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Identification and comparison of two nonlinear extended phenomenological models for an automotive ElectroRheological (ER) damper

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Abstract: This paper presents a comparative study between different nonlinear damper models, developed for a semi-active Electro-Rheological damper. The first model is developed by extending the hysteresis model to obtain the dynamic characteristic of the damper. To improve the accuracy of capturing the damper's behavior, a nonlinear Bouc-Wen model is developed by using parameter-varying functions of control input and damper state. One of the key points in the proposed models is to describe the dynamics of the controlled portion of the damper force as a first-order system. Hence, the proposed models are adequate for the design and synthesis of Linear Parameter Varying observers and controllers. The model parameters are identified using a simple nonlinear identification procedure. The experimental validation tests are performed on a real 1/5-scaled vehicle testbed of GIPSA-lab, namely INOVE (see www.gipsa-lab.fr/projet/inove). The results assess the ability and the accuracy of the proposed models to characterize the behaviors of ER dampers.

Keywords: Electro-rheological dampers, phenomenological model, Semi-active suspension, vehicle dynamics,

1. INTRODUCTION

Nowadays, vehicle dynamics are of paramount importance in the automotive industry. Indeed, over the last decade, automotive engineering has witnessed rapid growth in semi-active and active systems. There has been a lot of research, development, and implementation of such semi-active and active subsystems such as braking, steering or suspension actuators (see Savaresi et al. (2010)), Tanelli et al. (2014), Do et al. (2012), Yamamoto (2017), Priyandoko et al. (2009), Yoshimura et al. (2001)), Bremner (2005)). In particular, the automotive suspension systems have also received a lot of consideration from industry and academia due to their vital role in reducing the effects of the road disturbances on the on-board passengers (comfort) and in maintaining tire-road contact (safety) (see Gillespie (1992), Savaresi et al. (2010)).

The configuration of the suspension system includes three main components: 1) a coil spring; 2) a damper; 3) the mechanical elements to link the sprung mass (chassis) and unsprung mass (wheel). According to the damper

characteristics, the damper is broadly classified into a) passive b) semi-active c) active. Amongst these categories, the semi-active damper is a potential candidate in the automotive market due to the advantages of the semi-active damper compared to active and passive ones such as negligible power demand, safety characteristics, improvement of the vehicle performance and low cost and weight (see in Savaresi et al. (2010)). There are three major semi-active damper technologies available today: 1) Electro-Hydraulic (EH) dampers, i.e. hydraulic devices usually equipped with solenoid valves Aubouet (2010); 2) Magneto-Rheological (MR) Metered et al. (2010); de J Lozoya-Santos et al. (2012); Witters and Swevers (2010); Ashfak et al. (2013) and 3) Electro-Rheological (ER) shock absorbers, both damper types being filled with a Rheological fluid that varies its apparent viscosity under the action of a magnetic (see Carlson et al. (1996)) or an electric field (see Acerbi and Mingione (2002)), respectively. Many research works have therefore been devoted to the modeling, estimation, and control designs of the semi-active suspension system (see Choi et al. (2001), Poussot-Vassal et al. (2012), Dugard et al. (2012) and references therein). This paper is concerned with the modelling and identification of ER damper that are mounted in a 1/5-scaled vehicle.

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In order to capture the main characteristics of the ER damper, many models have been derived using several methodologies with different complexity and accuracy. The damper models may be classified in terms of static and dynamic characteristics:

- Static models include Bingham model with Coulomb friction (see Stanway et al. (1987)), hysteresis based model (see Guo et al. (2006), de J Lozoya-Santos et al. (2012)).
- Dynamic models considers the Bouc-Wen model in (Wen (1976), Ahmadian et al. (2004) and Spencer Jr et al. (1997)).
- Based on several approximators such as neural network (Chang and Roschke (1998), Chang and Zhou (2002)), fuzzy (Schurter and Roschke (2000)), polynomial (Du et al. (2005)) and among others (Savaresi et al. (2005)), proposed black box model can be also divided into static or dynamic groups, depending on the typical model.

It must be remarked that the parametric modeling approach can lead to reasonably simplified functions that describe the delivered ER damper force quite well. The Bingham model is representative, however, it only characterizes the static nonlinear behavior of semi-active suspensions. Therefore, one of our objectives here is to extend the models by taking into account both the nonlinear and dynamic behaviors of the ER dampers. In addition, the nonlinear dynamic model of ER damper based on the Bouc-Wen model is also presented. We aim at comparing the two models in capturing behaviors of the ER dampers.

It is worth noting that the proposed model is a key tool to develop Linear and NonLinear Parameter Varying approaches for control and observation of semi-active suspension system. The contribution of this paper can be summarized as follows:

- We propose two extended dynamical Bingham and Bouc-Wen for ER damper modelling.
- Using experimental data, we compare the extended dynamical Bingham model with the well-known Bouc-Wen model.

The rest of this paper is as follows. Section 2 presents the INOVE testbench which is used to validate the approaches. Section 3 provides two models of ER damper. In section 4 presents the identification procedures. Section 5 discusses the validation results and finally, section 6 give some concluding remarks.

2. INOVE TESTBENCH

For validation and identification purposes, as well as for many experimental tests to assess the proposed algorithms, a real $\frac{1}{5}$ -sized vehicle test-rig, INOVE testbench available in GIPSA-lab, is used in this work. The scaled car is equipped with a semi-active suspension system involving four ER dampers which have a force range of ± 50 N and have been designed by *Fludicon* ©. These ER dampers are adjusted using a controlled voltage inside the range of $[0, 5000]$ V, generated by amplifier modules. The control input for each module is a PWM signal at 25 kHz. In terms of capturing the vehicle's behavior, this testbed is equipped with a wide variety of sensors. In order to gener-

ate road profiles, four linear servomotors placed under the four wheels of the vehicle mimic the desired road profiles. A photo of the scaled car INOVE and of the front-left ER damper of the experimental platform is given in Figure 1.

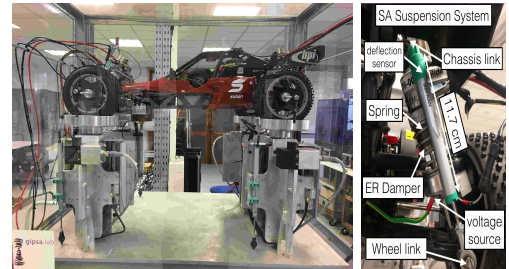


Fig. 1. The experimental testbed INOVE at GIPSA-lab (see www.gipsa-lab.fr/projet/inove)

Finally, this unique 1/5 scale automotive system has been developed to study the vertical dynamic of the ground car, therefore, neither the steering nor braking systems are mounted in the car.

3. ER DAMPER MODELING

A schematic diagram of ER shock absorbers is given in Figure 2: basically, it has two chambers (upper and lower), divided by a piston that provides the force, and electrodes that induce an electric field over an ER fluid. These chambers are fully filled with the Rheological fluid. As the piston moves, the ER fluid flows from one chamber to the other through the annular duct between inner and outer cylinders. In this section, two dynamic models of the ER damper are presented.

3.1 Extended Bingham model

Starting from this schematic configuration and knowing its operating principle, a quasi-static model can be derived on the basis of the Bingham Rheological laws of ER fluid Bird et al. (1983). From this point, a control-oriented dynamic model for the ER damper is then proposed in our previous work Morato et al. (2020). The static Bingham model of the ER damper is recalled as follows:

$$\begin{cases} F_d = k_0 x_p + c_0 \dot{x}_p + F_{er} \\ F_{er} = \sigma u^j \text{sign}(\dot{x}_p) \end{cases} \quad (1)$$

where F_d is the damper force, $x_p(t)$ is the piston displacement (suspension deflection), $\dot{x}_p(t)$ is the piston velocity, u is duty cycle of PWM channel. k_0 , c_0 , σ , j are the model parameters.

In order to improve the accuracy and to account for the dynamically-varying behavior of the ER fluid, the model (1) is extended with a first-order dynamical equation, as follows:

$$\begin{cases} F_d + \tau(u) \frac{dF_d}{dt} = k_0(u) x_p + c_0(u) \dot{x}_p + F_{er} \\ F_{er} = \sigma(u) u^j \text{sign}(\dot{x}_p) \end{cases} \quad (2)$$

where the parameters $k_0(u)$, $c_0(u)$, $\sigma(u)$, and the varying time constant $\tau(u)$ are polynomial functions of the control input u .

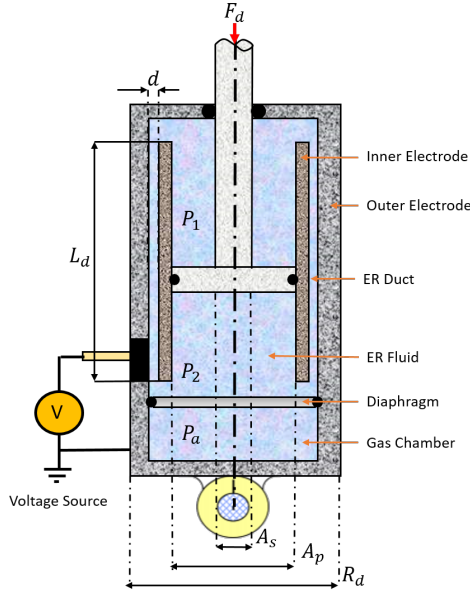


Fig. 2. Schematic Representation of an ER Damper, adapted from Choi and Han (2003)

3.2 Extended Bouc-Wen model

In this subsection, the ER damper is modelled by using the Bouc-Wen model. It is a multiple parametric mechanical model that is used for representing high nonlinearity and hysteretic systems. The Bouc-wen model-based ER damper is as the following:

$$\begin{cases} F_d = k_0 x_p + c_0 \dot{x}_p + \alpha \cdot z \\ \frac{dz}{dt} = A \cdot \dot{x}_p - \beta |\dot{x}_p| |z|^{(n-1)} - \gamma \cdot \dot{x}_p \cdot |z|^n \end{cases} \quad (3)$$

where, F_d is the damper force, z is the hysteretic deformation of the model; k_0 , c_0 , A , n , α , β , γ are Bouc-Wen model parameters

As previously mentioned, the model parameters differ depending on the control input u , which is as follows:

$$\begin{cases} F_d = k_0(u) x_p + c_0(u) \dot{x}_p + \alpha(u) \cdot z \\ \frac{dz}{dt} = A(u) \cdot \dot{x}_p - \beta(u) |\dot{x}_p| |z|^{(n-1)} - \gamma(u) \cdot \dot{x}_p \cdot |z|^n \end{cases} \quad (4)$$

where $k_0(u)$, $c_0(u)$, $\alpha(u)$, $A(u)$, $\beta(u)$, $\gamma(u)$ are the polynomials of the control input u .

Straightforward identification procedures are presented in the next section to get the parameter values of both proposed models.

4. IDENTIFICATION PROCEDURE OF THE MODEL PARAMETERS

In the above models, the nonlinear functions were used to describe the ER damper. Therefore, the identification procedures using nonlinear non-linear least square methods (the Trust-Region Reflective Algorithm is primarily used) are introduced to identify the model parameters. Then validation results are shown to assess the accuracy of the proposed models. This has been carried out using

the data from the front-left ER damper of the INOVE testbench, as explained in Section 2.

4.1 Data collection

Different experimental tests were carried out to estimate the parameters characterizing the proposed models (2) and (4). These tests consist of changing the magnitude and the frequency of the external inputs (control input and road profile) of the ER shock absorber (Velocity, Electric Field) in order to identify the response of the system. To identify the values, it is necessary to check the damper output (Force) with respect to the inputs (Velocity, Electric Field, etc).

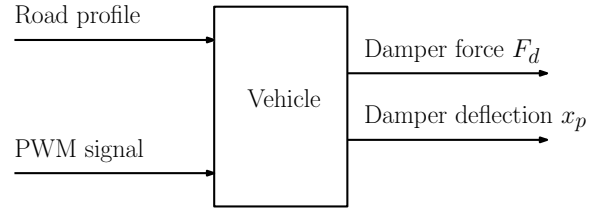


Fig. 3. Block diagram for the data collection in the identification procedures.

In order to collect the experimental data for the identification steps, the experimental scenarios (see Figure 3) are designed as follows:

- The PWM signals vary inside the set $\{0, 5, 10, 15, 20, 25, 30, 35, 40, 45, 50\}$ %.
- The road profile is the sine waves with different frequency.

It is worth noting that we run a lot of experiments with different duty cycle of PWM signal. Here we show the data of a test with $u = 0.1$.

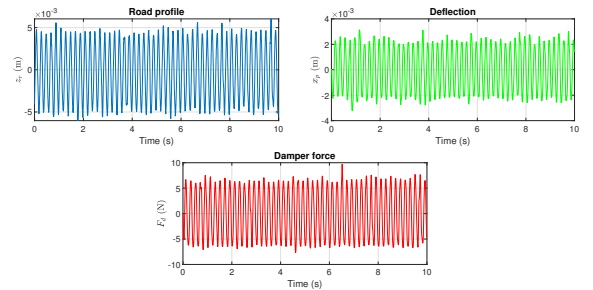


Fig. 4. Collection data $u = 0.1$: (top-left) Road profile, (top-right) displacement, (bottom) damper force.

The collected data of damper force, displacement and velocity are shown in Figure 5

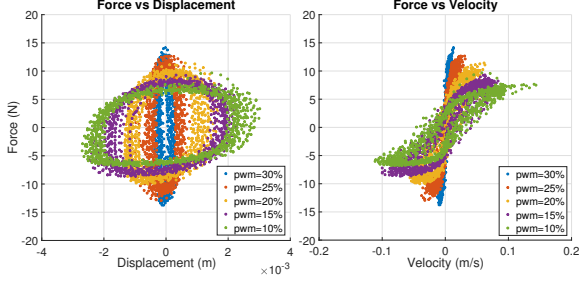


Fig. 5. Damper Force vs. Displacement and Damper force vs Velocity diagrams of the real data for $u = 0.1 - 0.3$.

Since the considered models are nonlinear phenomenological models, the identification procedure is carried out following the physical law of such models. Therefore a multi-step methodology is proposed instead of a global method. If this may lead to a non-global optimum (in terms of identification optimization cost), this is a guarantee to get realistic physical parameters of the phenomenological models.

4.2 Identification of the extended Bingham model

In order to identify the parameters in the extended Bingham model (2). The conducted identification procedure is explained in the following

Step 1: In the first step, The experimental tests consist of transmitting a null control signal ($u = 0$) to the damper. In this case, the damper force is generated from a passive uncontrolled damper. In order to find $k_0(0)$, $c_0(0)$ and $\tau(0)$ for $u = 0$, the trust region reflective algorithm is used on the data.

Step 2: In order to identify the functions ($k_0(u)$, $c_0(u)$, $\sigma(u)$, $\tau(u)$), the experimental tests considered the velocity input within the range $[-0.15, 0.15] \frac{m}{s}$ and the fixed PWM (i.e. u_{eq}) signals inside the set $\{5, 10, 15, 20, 25, 30, 35, 40, 45, 50\}$ %. Then, the fixed parameters $k_0(u_{eq})$, $c_0(u_{eq})$, $\sigma(u_{eq})$, $\tau(u_{eq})$ and j can be identified the Nonlinear Least square method. Notice that the stiffness value k_0 has a lower bound $k_0(0)$ found in Step 1.

Step 3: Based on the values given in steps 1 and 2, the relationship between control input u and k_0 , c_0 , σ , τ can be approximated by the following functions. The results are shown as follows:

$$k_0(u) = 263.1178 - 3958.09 \cdot u + 98674.77 \cdot u^2 - 661285.4 \cdot u^3 + 1624913 \cdot u^4 - 1338958 \cdot u^5 \quad (5)$$

$$c_0(u) = 31.26 + 7447.4 \cdot u - 160814.7 \cdot u^2 + 1027574 \cdot u^3 - 2276490 \cdot u^4 + 1656798 \cdot u^5 \quad (6)$$

$$\sigma(u) = 2.05 - 29.1 \cdot u + 388.16 \cdot u^2 - 1730.37 \cdot u^3 + 3057.79 \cdot u^4 - 1810.11 \cdot u^5 \quad (7)$$

$$j = 0.0012 \quad (8)$$

$$\tau(u) = 0.00067 + (0.0069)/(1 + (u/0.25)^{133.6}) \quad (9)$$

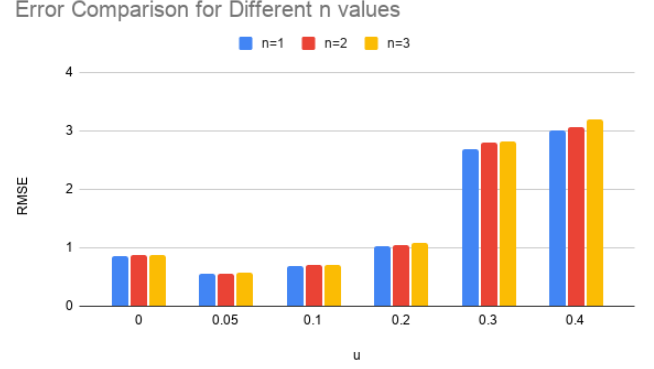


Fig. 6. Sum of Squared Error versus u (for $n = 1, 2, 3$)

4.3 Identification of the extended Bouc-Wen model

Notice that the $k_0(u)$ and $c_0(u)$ functions are similar ones in the Bingham model. Therefore, in this subsection, we propose the identification procedure to identify the parameters $\alpha(u)$, $\beta(u)$, $\gamma(u)$, and $A(u)$. The following procedure is carried out.

Step 1: The experimental data for the tests with the signals inside the set $\{0, 5, 10, 15, 20, 25, 30, 35, 40, 45, 50\}$ % are used. The value of n in (4) is chosen from the set $\{1, 2, 3\}$. We compare the Root-Mean-Square-Error (RMSE) of the models with different values of n to finalise the n value ($n = 1$) (shown in Figure 6). To find $\alpha(u_{eq})$, $\beta(u_{eq})$, $\gamma(u_{eq})$, and $A(u_{eq})$ for $u_{eq} \in \{0, 5, 10, 15, 20, 25, 30, 35, 40, 45, 50\}$, a Nonlinear least square approach is applied. The RMSE is defined as follows:

$$RMSE = \sqrt{\frac{\sum_{i=1}^n (F_{measured_i} - F_{model_i})^2}{n}} \quad (10)$$

The results are shown in Table 4.3.

Step 2: Based on the values shown in Table 4.3, the functions $\alpha(u)$, $\beta(u)$, $\gamma(u)$, and $A(u)$ can be approximated by the following functions. The results are shown as follows:

$$n = 1 \quad (11)$$

$$A = 95.30 + (23.76 - 95.30)/(1 + (u/0.262)^{34.627}) \quad (12)$$

$$\alpha = -0.47 + 6.54 \cdot u - 379.47 \cdot u^2 + 1835.07 \cdot u^3 - 42.65 \cdot u^4 - 3379.15 \cdot u^5 \quad (13)$$

$$\beta = 780.5 + 4.07 \cdot 10^4 \cdot u - 8.1 \cdot 10^5 \cdot u^2 + 5.3 \cdot 10^6 \cdot u^3 - 1.3 \cdot 10^7 \cdot u^4 + 1.06 \cdot 10^7 \cdot u^5 \quad (14)$$

$$\gamma = -1.03 \cdot 10^5 + (3.48 + 1.03 \cdot 10^5)/(1 + (u/60.26)^{2.62})^{827.06} \quad (15)$$

5. MODEL VALIDATION

Now, the final validation results of the proposed Electro-Rheological damper models are presented, considering the adjusted identified parameters for the *INOVE Soben-Car* mechatronic test-rig.

u	A	α	β	γ
0	26.512	-25.141	953.61	0.38287
0.05	24.049	-22.575	849.38	0.4082
0.1	25.114	-23.731	1075.5	0.38314
0.15	12.683	-36.896	930.96	1.3578
0.2	30.484	-30.154	2129.4	0.2699
0.25	34.805	-30.651	513.79	-87.479
0.3	94.635	-102.34	4721	-91.507
0.35	98.607	-106.01	2944.3	-100.54
0.4	105.59	-112.48	3412.2	-106.8
0.45	83.131	-91.652	400.08	-281.18
0.5	93.837	-101.27	1598.9	-281.21

Table 1. Parameter values obtained from Step 1.

5.1 Test 1

In case 1, two models are validated with the data collected for the identification in section 4. The new adjusted Force *vs.* Deflection and Force *vs.* Deflection-Velocity diagrams are given, respectively, in Figures 7 and 8, which compare the proposed damper models and the real data measured from available sensors on the platform. In the Figure, it is clear that there exists an overall good agreement between model and data.

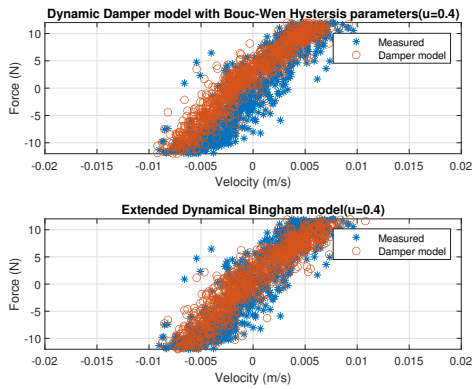


Fig. 7. Model validation: force versus deflection

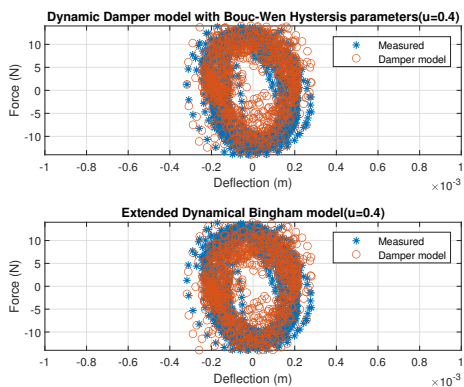


Fig. 8. Model validation: force versus velocity

Besides, the RMSE of two models comparing with real data of the experimental tests ($u = 0 - 0.5$) are shown in Figure 9.

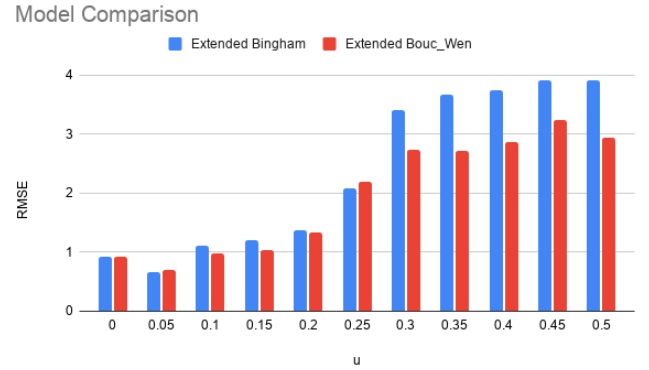


Fig. 9. RMSE comparison between Bouc-Wen model and extended Bingham model

5.2 Test 2

One considers the road profile of a vehicle running at 120 km/h in a straight line on a dry road with a sequence of 10 mm sinusoidal bumps and PWM signal $u = 0.2$. The results comparing the model-based computed force and the real (measured) force are given by Figure 10. Clearly, the model is well adjusted.

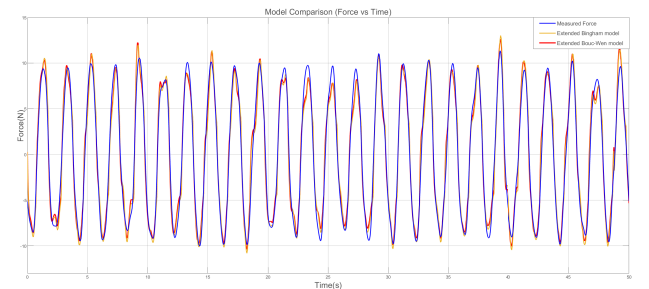


Fig. 10. Model validation: force versus time

6. CONCLUSION

This paper presented two models for an Electro-Rheological damper, in the case of (semi-active) automotive applications. The first model is derived from Bingham law, while the other is developed from Bouc-Wen model in order to catch the dynamical property of the ER damper. Then, the identification procedures using nonlinear least square methods proposed to identify the model parameters. The proposed models were validated with various tests on the INOVE experimental platform (small scaled car). The overall results assess the ability and the accuracy of the proposed models to represent the real damping force of the ER semi-active damper.

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