2022 19th International Conference on Electrical Engineering, Computing Science and Automatic Control (CCE). Mexico City, Mexico. November 9-11, 2022

Semi-Active Road-Vehicle Dynamics by implementing Neuro-Fuzzy Controlled damper

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***Abstract*— This paper investigates the semi-active road-vehicle behavior of full car model using neuro-fuzzy controlled magnetorheological dampers. The neuro-fuzzy is a neural network approach to create and optimize the fuzzy rules. The implemented neuro-fuzzy was based on the response of a PID controller. The effect of the neuro-fuzzy controller was studied on the different dynamics of the full car suspension model compared to the passive and PID controlled models that were used to generate the fuzzy rules. The control methods were applied on MATLAB simulation software. The neuro-fuzzy controller yielded promising results and enhanced the response of the system in all aspects, which was unobservable in the quarter car model. This shows the great potential in utilizing neuro-fuzzy as a robust controller.**

Keywords—suspension dynamics, magnetorheological damper, PID control, neuro-fuzzy control.

I. INTRODUCTION

Suspension systems have major importance in carrying the vehicle body and transmitting all vertical forces between body and road [1]. This statement shows the importance of studying suspension systems in terms of stability, optimization of parameters and control to achieve a greater ride experience. Moreover, road disturbances cause major shock vibrations and the suspension system task is to absorb most of these shocks. The two main functions of the suspension system are to ensure driver comfort and safety. The most common form used to model and analyze the suspension system dynamics is the spring damper method which divides the system into a spring element and a damping element.

In order to achieve the optimum ride with the best aspects of comfort and safety, several researches were conducted to fully analyze the suspension dynamics and many attempted different control systems implementations in order to enhance the driver's comfort and safety. Mohite et al. investigated suspension dynamics using linear mathematical model of the suspension system components proving the forces induced by every and single component [2]. Krishna studied the suspension dynamics using the 7-DOF full car model while applying fuzzy control to improve its response when applied to different road disturbances [3]. Also, Paksoy implemented a fuzzy control system but with adding the MR damper control for the semi-active suspension systems [4]. Saboo, discussed different types of road disturbances for completing the analysis of the road-vehicle interaction [5]. Yang established a fuzzy adaptive PID controller on quarter vehicle mathematical model based on the nonlinear model of semi-active suspension system [6]. Ahmed used a quarter car model with two degrees of freedom attempted to improve the performance of a car's active suspension system and control the vibrations that occurred in the car using LQR and fuzzy-PID controllers [7].

Vehicle suspension is usually divided into three categories passive, semi-active, active suspension systems. Passive suspension is known for several advantages, such as simple mechanical structure and good reliability, However, its performance is limited. The difference between the semi-active and active suspension systems is the main reason of change between them in terms of energy dissipation. The semi-active suspension mode of operation changes the system parameters by applying control techniques that alters the damping coefficient, this is accomplished by utilizing a device called a variable damper. Whilst active suspension provides extra force and is used for settling the roll, pitch motion and vibrations of the car body. This extra force increases the operation range of the suspension but it is extremely expensive and dissipates a lot of energy. The two main reasons why we used semi-active suspension instead of active and passive is that in case of failure of the actuating components, the passive suspension components are available to protect the driver and also it is cost effective compared to active suspension in real life applications.

As for the actuation of the semi-active suspension as discussed above a variable damper is used. The specific type of variable damper used is the MR damper (magnetorheological damper). This damper consists of an electromagnetic coil in the piston head. The choice of the material is tremendously important in this kind of damper as magnetic steel is used for the piston while non-magnetic stainless steel is used for the rod to avoid the flux leakage through the coil. The MR fluid is a type of oil that contains magnetic (mostly iron) micron sized particles. When the magnetic field is absent, the fluid is in a free-flowing state with low viscosity and thus low damping force. Applying current to the damper changes the fluid into a semi-solid state.

In this study a neuro-fuzzy controller is implemented based on PID-fuzzy rules for a 7-DOF full car suspension model to improve the sprung mass acceleration and pitch in terms of suspension dynamics. The choice of implementing the neuro-fuzzy controller on a 7-DOF full car model is due to the absence of the pitch and rolls degrees of freedom in the linear quarter car model. Improving the pitch dynamics offers a trade-off in the sprung mass acceleration as a higher overshoot occurs relative to the quarter car model's response. Where, implementing a neuro-fuzzy controller on a quarter car model takes into consideration only the sprung mass displacement and acceleration, which is considered insufficient for the full analysis of the neuro-fuzzy controller effect on the ride experience.

II. QUARTER CAR MODEL

Fig 1.1 shows a semi-active suspension system model of a quarter car.

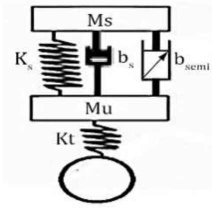

Figure 1 :Quarter car semi active suspension

Ks is the spring constant, bs is the damping coefficient which is considered constant at this case because its linear behavior. Kt is the tire stiffness. Furthermore, Zr represents road disturbance and $[Zs\ Zu]$ are the displacements of the sprung mass and unsprung mass. Road disturbances will be induced to analyze the system dynamics and optimize the controller response, as road disturbances result in undesired suspension dynamics. In order to control the dynamics of the quarter car suspension stated above we need first to formulate the equations of motion. Another damper is plugged between the road and the unsprung mass although it is mostly ignored due to its insignificance compared to the damping of the sprung mass. The dynamics equations stated below describe the passive model whilst adding the variable damper. The variable damper is added to the equations with the term F. Therefore, the equations of motion for the two degrees of freedom quarter car are:

$$Ms\ddot{Zs} = Ks(Zu - Zs) + bs(\dot{Zu} - \dot{Zs}) + F \quad (1)$$

$$Mu\ddot{Zu} = Kt(Zr - Zu) + bt(\dot{Zr} - \dot{Zu}) - Ks(Zu - Zs) - bs(\dot{Zu} - \dot{Zs}) - F \quad (2)$$

The symbolic values are replaced by numerical values. The numerical values are shown in Table 1.

Table 1: Quarter car parameters

Parameter	***Value***	***Unit***
Ms	400	Kg
Mu	48	kg
Ks	20000	N/m
Kt	190000	N/m
bs	1500	Ns/m
bt	10	Ns/m

The damping of the unsprung mass in the quarter car model will be different than the full car model due to the difference in weight.

III. FULL CAR MODEL

Most of the analysis discussed in the literature were made on quarter car models due to their simplistic nature and ease of modelling. Also, several control techniques are more difficult to implement on the full car model due to complexity of the mechanical structure and suspension behavior. The motions applicable in a full car suspension model are the sprung mass bounce, sprung mass pitch, unsprung mass bounce (wheel 1, 2, 3, 4).

Fig 2 represents the Full car model taken into consideration that each wheel has an effect of the spring and the damper of the other wheel and linearity of the suspension components.

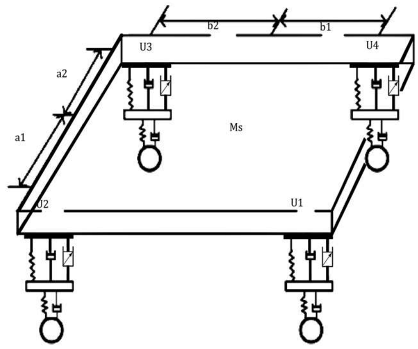

Figure 2: Full car Semi active suspension model

The equations of motion describing the 7 degrees of freedom are:

$$Ms\ddot{Xs} = -Ks1(Xs1 - Xu1) - Ks2(Xs2 - Xu2) - Ks3(Xs3 - Xu3) - Ks4(Xs4 - Xu4) - c1(\dot{Xs}1 - \dot{Xu}1) - c2(\dot{Xs}2 - \dot{Xu}2) - c3(\dot{Xs}3 - \dot{Xu}3) - c4(\dot{Xs}4 - \dot{Xu}4) \quad (3)$$

$$I_x\ddot{\emptyset} = -b1Ks1(Xs1 - Xu1) + b2Ks2(Xs2 - Xu2) + b2Ks3(Xs3 - Xu3) - b1Ks4(Xs4 - Xu4) - b1c1(\dot{Xs}1 - \dot{Xu}1) + b2c2(\dot{Xs}2 - \dot{Xu}2) + b2c3(\dot{Xs}3 - \dot{Xu}3) - b1c4(\dot{Xs}4 - \dot{Xu}4) \quad (4)$$

$$I_y\ddot{\theta} = a1Ks1(Xs1 - Xu1) + a1Ks2(Xs2 - Xu2) - a2Ks3(Xs3 - Xu3) - a2Ks4(Xs4 - Xu4) + a1c1(\dot{Xs}1 - \dot{Xu}1) + a1c2(\dot{Xs}2 - \dot{Xu}2) - a2c3(\dot{Xs}3 - \dot{Xu}3) - a2c4(\dot{Xs}4 - \dot{Xu}4) \quad (5)$$

$$Mu1\ddot{Xu}1 = -ct1(\dot{Xu}1 - \dot{Z}1) - Kt1(Xu1 - Z1) + Ks1(Xs1 - Xu1) + c1(\dot{Xs}1 - \dot{Xu}1) \quad (6)$$

$$Mu2\ddot{Xu}2 = -ct2(\dot{Xu}2 - \dot{Z}2) - Kt2(Xu2 - Z2) + Ks2(Xs2 - Xu2) + c2(\dot{Xs}2 - \dot{Xu}2) \quad (7)$$

$$Mu3\ddot{Xu}3 = -ct3(\dot{Xu}3 - \dot{Z}3) - Kt3(Xu3 - Z3) + Ks3(Xs3 - Xu3) + c3(\dot{Xs}3 - \dot{Xu}3) \quad (8)$$

$$Mu4\ddot{Xu}4 = -ct4(\dot{Xu}4 - \dot{Z}4) - Kt4(Xu4 - Z4) + Ks4(Xs4 - Xu4) + c4(\dot{Xs}4 - \dot{Xu}4) \quad (9)$$

$$Xs1 = Xs - a1\theta + b1\emptyset \quad (10)$$

$$Xs2 = Xs - a1\theta - b2\emptyset \quad (11)$$

$$Xs3 = Xs + a2\theta - b2\emptyset \quad (12)$$

$$Xs4 = Xs + a2\theta + b2\emptyset \quad (13)$$

Ms, Mu are the sprung mass and unsprung mass respectively. $Ks1$, $Ks2$, $Ks3$, $Ks4$ represent stiffness of the front and rear springs of the suspension system. $c1$, $c2$, $c3$, $c4$, $ct1$, $ct2$, $ct3$, $ct4$ model the damping coefficients which are considered constant in this case because of the linear behavior. Ix and 1y are the moments of inertia of the car body around two axes. Kt models the tire stiffness. Furthermore, $Z1$, $Z2$, $Z3$, $Z4$ represent road disturbances and $[Xs\ Xu]$ are the displacements of the sprung mass and unsprung mass.

The numerical values of the coefficients are shown in Table 2. The same car is used for both the quarter car and full car models. The damping between the road disturbances are much higher in this model due to the difference in weight as stated earlier as the full car's model weight is approximately equal four times the quarter car model's weight.

Table 2: Full car parameters

Parameter	***Value***	***Unit***
Ms	1600	Kg
I_x	530	Kg.m^2
I_y	2500	Kg.m^2
$Mu1$	48	Kg
$Mu2$	48	Kg
$Mu3$	74	Kg
$Mu4$	74	Kg
$Ks1$	20000	N/m
$Ks2$	20000	N/m
$Ks3$	20000	N/m
$Ks4$	20000	N/m
$c1$	1500	Ns/m
$c2$	1500	Ns/m
$c3$	1500	Ns/m
$c4$	1500	Ns/m
$Kt1$	190000	N/m
$Kt2$	190000	N/m
$Kt3$	190000	N/m
$Kt4$	190000	N/m
$ct1$	700	Ns/m
$ct2$	700	Ns/m
$ct3$	700	Ns/m
$ct4$	700	Ns/m
$a1$	1.1	m
$a2$	1.4	m
$b1$	0.7	m
$b2$	0.8	m

IV. PID CONTROLLER

Divekar stated that "A PID controller calculates an error value as the difference between a desired set point and a measured variable [8]. The controller makes an effort to minimize this error by adjusting the process control inputs". The name of the controller comes from the nature of the equations defining the controller and the nature of every gain the controller has. The PID controller possesses three constant values which are considered gains: K_p, K_i , K_d . These 3 parameters control the process inputs using the following equation.

$$U(t) = K_p e(t) + K_i \int e(t)dt + K_d \frac{de}{dt} \tag{14}$$

$U(t)$ is the PID control variable, K_p is the proportional gain, $e(t)$ is the error value, K_i integral gain, de is the change in error value and dt is the change in time.

In the control system the following three parameters are taken into consideration: The system speed, the oscillation frequency and the steady state deviation. Generally, the next table summarises the effect of PID gains on the system response:

Table 3 PID gains effect

Response	Rise Time	Overshoot	Settling time	Steady state error
$\boldsymbol{K_p}$	Decrease	Increase	Small variation	Decrease
$\boldsymbol{K_i}$	Decrease	Increase	Increase	Eliminated
$\boldsymbol{K_d}$	Small variation	decrease	Decrease	Small variation

V. PID CONTROLLER IMPLEMENTATION

The gains for the PID controller are adjusted, using the PID-tuner in MATLAB, benefiting from the linear suspension behavior. Moreover, the road disturbance used is shown in Fig 3.

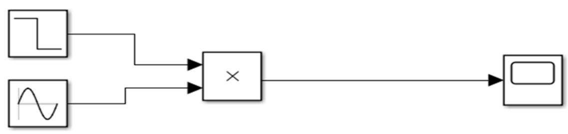

Figure 3:Road simulation

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Step time is π and the initial value is 1 and the final value is 0. The sine wave amplitude is 0.08 with frequency of 1 rad/s. The road disturbance is shown in Fig 4.

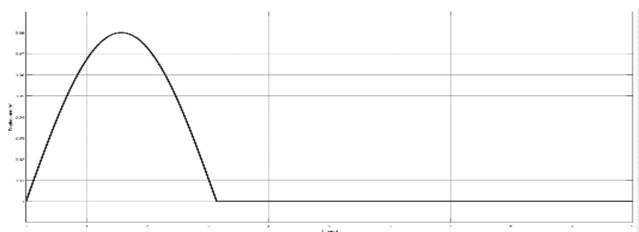

Figure 4: Road disturbance

The PID controller gains used in the quarter car are K_p= 4, K_i=92 and K_d=200. Moreover K_p=8, K_i=52 and K_d=0.3 are the adjusted gains in order to get the desired response in the full car model.

The PID controller gets its feedback from the sprung mass displacement. Then, calculates the error and adjusts the current of the MR damper ,changing the magnetic field and therefore the viscosity which in return modifies the damping force. However, There might be a slight delay so using The Field Programmable Gate Array (FPGA) which is suitable for fast implementation and quick hardware verification will be better but it is out of scope of this paper and is a future work for this implementation.

VI. NEURO-FUZZY CONTROLLER

Kurczyk stated that “Fuzzy controller is a linguistic base-oriented system, which allows for synthesis of the control signal on the basis of an expert knowledge. The expert knowledge is stored in the form of IF-THEN rules” [9]. The main advantage for this type of control is that it doesn’t require an accurate description of the system or any mathematical model to be given. However, the main problem of this method is that a lot of trial and error is required to optimize the fuzzy rules. To solve this problem the neuro-fuzzy technique is used to generate the fuzzy rules. The neuro-fuzzy technique uses machine learning and neural networks in combination with fuzzy logic to achieve human-like reasoning style. To perform this, the neuro-fuzzy has to be trained on data from an already existing controller.

Neuro-fuzzy hybridization results in an intelligent system that combines both human-like reasoning from fuzzy systems and the optimized learning structure of artificial neural networks. The main strength of such neuro-fuzzy systems is their ability to be accurate approximators, in this case the neuro fuzzy is approximating the PID response.

VII. NEURO-FUZZY CONTROLLER IMPLEMNTATION

The PID controller data was extracted from Simulink as shown in fig 5

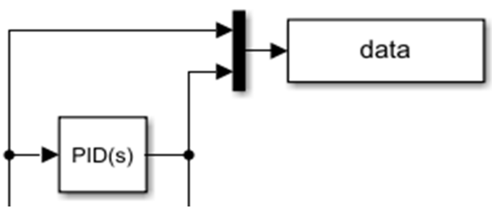

Figure 5 : PID data output

This scheme generates a table showing the inputs and outputs of the PID controller as a function of time. This data was inserted into the MATLAB neuro-fuzzy tool. Then empty fuzzy rules were generated with 1 input 1 output a 7 membership functions. The rules were then trained for 100 epochs until the optimal rules were generated.

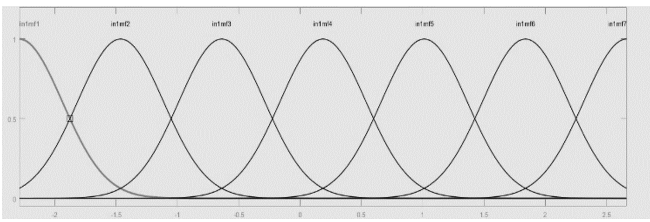
Figure 6: Input membership functions generated

VIII. RESULTS AND DISCUSSIONS

Fig 7 shows the sprung mass displacement passive simulated vs the PID-controlled in the quarter car model

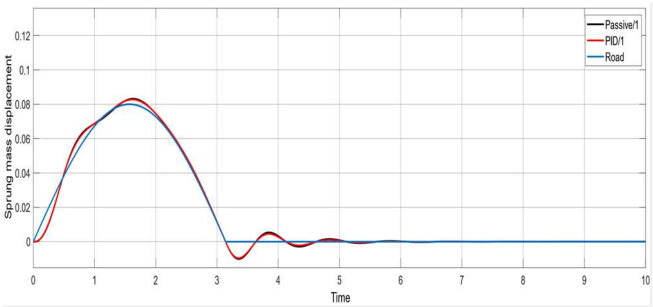

Figure 7:Quarter car sprung mass Displacement

Fig.7 here shows a slight improvement in the displacement as expected. Sprung mass displacement is simulated as to match the disturbance response because our main goal was to improve sprung mass acceleration.Fig.8 shows the sprung mass acceleration in passive and PID controlled systems.

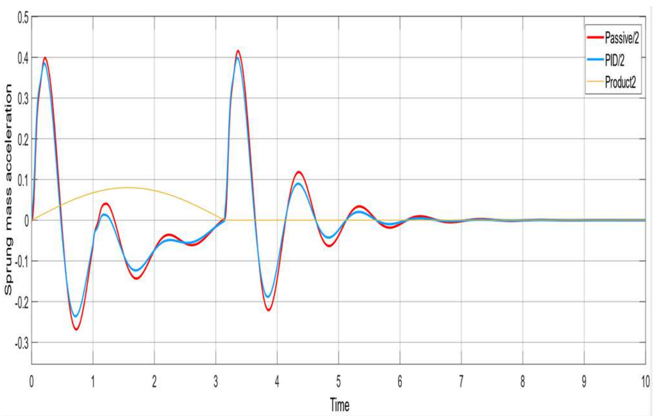

Figure 8:Quarter car sprung mass acceleration

Fig.8 shows improvement in the acceleration response when PID-controller is applied. The performance overshoot is improved by nearly 10%. The Neuro-Fuzzy Controller is implemented only on the full car model. Fig 9 shows the sprung mass displacement of the full car passive, PID controller and neuro-fuzzy controlled.

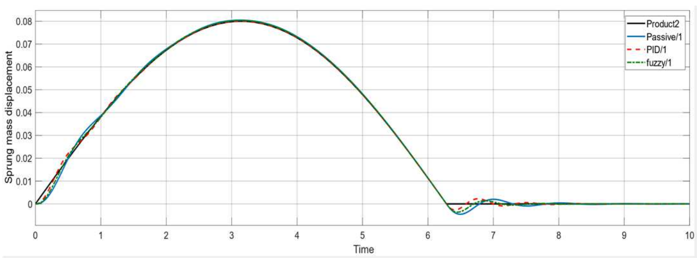

Figure 9 : Full car sprung mass displacement

We can clearly see that the responses of the quarter car and the full car models are similar as the full car model is simulated over π/2 step unlike the quarter car. This was conducted to clearly see that the system will respond the same way under any road disturbance. Fig.10 shows the sprung mass acceleration.

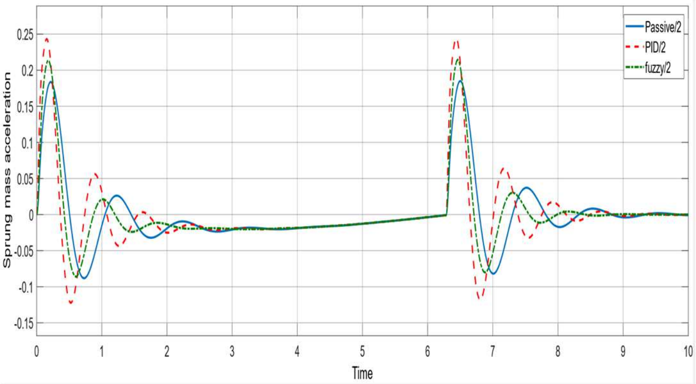

Figure 10 : Full car sprung mass acceleration

The acceleration here shows significant improvements in overshoot and settling time in neuro-fuzzy control, when compared to passive and PID controlled response except for the acceleration at the time of impact and this trade off as a result of improving the pitch of the system, because in full car suspension models, controlling the response of the pitch and the roll is very important to control the ride comfort and safety. Fig.11 shows the system pitch.

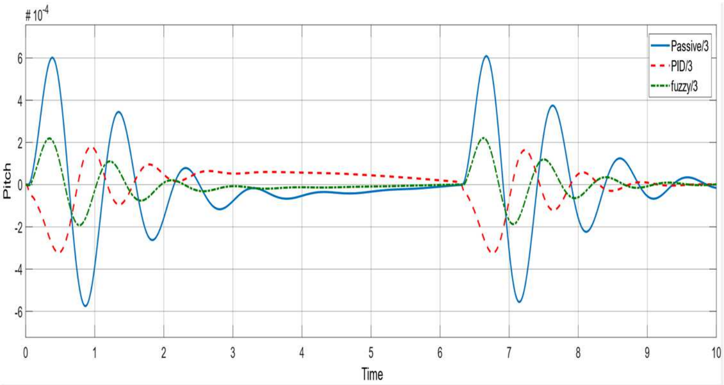

Figure 11 :Full car sprung mass pitch

The pitch response improved significantly in neuro-fuzzy compared to the passive response and PID-controlled response.

IX.Conclusion

The passive responses in both the quarter car and the full car models are similar in terms of the sprung mass displacement and acceleration. When PID controller was applied on the quarter car model, the response improved in the acceleration as desired. Neuro-fuzzy and PID controllers are then implemented on the full car model using similar numerical values and assuming vehicle symmetry and that each wheel has an effect of the spring and the damper of the other wheel and linearity of the suspension component. Neuro-fuzzy controller improved the response of the acceleration of the full car model with comparison to PID and passive models, although at the time of impact it resulted in a greater overshoot. This is tradeoff is done in order to improve the pitch of the suspension whereas in the quarter car suspension the overshoot of the acceleration in the point of impact is not existent due to the ignorance of the pitch in the two degree of freedom quarter car model.

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