Control system with magnetorheological fluid device for mitigation of the railway vehicle hunting oscillations

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Abstract The paper presents the opportunity of using a magnetorheological device to control the lateral oscillations of a passenger railway vehicle to increase its comfort and speed features. High speed generates large amplitude oscillations in horizontal plan at the axles, bogies and carbody levels which beyond a specific value – the critical speed – leads to unstable movement of the vehicle. The lateral dynamics of the vehicle is simulated using the multibody method. It is build a model with 17 degrees of freedom considering the lateral, yawing and rolling oscillations. An original device with magnetorheological fluid based on the balance logic strategy to control the anti-yaw damper is integrated in the secondary suspension of the railway vehicle. The conception and the design of the magnetorheological device are presented. It is demonstrated that the magnetorheological semi-active suspension improves the safety and the comfort of the railway vehicle.

1. Introduction

High speed trains are more and more present in the landscape of contemporary means of transport, being efficient, economic and environmental friendly. The dynamic response of the vehicle running on rails that have geometric irregularities is determined by the frequency of the random excitation from the rail and by the vehicle’s speed [1]. High speeds generate large amplitude oscillations of the axles, bogies and vehicle body which beyond a specific value – the critical speed – leads to unstable movement of the vehicle. To keep under control the oscillations of the vehicle, through the years there were developed different types of suspension systems able to assure stable running at high speeds. The conventional suspension system for a passenger vehicle usually has two levels: the axle’s suspension, stiffer, to provide safety and the suspension of the carbody, softer, to offer best ride quality [2]. Those suspension systems usually incorporate passive elements as springs and hydraulic dampers. There are possibilities to increase vehicle’s speed by an appropriate design of the passive suspension but this approach proves to have limits. To overcome this issue the present paper considers the possibility of using a semi-active secondary suspension to control the hunting phenomenon of the railway vehicle carbody at high speeds. The semi-active device which will be used is an anti-yaw hybrid magnetorheological damper.

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2. The mathematical model of the railway vehicle

The lateral oscillations of the railway vehicle - the yaw and lateral displacement coupled oscillations - are critical for the vehicle performances because when the speed exceeds a certain value the movement of the vehicle becomes unstable, phenomenon called hunting. A high speed vehicle should be provided with a suspension able to increase the critical speed and in the same time to keep the accelerations of the carbody in certain comfort limits. Having in view the importance of the lateral oscillations phenomenon numerous authors have dedicated studies to the stability issue [3–11]. The mathematical models used in the literature to represent the vehicle differ depending on the number of degrees of freedom taken into account, the vehicle type, the linear or non-linear treatment of the wheel–rail contact phenomenon, of the forces appearing at the wheel–rail contact, as well as the irregularities of the tracks.

In view of studying the vehicle’s response in horizontal plan, it was built up a mathematical model which simulates the lateral dynamics of a four axle railway vehicle [12, 13]. It is assumed that all the elastic and damping elements forming the classical suspension systems are weightless and have linear characteristics. Under conditions of geometrical, elastic and inertial symmetry, with identical wheel and rail patterns, the equilibrium position of the coach coincides with its median position in relation to the tracks. The rolling surfaces' contact angles are small and the radii of curvature for the rolling treads remain unchanged. Conicity has been considered as having an equal constant value with the rolling surfaces' effective conicity. The mechanical model's geometrical and elastic symmetry facilitates the decoupling of the lateral movements from the vertical ones [1, 5, 10, 11]. To study the vehicle's lateral oscillations, the mechanical model considers the following degrees of freedom: \( y_c, \psi_c, \phi_c, y_{bj}, \psi_{bj}, \phi_{bj}, y_i, \psi_i \), where \( j=1,2 \) represent the bogies and \( i=1–4 \) the wheelsets.

![Figure 1. The railway vehicle model for lateral oscillations.](image)
The model highlights the phenomenon of wheel-track contact at large amplitude movement of the axles, characterizing the stability limit cycles, because the flange-rail contact force is considered a linear spring with deadband [3]. The contact forces are expressed according to Kalker’s linear theory [14]. The resultant creep force cannot exceed the adhesion force. The nonlinear effect of the adhesion limit is treated according to the Vermeulen - Johnson's nonlinear theory [14] for the case which doesn’t consider the spin creepage. In the model, the tangent track’s irregularities were considered to be periodical, represented with a sinusoid type expression, according to [5, 10]. The isolate variations of the track geometry type bump, was simulated with a single period sinusoid function which considers the irregularity tangent to the track and provides the variation of its dimensions according to the vehicle’s speed.

The vehicle is considered as it is composed of a limited number of rigid bodies, simulating its main parts, connected in between through mechanical weightless linkages: the carbody, the bogies and the wheelsets [15, 16]. Lagrange's equation method was applied in order to establish the movement equations:

$$\frac{d}{dt} \left[ \frac{\partial(E-V)}{\partial \dot{q}_k} \right] - \frac{\partial(E-V)}{\partial q_k} + \frac{\partial D}{\partial \dot{q}_k} = Q_k$$

(1)

where, \( q_k \) is the generalized coordinate, \( \dot{q}_k \) is the generalized speed, \( E \) is the kinetic energy, \( V \) is the potential energy, \( D \) is the energy dissipation function; \( Q_k \) is the generalized force corresponding to the generalized coordinate \( q_k \).

The generalized forces that are considered refers to \( y_i \) and \( \psi_j \) degrees of freedom of the wheelsets. The equations of the generalized forces are established using the wheelset efforts diagram presented in [5]. The generalized forces are \( Q_{y_i} \) generated by the lateral forces and \( Q_{\psi_j} \) due to the yaw moments acting on the wheelsets. Applying Lagrange's equations for the model’s degrees of freedom, it can be obtained the motion equations for the body case, bogies and axles [12, 13]:

$$m_c \ddot{y}_c + 2k_{cy} [2(y_c + h_{cc} \phi_c) - (y_{h1} + y_{h2}) + h_{cb} (\phi_{h1} + \phi_{h2})] + 2 \rho_{cy} [2(\dot{y}_c + h_{cc} \dot{\phi}_c) - (\dot{y}_{h1} + \dot{y}_{h2})] + h_{cb} (\ddot{\phi}_{h1} + \ddot{\phi}_{h2}) = 0 \quad (2)$$

$$I_{cz} / 2 \ddot{\phi}_c + 2(\rho_{cy} l^2 + \rho_{cc} d_c^2) \ddot{\phi}_c + 2 [k_{cy} l^2 + k_{cc} d_c^2] \ddot{\phi}_c - \rho_{cy} l (\ddot{y}_{h1} + \ddot{y}_{h2}) + \rho_{cy} l (\ddot{\phi}_{h1} + \ddot{\phi}_{h2}) h_{cb} - \rho_{cc} d_c^2 (\ddot{\phi}_{h1} + \ddot{\phi}_{h2}) = 0 \quad (3)$$

$$I_{cz} / 2 \ddot{\phi}_c + 2(\rho_{cy} h_{cc}^2 + \rho_{cc} d_c^2) \ddot{\phi}_c + 2 [k_{cy} h_{cc}^2 + k_{cc} d_c^2] \ddot{\phi}_c + 2 \rho_{cy} h_{cc} \dot{y}_c - \rho_{cy} h_{cc} (\dot{y}_{h1} + \dot{y}_{h2}) + (k_{cy} h_{cc} + k_{cc} d_c^2) (\phi_{h1} + \phi_{h2}) = 0 + (\rho_{cy} h_{cc} h_{cb} - \rho_{cc} d_c^2) \ddot{\phi}_c + 2k_{cy} h_{cc} y_c - k_{cy} h_{cc} (y_{h1} + y_{h2}) \quad (4)$$

$$m_b \ddot{y}_{bj} + 2 \rho_{cy} \dot{y}_{bj} + 2[k_{cy} + 2k_{cy} l] y_{bj} - 2 \rho_{cy} y_{bj} - 2 \rho_{cy} h_{cc} \phi_{ej} - 2(-1)^{n_b + 1} \rho_{cy} l \dot{\psi}_c - 2(\rho_{cy} h_{cb}) \ddot{\phi}_{bj} - 2k_{cy} y_{bj} - 2k_{cy} h_{cc} \dot{\phi}_{ej} - 2(-1)^{n_b + 1} k_{cy} l \dot{\psi}_c - 2(k_{cy} h_{cb} + 2k_{cy} h_{ab}) \ddot{\phi}_{bj} - 2k_{cy} (y_{bj} + y_{bj}) = 0 \quad (5)$$

$$I_{hz} / 2 \ddot{\psi}_b + \rho_{cx} d_c^2 \ddot{\psi}_b + (k_{cx} d_c^2 + 2k_{ho} a^2 + 2k_{ao} d_o^2) \ddot{\psi}_b - \rho_{cx} d_c^2 \dot{\psi}_c - k_{cx} d_c^2 \psi_c - k_{cx} d_c^2 \psi_c - k_{cx} d_c^2 \psi_c - k_{cx} d_c^2 \psi_c = 0 \quad (6)$$
\[(I_{bx}/2)\ddot{\phi}_j + \left(\rho_{cy}h_c^2 + \rho_{cz}d_z^2 + 2\rho_{cx}d_x^2\right)\dot{\phi}_j + \left(k_{cy}h_c^2 + k_{cz}d_z^2 + 2k_{cx}d_x^2\right)\phi_j + \rho_{cy}h_c\dot{y}_c + \left(\rho_{cz}h_c^2 - \rho_{cx}d_x^2\right)\dot{\phi}_c + (1)^{\gamma_1/l}\rho_{cz}h_c\dot{y}_c - \rho_{cy}h_c\dot{\phi}_j + k_{cy}h_c\dot{y}_c + \left(k_{cy}h_c^2 - k_{cz}d_z^2\right)\dot{\phi}_c + (1)^{\gamma_1/l}k_{cz}h_c\dot{y}_c - \left(k_{cy}h_c + 2k_{cy}h_c\right)\phi_j - \left(k_{cy}h_c + 2k_{cy}h_c\right)\dot{y}_j + k_{cy}h_c\left(y_{j+1} + y_{j-1}\right) = 0 \tag{7}\]

\[I_{aw}\ddot{\psi}_j - 2\rho_{aw}d_w^2\dot{\psi}_j - 2k_{aw}d_w^2\psi_j + \left[I_{aw} v \frac{\dot{\lambda}}{r_0} - 2\frac{\partial f_{12}}{v} \left(1 + r_0 \frac{\dot{\lambda}}{e}\right)\right] \ddot{\psi}_j + 2\frac{\partial f_{33}}{v} e \frac{\dot{\lambda}}{r_0} \ddot{\psi}_j + 2\left(\frac{\partial f_{12}}{v} - eQ\dot{\lambda}\right)\psi_i = 2\frac{\partial f_{33}}{v} e \frac{\dot{\lambda}}{r_0} \psi_i \tag{8}\]

\[m_{aw}\ddot{y}_j + 2\rho_{aw}\dot{y}_j + 2k_{aw}\dot{y}_j + (1)^{\gamma_1/l} \cdot 2\rho_{aw}a\psi_j + (1)^{\gamma_1/l} \cdot 2k_{aw}a\psi_j + 2\rho_{aw}h_ah\phi_j + 2k_{aw}h_a\phi_j + 2\frac{\partial f_{12}}{v} \ddot{\psi}_j + 2\frac{\partial f_{11}}{v} \dot{\psi}_j - 2\frac{\partial f_{1}}{e} \psi_i = \left(\frac{\dot{\lambda}}{e} - 2k_{aw}\right) \eta_i \tag{9}\]

The significance of the symbols employed in the equations (2) - (9) is presented in the Table 1. The equation system representing the mathematical model of the lateral motion of the railway vehicle can be written in the following matrix form:

\[\begin{bmatrix} [M] \ddot{\psi} + [C] \dot{\psi} + [K] \psi = \{F(t)\} \end{bmatrix} \tag{10}\]

where \([M] \in \mathbb{R}^{17 \times 17}, [C] \in \mathbb{R}^{17 \times 17}\) and \([K] \in \mathbb{R}^{17 \times 17}\) are the mass, damping and stiffness matrixes of the vehicle’s system, \(\{\dot{\psi}\}, \{\psi\}\) and \(\{\dot{\psi}\}\) are the accelerations, speeds and displacements vectors and \(\{F(t)\} \in \mathbb{R}^{17}\) is the vector of the periodical excitation given by the tracks.

Through a variable change, the system can be transformed into an ordinary differential equation system [6] which it is treated using numeric methods:

\[\{\dot{\gamma}\} = [E] \{\gamma\} + \{F^*(t)\} \tag{11}\]

where,

\[E = \left[ \begin{array}{c} 0 \\ -[M]^{-1} [K] - [M]^{-1} [C] \end{array} \right] \]

\[\{F^*(t)\} = \left\{ \begin{array}{c} 0 \\ \{[M]^{-1} \{F(t)\}\} \end{array} \right\} \tag{12}\]

The equations (11) are employed both for the study of stability and the system’s response using a numerical integration method of the movement equations, the Runge – Kutta method of 4th order, for a simulation software has been designed. To simulate the vehicle’s response, there are used the construction characteristics presented in Table 1. As example, the response of the vehicle with the characteristics in Table 1, launched on a tangent track with periodic irregularities, running with 180 km/h – the maximal testing speed, is presented in the Figures 2 – 4, indicating that the tracks’ perturbations effect is slightly felt at the coach case level, as opposed to the bogie and axles where it
persists during the coach's circulation. The coach's main suspension acts correspondingly and meets the comfort demands inside the coach.

### Table 1. Construction characteristics of the vehicle.

| Characteristics                     | Symbols | Values   | Units    |
|-------------------------------------|---------|----------|----------|
| Body case mass                      | $m_c$   | 30760    | kg       |
| Bogie mass                          | $m_b$   | 2300     | kg       |
| Wheelset mass                       | $m_o$   | 1410     | kg       |
| Carbody moments of inertia          | $I_{cx}$| 53596    | kg m²    |
|                                     | $I_{cz}$| 1661732  | kg m²    |
| Bogie moments of inertia            | $I_{bx}$| 2240     | kg m²    |
|                                     | $I_{bz}$| 2965     | kg m²    |
| Axles moments of inertia            | $I_{ox}$| 980      | kg m²    |
|                                     | $I_{oz}$| 100      | kg m²    |
| Central suspension stiffness        | $k_{cx}$| 133      | kN m⁻¹   |
|                                     | $k_{cy}$| 133      | kN m⁻¹   |
|                                     | $k_{cz}$| 473      | kN m⁻¹   |
| Axle suspension stiffness           | $k_{ox}$| 256      | kN m⁻¹   |
|                                     | $k_{oy}$| 885      | kN m⁻¹   |
|                                     | $k_{oz}$| 904      | kN m⁻¹   |
| Central suspension damping          | $\rho_{cx}$|      | kN m⁻¹ s⁻¹ |
|                                     | $\rho_{cy}$|      | kN m⁻¹ s⁻¹ |
|                                     | $\rho_{cz}$|      | kN m⁻¹ s⁻¹ |
| Damping of the axle suspension      | $\rho_{ax}$| 3.67   | kN m⁻¹ s⁻¹ |
| Wheel tread nominal radius          | $r_0$   | 0.460    | m        |
| The track’s gauge                   | $2e$    | 1.435    | m        |
| The bogie's wheelbase               | $2a$    | 2.560    | m        |
| The distance between bogies         | $2l$    | 17.2     | m        |
| The distance between the secondary suspension springs | $2d_c$ | 2 m |
| The distance between the primary suspension springs | $2d_p$ | 2 m |
| The distance case center – central suspension | $h_{cc}$ | 1.24 m |
| The distance axles suspension - bogie center | $h_{cb}$ | 0.06 m |
| The distance central suspension - bogie center | $h_{cb}$ | 0.06 m |
| Load on wheel                       | $Q$     | 51250    | N        |
| The longitudinal creep coefficient  | $f_{11}$| 9430000  | N        |
| The lateral/spin creep coefficient  | $f_{12}$| 1200     | N m      |
| The spin creep coefficient          | $f_{22}$| 1000     | N m²     |
| The lateral creep coefficient       | $f_{33}$| 10250000 | N        |
| The effective wheel conicity        | $\lambda$| 0.14    | -        |
| The maximum testing speed           | $v_{max}$| 50      | m s⁻¹    |

The mathematical model of the vehicle is used to study the vehicle stability. The dynamic system is asymptotically stable if and only if all the eigenvalues of the matrix $E$ have a negative real part [1]. For the vehicle with the characteristics in Table 1, the value of linear critical speed is 245.6 km/h. The non-linear critical speed is determined by simulation [17] with the mathematical model and it has the value of 234.2 km/h. In Figure 5 it is shown the response of the leading axle of the trailing bogie at the stability limit and in Figure 6 the diagram of stability limit.
For the vehicle with the characteristics in the Table 1, the stationary solution of the dynamic system is asymptotically stable for speeds less than 245.6 km/h. For speeds greater than 245.6 km/h the solution of the system is unstable. The point of the diagram with the coordinates (245.6; 0) is consequently a bifurcation point – a point where an unstable periodic solution bifurcates subcritically.
The solution becomes unstable in the saddle-node bifurcation point with the coordinates (234.2; 9) where the non-linear speed is reached and the amplitude of the wheelset’s movement equals the rail-wheel clearance.

3. The limits of the classical suspension

The mathematical model of the vehicle previously presented can become a useful tool to improve the suspension design. A parametric study of the vehicle’s non-linear stability shows the possibilities of raising the vehicle’s speed through the passive suspension construction.

![Figure 7](image1.png)

**Figure 7.** Primary suspension stiffness influence on critical speed.

![Figure 8](image2.png)

**Figure 8.** Primary suspension damping influence on critical speed.

![Figure 9](image3.png)

**Figure 9.** Secondary suspension stiffness influence on critical speed.

![Figure 10](image4.png)

**Figure 10.** The lateral accelerations of the carbody.

It is investigated the influence of the suspension damping and stiffness lateral and longitudinal components on the critical speed of the vehicle. The critical speed of the vehicle is strongly influenced by the primary suspension parameters $k_{ox}$, $\rho_{oy}$ and by the lateral damping of the secondary suspension $k_{cy}$.

In the Figures 7, 8 and 9 are presented the influences of the mentioned parameters on the vehicle’s critical speed. The increase of the critical speed is not the unique criteria of the railway vehicle performances. According to UIC 518 leaflet, the assessment of the vehicle ride quality can be made by measuring the accelerations of the carbody on vertical and lateral directions. To establish the lateral acceleration of the carbody it is employed the mathematical model presented previously. The expression of the lateral acceleration is:
There are determined the accelerations of the car body for different values of the central suspension lateral damping coefficient. It can be seen that the result of using more rigid dampers in the central suspension is worse in terms of comfort, due to the coupling effect of those between the lateral oscillations of the bogies and of the car body – Figure 10.

4. The opportunity of a semi-active magnetorheological railway suspension

In the last years the magnetorheological dampers have been employed with predilection in the applications for low frequencies vibrations damping type vibration insulation or earthquake protection of civil constructions. There are also magnetorheological applications in the automotive, aeronautic or military industries.

In the railway field there are not many vehicles using magnetorheological devices. One of the reasons it may be the fact that active technologies are already widely used on high speed trains: the tilting devices which became standard since 1990, the servo-pneumatic and servo-hydraulic active suspensions or the piezoelectric active suspensions. Although extremely effectual, the active suspensions require high powers and sophisticated control technologies. The fact that the active systems bring supplementary energy to the vehicle may lead to system’s instability.

Semi-active suspension systems combine the advantages of the construction simplicity and of the low costs specific to the passive devices with those specific to the active systems, resulting very effective devices that assure high performing control with low power consumption, reliable and stable. The magnetorheological dampers valorize the ability of the material to radically modify its rheological properties in milliseconds when in the presence of a magnetic field. The short reaction time becomes another strong point towards the employment of the magnetorheological dampers to railway vehicles, having in view the complexity of the operational conditions of those: variable loads, speeds, track irregularities, weather conditions.

There are few articles treating the magnetorheological suspensions for railway vehicles. In [18] the authors implemented several control strategies used in the automotive industry to a railway vehicle suspension through a computer simulation program which accounts different operational conditions. The results showed that semi-active control improves ride quality of the vehicle. The study, carried on in mid-90th years, evokes the opportunity of using in the railway vehicle suspension electrorheological or ferrofluid dampers. A study of a magnetorheological damper in the secondary suspension of a locomotive is presented in [19]. The paper showed the fact that the semi-active strategy is more efficient than the passive one and the results may be closed to those obtained with an active damper. The design and construction of a magnetorheological double ended damper is presented in [20]. The damper model has been established using a testing bench and the ride quality of a vehicle using such a damper was assessed by means of a simulation software showed to improve. The papers [21, 22] are of interest because it is presented the implementation of a acceleration feed-back control strategy and the positive results of using a magnetorheological damper in vehicle’s secondary suspension.

A possibility to continue to increase the vehicle’s speed without affecting the comfort is the use of the controllable suspensions. In the present paper, the option for controlling the lateral oscillations of the vehicle’s car body is to use a semi-active control system with a magnetorheological fluid device.

The semi-active control systems have the following advantages:

- the oscillations insulation is accomplished by dissipation of energy, similar to the passive devices;
the system assures full adaptability as active systems does;
no additional energy is introduced in the system;
the energy needed to operate the semi-active device may be supplied by low power sources;
there are robust and reliable devices;
the systems are opened to the unconventional technical solutions.

As operational device of the semi-active suspension, a hybrid magnetorheological damper will be introduced in the secondary suspension to control the passive anti-yaw hydraulic damper.

The magnetorheological damper brings forward significant benefits in comparison to the classic versions:

the damping force is controllable in real time;
assures great value damping forces at low speeds which is practically impossible with hydraulic dampers;
the magneto-rheological fluid has its own viscosity which assures a minimal damping force even if the power supply fails;
the constructive solution is not sophisticated and easy to be implemented to replace classical dampers.

In order to increase the railway vehicle performance, four magnetorheological controlled dampers will be introduced between the bogies and the carbody to replace the passive dampers. The semi-active systems act to reduce the elastic forces resultant across the vehicle secondary suspension. The semi-active suspension system should continuously control the vehicle’s response and set the lateral damping force of the secondary suspension to the appropriate values.

If in the equations (10) are replaced the expressions of the lateral damping forces of the secondary suspension with those given by the magnetorheological dampers, it will result the following formulation of the matrix form of the vehicle’s model:

\[
[M][\ddot{q}] + [C][\dot{q}] + \{F_d(\Delta q, \dot{\Delta q}, i)\} + [K][q] = \{F(t)\}
\] (14)

where \([C]\) is the passive damping matrix, \(\{F_d(\Delta q, \dot{\Delta q}, i)\}\) is the controllable damping force of the secondary suspension vector and \(i\) is the intensity of the current applied to the driver. The controllable damping force depends on the relative lateral displacement \(\Delta q = \Delta y_c\) and the relative lateral speed \(\Delta \dot{y}_c\) in the secondary suspension and on the applied current intensity.

Using the notations according to Figure 1, the relative displacement and speed equations are:

\[
\Delta y_c = y_c + h_{cc}\phi_c + (-1)^{i+1} l\psi_c - y_{bj} + h_{cb}\phi_{bj}
\] (15)

\[
\Delta \dot{y}_c = \dot{y}_c + h_{cc}\dot{\phi}_c + (-1)^{i+1} \dot{l}\psi_c - \dot{y}_{bj} + h_{cb}\dot{\phi}_{bj}
\] (16)

4.1. The hybrid magnetorheological damper

The hybrid magnetorheological dampers are devices that use a hydraulic section to create the damping force and a magnetorheological section to control the hydraulic section. The control element is a small magnetorheological device. In literature this type of damper is known also as magnetorheological piloted hydraulic damper. This type of damper features several advantages as: reduced cost due to the small quantity of magnetorheological fluid required, reduced weight, aspect and overall dimensions similar to those of the passive damper. In fact, the design of the hybrid damper will start from the original passive damper used in the secondary suspension of the vehicle. Figure 11 presents the schematic diagram of a hybrid magnetorheological damper proposed by the authors to be included in the vehicle’s semi-active suspension.

The main parts of the hybrid damper, according to Figure 11, are as follows:
• the hydraulic cylinder;
• the control pressure selector;
• the magnetorheological piloting device;
• check valves.

Figure 11. The hybrid magnetorheological damper.

As control strategy for the magnetorheological damper, it is selected a sequential algorithm based on the balance of forces (balance logic) built to control the system’s vibrations exclusively on an energy dissipation basis. According to the balance logic, the damping force has to be controlled to balance the elastic force when the carbody is moving away from its equilibrium position – the elastic and damping forces act in opposite directions – and by setting the damping force to a minimal value when the forces have the same direction.

The construction principle of the hybrid magnetorheological damper, presented in Figure 11 fulfils those requirements. The pressures in the chambers I and II of the damper are controlled by the magnetorheological device (the controlled hydraulic resistance) through the pressure switch. The damping force is purely hydraulic. The control system uses speed and movement sensors. When the movement and the speed have opposite signs, the solenoid is supplied with current and the magnetorheological fluid is activated. The current intensity is supplied in such way that the cumulated forces in chamber B of the device and of the magnetorheological fluid resistance allow the pressure in chamber A to move the piston of the device until the flow between the chambers I and II of the damper would determine the pressure gradient which will produce the damping force according to the control algorithm. The solution proposed for the magnetorheological device is a double ended piston with a fixed solenoid mounted in a cylinder. This solution brings benefits because the solenoid supply wires do not have to move in operation.

4.2. The hybrid magnetorheological damper design
The development of the solution for the magnetorheological controlled damper begun from the original passive damper type T50/20x210 with the characteristics presented in the Table 2. Having in view that the active surface of the damper’s piston is \( A_p = 16.5 \, \text{cm}^2 \), the pressure drop is \( \Delta p_d = 30.31 \, \text{daN cm}^{-2} \).

To establish the dimensions of the controlled hydraulic resistance piston it will be calculated the surface of the opening, \( A_o \), necessary to transfer the flow \( Q \) with a pressure drop \( \Delta p_d \).
Table 2. The characteristics of the passive damper

| Characteristic          | Symbol | Value | Unit   |
|------------------------|--------|-------|--------|
| Piston diameter        | \(d_p\) | 0.05  | m      |
| Piston rod diameter    | \(d_t\) | 0.02  | m      |
| Maximal force          | \(F\)  | 5000  | N      |
| Velocity               | \(v_p\) | 0.1   | \(\text{m s}^{-1}\) |
| Fluid flow             | \(Q\)  | \(1.65 \times 10^{-4}\) | \(\text{m}^3 \text{s}^{-1}\) |

The fluid’s flow is considered turbulent. The following equation will be applied:

\[
Q = c_d A_o \left(\frac{2 \Delta \rho d^4}{\rho}\right)^{\frac{1}{2}}
\]  

(17)

where \(c_d \approx 0.61\) characterize the turbulent flow, \(\rho = 880\ \text{kg/m}^3\) is the density of the hydraulic fluid.

The minimal opening surface will be: \(A_o^{\text{min}} = 3.26\ \text{mm}^2\).

The maximal surface of the opening will be obtained for free flow between the damper chambers. The minimal pressure gradient is chosen to be 3 bar – the pressure drop estimated as being generated by the hydraulic resistances on the circuit - and the maximal surface of the opening is \(A_o^{\text{max}} = 10.36\ \text{mm}^2\).

Considering that the piston of the control device has a diameter of 12 mm, the minimal and maximal forces acting on the piston are:

- \(F^{\text{min}} = 34\ \text{N}\) – the force of the spring in chamber B necessary to close the circuit between the chambers I and II by placing the piston of the controllable device in a position that completely closes the hydraulic resistance;
- \(F^{\text{max}} = 343\ \text{N}\) – the minimal force that should be produced by the magnetorheological fluid device at the highest value of the applied current.

The opening is considered to be rectangular with the bigger leg, \(L_o\), perpendicular on the hydraulic resistance piston axle equal to 6 mm. The strokes of the two extreme positions of the piston are: \(c^{\text{min}} = 0.54\ \text{mm}\) and \(c^{\text{max}} = 1.73\ \text{mm}\). The maximal stroke is chosen to be 2 mm.

Table 3. The input data of the design.

| MR fluid type       | MRF-132LD       |
|---------------------|-----------------|
| Maximal velocity of the piston | \(v^{\text{max}} = 0.015\ \text{m s}^{-1}\) |
| Maximal stroke      | \(c^{\text{max}} = 5 \times 10^{-3}\ \text{m}\) |
| Interior diameter of the device | \(d = 22 \times 10^{-2}\ \text{m}\) |
| Diameter of the piston rod | \(d_p = 8 \times 10^{-3}\ \text{m}\) |
| Operation temperatures range | \(-20^\circ\ \text{C} + 150^\circ\ \text{C}\) |
| A wide range of elastomers compatibility | - |

The constructive model is the one of the double ended device. A design method of a double ended magnetorheological damper is presented in [23], [20] and more developed in [24]. The input data are presented in the Table 3.
To continue the dimensional design, following parameters are chosen according to the magnetorheological fluid characteristics supplied by Lord Corp. [25]. The chosen parameters are shown in the Table 4.

Table 4. Magnetorheological fluid parameters

| Parameter                        | Value          |
|----------------------------------|----------------|
| Shear rate                       | $\dot{\gamma} = 140 \text{ s}^{-1}$ |
| Maximal intensity of the magnetic field | $H = 250 \text{ kA m}^{-1}$ |
| Fluid viscosity                  | $\eta = 0.25 \text{ Pa s}$ |
| Yield stress                     | $c = 44.1 \text{ kPa}$ |

The aim is to maximize the controllability of the device by maximizing the dynamic range coefficient and to minimize the volume of magnetorheological fluid. Imposing those conditions it results the value of the thickness of the gap between the piston and the inner surface of the device [24].

Figure 12. The solenoid of the device

To design the constructive parameters of the solenoid there are calculated the dimensions of the magnetic circuit composed of the poles, the device case between the poles, the support of the solenoid and the magnetorheological fluid [24], considering an applied current of 1.5A.

The resulting parameters of the magnetorheological device shown in Figure 12 are presented in the Table 5.

The functional characteristics of the device with magnetorheological fluid in Figures 13 and 14 were experimentally determined for sinusoidal motion with 1.0 Hz frequency and 0.002 m displacement amplitude.

Table 5. The parameters of the magnetorheologic device

| Parameter                        | Value          |
|----------------------------------|----------------|
| Thickness of the gap             | 0.4 mm         |
| Total force of the device        | 406.7 N        |
| Number of poles                  | 2              |
| Pole width                       | 2 mm           |
| Maximal solenoid diameter       | 16 mm          |
| Solenoid length                  | 30 mm          |
| Wire diameter                    | 0.6 mm         |
As it can be seen in the figures, the maximal value of the force acting on the solenoid piston is bigger than the value presented in Table 5. This fact is generated by the number of windings that for technological reasons was bigger than the calculated one. The maximal value of the current intensity used in the tests, 1.6A is greater than the estimated 1.5A and this has also an increasing effect on the force. The force of the device is greater than the minimal needed value of 343 N and the device is suitable to control the hydraulic damper.

4.3. The semi-active suspension simulation

According to the balance logic control strategy [26], the equation of the semi-active damping force is:

\[
F_d(\Delta y_2, \Delta \dot{y}_2, i) = \begin{cases} 
F_e(\Delta y_e) \sgn(\Delta y_e) \Delta y_e \Delta \dot{y}_e < 0 \\
F_{d_{min}}(\Delta y_e, \Delta \dot{y}_e), \Delta y_e \Delta \dot{y}_e \geq 0 
\end{cases}
\]  \tag{18}

where \(F_e(\Delta y_e)\) is the lateral elastic force across the secondary suspension and \(F_{d_{min}}(\Delta y_e, \Delta \dot{y}_e)\) is the minimal damping force, due for example to the inherent energy dissipation when the hydraulic fluid flows through the control pressure selector’s orifices, reached when the magnetic field isn’t applied (\(i = 0\)). Having in view the conception of the hybrid damper, in the next pages \(F_{d_{min}}\) will be considered negligible in comparison to the damping forces in semi-active mode. The equation of the damping force will become:

\[
F_d(\Delta y_e, \Delta \dot{y}_e, i) = \begin{cases} 
2\alpha k_c |\Delta y_e| \sgn(\Delta y_e), \Delta y_e \Delta \dot{y}_e < 0 \\
0, \Delta y_e \Delta \dot{y}_e \geq 0 
\end{cases}
\]  \tag{19}

where \(\alpha\) is a dimensionless gain factor.

To simulate the dynamics of the device with magnetorheological fluid of the hydraulic resistance, it will be used the modified Bouc-Wen model, proposed by Spencer et al. in 1997 [27]. This model improves the simulated answer of the magnetorheological damper in the yield region.

The Bouc-Wen models have been conceived aiming from the start to represent the magnetorheological dampers. The modified Bouc-Wen model, shown in Figure 15, consist in a Bouc-Wen operator which represents the hysteretic behaviour of the damper set in parallel with a viscous damper and a spring simulating the viscous damping and the stiffness of the damper at high velocities, combined with a spring representing the pneumatic accumulator stiffness and a viscous damper to represent the viscous damping effects at high velocities.

The governing equations of the Bouc-Wen model [27] are:

\[
F(t) = c_i y + k_i (x - x_0)
\]  \tag{20}
\[
\dot{y} = \frac{1}{c_0 + c_1} [\alpha z + c_0 \dot{x} + k(x - y)]
\]

\[
\ddot{z} = -\gamma |\dot{x} - \dot{y}| \cdot |z|^{n-1} \cdot z - \beta (\dot{x} - \dot{y}) \cdot |\dot{z}| + A(\dot{x} - \dot{y})
\]

where \( y \) is the internal displacement of the damper, \( A, \gamma, \beta \) are the parameters that determine the shape of the loop, \( n \) is the parameter that determines the smoothness of the force-displacement diagram, \( x_0 \) is the initial displacement representing the pneumatic accumulator and \( z(x) \) is the hysteretic component that represents a function of the time history of the displacement.

To enlighten the influence of the applied current on the magnetorheological damper behavior, it is assumed that the parameters \( c_0, c_1, \alpha \) depend on the voltage applied to the current driver. There are several formulations of this dependence: linear, third grade polynomial or asymmetric sigmoid function. The linear dependence according to [27] can be expressed with the following equations:

\[
\alpha(u) = \alpha_a + \alpha_b u
\]

\[
c_i(u) = c_{ia} + c_{ib} u
\]

\[
c_0(u) = c_{0a} + c_{0b} u
\]

where \( \alpha_a, c_{ia} \) and \( c_{0a} \) are the values of the parameters at 0 V – free flow of the magnetorheological fluid.

The Bouc-Wen model involves 9 parameters that characterize the damper’s behavior. Those parameters should be determined for each value of the applied current in such way that the predicted response of the damper using the model should be as close as possible to the experimental results obtained through tests. This represents an optimization problem that can be solved through genetic algorithms method. In [26] it is recommended a sequential formulation of the voltage control law similar to the balance logic strategy that would simplify the identification process:

\[
u(x, \dot{x}) = \begin{cases} 
\varepsilon K \varepsilon \sgn(\dot{x}), & x \dot{x} < 0 \\
0, & x \dot{x} \geq 0
\end{cases}
\]
where $\varepsilon$ is a dimensionless gain factor experimentally determined to obtain a convenient transmissibility or a minimum r.m.s. acceleration output and $K$ is the measuring calibration factor expressed in $Vmm^{-1}$.

**Figure 16.** Carbody acceleration ($v=220 \text{ km/h}, \eta_0 = 5\text{mm}$).

**Figure 17.** Carbody acceleration ($v=220 \text{ km/h}, \eta_0 = 10\text{mm}$).

**Figure 18.** Carbody acceleration ($v=250 \text{ km/h}, \eta_0 = 10\text{mm}$).

**Figure 19.** Carbody acceleration PSD.

Several simulations of the employment of the semi-active suspension compared with the use of the classic solution in terms of carbody accelerations are made for different speeds and periodic irregularities amplitudes. To perform those simulations, the controlled damping force generated by the semi-active dampers was included in the mathematical model of the vehicle.

As presented in Figures 16 - 18, the use of a semi-active control strategy reduces significantly the values of the lateral accelerations improving the railway vehicle’s comfort.

Figure 19 shows the influence of the semi-active control strategy on the railway vehicle in terms of the p.s.d. value of the carbody acceleration response. There are opened perspectives to study the improvement of passengers comfort using the hybrid magnetorheological damper described previously which offers a simple and reliable control of transmissibility of vehicle’s oscillations from the rolling gear to the carbody.
5. Conclusions
Having in view that for the passenger trains the safety and comfort conditions are essential, the paper propose a new type of semi-active suspension controlled by a device with magnetorheological fluid. The proposed system may transform any hydraulic damper in a damping system with semi-active control. This application shows the capability of the intelligent fluids to develop a control system more reliable and cost effective than the electro-hydraulic devices.

The design of the piloting device with magnetorheological fluid offers possibilities of further developments, being easy adaptable to the specific requirements of the application. For example, if the controlled damper should operate within a wider range of damping forces, the dimensions of the opening and the piston’s stroke can be modified accordingly. The control strategy, thought simple, can be very flexible through the dimensionless gain factor which can be adapted according to the specific operational conditions of the vehicle.

The simulation of the employment of a semi-active secondary suspension was performed by means of the mathematical model of the vehicle lateral oscillations developed by the authors. The comparison between the carbody accelerations in passive and semi-active modes shows that the semi-active suspensions represent a reliable option for the improvement of the railway vehicle’s comfort and safety.

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