University of Huddersfield Repository

Wang, R., Tran, Van Tung, Gu, Fengshou and Ball, Andrew

An Investigation on Energy Recovery Analysis of Active Suspension System

Original Citation

Wang, R., Tran, Van Tung, Gu, Fengshou and Ball, Andrew (2013) An Investigation on Energy Recovery Analysis of Active Suspension System. In: Proceedings of the 19th International Conference on Automation and Computing (ICAC) 2013: Future Energy and Automation. ICAC 2013 . Brunel University, London, UK. ISBN 978-1908549082

This version is available at http://eprints.hud.ac.uk/id/eprint/18556/

http://eprints.hud.ac.uk/

An Investigation on Energy Recovery Analysis of Active Suspension System

R. Wang, V. T. Tran, F. Gu and A. D. Ball

Centre for Efficiency and Performance Engineering, University of Huddersfield

Huddersfield, UK

Abstract—**Currently active suspension system is more applicable than passive for improving the suspension performance, ensuring the stability and passenger safety in modern automotive suspension system. However, high energy consumption is one of the main disadvantages of implementing this system in real applications. In this paper, the energy consumption of electromagnetic actuators used for an active suspension system controlled by proportional-integral-derivative (PID), fuzzy adaptive PID, and neuron adaptive PID are investigated through simulation studies. Based on the energy consumption and the performance analysis, it has found that it is potential to develop a vibration energy recovery system to achieve the energy balance requirement in active suspension systems.**

Keywords-; active suspension; electromagnetic; energy recovery; controls

I. INTRODUCTION

Vehicles travelling on the road are always subjected to the excitation from road irregularities, braking forces, acceleration forces, and inertial forces on a curved track. This excitation not only causes the harmfulness for the passengers in terms of comfort but also influences the manoeuvrability. Therefore, the main task of suspension systems which have been widely used for vehicles is to ensure ride comfort and road holding for a variety of road conditions. Generally, the suspension systems in use could be classified into three categories: passive, semi-active and active [1]. Over the years, both passive suspension system (PSS) and active suspension systems (ASS) have been proposed to optimize the vehicle quality [2][3].The PSS is commonly used for vehicle due to its low price and high reliability. However, this system cannot assure the desired performance and eliminate the trade-off between ride comfort and suspension deflection. Moreover, the vibration energy of PSS is wasted by dissipating as the form of heat [4].

To achieve better performance, the active type has been proposed by many researchers [5]–[8] [9]. The ASS is more elastic and efficient than the passive one, making it capable of providing better road-holding ability and ride quality. Nevertheless, ASS is commonly expensive, complex, and high energy consumption which lead to its limitation in implementations. Hence, regenerative suspension systems have attracted the consideration of numerous researchers in using vibration energy harvester while improving the ride and the holding quality of vehicles.

The potential benefits of recovering the vibration energy from ASS controlled by different methods have been studied by many researchers [4]–[7]. Suda et al. [4] employed two DC motors for a self-powered ASS controlled by skyhook method. In their study, one motor rotated inverse to generate power supplying to the other motor which was used as an actuator to control the vibration performance. Nakano et al. [6] proposed a self-powered active suspension system for truck cabin, where the skyhook was used as controller. Based on the results of energy balance analysis, they concluded that the energy balance depends on the dynamical properties of the suspensions, power spectral density of the road roughness, feedback gain of the active controller, and the specifications of the energy regenerative damper and the actuator. In the study of Hsu [7], electric motor was also used as regenerative damper in ASS which was controlled by linear quadratic optimal control law. The conclusion of Hsu's study was that the recoverable energy depended not only on the driving and road conditions but also on the type of control algorithm. Other outstanding studies of regenerative suspension system could be found in [8]–[13].

As mentioned in [7], the energy that could be recovered from ASS depends on the road conditions and the type of control algorithm. Many control methods have been proposed to apply for ASS such as linear quadratic Gaussian, optimal control, adaptive control, non-linear control and fuzzy control [14]–[19]. In this paper, an investigation of the energy consumption of electromagnetic actuator implemented in quarter-car model was carried out. The simulation was conducted by three control methods including: PID, fuzzy adaptive PID (FA-PID), and neuron adaptive PID (NA-PID). Furthermore, based on the simulation results, vibration energy harvester will be designed to satisfy the energy balance requirement of self-power systems.

II. QUARTER-CAR MODEL AND CONTROLLERS

A. Quarter-Car Model

In order to investigate the vibration recoverable energy of ASS, a quarter-car model which is shown in Fig.1 was employed for this study. The model was developed with two degrees of freedom vibration model which contains the sprung mass m_1 (vehicle body mass) and the un-sprung mass m_2 (wheel mass). The suspension stiffness, the tire stiffness, and the suspension damping coefficient are denoted as k_1, k_2, and c respectively. The

tire damping coefficient is small and eliminates in this study. x_1 and x_2 are the displacement of sprung mass and unsprung mass. An electromagnetic actuator is used to produce control force u between m_1 and m_2 to ensure the desired performance of vehicle.
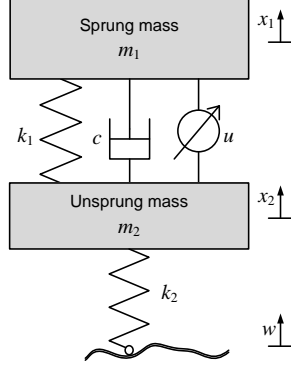

Figure 1. Active quarter-car model

The dynamic equations of motions are written as follows:

$$m_1\ddot{x}_1 = -c(\dot{x}_1 - \dot{x}_2) - k_1(x_1 - x_2) + u$$
$$m_2\ddot{x}_2 = c(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) + k_2(w - x_2) - u$$
(1)

Equation (1) can be rewritten in matrix form using Laplace transform:

$$\begin{bmatrix} m_1 s^2 + cs + k_1 & -(cs + k_1) \\ -cs + k_1 & m_2 s^2 + cs + (k_1 + k_2) \end{bmatrix} \begin{bmatrix} X_1(s) \\ X_2(s) \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ -1 & k_2 \end{bmatrix} \begin{bmatrix} U(s) \\ W(s) \end{bmatrix} \quad (2)$$

This is a MIMO system which the inputs are the road disturbance (w) and the force control signal (u). The performance index for the dynamic response is selected as the relative displacement x_1-x_2. Due to the fact that this is a linear system, the superposition principle could be applied: the output x_1-x_2 is the combined effect of the signal w and u. In these terms, the transfer functions are obtained from (2) as:

$$G_1(s) = \frac{X_1(s) - X_2(s)}{U(s)} = \frac{(m_1 + m_2)s^2 + k_2}{\Delta} \quad (3)$$

$$G_2(s) = \frac{X_1(s) - X_2(s)}{W(s)} = -\frac{m_1 k_2 s^2}{\Delta} \quad (4)$$

Where：

$$\Delta = (m_1 s^2 + cs + k_1)(m_2 s^2 + cs + k_1 + k_2) - (cs + k_1)^2 \quad (5)$$

And s is the variable known as Laplace operator in the form of $s = \alpha + i\beta$.

In the passive suspension system, most of energy is dissipated where the instant damping force is generated. The instant damping force is proportional to the square of the relative velocity between m_1 and m_2. Therefore, the dissipation energy could be recovered under the roughness road conditions:

$$P = c(\dot{x}_1 - \dot{x}_2)^2 \quad (6)$$

In active suspension system, the control strategies are used to reduce vibration by using electromagnetic actuator. The equation of energy consumption of this actuator can be written as:

$$P_e = \frac{4\pi\phi}{l}(v_s - v_u)I \quad (7)$$

Where Φ is the flux linkage, l is the lead of the ball-screw, (v_s-v_u) is the relative velocity between sprung mass and unsprung mass and I is the electric current flow through the motor's coils. If the power requirement is positive, the motor operates in electromotor mode and the current flows from the power supply (battery) into the positive terminal of the motor for consumption. The motor normally operates in clockwise when the m_1 and m_2 approaching each other with negative relative velocity for electromotor mode. On the contrary, if the motor operates inversely and transforms into power generation, the motor operates in anticlockwise when the m_1 and m_2 separating each other with positive relative velocity for generation mode. The current flows directly to the positive electrodes in the battery and the motor charges the battery as a generator with recovered energy from vibration of the suspension system [20].

B. *PID and Fuzzy Adaptive PID Controllers*

In order to achieve the stability and passenger safety, the force u generated by controller reduces the relative vibration x_1-x_2 corresponding with the road condition w. The block diagram of ASS is shown in Fig. 2.
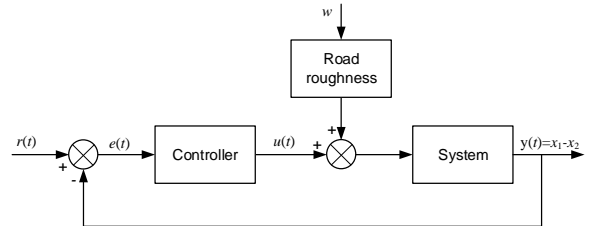

Figure 2. Block diagram of ASS

The controllers depicted in Fig. 2 could be PID, FA-PID, and NA-PID. The PID controller being a common feedback loop has been used to control various engineering systems. The error signal $e(t)$ is then used to adjust some inputs to the system in order to bring the system output back to its desired set-point. PID controllers do not require advanced mathematics to design and can be easily adjusted to the desired application. The block diagram of the PID controller is depicted in Fig. 3, where K_p, K_i, and K_d are the proportional gain, the integral gain, and the derivative gain, respectively. The values of these gains are determined by the methods explained in [21].
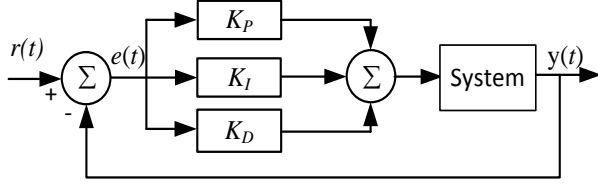
Figure 3. The diagram of PID controller

FA-PID controller applies fuzzy control rules to modify PID parameters on-line using error $e(t)$ and change-in-error $de(t)/dt$ as inputs, which can meet the request of $e(t)$ and $de(t)/dt$ in different time for self-tuning PID parameters. The core of FA-PID control system consists of fuzzy inference system and traditional PID controller as shown in Fig. 4. The fuzzy inference processes the inputs ($e(t)$ and $de(t)/dt$) and computes the outputs which are K_p, K_i, and K_d in cope with the rule base and data base. In this study, seven (7) linguistic variables are used for each input and output. Totally, 49 fuzzy rules are created from these input linguistic variables. Fig. 5 is an example of output surface of K_d.
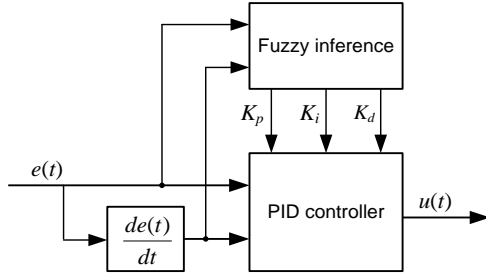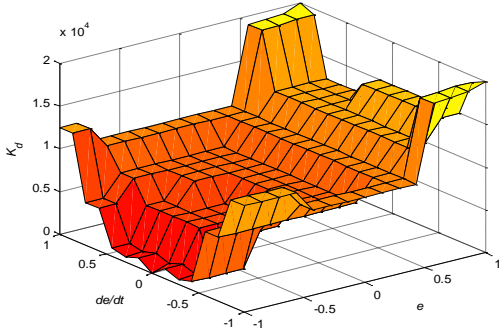

Figure 4. The diagram of FA-PID controller

Figure 5. The output surface of K_d

C. Neuron Adaptive PID Controller

The block diagram of the NA-PID controller is shown in Fig. 6, where the outputs of the neural network are proportional K_p, K_i, and K_d. The inputs are the error and relevant parameters $e(t)$, $\Delta e(t)$ and $\Delta^2 e(t)$. The adaptive control algorithm can be given as:
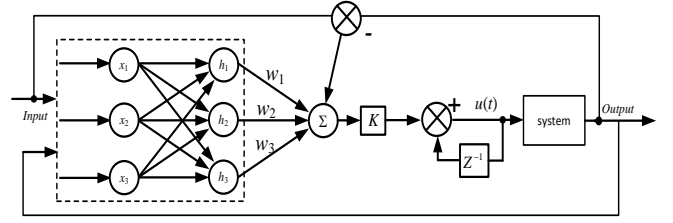

Figure 6. The structure of NA-PID controller

$$u(t) - u(t-1) + K\sum_{i=1}^{3} w_i(t)x_i(t) \qquad (8)$$

$$w_i(t) = \frac{w_i(t)}{\sum_{i=1}^{3}|w_i(t)|} \qquad (9)$$

$$w_1(t) = w_1(t-1) + \eta_I z(t)u(t)(e(t) + \Delta e(t)) \quad (10)$$

$$w_2(t) = w_2(t-1) + \eta_P z(t)u(t)(e(t) + \Delta e(t)) \quad (11)$$

$$w_3(t) = w_3(t-1) + \eta_D z(t)u(t)(e(t) + \Delta e(t)) \quad (12)$$

$$x_1(t) = e(t) \qquad (13)$$

$$x_2(t) = e(t) - e(t-1) \qquad (14)$$

$$x_3(t) = \Delta^2 e(t) = e(t) - 2e(t-1) + e(t-2) \quad (15)$$

$$z(t) = e(t) \qquad (16)$$

Where was the bias of proportion η_P, integral η_I, and differential η_D at the sample time t. it adopt variable learning rate of integration, proportion, and differential to adjust different weighted coefficient. Furthermore, K is the ratio coefficient of neuron which used to adjust the weight of integration, proportion and differential.

III. SIMULATION RESULTS AND DISCUSSION

Some initial parameters of ASS and electromagnetic actuator for this simulation are given in Table 1 and Table 2. The road disturbance is single bump input (pulse signal) which peak amplitude is around 100mm as shown in Figure 7.
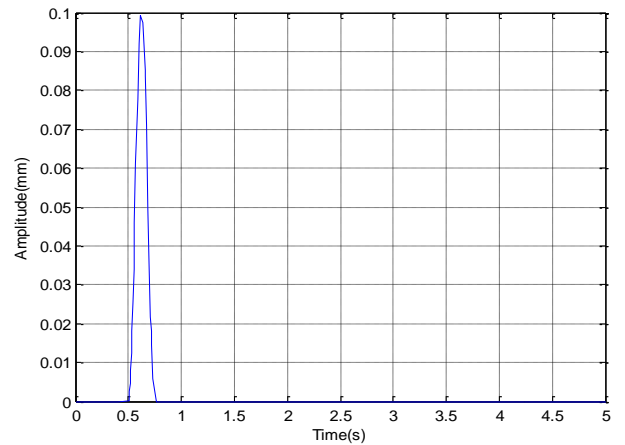

Figure 7. Road disturbance input

$$r(t) = \begin{cases} \alpha(1-\cos\pi t), 0.5 \leq 0 \leq 0.75s \\ 0, else \end{cases} \qquad (17)$$

TABLE 1. PARAMETERS OF QUATER-CAR MODEL

Quarter-car model parameters				
m_1 (kg)	m_2 (kg)	k_1 (N/m)	k_2 (N/m)	c (Ns/m)
2500	320	150,000	300,000	350

TABLE 2. PARAMETERS OF ENERGY STRATEGY MODEL

Electromagnetic actuator & DC motor			
Φ (m/wb)	l (mm)	I (A)	V (Volt)
9.62	16	3.5	12

Based on the simulation analysis, the relative displacements (suspension travel, x_1-x_2) of these controllers are depicted in Fig 8. It shows that NA-PID is the best controller in comparison with the others. The relative displacement of NA-PID can reach the stability in a short of settling time (3.5s) compared with FA-PID 4s and PID 6s. The overshoot amplitude for suspension is significant lower than that of either PID or FA-PID, and most of suspension travel acts on the wheels. This indicates that NA-PID can create a better ride comfort for vehicle and passenger safety.
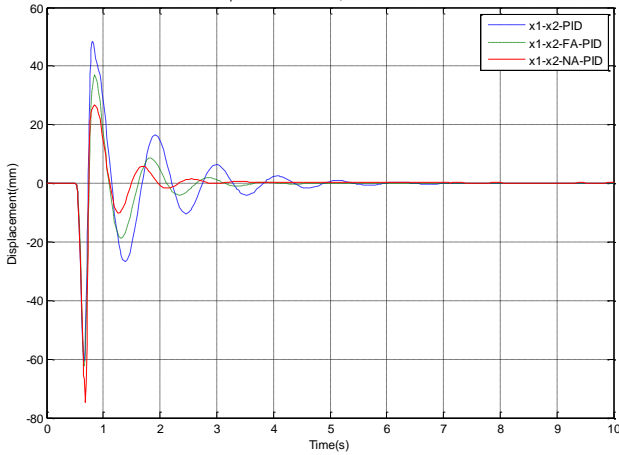

Figure 8. The system relative displacement of PID, FA-PID, and NA-PID

The sprung mass displacements x_1 of these three controllers shown in Fig 9, it can be seen that NA-PID is more reliable and stable than FA-PID and PID for the vehicle's body. NA-PID provides much lower overshoot amplitude in the travelling times and ensures the vehicle body comfort. And the unsprung mass displacement x_2 is shown in Fig. 10.

NA-PID is still obtained the shortest settling time, and the tire deflection is higher than the others. The tire deflection in NA-PID absorbs much more suspension travel than FA-PID and PID in order to reduce the

deflection of car body and supply a more comfortable driving condition. In FA-PID, the absorption of body deflection is quite better than tire deflection. The amplitude of tire deflection is the worst in those of three, usprung mass hasn't help suspension traveling enough to resist road disturbance, but it responses fast enough to make both body and tire to become stable. Both sprung and unsprung mass have dedicated to reduce suspension travel.
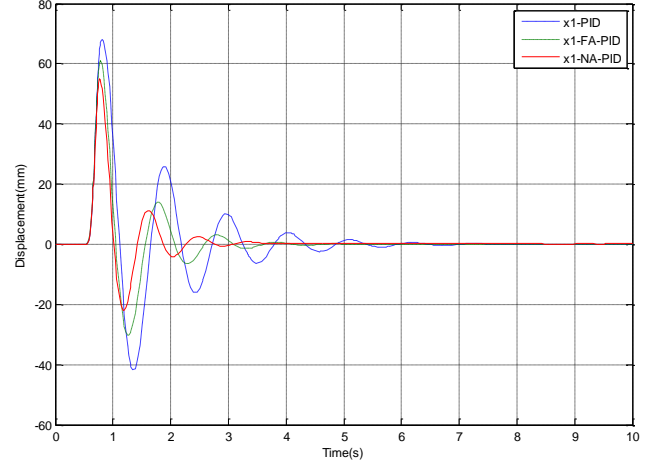

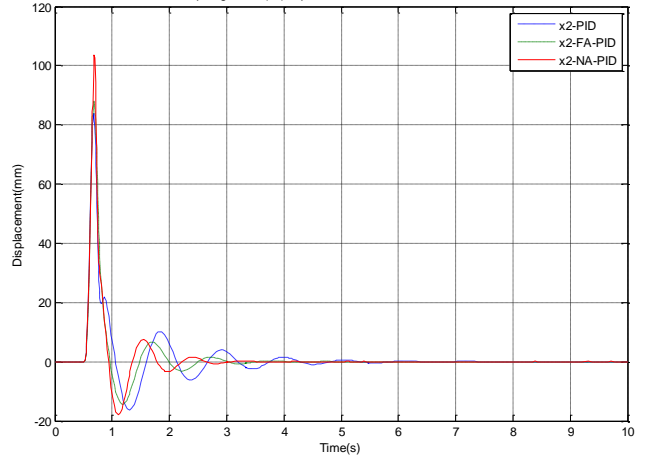Figure 9. The sprung mass displacement

Figure 10. The unsprung mass displacement

Fig. 11 presents the relative velocity between sprung mass and unsprung mass. From the (7), the power consumption and power recovery are both proportional to the relative velocity. The actuator with motor is as a connection between m_1 and m_2, and the motor's operations (electromotor mode and generation mode) depend on the direction (compression and extension) and value of relative velocity. For the electromotor mode, motor power requirement is positive with negative relative velocity. NA-PID obtains the lowest value than the others and obviously reduces the power consumption of actuator. For the generation mode, motor power is negative with positive relative velocity. NA-PID is also obtains the highest value for positive relative velocity of energy recovery by motor. Although FA-PID can restrain

suspension travel, the relative velocity decides its energy strategy with electromagnetic actuator in both consumption and regeneration modes, and the highest negative velocity and lowest positive velocity.
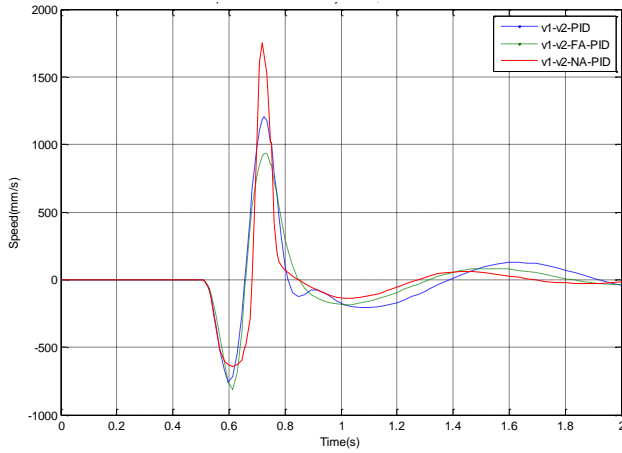

Figure 11. the system relative velocity

The energy consumption of electromagnetic actuator of controllers is shown in Fig. 12 and Fig. 13. Obviously, due to the fact that the overshoot of FA-PID is higher than those of the others, the electromagnetic actuator with NA-PID consumes lower energy in comparison with PID and FA-PID. NA-PID strategy is definitely achieved the purpose of reducing the power consumption of electromagnetic actuator. From the simulation analysis, the peak of PID consumption is 20.91W, FA-PID is 22.09W and NA-PID is 16.91W. The maximum NA-PID energy consumption is lower than PID and FA-PID. For the total energy consumption, the PID is higher than those of FA-PID and NA-PID. The energy which can be recovered (46.56W) is higher than PID (32.37W) and similar with FA-PID (24.94W) as shown in Fig 13. As a result of the first overshoot of NA-PID is large enough; the total energy recovery of NA-PID is still larger than those of PID and FA-PID.
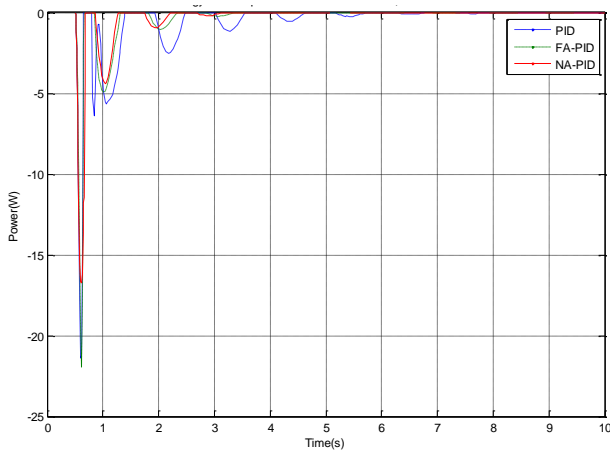

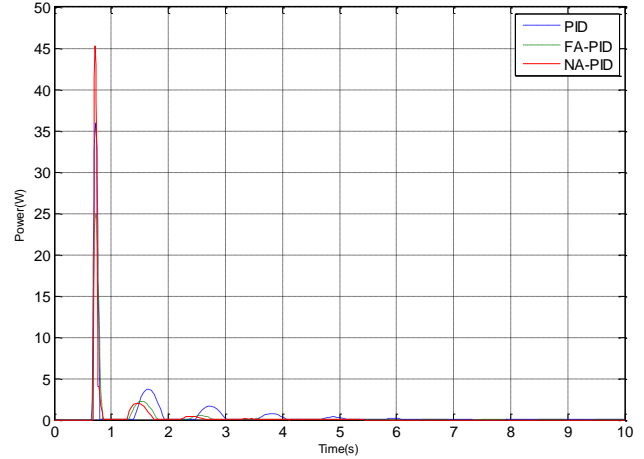Figure 12. The energy consumption with controllers

Figure 13. The energy regeneration with controllers

In above, the simulation results show that NA-PID has the best energy performance undoubtedly. FA-PID with favourable performance of suspension travel, and energy comsuption and regeneration have almost achieved a near-subsistence status. In contrast, both energy strategy and suspension performance in PID are not the best for car holding and energy saving, but PID control is easy to install in actual applications.

IV. CONCLUSION

The simulation results show that NA-PID is featured by reliable control, strong robustness and fewer adjustable parameters in obtaining good performance of suspension system and energy strategy in an active suspension with regenerative devices for a quarter car models. In addition, there is also a limitation of using FA-PID and PID controls in that they cannot generate good car handling, ride comfort and energy saving compared with NA-PID. NA-PID shows more advantages in linear model than FA-PID and PID to obtain energy saving and recovery.

In future work, both NA-PID and FA-PID will be optimized to gain their benefits for linear model and nonlinear model to accord with hybrid energy recovery suspension model and conduct tests to evaluate developed methods.

V. REFERENCES:

[1] J. Lin, R.J. Lian, C.N. Huang, and W.T. Sie, "Enhanced fuzzy sliding mode controller for active suspension systems," Mechatronics, vol. 19, no. 7, pp. 1178–1190, Oct. 2009.

[2] D. Karnopp, "Analytical Results for Optimum Actively Damped Suspensions Under Random Excitation,"Journal of Vibration, Acoustics Stress and Reliability in Design, vol. 111, no. 3 , pp. 278–282Jul, 1989.

[3] L. Sun, "Optimum design of 'road-friendly' vehicle suspension systems subjected to rough pavement surfaces," Applied Mathematical Modelling, vol. 26, no. 5, pp. 635–652,May 2002.

[4] S. N. Yoshihiro Suda, "Hybrid Suspension System with Skyhook Control and Energy Regeneration (Development

of Self-Powered Active Suspension),” Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, vol. 29, pp. 619–634, 1998.

[5] D. A. Crolla and R. S. Sharp, “Road Vehicle Suspension System Design - a review,” Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, vol. 16, no. 3, pp. 167–192, 1987.

[6] K. Nakano, “Combined Type Self-Powered Active Vibration Control of Truck Cabins,” Vehicle System. Dynamics: International Journal of Vehicle Mechanics and Mobility, vol. December, pp. 449–473, 2010.

[7] P. Hsu, “Power Recovery Property of Electrical Active Suspension Systems,” Energy Conversion Engineering Conference, vol. 3, pp. 1899 – 1904, vol.3, Aug 1996.

[8] Y. Okada and H. Harada, “Regenerative Control of Active Vibration Damper and Suspension Systems,” Proceedings of the 35th IEEE Conference on Decision and Control, vol. 4, pp. 4715–4720, Dec 1996.

[9] M. R. Jolly and D. L. Margolis, “Regenerative Systems for Vibration Control,” Journal of Vibration and Acoustics., vol. 119, no. 2, pp. 208–215, Apr. 1997.

[10] T. Shiiba and Y. Sude, “A New Hybrid Suspension System with Active Control and Energy Regeneration,” Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, vol. 25, pp. 641–654, 1996.

[11] G. R. Wendel and G. L. Stecklein, “A Regenerative Active Suspension System,” SAE International Congress & Exposition, SAE Technical Paper 910659, Feb. 1991.

[12] Y. Kawamoto, Y. Suda, H. Inoue, and T. Kondo, “Modelling of Electromagnetic Damper for Automobile Suspension,” Journal of System Design and Dynamics, vol. 1, no. 3, pp. 524–535, 2007.

[13] K. Huang, F. Yu and Y. Zhang, “Active Controller Design for an Electromagnetic Energy-regenerative Suspension,” International Journal of Automotive Technology, vol. 12, no. 6, pp. 877–885, Dec 2011.

[14] T. J. Gordon, C. Marsh, and M. G. Milsted, “A Comparison of Adaptive LQG and Nonlinear Controllers for Vehicle Suspension Systems,” Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, vol. 20, no. 6, pp. 321–340, 1991.

[15] A. Alleyne and J. K. Hedrick, “Nonlinear Adaptive Control of Active Suspensions,” IEEE Transactions, Control Systems Technology, vol. 3, no. 1, pp. 94–101, Mar 1995.

[16] S. Liu, Z. Huang, and Y. Chen, “Automobile active suspension system with fuzzy control,” Journal of Central South University of Technology., vol. 11, no. 2, pp. 206–209, Jun. 2004.

[17] F. J. D’Amato and D. E. Viassolo, “Fuzzy control for active suspensions,” Mechatronics, vol. 10, no. 8, pp. 897–920, Dec. 2000.

[18] C. Tang, L. Yue, L. Guo, S. Zhou, W. Zhou, and Z. Wang, “Fuzzy Logic Control for Vehicle Suspension Systems,” Intelligent Robotics and Applications, C. Xiong, H. Liu, Y. Huang, and Y. Xiong, Eds. Springer Berlin Heidelberg, 2008, pp. 197–206.

[19] S. J. Huang and W. C. Lin, “A Self-Organizing Fuzzy Controller for an Active Suspension System,” Journal of Vibration and Control, vol. 9, no. 9, pp. 1023–1040, Sep 2003.

[20] A. A. Aldair and W. J. Wang, “The Energy Regeneration of Electromagnetic Energy Saving Active Suspension in Full Vehicle with Neuron-fuzzy Controller,” International Journal of Artificial Intelligence & Applications, vol. 2, no. 2, pp. 32–43, Apr 2011.

[21] K. J. Åström and T. Hägglund, Automatic tuning of PID controllers. Research Triangle Park, North Carolina: Instrument Society of America, 1995.