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PROTOTYPING, TESTING AND CONTROL ENERGY FOR ACTIVE-REGENERATIVE ELECTROMAGNETIC SHOCK ABSORBER BASED QUARTER CAR MODEL

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ABSTRACT

In this paper, electromagnetic shock absorber for passenger car is designed and fabricated. Series of experiment is conducted to attain damping and current constant. Controllers are designed for energy regeneration and comfort based quarter car model. Shock absorber is designed and prototyped to absorb vibration energy and dissipate the energy as control actuation. The shock absorber use DC permanent magnet motor to absorb and dissipate power. Prototype is tested on Auto Damping Test Machine (ADFT). With the decreasing of external load, damping force and generated current are increased. Formulation of DC motor for electromagnetic damper is developed. Model of active-regenerative electromagnetic suspension and some controller strategies is simulated based to test results. Tracking reference, PI-tracking reference and Multi objective H∞ controller are presented and compared to determine their performance due to comfort, energy regeneration and consumption. Simulations are carried out with unevenness road input. Amount of regenerated energy is harvested when system at passive mode with high body acceleration. Multi objective controller is satisfy to maintain body acceleration, suspension travelling and tire deflection with minimum power requirement. PI-tracking reference controller has less body acceleration, suspension travelling and tire deflection compare to tracking and H∞ multi objective controller but it need the highest power consumption. Averages of RMS body acceleration are: 1.81, 0.59, 0.42, 0.46 m/sec^2 respectively for passive, tracking reference, PI-tracking reference and multi objective H∞ controller and power consumption are: 49.58 Watt, 72.11 Watt, 51.49 Watt, respectively for tracking reference, PI-tracking reference and multi objective H∞ controller when 20Ω electric load applied for the prototype with C Class road input and 50km/h vehicle speed. Average power regeneration at passive mode is: 19.83Watt that complies with experiment result.

Keywords: *Electromagnetic Shock Absorber, Passenger Car, Control Energy, Active-Regenerative.*

1. INTRODUCTION

Vehicle suspension is developed to increase ride comfort and handling performance. Active suspension was implemented to improve ride index by control damping force. Unfortunately it is much power required to compensate active actuation. Hydraulic active suspension requires power up to 370W by using LQR controller [1] and the maximum power consumption for electromagnetic shock absorber is 108 W for the best comfort controller [2] on a quarter car test rig experiment. In the other hand, it is possible to regenerate energy from vehicle suspension vibration.

A regenerative electromagnetic shock absorber will be able to harvest 16–64 W power at 0.25–0.5 m/s RMS suspension velocity [3]. A peak power of 45 Watt and average power of 11.43 Watt with 10Ω electrical load are attained from the prototype when oscillating speed of bench test at 0.1 m/s, the RMS of suspension velocity when vehicle pass C class road with speed 50 km/h [4]. Prototyping and bench test has also conducted by Zhang [5]. He varies three electrical loads, 3.3, 4.7 and 10Ω. 62.4%

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efficiency, 12.59W harvested energy and 2475 Ns/m damping forces were achieved with 3.3Ω external load. From the above results, electromagnetic shock absorber is capable for harvesting energy and has controllability for active suspension. There are some ideas to integrate between actuator regeneration and activation of electromagnetic suspension as a control strategy. But there is a trade-off between harvesting energy and ride comfort.

Electromagnetic is proper as an actuator because electromechanical construction enable to support for switching between regeneration mode and dissipation mode, low energy consumption and high power to weight ratio. A novel study of energy regenerative active suspension by two modes was studied as an initial research [6]. Simulation study of novel self-powered active suspension system for automobile was conducted to compare three condition of a single electromagnet motor-generator, such: passive, fully active and self active with zero energy consumption [7]. Multi objective H∞ control strategy for energy harvesting while damping actuation was purposed [8]. Simulation results with quarter car model, average power of 40 Watt has harvested by using regenerative damper control strategy.

According to the above researches, there is rare discussion about controller performance due to ride comfort, ride handling and the driving of harvested energy based to prototype characteristic.

In this paper, we propose a prototype of electromagnetic shock absorber with DC permanent magnet actuator. It is designed for passenger car that light in weight. Prototype parameters are obtained by series of testing on ADFT-2NH test machine. The parameters are used to simulate controllers. Tracking reference, PI-tracking reference and H∞ multi objective controller are compared to investigate their performance due to comfort and handling.

2. PROTOTYPING

Electromagnetic shock absorber consists of outer tube as moving part and inner tube static part. The two components connected by rack and tapered gear. Prototype dimension is described by Fig 1 and captured as Fig 2.

Rack and pinions convert linear motion from suspension stroke into angular motion for motor current input/output.

Tube was made from 6061 alluminium alloy. Tapered gear, spur gear and rack are from Ø 20 mm, Ø 18 mm , Ø 16mm ST 52 low alloy steel respectively. Rack has maximum 30 mm stroke with 1.5 mm of lead. Specifications of motor are listed on Table 1.

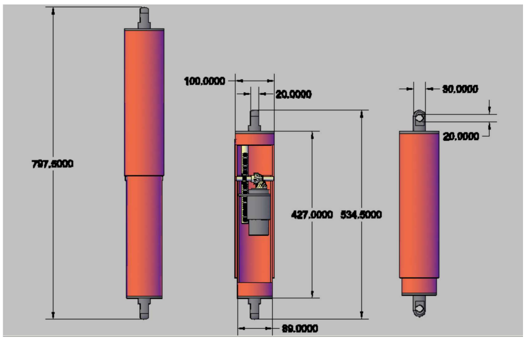

Figure 1: Prototype Dimension

Table 1: Specifications of DC motor

Parameter	Value
Rated voltage (V)	24 V
Max Current (I)	5 Amp
Armature Resistance (r_i)	2 Ω
Rotor Inductance (L)	0.42 mH

Figure 2: Prototype Photograph

3. PROTOTYPE TESTING

Test is aimed to characterized mechanical and electrical properties of prototype. The test is conducted on ADFT-2NH, Auto Damping Force Testing machine as shown by Fig 3. A series of experiments were carried to find out damping force characteristics for different electrical external load related to current of DC motor. Sinusoidal displacement input with different velocities was imposed to the prototype. Force-distance diagram for different electrical load and velocities are shown by Fig 4 and 5. Generated currents for different electrical loads (R_{ext}) with velocity oscillation v=0.2m/s is shown by Fig 6. Since power output follows $P = i^2 R_{ext}$, average power will be: 10.59 W, 13.12 W, 17.07 W and 19.50 W respectively for 50, 40, 30 and 20 Ω external electrical loads.

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Figure 3: Conducted Test on ADFT-2NH

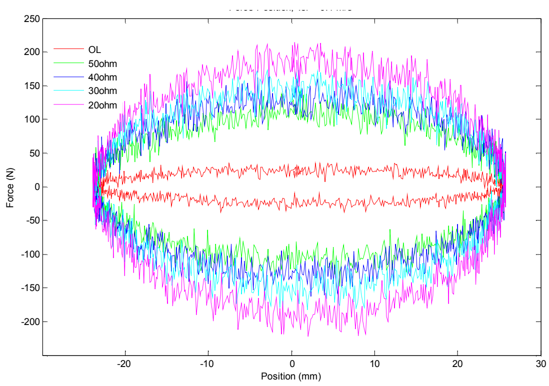

Figure 4. Force-distance diagram for different electrical loads with v=0.2m/s

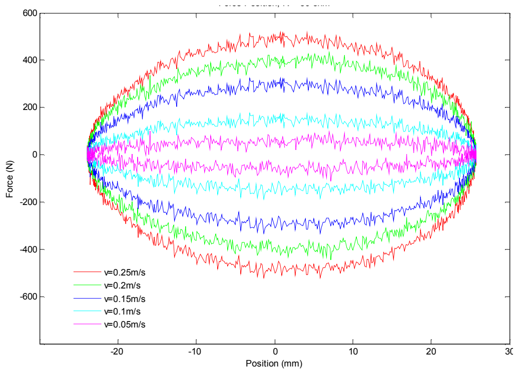

Figure 5. Force-distance diagram for different velocities, R=20ohm

Relation between damping force and currents tabled by table 2 that shows current constants (Φ) for any electrical load.

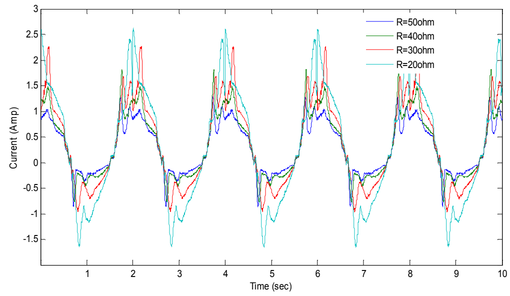

Figure 6.Generated current, v=0.2m/s for different electrical loads.

Table 2. Current Constant

Load (Ohm)	Max Current (A)	Force (N)	Current Constant (Φ) N/A
50	1.27	116.7	91.89
40	1.82	133.33	73.26
30	2.26	150	66.37
20	2.59	183.3	70.77

4. MODELING AND CONTROL

4.1. Quarter Car Model

Electromagnetic suspension is developed as electro-mechanical system in a quarter car model with 3 Degree of Freedom (DOF) as Fig 7. C_m represents Electromagnetic damper which works as damper that regenerate energy and use the energy for sharing with external power source to generate force actuation f_d for comfort and handling.

Mathematical equations can be written as:

$$m_s\ddot{z}_s = -k_s(z_s - z_u) - k_r(z_s - z_u) - c_m(\dot{z}_s - \dot{z}_r)$$
$$m_r\ddot{z}_r = k_r(z_s - z_r) + c_m(\dot{z}_s - \dot{z}_r) + f_d$$
$$m_{us}\ddot{z}_{us} = k_s(z_s - z_u) - k_w(z_u - z_0) - f_d \qquad (1)$$

where :

m_s : sprung mass, m_r : mass of gear rack, m_{us} : sprung mass, k_s : spring constant, k_r : spring constant of rack-gear, k_w : spring constant of wheel, c_m : motor damping constant, z_s : sprung mass displacement, z_r : rack displacement, z_u : unsprung mass displacement, z_0 : contour of road, f_d : applied force.

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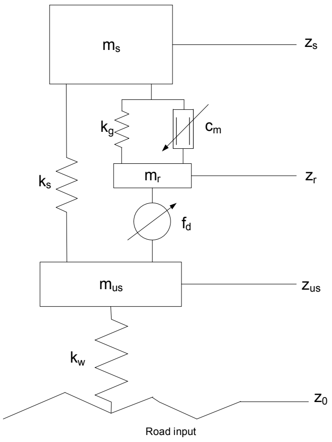

Figure 7: Quarter Car Model with Electromagnetic Damper

Excitation of suspension by road input can be expressed as random Gaussian signal :

$$\dot{z}_0 = -2\pi f_0 z_0 + 2\pi\omega_{road}\sqrt{vG_0} \qquad (2)$$

where:

f_o is lower limiting frequency, ω is zero mean white noise process, v is vehicle speed, G_0 is unevenness road coefficient [6].

4.2. Electromagnetic Damper Model

Stroke of suspension convert to angular motion of motor by:

$$\omega = \frac{2\pi}{l}\dot{z} \qquad (2)$$

where, $z = z_r - z_u$

Shock absorber damping force is

$$f_d = u = \Phi i \qquad (3)$$

Electrical circuit for DC motor as a damper with supplied voltage described by Fig 8 and circuit equation as:

$$v = L\frac{di}{dt} + Ri + \Phi\dot{z} \qquad (4)$$

where :

$R = r_i + R_{ext}$

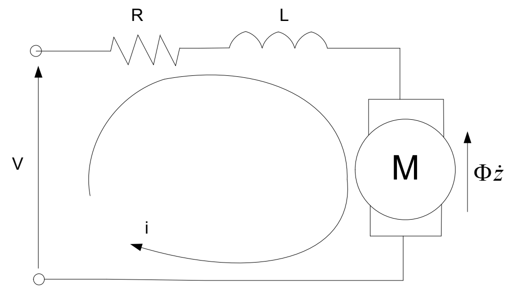

Figure 8: Motor Circuit

For active system, energy consumption is the product of voltage of power supply (v), and current of motor circuit (i), and also writes as required output force (u_{ref}) as:

$$E = iv = \frac{u^2}{C_\Phi} + u\dot{z}, \text{ with } C_\Phi = \frac{\Phi^2}{R} \qquad (5)$$

5 . Control Strategy

5.1 Tracking Reference controllers

Current of DC motor actuator is controlled follows required output force u_{ref}. The required output force as a reference control signal follows [9]:

$$u_{ref} = -C_s\dot{x}_s - C_u\dot{x}_u, \qquad (6)$$

Where's, C_s and C_u are feedback gain velocity of sprung and un-sprung mass respectively. System is stable if $C_s \geq 0$ and $C_u \geq 0$. The positive feedback gain is for sprung mass velocity skyhook isolation and the negative gain is for skyhook road holding.

Equation (5) can be written:

$$u_{ref} = -(C_s + C_u)\dot{x}_s + C_u\dot{z} \qquad (7)$$

Substituting equation (5) to (7), energy consumption is follow:

$$E = \frac{(C_s+C_u)^2}{C_\Phi}\dot{z}_s^2 - \frac{(C_s+C_u)}{C_\Phi}(2C_u + C_\Phi)z_s\dot{z} + \frac{C_u}{C_\Phi}(C_u + C_\Phi)\dot{z}^2 \qquad (8)$$

The first time is energy consumption for reducing vibration of sprung mass, the second term is gain setting for comfort and handling and the third term is regenerative energy.

5.2. PI-Tracking Reference

In this control strategy, we insert Proportional Integral controller to tracking controller that introduced 5.1. PI controller describes by Figure 9.

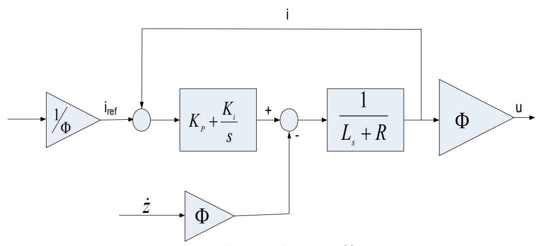

Figure9. PI Controller

5.1.1 Multi Objective H∞ Control

In this control strategy, ride comfort, handling and energy driving are optimized. The control objective is: minimizing both of sprung mass acceleration and actuator current and also maximizing harvesting energy [8]. The objective functions are filtered by simple filter as described by:

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$$\alpha(s) = \frac{K_f}{s\tau + c} \quad (9)$$

K_f is fileter gain and c is a constant.
Define state variable as :

$$\begin{bmatrix} z_s - z_u \\ z_s - z_r \\ z_u - z_0 \\ \dot{z}_s \\ \dot{z}_r \\ \dot{z}_u \\ \dot{z}_0 \end{bmatrix} \quad (10)$$

By considering the filter, state space representation becomes:

$$\dot{x}(t) = Ax(t) + Bu(t) + E\dot{z}_r(t) \quad (11)$$

with:

$$A = \begin{bmatrix} 0 & 0 & 0 & 1 & 0 & -1 & 0 \\ 0 & 0 & 0 & 1 & -1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ -\frac{k_s}{m_s} & -\frac{k_r}{m_s} & 0 & -\frac{c_m}{m_s} & \frac{c_m}{m_s} & 0 & 0 \\ 0 & -\frac{k_r}{m_r} & 0 & \frac{c_m}{m_r} & -\frac{c_m}{m_r} & 0 & 0 \\ \frac{k_s}{m_u} & 0 & -\frac{k_w}{m_u} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & -\frac{1}{\tau_s} \end{bmatrix}$$

$$B = \begin{bmatrix} 0 & 0 & 0 & 0 & {K_T}/{m_r} & {-K_T}/{m_u} & 0 \end{bmatrix}^T$$

$$E = \begin{bmatrix} 0 & 0 & -1 & 0 & 0 & 0 & {K_f}/{\tau_f} \end{bmatrix}^T$$

The following objective can be defined as a state vector:

$$y_p(t) = [\ddot{z}_s(t) \quad i(t) \quad \dot{z}_u]^T \quad (12)$$

that can be written as :

$$y_p(t) = Cx(t) + Du(t) + F\dot{z}_r(t) \quad (13)$$

with:

$$C = \begin{bmatrix} -\frac{k_s}{m_s} & -\frac{k_r}{m_s} & 0 & -\frac{c_m}{m_s} & \frac{c_m}{m_s} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 1 \end{bmatrix}$$

$$D = [0 \quad 1 \quad 0]^T, F = [0 \quad 0 \quad 0]^T$$

Figure 10 shows the overall control scheme for multi objective controller:

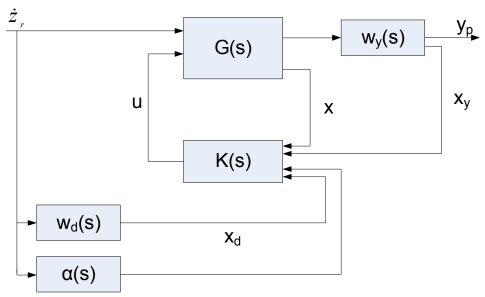

Figure10. Multi objective H∞ control scheme

Denote that state space realization for W_d (s) and W_y (s) respectively as : (A_d,B_d,C_d,D_d), (A_y,B_y,C_y,D_y), the augmented plant follows Fig.10 can be written as:

$$\dot{x}_g(t) = A_g x_g(t) + B_{ug} x_g(t) u(t) + B_{zg}\dot{z}_r(t) \quad (14)$$

$$y(t) = C_g x(t) + D_{ug} u(t) + D_{zg}\dot{z}_r(t) \quad (15)$$

where :

$$A_g = \begin{bmatrix} A & 0 & EC_d \\ B_y C & A_y & B_y F C_d \\ 0 & 0 & A_d \end{bmatrix}, B_{zg} = \begin{bmatrix} ED_d \\ B_y F D_d \\ B_d \end{bmatrix}$$

$$B_{ug} = [B \quad B_z D \quad 0]^T, C_g = [D_z C \quad C_z \quad D_z F C_d]$$

$D_{zg} = D_z F D_d$, $D_{ug} = D_z D$, then state controller written as:

$$u(t) = K_{opt} x_g(t), \quad (16)$$

The result of close loop system is:

$$\begin{cases} \dot{x}_g(t) = (A_g + B_{ug}K)x_g(t) + B_{zg}\dot{z}_r(t) \\ y(t) = (C_g + D_{ug}K)x(t) + D_{zg}\dot{z}_r(t) \end{cases} \quad (17)$$

and the objective is minimizing of H∞ norm of the following matrix:

$$T(s) = (C_g + D_{ug}K)(sI - A_g - B_{ug}K)^{-1} B_{zg} + D_{zg} \quad (18)$$

By using LMI programming to solve the problem,

$$\min_{X,Y,\gamma} \gamma \begin{cases} \begin{bmatrix} A_{cl} + A_{cl}^T & B_{ug} & (C_g X + D_{ug} Y)^T \\ * & -\gamma I & D_{zg}^T \\ * & * & -\gamma I \end{bmatrix} < 0 \\ X = X^T \\ \gamma > 0 \end{cases} \quad (19)$$

where: $A_{cl} = A_g + B_{ug}Y$, and the state feed-back controller is :

$$K_{opt} = YX^{-1} \quad (20)$$

6. Simulation Result

Simulations have been undertaken for testing: regenerated power, controller performance and energy consumption of each controller. Simulation result of regenerated power also compare to shock absorber test result. Simulation parameters are listed at table 3 and table 4 for controller parameters.

Table 3. Simulation Parameter

Parameter	Value
Mass-spring system	
Sprung mass (m_s)	250 Kg
Un-sprung mass (m_u)	40 Kg
Mass of rack (m_r)	20 Kg
Spring stiffness (k_s)	16 kN/m
Rack stiffness (k_g)	250 N/m
Wheel stiffness (k_w)	160 kN/m
Electromagnetic Damper	

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Motor Constant (Φ)	70.77 N/A
Armature Resistance (R_i)	2 Ω
Motor Inductance (L)	0.42 mH
External Resistance (R_{ext})	20 Ω

We use bode gain phase margin for tuning proportional gain (K_p) and integral gain (K_i) for PI Controller.

Table 4. Controller parameter

Parameter	Value
Tracking and PI Tracking Controller	
Sprung mass velocity feed-back gain (C_s)	0.3
Un-sprung mass velocity feed-back gain (C_u)	2
Proportional Gain (Kp)	1.8
Integral Gain (K_i)	0.2
Multi objective H∞ Controller	
Road filter (K_d, τ_d)	(2, 0.0005)
Acceleration Filter (K_a, τ_a)	(20, 0.07)
Current Filter (K_u, τ_u)	(20,0.08)

With the above parameters, optimal gain eq (20) is :

K_{opt} = [-238.18 278.74 -72.65 -110.67 -14.90 2.95 1.97e-5 -1.72 0.61 0 0.035]

a. Road model

For simulation, we use road model as Eq (2) with choosing C class road input that :

f_o =0.1Hz, $G_0 = 5x10^{-3}$ m^3/sec, ω_{road} = zero mean white noise with 0.01 variance and vehicle speed v =14 m/sec. Road contour is depicted by Fig 11.

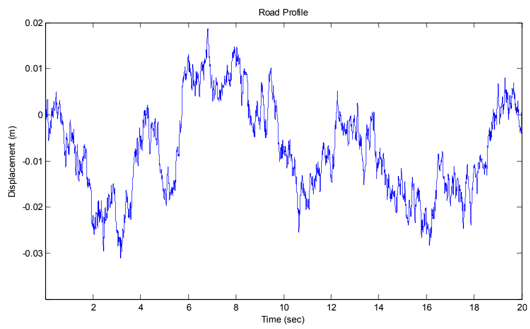

Figure11. Regenerated Power

b. Power regeneration

It is possible to harvest vibration energy when the system in passive mode. In this condition u_{ref} set to zero and average root mean square (RMS) of regenerated power is 19.83 Watt. Regenerated power is depicted by Fig. 12.

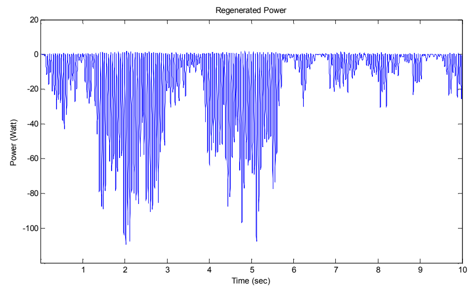

Figure12. Regenerated Power at Passive Mode

c. Controller performance

Open loop, tracking controller, PI-tracking and multi objective H∞ controller response of vertical body acceleration is shown by Figure 13. Suspension working space and dynamic tire load response are shown by Figure 14 and Figure 15.

Three controller performance are :

- Average root mean square of Body Acceleration (BA) are: 1.81, 0.59, 0.42, 0.46 m/sec^2.
- Maximum Suspension working space (SWS) are: 69.2, 29.7, 20.2, 16.4 mm.
- Maximum Dynamic tire deflection (DTD) are : 60. 6 , 20.2, 6.1, 1.8 mm.

respectively for passive, tracking, PI-tracking and multi objective H∞ controller

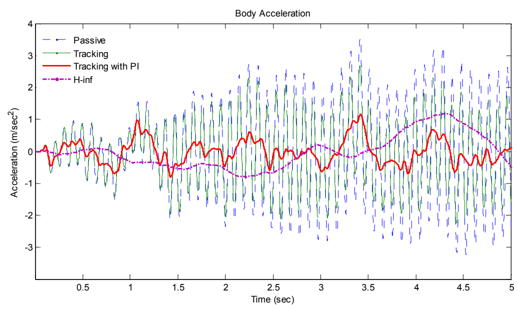

Figure13. Sprung-mass body acceleration

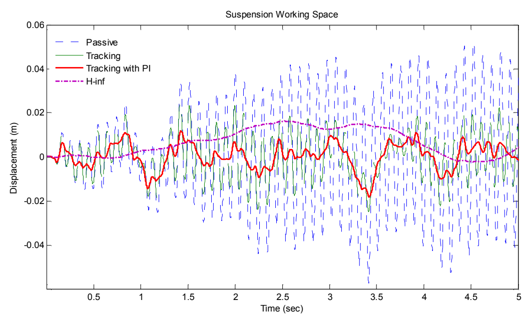

Figure14. Suspension working space

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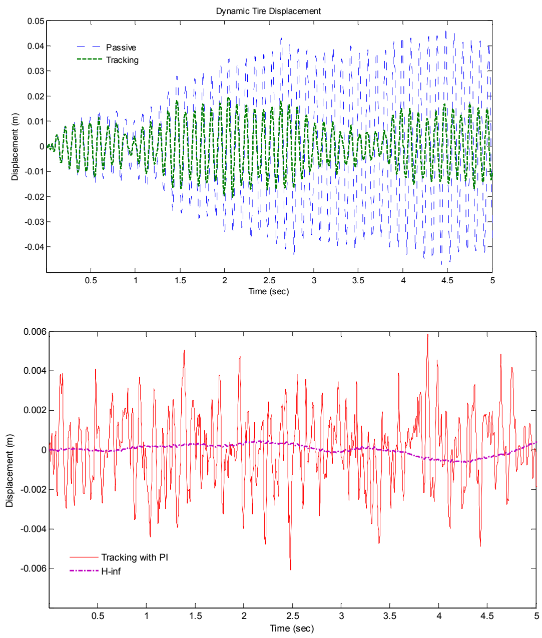

Figure15. Dynamic Tire Deflection

d. Power Consumption

Required power consumption is the first term of Eq (8) for Tracking /PI-tracking controller or product of squared output current (i^2) and load (R) for H∞ multi-objective controller. The average of power consumption are : 49.58 Watt, 72.11 Watt, 51.49 Watt respectively for Tracing, PI-Tracking and Multi objective H∞ with maximum required power are : 247.32 Watt, 160.97 Watt and 216.37 Watt.

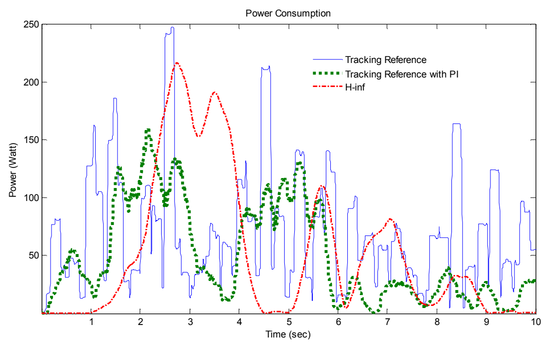

Figure 16.Power Consumption

7. CONCLUSION

Much of energy was required to achieve comfort and handling for vehicle active suspension system. Electromagnetic is capable to convert vibration energy into electrical energy. In other hand it also converts electrical energy becoming mechanical force. It is properly to design active suspension system which supported by vibration energy regeneration.

Prototype of DC motor electromagnetic regenerative shock absorber is developed and tested on ADFT instrument. Sinusoidal input with 25mm maximum displacement is imposed to prototype. The results indicate that damping coefficient (c_m) and regenerated current are increase with decreasing applied external load. We also simulate three controller based to prototype properties. They are: Tracking, PI Tracking and multi objective H∞ controller. The result shows that multi objective H∞ controller need moderate energy consumption to achieve comfort performance near comply with ISO 2631-1 [10] that require acceleration of passenger is under 0,315 m/s^2 and superior handling compare to tracking and PI-tracking controller.

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