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Experimental investigation on the dissipative and elastic characteristics of a yaw colloidal damper destined to carbody suspension of a bullet train

B Suciu\(^1\) and T Tomioka\(^2\)

\(^1\)Department of Intelligent Mechanical Engineering, Fukuoka Institute of Technology, 3-30-1 Wajiro-Higashi, Higashi-ku, Fukuoka-shi, Fukuoka 811-0295 Japan
\(^2\)Planning Division, Railway Technical Research Institute, 2-8-38 Hikari-cho, Kokubunji-shi, Tokyo 185-8540 Japan

E-mail: suciu@fit.ac.jp

Abstract. Yaw damper represents a major source of excitation for flexural vibration of the railway carbody. In order to reduce transmissibility of such undesired excitation, yaw damper should allow for large force transmission at low working frequencies, but should behave as vibration isolator at high working frequencies. Unfortunately, the yaw oil damper (OD), which is nowadays in service, has poor intrinsic elastic capabilities and provides damping forces varying as a power function versus the piston speed. Since colloidal damper (CD) has intrinsic elastic capabilities and larger damping forces at lower excitation frequencies, it occurs as an attractive alternative solution to traditional yaw dampers. In this work, a yaw CD destined to carbody suspension of a bullet train was designed and manufactured; then, its dynamic characteristics, produced by both the frictional and colloidal effects, were evaluated from the experimental results, obtained during horizontal vibration tests, performed on a ball-screw shaker. Compared to the corresponding classical yaw OD, the trial yaw CD allowed for: weight reduction of 31.6 \%; large damping force, dissipated energy and spring constant at long piston stroke under low excitation frequency; low damping force, dissipated energy and spring constant at short piston stroke under high excitation frequency. Elastic properties were justified by introducing a model for the spring constant that included the effect of pore size distribution.

1. Introduction

Oil damper (OD) is nowadays used in construction of railroad vehicles in general, and in construction of the high speed trains in particular. E.g., in the case of Japanese bullet train, called “SHINKANSEN”, vertically working ODs (4 axle dampers per bogie) and horizontally working ODs, i.e., 2 left-right lateral dampers per bogie, 2 yaw dampers per bogie, as well as a set of 2 yaw dampers between successive cars, are used [1]. Each axle damper is connected in parallel with an axle spring, and has the main purpose to attenuate vibration produced by the vertical irregularities of the railway. Left-right lateral dampers are centrally connected between bogie and carbody, but perpendicularly to railways direction. They attenuate relative vibration occurring between the bogie and carbody. Yaw dampers are laterally connected between bogie and carbody, but parallelly to the railways direction. Their main purpose is to attenuate vibration produced by the left-right lateral irregularities of the railways, as well as...
as the lateral winds. Thus, they are used to suppress the bogie’s hunting motion, and to improve the travelling stability of the vehicle. *Yaw dampers between successive cars* are connected parallelly to the railways direction. They attenuate relative vibration occurring between cars due to the same excitation mechanisms as in the case of yaw dampers, and thus, they improve the ride comfort of the vehicle [1].

Mass unbalance of wheels is well-known as one of the main sources of excitation for the vertical bending vibration of large train carriages, leading to worsened ride comfort. Thus, during travel, bogie is excited by wheels at the corresponding rotational frequency, even if wheels have quite small mass unbalances. Amplification of bending vibration of the train carriage can be achieved if this excitation is transmitted from bogie to carbody, at a frequency close to the natural frequency of the flexural mode.

On the other hand, for high speed trains, wheelset rotational frequency, e.g., about 28 Hz at a speed of 270 km/h, is considerably higher than the natural frequency of carbody bending mode. However, bogie pitching motion induced by vertical track irregularities with a wavelength of 5 m, equalling the interspace between concrete slabs placed underneath rails, often significantly excites the carbody bending mode. Thus, a bullet train, running at 270 km/h, suffers a forced excitation of about 15 Hz, produced by vertical track irregularities. This excitation manifests firstly as a bogie pitching motion, which is then transmitted to carbody.

In order to improve the ride comfort of large train carriages in general, and of the bullet trains in particular, transmissibility of vibration from bogie to carbody should be reduced. Since *yaw dampers* and traction links are the main paths of vibration transmission from bogie to carbody, they should be able to allow for large force transmission at lower frequencies, but to behave as vibration isolators in the domain of relatively higher frequencies.

Concerning the traction links, a vibration isolation solution was proposed and its efficacy was validated [2]. Concretely, when rubber buffer couplings of variable stiffness were applied to the traction links, improvement of the ride comfort was experimentally confirmed [2]. Thus, if the elastic characteristics of the rubber buffer couplings were selected to achieve a stiff link, in the range of low frequencies and long amplitudes of vibration, but a soft link, in the range of high frequencies and short amplitudes of vibration, without diminishing the transmissibility of traction force and travelling stability, a reduction of the carbody bending vibration was experimentally observed [2].

However, until now, an effective solution to suppress vibration transmissibility via yaw dampers has not been found, especially, for relatively higher frequency vibration, i.e., higher than about 10 Hz [2-4]. One reason is that the commonly used yaw OD has quite poor intrinsic elastic capabilities, by itself [5-8]. Thus, elasticity of the whole yaw damper system is mainly produced by the rubber buffer couplings, which are placed at its extremities [2], [9].

In this work, as possible alternative solution, a yaw colloidal damper (CD) is proposed, since it has intrinsic elastic capabilities [10-14] and attractive features [10], [12], [15-16]. Thus, CD provides large damping forces, dissipated energies and spring constants in the range of low frequencies and long amplitudes of vibration, but small damping forces, dissipated energies and spring constants in the range of high frequencies and short amplitudes of vibration. Besides, CD is an oil-free environmental friendly absorber, since dissipation is obtained at penetration/exudation of water in/from the nanopores of a silica-gel matrix (artificial sand with controlled nanoporous architecture).

Although cases of study for CD used as autovehicle suspension can be found in literature [10-11], [13-14], [16-19], CD used for train carriage suspension, is a subject left opened to investigation. For this reason, in this work, a yaw CD destined to the carbody suspension of a bullet train, was designed and manufactured. From experimental results obtained during horizontal vibration tests of the trial yaw CD, performed on a ball-screw shaker, variation against frequency of its dynamic characteristics, induced by both the frictional and colloidal effects, are evaluated.

2. Structure of the proposed yaw colloidal damper (CD)

Figure 1 shows the schematic view and main dimensions of a classical yaw OD mounted between the bogie and carbody of a bullet train, by using rubber buffer couplings, bolts and nuts. Maximal diameter of the yaw OD ($D = 143 \text{ mm}$), external diameter of the cylinder ($d = 127 \text{ mm}$), and the mean
distance between centres of the left-right rubber buffer couplings ($L = 660$ mm) are used as reference specifications for the design and manufacturing of the trial yaw CD.

Figure 2 illustrates the assembly drawing of the trial yaw CD, represented without rubber buffer couplings. Table 1 shows the main characteristics of the trial yaw CD in comparison with those of the classical yaw OD. Figures 3 and 4 present photos of the manufactured yaw CD without and with rubber buffer couplings. Table 1 indicates that the most important characteristics, i.e., mean distance between centres of the left-right rubber buffer couplings ($L = 660$ mm), maximal piston speed (200 mm/s), as well as the maximal/minimal loading force in tension/compression ($\pm 19.6$ kN) are the same for the classical yaw OD and the trial yaw CD. Due to design circumstances, the external diameter of the cylinder and the maximal piston stroke are slightly different, but this does not significantly affect the overall functioning of the yaw damper. Compared to the classical yaw OD (mass of 30.4 kg), a weight reduction of 31.6 % was achieved for the trial yaw CD (mass of 20.8 kg).

Trial yaw CD is consisted of the following parts: cylinder, piston, support plate, left- and right-side connecting rings which are necessary to fit the rubber buffer couplings, left- and right-side seal covers, guide-way unit, filters, radial and axial plugs, high-pressure seals, O-rings, etc. (see figure 2). Central part of the piston has a diameter of 28.3 mm, and the piston-rods have a diameter of 20 mm.

A quantity of 7.2 g of water-repellent nanoporous silica-gel is supplied in each of the 8 rooms cut in the left-side, and 8 rooms cut in the right-side of the cylinder. Hence, a total amount of 115.2 g of silica-gel was used; its physical properties are described elsewhere [20].

By using a manually operated pump (not shown in figures 2-4), a total amount of 350 g of water was injected in the left- and right-side centrally placed chambers of the cylinder, via the supply orifices and valves indicated in figures 2 and 3, respectively. Further, water is able to pass through the filters located in the radially disposed holes, in the vicinity of the left- and right-side seal covers, and then, to penetrate inside all the 16 rooms accommodating silica-gel grains.

![Figure 1. Schematic view and main dimensions of the yaw OD mounted between bogie and carbody.](image1)

![Figure 2. Schematic view of the trial yaw CD represented without rubber buffer couplings.](image2)
Table 1. Comparison between the classical yaw OD and the trial yaw CD.

<table>
<thead>
<tr>
<th>Parameter of comparison</th>
<th>Yaw OD (classical)</th>
<th>Yaw CD (trial)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximal distance between centres of the rubber buffer couplings, ( L_{\text{max}} )</td>
<td>775 mm</td>
<td>774 mm</td>
</tr>
<tr>
<td>Minimal distance between centres of the rubber buffer couplings, ( L_{\text{min}} )</td>
<td>545 mm</td>
<td>546 mm</td>
</tr>
<tr>
<td>Mean distance between centres of the rubber buffer couplings, ( L )</td>
<td>660 mm</td>
<td>660 mm</td>
</tr>
<tr>
<td>Maximal stroke of the piston ( \pm 115 ) mm</td>
<td>127 mm</td>
<td>125 mm</td>
</tr>
<tr>
<td>Maximal/minimal loading force (tension/compression) ( \pm 19.6 ) kN</td>
<td>200 mm/s</td>
<td>200 mm/s</td>
</tr>
<tr>
<td>Mass of the damper (excluding mass of the rubber buffer couplings)</td>
<td>30.4 kg</td>
<td>20.8 kg</td>
</tr>
</tbody>
</table>

Thus, when compressed through the action of the pump or the piston, water passes through the filter orifices with negligible flow resistance, and penetrates inside the rooms accommodating silica-gel micro-particles, producing a colloidal solution. On the other hand, filters with diameter of orifices of 1 µm, are able to prevent permeation of silica-gel grains, which sustained fatigue fracture, inside the central chamber of the cylinder [21]. By using such liquid-permeable encapsulation of the silica-gel, a certain desired lifetime of the yaw CD can be achieved.

Colloidal tanks placed in the right-, and respectively, left-side of figure 2, are compressed when the yaw CD is loaded by tensile, and respectively, compressive forces. In this way, the elastic and dissipative characteristics through colloidal effect are achieved. High-pressure seals of the yaw CD were selected to allow for usual working pressures in the range of 20-40 MPa, and a maximal allowable pressure of 70 MPa.

3. Vibration test rig of the yaw colloidal damper (CD)

Vibration tests of the trial yaw CD were performed on a made-in house ball-screw shaker (figure 5). Details concerning the shaker used and its performances are given elsewhere [22]. A load cell, placed at the piston end of the yaw CD, is used to measure the excitation force \( F \). A laser displacement sensor is employed to determine the piston displacement \( S \) and the amplitude of vibration \( S_{\text{max}} \). Since the maximal piston stroke of the trial yaw CD (±114 mm) is larger than the measurement range (100 ± 40 mm) of the laser displacement sensor, used in our previous work [22], in this paper, a new laser displacement sensor, with very long measurement range of 300 ± 140 mm, was employed.

In order to evaluate the dynamic features of the yaw CD itself, left- and right-side rubber buffer couplings, as well as their corresponding connecting rings, were omitted. Instead, at the right end of the piston rod a screw-coupling was used. On the other hand, absorber’s cylinder was directly fixed on the shaker, by using its support disk, and a block-coupling (see figure 5). In this way, the left piston rod is able to travel through an inner hole, drilled both into the support disk and the block-coupling.

Dissipated energy of the yaw CD can be calculated as the area of its hysteresis loop (figures 6-7):

\[
E = \int F(S) dS ,
\]

and by modelling the yaw damper as a dashpot, its damping coefficient \( C \) can be computed as the slope of the graph showing the variation of the damping force \( F \) versus the piston speed \( V \):

\[
C = dF / dV .
\]
Figure 5. Photo of the test rig used during vibration experiments on the proposed yaw CD.

For classical yaw OD, graph of the damping force versus the piston speed shows almost linear lines, of constant but different slopes, in the left and right sides of a cracking point. Thus, relative to the cracking point, in the range of low piston speeds, a constant but relatively higher damping coefficient, and in the range of high piston speeds, a constant but relatively lower damper coefficient is gained [5].

However, the same value of the mean piston speed ($V = 2S_{\text{max}}f$) can be obtained either in the case of a lower frequency $f$ and longer amplitude of vibration $S_{\text{max}}$, or in the case of a higher frequency and shorter amplitude of vibration. Hence, the influence of frequency, on the damping coefficient, remains unsettled [9]. To clarify the effect of frequency and amplitude of vibration on the damping coefficient, in this work, both the damping coefficient $C$, as given by equation (2), and the equivalent damping coefficient $C_e$, as given by equation 3, are calculated and the results are comparatively discussed.

$$C_e = 2E / (\pi^2 f S_{\text{max}}^2)$$

With this aim, two types of vibration tests are conducted: experiments at fixed frequency but variable amplitude of vibration, and experiments at fixed amplitude but variable frequency of vibration.

Thus, considering the trapezoidal excitation cycle of the shaker (trapezoidal variation of the ball-screw rotation speed versus the excitation time), an experiment conducted at fixed frequency but variable amplitude of vibration can be defined as follows [23]. Fixing, for instance, the excitation period to $T = 2s$, a constant frequency, of $f = 0.5$ Hz, is achieved. Then, by setting the speed of rotation of the ball-screw to $n = \pm 2.5, 5, 10, 15, 20, 25, 30, 35, 40, 45, 50, 55, 60, 70$ rps, it becomes possible to change, together with the amplitude of vibration $S_{\text{max}}$, the mean speed of the piston $V$. For $f = 0.5$ Hz, if the rotation speed is set to $n = \pm 70$ rps, the amplitude of vibration becomes $S_{\text{max}} = \pm 113$ mm, i.e., it approaches the design allowable amplitude, of ±114 mm (see table 1).

During experiments at fixed amplitude but variable frequency of vibration, the amplitude is set to a certain desired value, and the speed of rotation is adjusted in the range of 2.5-100 rps [23]. For instance, if amplitude of vibration is set to $\pm 4, 8,$ and $16$ mm, it becomes possible to change the frequency of vibration in the range of $f = 0.5$-6.8 Hz, 0.5-4.2 Hz, and 0.5-2.4 Hz, respectively.

Since the rotation speed of the ball-screw is limited to 100 rps, as the amplitude of vibration increases, the possible range of variation for the frequency becomes narrower. For this reason, standardization of the experimental conditions, for vibration tests conducted on yaw dampers, was recently proposed [9]. E.g., if the vibration amplitude is fixed to $\pm 5$ mm, it is advisable to set the frequency of vibration to the following values: 0.3, 0.6, 1.6, 3.2, and 5.1 Hz [9]. From this standpoint, the testing technique of the yaw CD, adopted in this work, is similar to the standardized testing technique, previously suggested [9].
4. Experimental results: Damping characteristics of the yaw colloidal damper (CD)

Firstly, tests are performed on the trial yaw CD without supplying water inside the cylinder. In this way, damping characteristics, determined only by the frictional effects, are estimated. Then, water is supplied inside the cylinder, and in such conditions, global damping characteristics of the yaw CD determined by the cumulative colloidal and frictional effects, were estimated.

Figure 6 presents the frictional hysteresis loop, obtained for tests at fixed frequency of 0.5 Hz, and various amplitudes of vibration (± 4, 8, 16, 24, 32, 40, 48, 56, 64, 72, 80, 88, 96, and 104 mm). A Coulomb friction-like rectangular hysteresis loop, having the compression force slightly larger than the extension force, was obtained. This can be explained by the additional viscous friction, produced by the guide-way used to support the piston, when the damper is compressed (see figure 2). Since the extreme values of friction force are weakly depending on the piston speed, equation (2) predicts an almost zero damping coefficient (triangular symbols on figure 9). However, since dissipation is gained, equation (3) predicts a nonzero equivalent damping coefficient (see rhombic symbols on figure 9).

Figure 7 shows the global hysteresis loop, involving both colloidal and frictional effects, obtained for tests at fixed frequency of 0.5 Hz, and various amplitudes of vibration (± 1.8, 3.6, 7.3, 14.9, 18.1, 21.8, 25.4, 29.5, 33.0, 36.7, 40.5, 44.5, 50.2, and 55.6 mm). A Coulomb friction-like hysteresis loop is obtained in the case of short excitation amplitudes, e.g., ± 1.8 mm. For long excitation amplitudes, an asymmetric hysteresis loop, resembling an inclined eight number, is produced by the colloidal effects.

Figure 8 presents the variation of the dissipated energy, and figure 9 shows the variation of the damping coefficients versus the excitation amplitude (mean travelling speed), for $f = 0.5$ Hz.

An almost linear graph $E \approx a S_{\text{max}}$ (where $a$ is a constant) is obtained for the frictional dissipated energy versus the excitation amplitude, at fixed excitation frequency. This can be explained by the increase of the width of the hysteresis loop, at augmentation of the excitation amplitude.

A nonlinear graph, resembling that of the function $S_i^m$ (for an index $i > 1$), is obtained for the global dissipated energy, at fixed excitation frequency. This can be explained by the fact that, at augmentation of the excitation amplitude, both the width and height of the hysteresis loop increases.

For short amplitudes, the colloidal is not sufficiently pressurized, and the dissipation is produced mainly by the frictional effect. Hence, the index $i$ displays a small value close to 1. At augmentation of the excitation amplitude, dissipation produced by colloidal effect largely exceeds dissipation produced by frictional effect. Hence, the index $i$ gradually increases, and finally exceeds the value of 2.

Figure 9 shows that, the equivalent frictional damping coefficient decreases versus amplitude, as a $b S_{\text{max}}^{-1}$ type function, where $b$ is a constant. On the other hand, the equivalent global damping coefficient varies against amplitude, as a $S_{\text{max}}^{i-2}$ type function.
variable frequency, and excitation amplitudes of speed of the piston. Triangular, square, and circular symbols illustrate the results obtained for tests at present results obtained for tests at variable excitation amplitude, and

1), but at higher frequencies, the hysteresis loop resembled a parallelogram cycle, typical for the reciprocating sliding fretting (0 < slip ratio < 1).

Figure 10 shows the variation of the equivalent damping coefficient versus the mean travelling speed of the piston. Triangular, square, and circular symbols illustrate the results obtained for tests at variable frequency, and excitation amplitudes of ± 16, 8, and 4 mm, respectively. Rhombic symbols present results obtained for tests at variable excitation amplitude, and \( f = 0.5 \text{ Hz} \).

Against the mean travelling speed of the piston, results obtained in tests at fixed frequency but variable amplitude of excitation are superposed on the results obtained in tests at fixed amplitude but variable frequency of excitation.

However, by observing the same results as expressed against the excitation frequency (figure 11), one obtains distinctive graphs corresponding to various amplitudes. Moreover, at augmentation of the excitation amplitude, reduction of the equivalent damping coefficient can be perceived.

In conclusion, in order to satisfactorily clarify the damping features of a yaw damper, not only the dependence against the mean piston speed (figure 10), but also the variation versus the excitation frequency (figure 11), should be specified. The reason for this is that, the same value of the mean travelling speed of the piston can be obtained for low excitation frequency but long amplitude, for high excitation frequency but short amplitude, as well as for middle level of excitation frequency and amplitude. Thus, information on the frequency behaviour is vital for proper selection of a yaw damper.

Figure 10. Damping coefficient versus speed. Figure 11. Damping coefficient versus frequency.
5. Experimental results: Elastic characteristics of the yaw colloidal damper (CD)

Since during the unloading phase of a CD, water naturally exudes from the nanopores of the liquid-repellent silica-gel matrix, the restoration force is achieved. Thus, CD behaves as a machine element with dual function, of absorber (dissipative element) and spring (elastic element). From the standpoint of the primarily function of a yaw damper, i.e., prevention of the yawing movement of the railroad carbody, the elastic behaviour cannot be considered as a desirable characteristic. However, from the point of view of the secondary function of a yaw damper, i.e., prevention of the vibration transmission from bogie to carbody, the elastic behaviour can be considered as an advantageous characteristic.

In order to evaluate the elastic features of the trial yaw CD, in one possible simplified definition, the equivalent spring constant can be calculated as the slope of the central segment associated with the hysteresis loop (see figure 12). Thus, although the graph of the mean damping force versus the piston displacement has a variable slope, the equivalent spring constant can be taken, in a first approximation, as the slope of the central segment of the hysteresis loop.

In the range of short piston displacements, since dissipation is mainly produced through frictional effect, slope of the central blue segment of the hysteresis loop is comparatively low. Accordingly, a relatively soft spring elastic behaviour, with a spring constant of 96.2 N/mm for compression, and 92.6 N/mm for extension, is achieved. Under the limitation of the maximal piston speed, imposed by the seal manufacturers, in this working domain, higher excitation frequencies can be achieved.

In conclusion, although the equivalent spring constants during compression and extension are slightly different, the mean spring constant measured for long piston displacements, is about 6 times larger than the average spring constant determined for short piston displacements.

\[
\text{Piston force, } F \begin{cases} \text{[kN]} \\ \text{Long stroke} \end{cases} \begin{cases} \text{Short stroke} \end{cases} \begin{cases} \text{Long stroke} \end{cases} \begin{cases} \text{Low frequency} \end{cases} \begin{cases} \text{High frequency} \end{cases} \begin{cases} \text{Low frequency} \end{cases}
\]

\[
\text{Mainly frictional effect: Lower spring constant, and dissipated energy}
\]

\[
\text{Mainly colloidal effect: Higher spring constant, and dissipated energy}
\]

\[
\text{Cracking point}
\]

**Figure 12.** Hysteresis loop of the yaw CD and definition of the equivalent spring constant.

In order to explain the experimental results shown by figure 12, a model for the spring constant of the yaw CD is proposed. In the absence of silica-gel particles, the yaw CD behaves as a classical liquid spring of stiffness \( k = A_p E_t / l_t \) [24] (where \( A_p = \pi (28.5^2 - 20^2) / 4 = 314.9 \text{ mm}^2 \) is the cross-sectional...
area of the piston, \( l \) is the length of the liquid column, and \( E_l \) is the liquid bulk modulus of elasticity. However, by mixing silica-gel particles in water, a beneficial reduction of the bulk modulus of elasticity of the resulting colloidal solution is obtained. Thus, for a colloidal mixture of silica-gel particles in water, the spring constant \( k_{CD} \) of the resulting yaw CD can be written as [24]:

\[
k_{CD} = \frac{A_p E_w}{S_{\text{max}}} \frac{E}{\bar{S} + E(\beta - \bar{S})}
\]

where \( E_w = 2.15 \, \text{GPa} \) is the water bulk modulus of elasticity, \( S_{\text{max}} \) is the amplitude of vibration, \( \bar{S} = S / S_{\text{max}} \in [0;1] \) is the dimensionless piston displacement, \( \beta \) is the coefficient of water surplus, and \( E = E_s / E_w \) is the dimensionless elastic modulus. Here \( E_s \) is the silica-gel equivalent modulus of elasticity, which can be calculated as [24]:

\[
E_s = \frac{2\sigma \cos \alpha}{r} \in \{E_{s,\text{min}} = -\frac{2\sigma \cos \alpha}{r_{\text{max}}}; E_{s,\text{max}} = -\frac{2\sigma \cos \alpha}{r_{\text{min}}}\}
\]

where \( \sigma = 0.0728 \, \text{N/m} \) is the surface tension of water, \( r \in [r_{\text{min}}; r_{\text{max}}] \) is the radius of the nanopores, and \( \alpha = 128 \, \text{deg} \) is the contact angle at the interface of water, and water-repellent silica-gel wall.

Since the equivalent modulus of elasticity of silica-gel equals the pressure (capillary pressure) created inside the cylinder, it occurs as variable parameter during the compression-decompression cycle of the yaw CD.

Minimal pore radius can be calculated as: \( r_{\text{min}} = -\frac{2\sigma \cos \alpha}{p_{\text{max}}} \approx 1.4 \, \text{nm} \), where the corresponding maximal pressure is estimated from figure 12 as: \( p_{\text{max}} = F_{\text{max}} / A_p = 62 \, \text{MPa} \). This result is in good agreement with the pore radius distribution shown in figure 13. Since the specific pore volume has a maximal value of 0.3 cc/g (see figure 13), and the amount of silica-gel for a half-damper is of 57.6 g, the maximal quantity of water penetrating the pores can be estimated as: \( A_p S_{\text{max}} \approx 17.3 \, \text{cc} \). This result is in good agreement with figure 12, since the corresponding maximal stroke is: \( S_{\text{max}} \approx 55 \, \text{mm} \). On the other hand, the coefficient of water surplus can be found as: \( \beta = 17.3 / 175 \approx 10 \), where 175 cc is the amount of water for a half-damper.

Cracking point, corresponding to the spring constant change (figure 12), seems to be related to the sudden variation of the pore radius distribution, from \( r_{\text{max}} \approx 22 \, \text{nm} \) to \( r_{\text{min}} \approx 1.4 \, \text{nm} \) (figure 13). Thus, equation (4) can be applied to calculate the spring constants in the left-side (low spring constant) and in the right-side (high spring constant) of the cracking point. Concretely, by taking into account that the dimensionless piston displacement corresponding to the cracking point is about \( \bar{S} \approx 0.48 \), the ratio of the high spring constant \( k_{CD,h} \) to the low spring constant \( k_{CD,l} \) can be calculated as:

\[
\frac{k_{CD,h}}{k_{CD,l}} = \frac{k_{CD}(\bar{S} = 1)}{k_{CD}(\bar{S} = \bar{S})} = \frac{\bar{S} + E_{s,\text{max}} r_{\text{min}} (\beta - \bar{S})}{E_{w} r_{\text{max}} + E_{s,\text{max}} r_{\text{min}} (\beta - 1)} \approx 6.
\]

This result agrees quite well with the ratio estimated from experimental data, as shown by figure 12.

Experimental and theoretical analysis, developed in this section, proves that in the range of low frequencies and large amplitudes, the damping forces, spring constant and dissipated energies are large. Oppositely, in the range of high frequencies and small amplitudes, the damping forces, spring constant and dissipated energies are small. Such findings agree with the expected dynamic characteristics.

In conclusion, by using the proposed yaw CD for the carbody suspension of a bullet train, it is plausible to expect improvement of the travelling stability and vibration isolation of the carbody from the bogie, in the range of higher frequencies. However, it is necessary to verify during real travel tests of the railroad vehicle, if the expected improved properties, are indeed to be achieved. Such aspects will be addressed in our future investigations on the subject of yaw CDs.
Tests conducted on the proposed yaw CD revealed that: results obtained for compression and extension are slightly different; the pressure inside the cylinder depends on the position of the piston; and, the maximal allowable loading is achieved at about half of the imposed maximal piston stroke. Such problems can be solved, by considering silica-gels with higher specific pore volumes, and by adding to the present yaw CD a pressure controlling system. Such device, should allow for pressure relief when long piston strokes are required, and for pressure adjustment in rooms placed in the left- and right-sides of the cylinder, in order to balance the damper characteristics during compression and extension, when necessary.

6. Conclusions
In this work, a yaw colloidal damper (CD) destined to carbody suspension of a bullet train was designed and manufactured. Then, its dynamic characteristics, produced by both the frictional and colloidal effects, were evaluated from experimental results, obtained during horizontal vibration tests, performed on a ball-screw shaker. Compared to the corresponding classical yaw oil damper (OD), the trial yaw CD allowed for:
1) Weight reduction of 31.6 %;
2) Large damping force, dissipated energy and spring constant at long amplitude of vibration under low excitation frequency;
3) Small damping force, dissipated energy and spring constant at short amplitude of vibration under high excitation frequency.

Damping performances were illustrated not only against the mean piston speed, but also versus the frequency and amplitude of excitation. Such information on the frequency behaviour seems to be of great importance for the proper selection of a yaw damper.

Elastic characteristics were justified by introducing a model for the spring constant that included the effect of silica-gel pore size distribution.

Dissipative and elastic properties were found to be in quite good agreement with the expected dynamic characteristics. By using the proposed yaw CD for the carriage suspension of a bullet train, improvement of the travelling stability and vibration isolation of the carbody from the bogie, in the range of higher frequencies, can be reasonably expected. However, these prospects should be verified during real travel tests of the railroad vehicle. Such aspects will be addressed in our future work.

In the end, some material (usage of a silica-gel with higher specific pore volume) and structural (addition of a pressure controlling device) improvements of the proposed yaw CD were suggested.

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