt; 6.3 )</td>
<td>Strong perception</td>
</tr>
<tr>
<td>( 6.3 \leq a_K &lt; 15 )</td>
<td>Very strong perception</td>
</tr>
<tr>
<td>( 15 \leq a_K &lt; 40 )</td>
<td>Influence on sickness</td>
</tr>
<tr>
<td>( 40 \leq a_K &lt; 90 )</td>
<td>Influence on concentration</td>
</tr>
<tr>
<td>( a_K \geq 90 )</td>
<td>Influence on health</td>
</tr>
</tbody>
</table>

### 2.3. Comparison of the Filters used by the \( K \) Factor and ISO 2631 Methods

Figure 1 illustrates the variation of the frequency weightings \( G \) (used by the \( K \) factor method for ride-comfort evaluation) and \( W_x, W_y, W_z \) (used by the ISO 2631 method for ride-comfort evaluation) versus the excitation frequency. For vertical vibrations of the vehicle’s body, the frequency band of the filter \( G \) is 0.1-100 Hz, and a maximum gain of 0.708 is displayed in the frequency range of 4-8 Hz. Oppositely, frequency band of the filters \( W_x, W_y, W_z \) is 0.1-400 Hz and the filter selection is connected to location of the accelerometer, i.e., feet, seat and seat-back surfaces, as well as to the specific axis of the 3D Cartesian system of coordinates (see Eq. (2)). Thus, for vertical vibration of the vehicle’s body (see the \( z \) axis in Fig. 4), filter \( W_z \) is used for acceleration measured at the feet and seat surfaces, but filter \( W_d \) is applied to acceleration measured at the seat-back surface.
Maximum gains of the filters $W_c, W_d, W_k$ are 1.024, 1.013, and 1.054 at a frequency of 3.7 Hz, 1.1 Hz, and 6.2 Hz, respectively. Although all the filters shown by Fig. 1 represent vibration transfer functions of the human body along the vertical, horizontal and lateral axes, note that, relative to a maximum value of 0.708 displayed by the filter $G$, the maximum values corresponding to ISO filters ($W_c, W_d, W_k$) slightly exceed the unity. Moreover, one expects more accurate evaluation of the vehicle comfortableness to be obtained by using the ISO 2631 method, which is more elaborate than the $K$ factor method.

Fig. 1  Variation of the frequency weighting $G$, used by the $K$ factor method, and variation of the frequency weightings $W_c$, $W_d$ and $W_k$, used by the ISO 2631 method, versus the excitation frequency

### 3. Characteristics of the Evaluated Suspensions

In this work, three types of suspensions are evaluated as follows: actual suspension consisted of an oil damper mounted in parallel with a compression helical spring, and proposed suspensions consisted of colloidal dampers with and without attached springs (see Table 3). Note that the outer diameter reduces from 135 to 55 mm (59 % reduction) and the mass reduces from 5.9 to 4.6 kg (22 % reduction) in the case of a frontal colloidal suspension without spring. Moreover, 120 g of oil are replaced by an ecological mixture of silica gel (8 g) and water (22 g). Similarly, the outer diameter reduces from 105 to 40 mm (62 % reduction) and the mass reduces from 2.8 to 1.9 kg (32 % reduction) in the case of a rear colloidal suspension without spring. Besides, 80 g of oil are replaced by 20 g of ecological colloid. Thus, the structural simplification (spring omission) is accompanied by a compact and lighter design of such environmental friendly suspension.

<table>
<thead>
<tr>
<th>Type of suspension</th>
<th>Frontal suspension</th>
<th>Rear suspension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actual suspension</td>
<td>Oil damper in parallel with spring</td>
<td>Oil damper in parallel with spring</td>
</tr>
<tr>
<td>Photo, mass and outer diameter</td>
<td>5.9 kg 135 mm</td>
<td>2.8 kg 105 mm</td>
</tr>
<tr>
<td>Proposed suspension</td>
<td>Colloidal damper in parallel with spring</td>
<td>Colloidal damper in parallel with spring</td>
</tr>
<tr>
<td>Photo, mass and outer diameter</td>
<td>6.2 kg 135 mm</td>
<td>3.4 kg 105 mm</td>
</tr>
<tr>
<td>Proposed suspension</td>
<td>Colloidal damper without spring</td>
<td>Colloidal damper without spring</td>
</tr>
<tr>
<td>Photo, mass and outer diameter</td>
<td>4.6 kg 55 mm</td>
<td>1.9 kg 40 mm</td>
</tr>
</tbody>
</table>
Figure 2 illustrates the two degrees of freedom vibration model corresponding to a vehicle suspension consisted of an absorber mounted in parallel with a compression spring and Table 4 shows the associated mass elements, spring constants and damping coefficients. One takes $k_{cs} = 0$ for colloidal suspensions mounted without compression helical springs. Moreover, spring constant of the oil damper can be neglected ($k_{od} = 0$). Table 5 illustrates variation of the tire spring constant versus the inflation pressure.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Frontal suspension</th>
<th>Rear suspension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Body mass, $M_b$ [kg]</td>
<td>240</td>
<td>240</td>
</tr>
<tr>
<td>Wheel mass, $M_w$ [kg]</td>
<td>27</td>
<td>23</td>
</tr>
<tr>
<td>Constant of the compression helical spring, $k_{cs}$ [N/mm]</td>
<td>21.6</td>
<td>18.6</td>
</tr>
<tr>
<td>Spring constant of the colloidal damper, $k_{cd}$ [N/mm]</td>
<td>35</td>
<td>25</td>
</tr>
<tr>
<td>Damping coefficient of the oil damper, $c_{od}$ [Ns/m]</td>
<td>730</td>
<td>675</td>
</tr>
<tr>
<td>Damping coefficient of the colloidal damper, $c_{cd}$ [Ns/m]</td>
<td>1,450</td>
<td>735</td>
</tr>
<tr>
<td>Damping coefficient of the tire, $c_t$ [Ns/m]</td>
<td>81</td>
<td>81</td>
</tr>
</tbody>
</table>

Table 5  Variation of the tire spring constant versus the inflation pressure

<table>
<thead>
<tr>
<th>Inflation pressure [kPa]</th>
<th>150</th>
<th>175</th>
<th>200</th>
<th>225</th>
<th>250</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tire spring constant, $k_t$ [N/mm]</td>
<td>195</td>
<td>205</td>
<td>215</td>
<td>225</td>
<td>235</td>
</tr>
</tbody>
</table>

Fig. 2  Two degrees of freedom vibration model corresponding to the evaluated vehicle suspensions

4. Evaluation of the Ride-Comfort from Impulse Tests of the Vehicle

Vehicle’s ride-comfort was evaluated during travel tests on a normal road with a single semi-sinusoidal asphalt step of 37 mm height and 405 mm width. Travel speed was adjusted in the range of 5-40 km/h with an incremental step of 2.5km/h by using a GPS speedometer. An impulse-like excitation was obtained when the vehicle passed the asphalt step at certain speed, and the ride-comfort was evaluated from the measured acceleration, under the conditions specified by the $K$ factor (Fig. 3) and ISO 2631 (Fig. 4) methods. Also the ride-comfort was evaluated for halted vehicle, in which case the excitation was induced by the engine and air-conditioning system. Tire inflation pressure was adjusted in the range of 150-250 kPa with an incremental step of 25 kPa (see Table 5). Figure 3 illustrates that during the halt and travel tests, according to the $K$ factor method, the acceleration in vertical direction was measured at 5 locations: upper and lower parts of the left frontal suspension, upper and lower parts of the right rear suspension, and at the seat surface. Amplified analog signals were transformed into digital signals by a mobile recorder, and then the equivalent
acceleration \( a_K \) was computed by using an in-house made program under MATLAB. On the other hand, Fig. 4 shows that during halt and travel tests, according to the ISO 2631 method, the acceleration in vertical, horizontal and lateral directions (\( z, x \) and \( y \)) was measured at 3 locations: seat, seat-back and feet surfaces. Recorded signal was automatically processed by using the commercially available DEICY system for ride-comfort evaluation, which directly computed the equivalent acceleration \( a_W \).

Fig. 3  Vibration measurement equipment to evaluate the vehicle’s ride-comfort (\( K \) factor method)

Fig. 4  Vibration measurement equipment to evaluate the vehicle’s ride-comfort (ISO 2631 method)

5. Experimental Results and Discussions

Firstly, variation of the RMS equivalent acceleration versus the travel speed of a vehicle equipped at frontal and rear suspensions with oil dampers mounted in parallel with helical springs, for various values of the inflation pressure (150-250 kPa) is given in Fig. 5, according to the \( K \) factor method, and in Fig. 6, according to the ISO 2631 standard. From Figs. 5 and 6 one observes that the RMS equivalent acceleration increases, i.e., the ride-comfort decreases at augmentation of the travel speed and the tire inflation pressure. This can be explained by the fact that, as the travel speed \( V \) increases, the excitation frequency \( f = V/\lambda \) increases and becomes closer to suspension natural frequency \( f_n \). Here \( \lambda \) is the wavelength of the road roughness. Oppositely, if the travel speed exceeds the critical speed...
\( V_n = \lambda f_n \) corresponding to natural frequency, the equivalent acceleration decreases, as illustrated by Fig. 6. As expected, the vehicle’s ride-comfort worsens at augmentation of the inflation pressure since the tire stiffness increases at higher pressurization values (Table 5).

Based on Tables 1 and 2, in the right side of Figs. 5 and 6 one illustrates the corresponding domains for human perception of vibration. In this way, although the numerical values of the equivalent accelerations \( a_k \) and \( a_w \) given by the \( K \) factor and ISO 2631 methods are different, one finds the relationship between the travel speed and the level of vibration perception, as well as the influence on the sickness, concentration and health, for various values of the inflation pressure. Thus, for \( 0 \leq V \leq 2 \) km/h the vibration is “perceptible” (\( K \) factor) or “not uncomfortable” (ISO 2631), for \( 2 \leq V \leq 4 \) km/h the vibration is perceived as “agreeable” (\( K \) factor) or “a little uncomfortable” (ISO 2631), for \( V \leq 9-13 \) km/h the vibration is felt as “strong” (\( K \) factor), for \( V \leq 13-15 \) km/h the vibration is “fairly uncomfortable” (ISO 2631), for \( V \leq 20-26 \) km/h the vibration is perceived as “very strong” (\( K \) factor), and for \( V \leq 23-30 \) km/h the vibration is “very uncomfortable” (ISO 2631). One observes that the vibration perception levels predicted by the \( K \) factor and ISO 2631 methods are in fairly good agreement, relative to the travel speed of the vehicle. Since for travel speeds larger than 20-26 km/h, the \( K \) factor method predicts “influence on the sickness”, one concludes that the maximum speed of 40 km/h for the performed impulse-like excitation tests is too high, but on the other hand, influence on concentration
and health is not to be expected. Based on such reference results, obtained in the case of actual suspension (Figs. 5 and 6), one concludes that our in-house made program ($K$ factor method) is fairly validated by the commercial software (ISO 2631 method).

In order to compare our findings with those reported by other researchers, Table 6 shows variation of the PSD body acceleration versus frequency for a V8 4.3L autovehicle equipped with classical and colloidal suspensions, traveling at a speed of 80 km/h (tabulated data after graphical results presented by Ref. (16)). Natural frequency of the actual suspension was $f_a = 1.2$ Hz, for a helical spring constant of 36.1 N/mm and a damping coefficient of 1,120 Ns/m (oil damper). Natural frequency of the tested colloidal suspension without helical spring was $f_a = 2.8$ Hz, for a colloidal spring constant of 213.2 N/mm and a damping coefficient of 1,742 Ns/m. Based on data from Table 6, one calculates the equivalent accelerations $a_k$ and $a_w$ according to $K$ factor and ISO 2631 methods as follows. In the case of classical suspension, one arrives to $a_k = 10.1$ m/s$^2$, meaning “very strong perception” after the $K$ factor method, and $a_w = 0.8$ m/s$^2$, meaning “fairly uncomfortable” after ISO 2631 method. On the other hand, for colloidal suspension, one arrives to $a_k = 27.8$ m/s$^2$, meaning “influence on sickness” after the $K$ factor method, and $a_w = 2.1$ m/s$^2$, meaning “very uncomfortable” after the ISO 2631 method. Since the colloidal spring constant was 6 times larger than the constant of the compression helical spring, one concludes that the colloidal suspension is too stiff. For this reason, the colloidal suspension was found to provide inferior comfortableness than the classical suspension, meaning 1 rank lower ride-comfort according to $K$ factor method, and 2 ranks lower ride-comfort according to ISO 2631 method.

In order to get a deeper understanding on the differences of behavior between the classical and colloidal suspensions, for 200 kPa inflation pressure of the tire, one compares the variation of equivalent acceleration $a_w$ versus the travel speed, for the following 4 types of combinations (see Fig. 7):

1) Frontal suspension: oil damper and spring; Rear suspension: oil damper and spring (red line graph in Fig. 7);

2) Frontal suspension: oil damper and spring; Rear suspension: colloidal damper and spring (blue line graph in Fig. 7);

3) Frontal suspension: colloidal damper and spring; Rear suspension: oil damper and spring (brown line graph in Fig. 7); and

4) Frontal suspension: colloidal damper without spring; Rear suspension: oil damper and spring (green line graph in Fig. 7).

Figure 7 illustrates, in agreement with Ref. (16), that the ride-comfort worsens when colloidal suspensions are used. However, one observes a different behavior for colloidal suspensions placed at the front and rear of the tested vehicle, meaning that the comfort worsening is
more prominent for frontal colloidal suspensions. For classical suspensions at all wheels, and for colloidal suspensions used only at the rear wheels, on the graph of equivalent acceleration versus the travel speed, a single peak can be seen in the higher speeds region. Oppositely, for colloidal suspensions at the frontal wheels, two different peaks can be observed, one for lower speeds and the other in the region of higher speeds (see Fig. 7).

In order to explain these results, one should consider the vehicle behavior during frontal and rear impact excitations (see Fig. 8), in correspondence with Fig. 9, which illustrates variation versus the travel speed of the main frequency weighting $W_f$ used by the ISO 2631 method for ride-comfort evaluation. Thus, during the travel test, firstly the frontal suspension receive the impact from the semi-sinusoidal asphalt step, this producing an impulse-like excitation of higher frequency $f_f = V/b$, where $b$ is the width of the step (Fig. 8). Then, the rear suspension receive an impact from the asphalt step, this producing an impulse-like excitation of lower frequency $f_r = V/L$, where $L$ is the distance between the frontal and rear wheels (Fig. 8). Complementarily, Fig. 9 shows variation of the filter $W_f$ versus the travel speed, obtained for the frontal impact excitation, rear impact excitation, and the superposed or global excitation, respectively. A comparison of Figs. 7 and 9 reveals that for combinations 1) and 2), i.e., in the case when classical suspensions are used at the frontal wheels, mainly the rear impact excitation is responsible for the recorded response.

![Frontal impact excitation of higher frequency: $f_f = V/b$](image1)

![Rear impact excitation of lower frequency: $f_r = V/L$](image2)
However, for combinations 3) and 4), i.e., in the case when colloidal suspensions are used at the frontal wheels, a global or superposed frontal-rear impact excitation is responsible for the recorded response. In other words, while the vehicle body still vibrates due to the frontal impact, it receives a second rear impact, this producing a superposition of effects.

Next, one considers the peaks observed in Fig. 7 at 8 km/h for combination 4) and 10.5 km/h for combination 3). Thus, Fig. 10 illustrates variation of acceleration versus frequency for combinations 3) and 4), calculated with and without the filter \( W_k \), used by the ISO 2631 method. Variation of the filter is also shown for easy understanding. Critical frequencies are obtained at 1.77 Hz for combination 4) and 2.36 Hz for combination 3).

On the other hand, based on the modal analysis and data from Table 4 one calculates the critical frequencies as follows. Thus, for colloidal damper without spring (combination 4)) the critical frequency is given by:

\[
f_{CD} = \frac{1}{2\pi} \sqrt{\frac{(k_{CD} + k_1)M_a + k_{CD}M_s}{2M_sM_a}} \left(1 - \frac{4k_{CD}k_1M_aM_s}{[(k_{CD} + k_1)M_a + k_{CD}M_s]^2}\right) = 1.78 \text{ Hz}, \tag{3}
\]

and for the colloidal damper with spring (combination 3)) the critical frequency becomes:
These values are in relatively good agreement with those shown in Fig. 10 for responses without frequency weighting.

6. Upon the Optimal Spring Constant of a Colloidal Suspension

With the purpose to improve the vehicle’s comfortableness when equipped with colloidal suspensions, one searched for the optimal stiffness ratio, i.e., for the optimal ratio of the colloidal spring constant $k_{CD}$ to the compression spring constant $k_{CS}$ (21), (22). In order to do this, by neglecting the lower part of Fig. 2, i.e., by neglecting the resonant peak connected to the tire, the two degrees of freedom vibration model shown by Fig. 2 was further simplified to a quarter-vehicle with one degree of freedom (21), (22). In this way, one considers only the upper part of Fig. 2, where the elastic element $k_{CS}$ corresponding to the compression helical spring is mounted in parallel with a colloidal damper, modeled as a Maxwell unit (a dashpot $c_{CD}$ serially connected with the elastic element $k_{CD}$). Analysis of the vibration transmissibility function for such a system reveals that, as the stiffness ratio $k_{CD}/k_{CS}$ increases, the resonant peak decreases, but the transmissibility in the higher frequency domain increases. In order to maximize the vehicle’s ride-comfort, i.e., in order to minimize the transmissibility of vibration from the rough road to the vehicle’s body in the whole frequency domain, through numerical integration one firstly finds the area below the graph of transmissibility versus the excitation frequency, for various values of the stiffness ratio $k_{CD}/k_{CS}$ (see Fig. 11). On the convex (valley-like) graph of the previously defined integral parameter versus the stiffness ratio, one finds the optimal stiffness ratio, as corresponding to the deepest point of the valley (21), (22). The optimal values of the stiffness ratio were found as follows: $(k_{CD}/k_{CS})_{opt} = 1$ in the case without filter; $(k_{CD}/k_{CS})_{opt} = 0.5$ in the case when filter $G$ (K factor method) was used, and $(k_{CD}/k_{CS})_{opt} = 0.6$ in the case when filter $W_k$ (ISO 2631 method) was used to account for the effects of vibration on the human body (21), (22). Colloidal suspensions redesigned based on such optimal values of the stiffness ratio are expected to provide better vehicle comfortableness.

Fig. 11  Optimization of the stiffness ratio to minimize the transmissibility of vibration from the rough road to the vehicle’s body in the whole frequency domain, i.e., to maximize the vehicle’s ride comfort (Cases without filter, with filter $G$ and with filter $W_k$)
7. Conclusions

In this work, a relatively inexpensive method was proposed to evaluate the ride-comfort of a vehicle equipped with colloidal dampers at its frontal and rear suspensions. Concretely, impulse-like excitation was obtained when the vehicle traveled on a normal road with a single asphalt step. Such experimental method allows comfortableness evaluation without using an expensive test rig on which the vehicle is placed over four actuators and excited to simulate the real road conditions. From the impulse response of tested vehicle one evaluated its comfortableness, by calculating the equivalent accelerations recommended both by the $K$ factor and ISO 2631 methods.

Results obtained were validated in the case of actual suspensions consisted of oil dampers mounted in parallel with compression helical springs. Relationship between the travel speed of the vehicle and the level of vibration perception, as well as the influence on the sickness, concentration and health was obtained for various values of the tire inflation pressure. Levels of vibration perception predicted for various travel speeds of the vehicle both by the $K$ factor and ISO 2631 methods were in good agreement. Ride-comfort decreases at augmentation of the travel speed and the tire inflation pressure.

Although the colloidal suspension was found to provide inferior comfortableness than the classical one (1 rank lower ride-comfort according to $K$ factor method, and 2 ranks lower ride-comfort according to ISO 2631 method), results obtained so far are encouraging, since better performances are to be expected by softening the colloidal suspension. Results obtained were explained by taking into account the vehicle behavior during frontal, rear and superposed impact excitations, in correlation with the variation versus the travel speed of the frequency weighting, used by the ISO 2631 method for ride-comfort evaluation.

As a future development of this work, one should experimentally investigate the ride-comfort of a vehicle equipped with colloidal suspensions redesigned to achieve the optimal values of the stiffness ratio. Additionally, in order to fully validate the proposed evaluation technique, one should compare the ride-comfort results obtained both based on the $K$ factor method and based on the ISO 2631 standard, during inside tests of a vehicle excited on all wheels by using actuators able to simulate the real road conditions, and also during outside tests of a vehicle travelling on various types of rough roads.

Acknowledgements

This research was supported by the Japanese Ministry of Education, Grant-in-aid for scientific fundamental research, Project C-22560151. Authors would like to acknowledge the technical support of Messrs. N. Kitagawa @ DEICY Instruments Ltd. and H. Takahashi @ MIZUKAMI Measurement Ltd. to accurately evaluate the autovehicle comfortableness.

References


