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PITCH-PLANE MODELS OF MR VEHICLE SUSPENSION FOR EXPERIMENTAL TESTING

SUMMARY

The study concerns 2 DOF and 3 DOF pitch-plane models of a semi-active vehicle suspension comprising magnetorheological (MR) dampers. The 2 DOF pitch-plane model with two independent spring – MR damper suspension sets enables us to study bounce and pitch motions of the suspended body. The 3 DOF pitch-plane model takes into account also a model of driver's seat and for this reason the third spring – MR damper suspension set was introduced. The mechanical structure, data acquisition and control equipment of the 2 DOF and 3 DOF system configurations are reviewed. Finally, selected results of functional testing conducted in the presented setups are provided.

Keywords: *vehicle suspension, pitch-plane model, MR damper, experimental setup*

PŁASKIE MODELE MAGNETOREOLOGICZNEGO ZAWIESZENIA POJAZDU DO BADAŃ EKSPERYMENTALNYCH

W artykule opisano płaskie modele semiaktywnego zawieszenia pojazdu o dwóch (2 DOF) i trzech (3 DOF) stopniach swobody. W modelach tych wykorzystano magnetoreologiczne (MR) tłumiki drgań. Model 2 DOF posiadający dwa niezależne układy zawieszenia: sprężyna – tłumik MR, umożliwia analizę ruchu pionowego i przechyłów wzdłużnych resorowanego nadwozia pojazdu. Model 3 DOF uwzględnia dodatkowo fotel kierowcy wraz z zawieszeniem, które stanowi trzeci układ sprężyna – tłumik MR. Dla obydwu modeli przedstawiono strukturę mechaniczną stanowisk badawczych wraz z aparaturą pomiarowo-sterującą. Zaprezentowano wybrane wyniki badań funkcjonalnych przeprowadzonych na obu stanowiskach.

1. INTRODUCTION

The type of the model selected for the purpose of vibration analysis is largely dependent on the objective of the analysis. The suspended body (sprung mass) of a vehicle may possess flexibility, which increases the amplitude of structural vibration modes, characterised by significantly higher frequencies than road-induced modes [2, 3]. For the ride vibration analysis, therefore, all wheeled road and off-road vehicles can be described by a 7 DOF suspension model, where respective DOFs represent bounce, roll and pitch motions of sprung mass, and bounce motions of the four unsprung masses of wheels assemblies (independent wheel suspensions are considered). At low frequencies the vehicle body, represented by its sprung mass, moves as an integral unit supported on the primary suspension system. Wheels, axles and brake hardware are represented by the unsprung masses in contact with the road surface through the tires. In response to the road roughness, the unsprung masses move as rigid bodies acting on (exciting) the sprung mass. The passengers or goods occupying the sprung mass are directly subjected to its vibrations. Thus the motion of the sprung mass and also unsprung masses is the primary concern of the ground vehicles vibration analysis [1].

Typical vehicle suspensions with passive dampers are characterised by unavoidable compromise between road roughness attenuation and drive stability. That is why active and semi-active vehicle suspensions are used. In semi-active suspensions the conventional springs are retained but passive dampers are replaced by controllable dampers [5]. These suspensions systems use external power supply only to adjust the damping levels and operate the controller and sensors.

The review of simple but credible models that can be useful for fundamental vibration analysis in terms of resonant frequencies and forced vibration response of sprung and unsprung masses reveals, that the vibration response of vehicles to different excitations can be investigated through the analysis of various in-plane models. Because the wheelbase of the majority of ground vehicles is significantly larger than the track width, the roll motions can be considered negligible compared to the magnitudes of vertical and pitch motions. That is why we focused on 2 DOF pitch-plane model of suspension [4, 5] equipped with MR dampers and its extension – 3 DOF pitch-plane model with the driver's seat and its suspension. These models assume negligible contributions due to the wheel-axle-brake assemblies' masses (compared to the vehicle body mass), and tires stiffness, which is 7–10 times higher than that of primary suspension. That means the road input is taken to be the same as the wheels input. Such models are also considered applicable for study of off-road vehicles without primary suspension – then stiffness and damping factors of the 2 DOF model apply to the tires properties.

The 2 DOF model with two independent spring – MR damper suspension sets enables us to study bounce and pitch motions of the suspended body and estimate natural frequencies (assuming negligible contributions due to

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unsprung assembly). For the purpose od 3 DOF pitch-plane model implementation, the third spring – MR damper set that represents the suspension of driver's seat has to be considered. These two models provide the background for the analysis of vibration reduction due to the MR dampers implementation.

2. PITCH-PLANE MODELS

The diagram of 2 DOF pitch-plane model is depicted in Figure 1. The suspended body is simulated by a rigid rectangle-intersection beam of mass m, moment of inertia I, total length L, width a, height b and centre of gravity (c.o.g.) in P_g (Tab. 1). The beam is supported in points P_f and P_r by two identical spring – MR damper sets (hereinafter called suspension-sets), which are subject to bottom displacement excitations similar to these acting upon conventional vehicle suspension. Distances from P_g to P_f and from P_g to P_r are denoted by l_f and l_r. Elasticity factors of the front and rear springs are denoted by k_f and k_r. Similarly, i_f and i_r denote currents in the front (d_f) and rear (d_r) MR dampers' coils. Excitations applied to the bottom of front and rear suspension-sets are denoted by w_f and w_r, displacements of points P_f and P_r by x_f and x_r. This model possesses 2 DOFs: vertical (bounce) displacement x and pitch displacement φ of the beam's c.o.g. [6, 7].

The 3 DOF pitch-plane model (Fig. 1) consists of 2 DOF model described above and, additionally, a mass m_s (modelling the driver and the seat), suspended in P_g on spring – MR damper set. The elasticity factor of the spring is k_s, current in the MR damper's d_s coil is i_s. The displacement of m_s in relation to beam's c.o.g. is x_s.

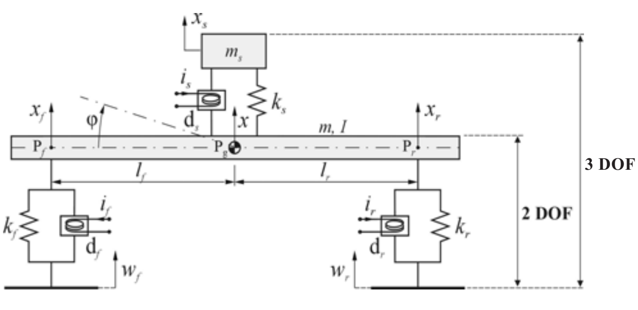

Fig. 1. 2 DOF/3 DOF pitch-plane model

Table 1. 2 DOF/3 DOF* – technical specifications data

Parameter	Designation
Distance (P_g , P_f)	l_f [m]
Distance (P_g , P_r)	l_r [m]
Total length of the beam	L [m]
Width of the beam	a [m]
Height of the beam	b [m]
Mass of the beam	m [kg]
Moment of inertia of the beam	I [kgm^2]
Elasticity factor of the front (rear) spring	$k_f(k_r)$ [N/m]
Mass of driver and seat*	m [kg]
Elasticity factor of the seat spring*	k_s [N/m]

3. EXPERIMENTAL SETUP CONFIGURATIONS

For the purpose of the pitch-plane suspension model analysis, an experimental setup was devised, comprising data acquisition and control equipment connected to PC-based communication and control software.

3.1. Mechanical structure

As our analysis is limited to pitch-plane oscillations, the experimental set-up should conform to the appropriate construction demands, namely transverse rigidity. All the motion components orthogonal to the pitch-plane, or other than pitch and bounce, should be eliminated. This implies introduction of appropriate central guiding elements with adequately small friction forces. All the joints should have high transversal rigidity. Another demand is that the excitations have to be stationary. Limited output of the excitation sources available in the laboratory conditions implies constraints to the total mass and moment of inertia of the sprung system.

The experimental setup of 2 DOF model (Fig. 2) consists of steel beam with rectangular intersection 1 as a load element (vehicle body), two identical suspension-sets: spring 2 – MR damper 3, central stabilizing guiding columns with car elements 4 moving inside them, and two kinematic shakers (6 and 7). Each suspension-set is built as a parallel connection of a vertically mounted MR damper inside and outer screw-cylindrical reflex spring guided onto two thin-wall sleeves 5. The sleeves are guided one inside the other with teflon slide ring between them. Both sleeves possess outer flanges as the spring support. A suspension-set is connected at the top with the beam and at the bottom with the shaker by means of pin joints. Two types of shakers were available in the experiments. One of them is an electro-hydraulic cylinder 6 type PZ 25/50 with maximum force output $2.5 \cdot 10^3$ N and maximum stroke $50 \cdot 10^{-3}$ m, supplied and controlled by F. Heckert testing machine type SHA 130.

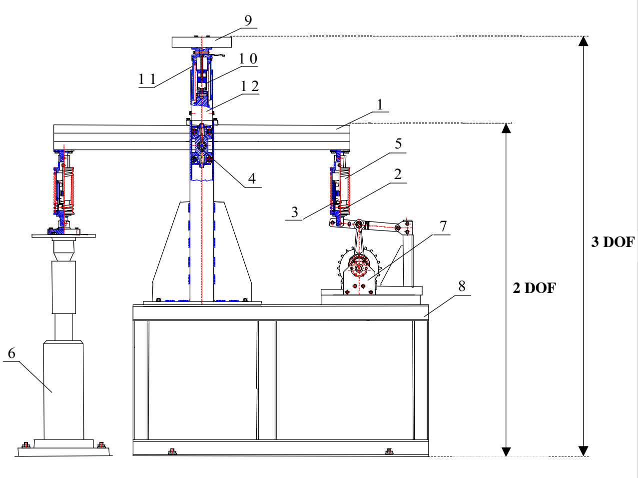

Fig. 2. 2 DOF/3 DOF experimental setup drawing (see text)

It is anchored to the foundation with bolts. The other shaker is a $3.5 \cdot 10^3$ W asynchronous electric motor with controllable circular cam crank mechanism 7. The motor is supplied by the electronic inverter type SELS, enabling smooth control of rotation speed and excitation frequency in the range of (3.5, 10) Hz. The c.o.g. of the beam is guided and vertically stabilized by means of two car elements located symmetrically on both sides of the beam 4. These elements (and c.o.g. of the beam) are mounted on the common axle and move during system operation inside vertical steel guides. This guarantees longitudinal and transversal rigidity of the system. To minimize friction, bearing systems offering backlash elimination were implemented. The guide assemblies and the kinematic crank shaker 7 are all mounted to the rigid cuboid frame (welded section steel) 8.

The experimental setup of 3 DOF model contains all the elements of 2 DOF setup and an additional sprung mass 9 (modelling the driver and the seat) supported in the central part of the beam (Fig. 2). The suspension of the driver and seat masses includes vertically, cylindrically mounted MR damper 10 inside and screw-cylindrical reflex spring 11 outside, guided by two outer metal sleeves 12, moving one inside another. One of the sleeves is rigidly mounted to the seat, the other one to the beam.

The experimental setups are shown in Figure 3.

a)
b)

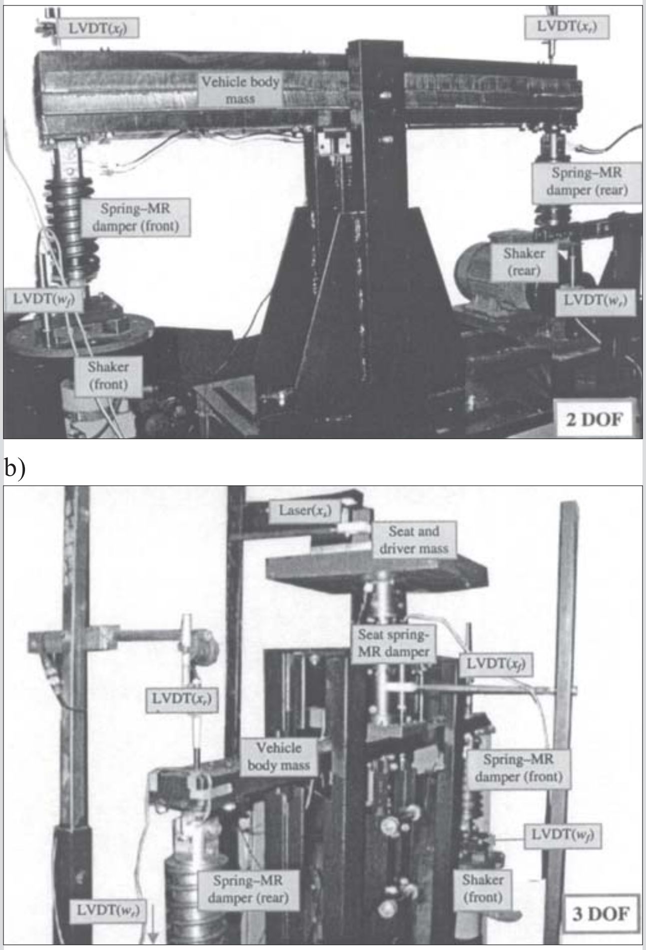

Fig. 3. General view of: 2 DOF experimental setup (a), 3 DOF experimental setup (b)

3.2. Data acquisition and control equipment

The diagram of the data acquisition and control equipment for 2 DOF and 3 DOF system setups is shown in Figure 4 (additional signal labels corresponding to 3 DOF system only are placed in brackets).

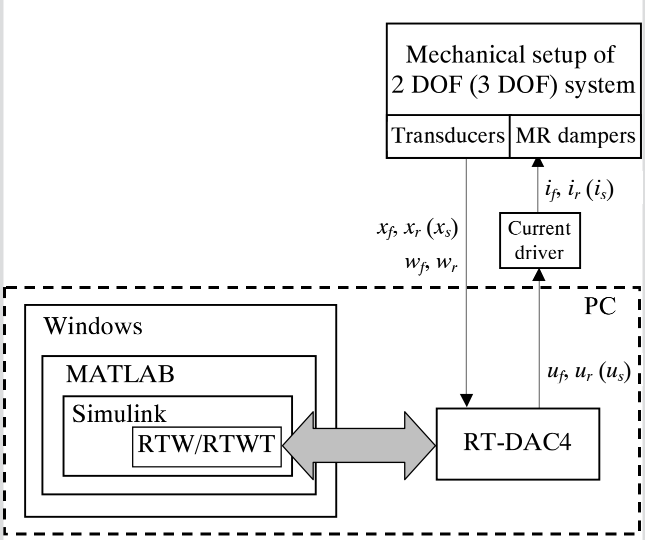

Fig. 4. Diagram of the data acquisition and control equipment

The measurements were conducted by means of four PSz-20 LVDTs (two of them located on the beam – x_f and x_r, the other two on the shakers – w_f and w_r), one laser displacement transducer (in 3 DOF system configuration only for x_s measurement), and a multi I/O board of RT-DAC4 series placed in standard PC. MATLAB/Simulink environment with RTWT (Real-Time Windows Target) extension of RTW (Real-Time Workshop) toolbox running on Windows 2000 operating system is chosen. MR dampers coils' currents i_f, i_r (for 3 DOF system also i_s) calculated in MATLAB/Simulink were output by means of RTWT/RTW, the RT-DAC4 board and the current driver.

4. FUNCTIONAL TESTING

On the basis of x_f and x_r measurement, vertical displacement x (bounce) and pitch displacement φ of the vehicle body's c.o.g. were obtained. For the purpose of 2 DOF and 3 DOF model behaviour analysis, frequency acceleration transmissibilities were determined [6].

Figure 5 presents bounce root-mean-square (RMS) acceleration transmissibility of 2 DOF system for front excitations and current levels in the MR dampers' coils set as 0.0 A (OS1) and 0.2 A (OS2). The resonance frequencies of the open loop 2 DOF system OS1 are: f_x = 3.1 Hz (bounce), f_φ = 5.0 Hz (pitch). Besides, for one-side excitation (consisting of both: bounce and pitch), the acceleration transmissibility (see Fig. 5) has its global maximum at 4.0 Hz, but also local maximum is present (at 3.1 Hz). The shift of the global maximum is due to the internal coupling between bounce and pitch DOFs occurring for the system equipped with nonlinear MR dampers. For the similar system, equipped with viscous dampers, the two DOFs (bounce, pitch) are independent: coupling elements in the state-space matrix of such (linear) system are all equal to zero.

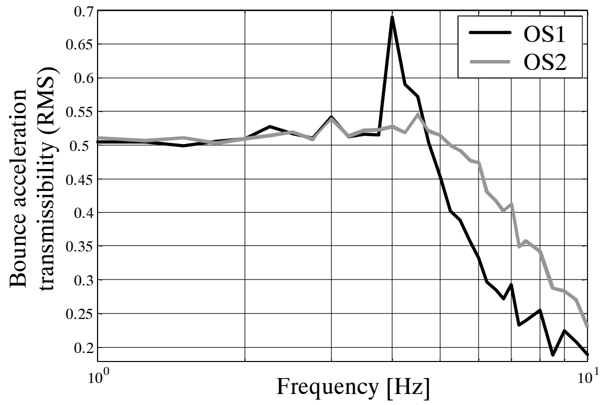

Fig. 5. Bounce acceleration transmissibility of 2 DOF system

Figure 6 presents seat (driver) bounce RMS acceleration transmissibility in 3 DOF system for d_s current set to 0.0 A (OS1). The resonance frequency of bounce is f_s = 1.8 Hz.

Relative locations of all the resonance frequencies of 3 DOF system ($f_s << f_x$) guarantee the best attenuation of the driver vibrations of the open loop system with no MR dampers current, in the wide range of frequencies.

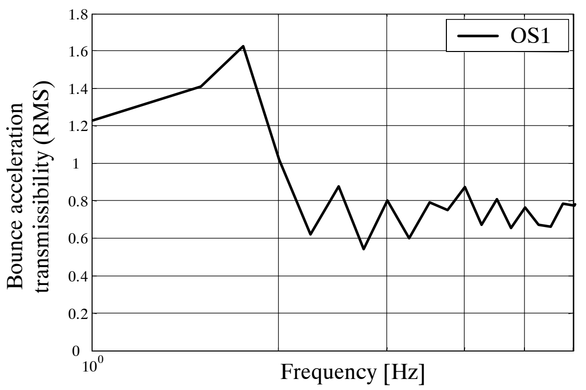

Fig. 6. Bounce acceleration transmissibility of seat (driver) in 3 DOF system

5. SUMMARY

The experimental setups for 2 DOF and 3 DOF pitch-plane models testing satisfy the assumed requirements, although the first versions suffered significant deficiencies concerning longitudinal and transversal rigidity. Following the redesign as regards to central guiding elements, this problem was eliminated. Compared to the preliminary version of the setups, the mass *m* had to be increased (for the available set of MR dampers) to reduce the system OS1 internal damping below the critical value. On the other side, mass enlargement was limited by the efficiency of available shakers.

Results of the analysis conducted for the open loop system prove the constructed setups can properly portray plane behaviour of the rigid vehicles like: trucks, tractors, trailers, mobile machines, tanks, etc., most of them possessing additional suspension of driver's seat.

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