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Semi-Active Damping System for Secondary Suspension of Electric Multiple Unit

T. Michálek[1], F. Jeniš[2], Z. Strecker[2], I. Mazůrek[2], O. Macháček[2], P. Staněk[3], Z. Malkovský[3] and J. Chvojan[4]

[1]Faculty of Transport Engineering, University of Pardubice Czech Republic
[2]Faculty of Mechanical Engineering, Brno University of Technology Czech Republic
[3]Testing laboratory of railway rolling stock, VÚKV, Praha, Czech Republic
[4]Dynamic testing and force laboratory, Research and Testing Institute in Pilsen, Czech Republic

Abstract

In this paper, a project of research and development of a semi-actively controlled damping system for an electric multiple unit and its current results are presented. The aim of the project is to develop and manufacture a set of fast magnetorheological dampers including its control, which can be applied in secondary suspension of an (inter)regional electric multiple unit as vertical and lateral dampers, and validate the expected contribution of this system for improvement of vehicle running comfort by operational tests. For purposes of assessment of efficiency of the considered control algorithms and tuning of their parameters, multi-body simulations were used. Besides to that, full scale laboratory tests of secondary suspension system in vertical direction on a dynamic test stand were used to verify the expected effects of the semi-actively controlled dampers.

Keywords: electric multiple unit, secondary suspension, magnetorheological damper, semi-active control, multi-body simulation, full scale test, running comfort.

1 Introduction

Increasing speed of railway vehicles on new and modernized railway lines as well as increasing demands of passengers on ride comfort lead to higher requirements on design of suspension system of the vehicles. Besides to that, changes in concept of the vehicles (long low floor coaches with aluminium vehicle body and traction equipment situated on the vehicle roof) together with the increasing quality of the modernized tracks (welded rail joints, low level of track irregularities) lead to changes in excitation of the running vehicles. Both these aspects influence optimal properties of suspension dampers. In case that the requirements on the ride comfort can be met with using of traditional hydraulic dampers only with difficulties, application of new technologies can be considered. For purposes of ride comfort improvement, especially the active and semi-active systems can be used. Because of energy demands and safety aspects of actuators, semi-actively controlled systems offer an interesting possibility.

In framework of solving the R&D project No. TN02000054 “BOVENAC” of the Technology Agency of the Czech Republic, a research, development and operational verification of a semi-actively controlled damping system for an electric multiple unit (EMU) is realized in the relevant work package. Within this project, STROJÍRNA OSLAVANY, spol. s r.o. (STOS) as the manufacturer of damping systems for railway applications cooperates with the Faculty of Mechanical Engineering of the Brno University of Technology (BUT), the Faculty of Transport Engineering of the University of Pardubice (UPCE) and two Czech research institutes – VÚKV a.s. and Výzkumný a zkušební ústav Plzeň s.r.o. (VZÚ). Activities of BUT are focused on a proposal of magnetorheological dampers with a short time response and their control, UPCE realizes multi-body simulations of running performance of the EMU coach equipped with the semi-actively controlled dampers in order to find optimal characteristics of the semi-active dampers and verify the proposed control algorithms, VZÚ participates on full scale tests of the damping system using a dynamic test stand and VÚKV ensures especially running tests and measurements with the instrumented EMU on the Czech railway network. The aim of this paper is to present the actual state of research and obtained results.

2 Development of magnetorheological dampers

The design of the new dampers is based on the magnetorheological (MR) bogie yaw damper developed earlier [1]. The MR damper is a single shell design with an external expansion tank, as can be seen in Fig. 1. The damper is mounted in rubber bushings with a nominal stiffness of 20 kN/mm. An accelerometer and position sensor are mounted on the outer cover. A second accelerometer can be fitted to the lower mount if required. The piston forms a closed electromagnetic circuit with a wound coil. Its power supply wires pass through the piston rod and are routed to a socket on the upper mounting bracket for the control unit, which also receives the signals from the aforementioned sensors. Based on the selected control algorithm, the control unit supplies the damper with an electric current of 0–7 A.

As the damper piston moves, the MR fluid flows through the gap in the piston, past the magnetic circuit coil. When the coil is de-energised, the fluid passes through the piston with minimal resistance and the damping force is low. When it is beneficial to increase the damping force the controller switches on the current in the coil. This creates a magnetic field in the active area of the piston which increases the apparent viscosity of the MR fluid and, therefore, the damping force. The controller algorithms work with signals from the accelerometers and position sensor.

The magnetic circuit is made of appropriate cutting steel and is grooved (Fig. 2) to achieve good transient behaviour of the damper [2]. From this point of view, the key characteristics of the damper are the force-velocity-current (F-v-I) map and the time of transient response. The F-v-I map expresses the dependence of the damper force on the actual piston velocity and the electric current in the coil. The measured F-v curves are shown in Fig 3; the graph also shows the F-v curve of the original passive damper which are replaced by the MR dampers on the investigated EMU.

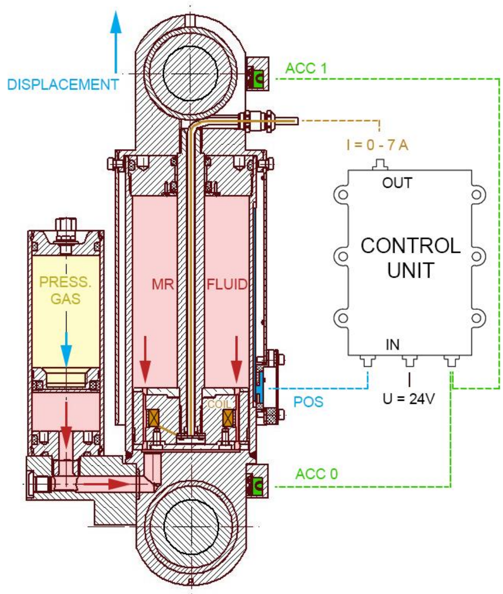

Figure 1: Scheme of MR damper.

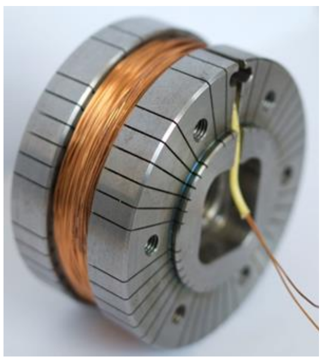

Figure 2: Unwound core before pouring in insulation material.

For a real damper, there is a delay between the change of electric current and the damping force. With a step change in the current, the damping force increases or decreases to the desired value exponentially. Therefore, the transition between the F-v curves in time is modelled in the simulation program as in a first-order dynamic system [3], using the equation of exponential function (3):

$$F(v,t) = F_0(v) + \left(F_1(v) - F_0(v)\right) \cdot \left(1 - \mathrm{e}^{-\frac{t}{\tau_{63}}}\right) \tag{1}$$

where $F_0(v)$ is the force at $t = 0$, and $F_1(v)$ is the force corresponding to the desired force for the given velocity and current, v is the piston velocity, t is the time from the step of the control signal, and τ_{63} is the damper time constant. This constant is defined as the time required to reach 63.2% of the required value [3]. The time of transient response depends on piston velocity. Measured dependency for a current rise from 0 A to 5 A and a drop from 5 A back to 0 A is shown in Fig. 4. It is evident that the drop in force when the current is switched to 0 A is much faster than the rise in force. The methodology of measuring of the F-v-I map and the transient response time is described in detail in [4].

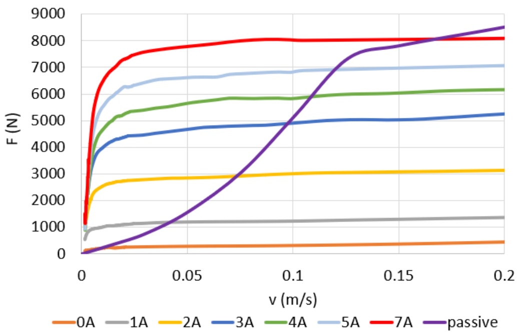

Figure 3: Measured F-v curves of the MR damper for electric currents 0–7 A.

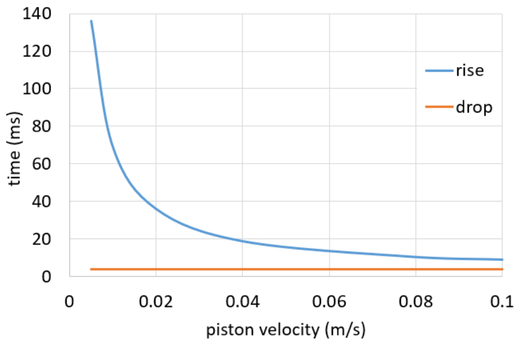

Figure 4: Force response time as a function of piston velocity.

3 Multi-body simulation of EMU with semi-active dampers

In order to investigate a potential contribution of the semi-actively controlled dampers in the secondary suspension stage of a real railway vehicle to the ride comfort, a multi-body model of an intermediate traction coach of the interregional single-deck ŠKODA 10Ev EMU was created. The model was built in the SJKV multi-body simulation tool which is being developed at UPCE. From the point of view of application of the semi-active damping system, the secondary suspension of the investigated vehicle plays a significant role. This suspension consists of a pair of (pneumatically interconnected) air springs with additional air reservoirs and an anti-roll bar per bogie (Fig. 5) and is supplemented with hydraulic vertical, lateral and anti-yaw dampers. A more detailed description of the vehicle as well as the computational model can be found in [5].

Figure 5: Trailer bogie of a single-deck ŠKODA EMU.

The basic semi-active control strategy of the MR dampers is based on the Skyhook Linear algorithm [6]. This algorithm has been simplified as follows:

$$I = \begin{cases} \mathrm{sat}\left(I_{\max} \dfrac{\dot{y}_2}{\dot{y}_2 - \dot{y}_1}\right), & \dot{y}_2\,(\dot{y}_2 - \dot{y}_1) \geq 0 \\ I_{\min}, & \dot{y}_2\,(\dot{y}_2 - \dot{y}_1) < 0 \end{cases} \qquad (2)$$

where I is a required electric current, $I_{\max}$ is the maximal current, $I_{\min}$ is the minimal current, $\dot{y}_2$ is the vehicle body (lateral, or vertical) velocity, $\dot{y}_1$ is the relevant bogie frame velocity, and the sat function denotes that $I \in \langle I_{\min}; I_{\max} \rangle$. The actual electric current defines the F-v curve (see the F-v-I map in Fig. 3). The simulation works with the “required current” (see Eq. (2)) and the “computational current” (corresponding to the actual force) which allows respecting the transient behaviour (analogically to Eq. (1)). To find the optimal setting of the dampers (the maximal current), extensive simulations of vehicle running on tracks with different track quality (given by track irregularities) at different speeds were performed; the vehicle response was evaluated.

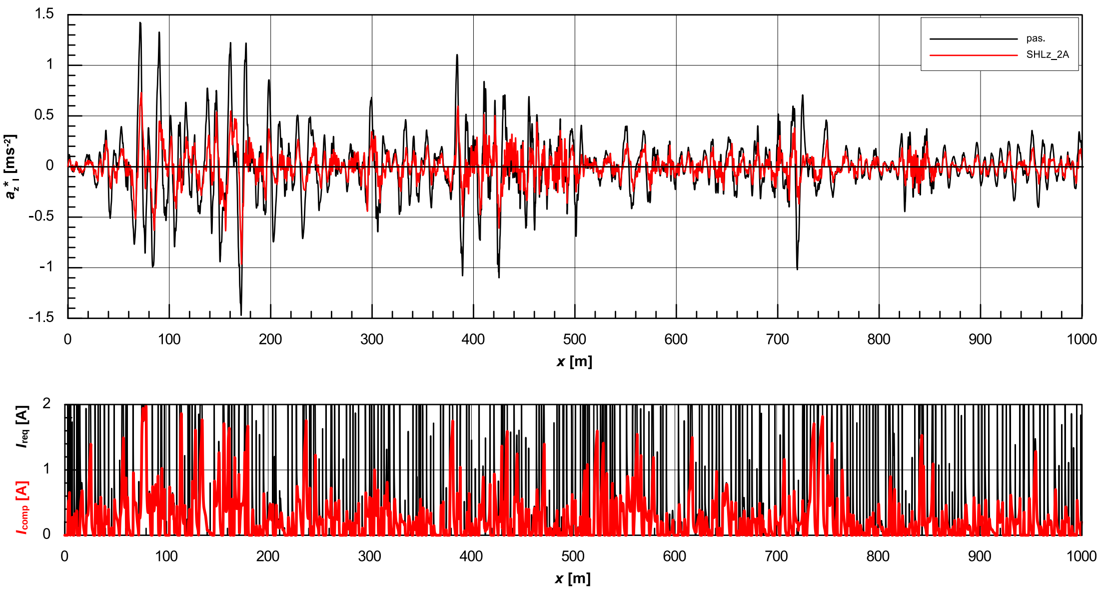

Figure 6: Comparison of vertical acceleration records at simulation of the investigated vehicle at 120 km/h with passive (black) and semi-actively controlled (red) vertical secondary dampers running on a track with measured irregularities (top) and relevant switching of the right MR damper of the front bogie (bottom).

The upper graph in Fig. 6 shows a comparison of vertical acceleration records in the vehicle body (on the floor over the front bogie pivot) as the simulation results of vehicle running in straight track at 120 km/h. The black line represents the model with passive dampers; the red line corresponds to the model equipped with semi-actively controlled vertical dampers, considering Skyhook Linear algorithm with a maximal current of 2 A. Then, the bottom graph in Fig. 6 demonstrates relevant switching of the right vertical damper on the front bogie at the simulation (the black line shows the "required electric current" and the red line the "computational electric current").

4 Experimental verification of semi-actively controlled dampers at the dynamic test stand

For purposes of verification of the contribution of the semi-active damping system, which is predicted by means of the simulation results, as well as for optimization of the applied control strategies and also estimation of real dynamic characteristics of the spring-damper system, physical tests of the vertical secondary suspension were realized on a dynamic test stand.

The test stand for the air springs (see Fig. 7), which was designed at the VÚKV and is placed in the VZÚ, was used to measure dynamic characteristics. The test stand consists of a supporting structure, an excitation cylinder, a distributor and weight. The supporting frame provides the motion control of the moving parts, carries the weight and also serves to support the weights by means of supports, if necessary. The air spring is placed between the weight and the distributor via the supporting structure. The distributor contains an additional air reservoir of variable size, an orifice slot is situated in it and allows the spring bellow to be connected to the additional volume by means of any pipe. The distributor is connected to the cylinder by a spherical joint and moves vertically with the excitation cylinder.

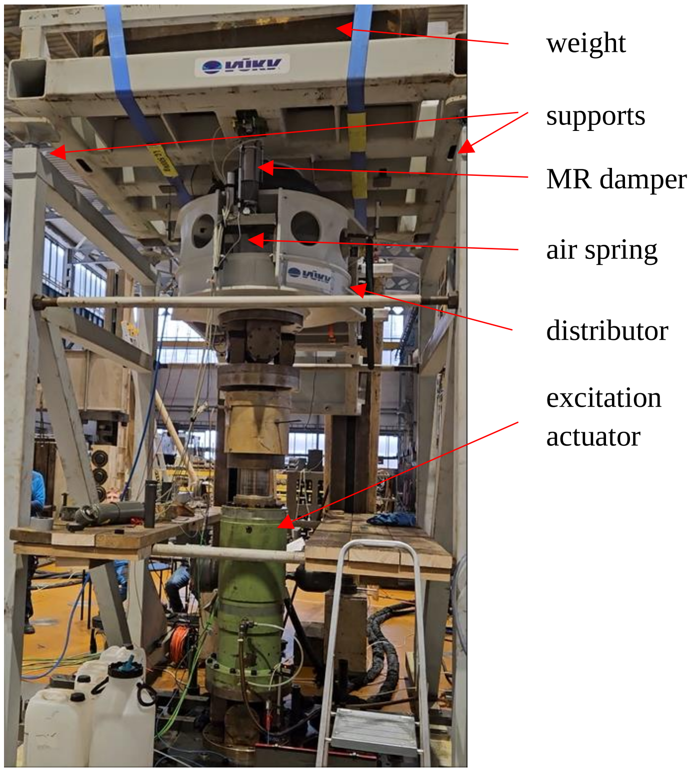

Figure 7: Test stand for the air springs.

Thanks to the excitation electro-hydraulic loading actuator used, the test stand allows measurement over a wide range of excitation frequencies and amplitudes. At the same time, the test stand allows the air spring settings to be changed easily, i.e. the size of the additional air reservoir, the orifice, the pipe length and the pipe diameter can be changed as well as the spring static load can be changed by changing the weight. With these capabilities, the dynamic characteristics of air springs at different settings can be measured on the test bench. Due to its characteristics of being able to faithfully simulate the secondary suspension stage of a real vehicle, it was possible to use this test stand to verify the characteristics of the magnetorheological damper.

For the relevant experiments, the arrangement of the test bench corresponded in its parameters to the vertical secondary suspension stage of the investigated EMU. For excitation of the test stand, the simulated records of vertical bogie motion in place of the vertical secondary damper in the SJKV model were used. As an example of results, a characteristic value of vertical acceleration of the sprung mass is presented for both cases – the simulation and the experiment – and for two different track sections; in the S/A mode, the maximum current was set to 3 A. The acceleration record is filtered by the weighting function Wb, according to EN 12299, a floating RMS with a 5 s window is calculated and its 95th percentile is evaluated. These values are presented in Fig. 8 and demonstrate a relatively good agreement of the simulation and experimental results. A relative (percentage) improvement in this parameter for both considered tracks and for simulation as well as experimental results is also presented in Tab. 1.

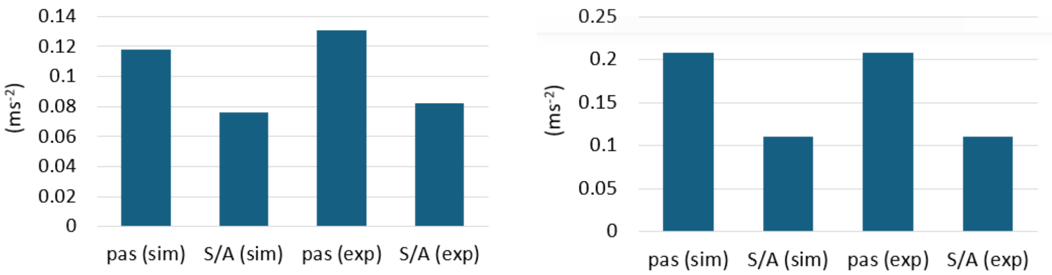

Figure 8: 95th percentile of moving RMS from filtered sprung mass vertical acceleration for passive and semi-active (S/A) modes, for simulation and experimental results and for tracks with better (left) and lower (right) quality.

Improvement	Track 1 (better quality) (160 km/h)	Track 2 (lower quality) (120 km/h)
Simulation	36%	47%
Experiment	37%	47%

Table 1: Improvement in ride comfort reached by means of the semi-active control.

5 Conclusions

The current results of this research show that the semi-actively controlled secondary dampers can improve the ride comfort of the investigated vehicle significantly. The next stage of this research is an experimental verification of contributions of the semi-actively controlled damping system by means of on-track tests which are currently in progress.

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