**Journal of Theoretical and Applied Information Technology**
30th September 2012. Vol. 43 No.2

ISSN: 1992-8645 www.jatit.org E-ISSN: 1817-3195

# AN APPROACH ON PERFORMANCE COMPARISON BETWEEN AUTOMOTIVE PASSIVE SUSPENSION AND ACTIVE SUSPENSION SYSTEM (PID CONTROLLER) USING MATLAB/SIMULINK

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**ABSTRACT**

Suspension system design is a challenging task for the automobile designers in view of multiple control parameters, complex objectives and stochastic disturbances. For vehicle, it is always challenging to maintain simultaneously a high standard of ride, handling and body attitude control under all driving conditions. The problems stem from the wide range of operating conditions created by varying road conditions, vehicle speeds, and loads. A good vehicle suspension system should have satisfactory road holding ability, while still providing comfort when riding over bumps and holes in the road. When the vehicle is experiencing any road disturbance such as pot holes, cracks, speed breaker and uneven pavement, the vehicle body should not have large oscillations, rather the oscillations should dissipate quickly. This research is carried out to study the performance of two basic suspension systems with a different approach, passive and active suspension system. For the simplicity, mathematical modeling is done by assuming 2 degree of freedom (2 DOF) system. Quarter car model is used to simplify the system. To analyze the model, simulation software MATLAB/SIMULINK is used. Results show that active suspension system has better ability to reduce the pick overshoot of sprung mass and also provides better damping quality than passive suspension system.

**Keywords:** *Active Suspension system, Passive Suspension system, PID controller, Quarter car model, Matlab/Simulink*

## 1. INTRODUCTION

Traditionally, automotive suspension designs have been a compromise between three conflicting criteria of road holding, load carrying and passenger comfort. The suspension system must support the vehicle, provide directional control during handling maneuvers and provide effective isolation of passenger payload from road disturbances. Good ride comfort requires a soft suspension whereas for intensity to applied load requires stiff suspension. Good handling requires a suspension setting somewhere between the two.

Due to these conflicting demands, suspension design has had to be something of a compromise, largely determined by the type of use for which the vehicle was designed. Active suspensions are considered to be a way of increasing the freedom one has to specify independently the characteristics of load carrying, handling and ride quality. A good suspension system should provide good vibration isolation, i.e. small acceleration of the body mass, and a small "rattle space", which is the maximal allowable relative displacement between the vehicle body and various suspension components [1]. The goal is to simultaneously maintain the suspension travel within the rattle space and to minimize car-body rate-of-change of acceleration. The vehicle suspension system is responsible for driving comfort and safety as the

Suspension carries the vehicle-body and transmits all forces between body and road [2].

It is well known that the ride characteristics of passenger vehicles can be characterized by considering the so-called 'quarter-car' model [2]. This method has been widely used to investigate the performance of passive [3], semi-active and fully active [3] suspension system.

## 2. MATHEMATICAL MODELING

A full modeling of the system dynamics related to the vehicle suspension system, road disturbances

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are described. This shall provide the basis for the rigorous computer simulation study to be carried out using MATLAB/Simulink. The mathematical modeling of the dynamic system is performed using the Newtonian mechanics. The suspension system is modeled based on a quarter car configurations. The active suspension system is specifically designed and modeled with the feedback control element embedded into the system which is known as an actuator.

### 2.1 Modeling: Quarter Car

Quarter car model [3] is a popularly used for suspension system analysis and design for its simplicity and yet ability to capture many important parameters.

Figure 2.1.1 shows a quarter car vehicle passive suspension system. Car body is denoted as sprung mass and the tire is denoted as un-sprung mass. Single wheel and axle is connected to the quarter portion of the car body through a passive spring and damper. The tire is assumed to have only the spring feature and is in contact with the road terrain at the other end. The road terrain serves as an external disturbance input to the system.

Based on Newtonian mechanics the equations of the motion for the passive suspension system are given as Reference[4]:

$$m_s\ddot{z}_s = -k_s(z_s - z_u) - C_s(\dot{z}_s - \dot{z}_u) \quad (1.1)$$

$$m_u\ddot{z}_u = k_s(z_s - z_u) + C_s(\dot{z}_s - \dot{z}_u) - k_t(z_u - z_r) \quad (1.2)$$

Where,

| | | |
|---|---|---|
| $z_r$ | - | Road displacement |
| $z_s$ | - | Car body displacement |
| $z_u$ | - | Un-sprung mass displacement |
| $C_s$ | - | Damping coefficient |
| $m_s$ | - | Sprung mass |
| $m_u$ | - | Un-sprung mass |
| $k_s$ | - | Spring stiffness constant |
| $k_t$ | - | Tire stiffness constant |

Quarter car model for active suspension system can be constructed by adding an actuator parallel to spring and damper. Figure 2.1.2 shows a schematic of a quarter car model for active suspension system.

The equations of motion for active suspension system are given as: [4]

$$m_s\ddot{z}_s = -k_s(z_s - z_u) - C_s(\dot{z}_s - \dot{z}_u) + f_a \quad (2.1)$$

$$m_u\ddot{z}_u = k_s(z_s - z_u) + C_s(\dot{z}_s - \dot{z}_u) - k_t(z_u - z_r) - f_a \quad (2.2)$$

Where, $f_a$-Actuator force

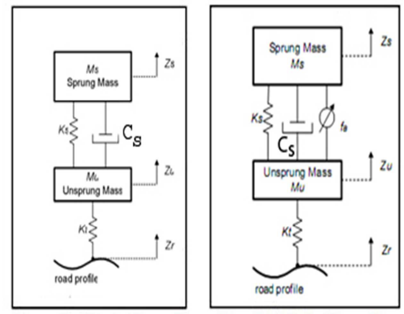


Figure 2.1.1 Passive Suspension Figure 2.1.2 Active Suspension

### 2.2 Disturbance Model

A bump introduced as a disturbance to the vehicle system in this study. Bump was assumed as a sinusoidal curve for the simplicity of the study.

The following figure illustrates the dimension of the disturbance model:

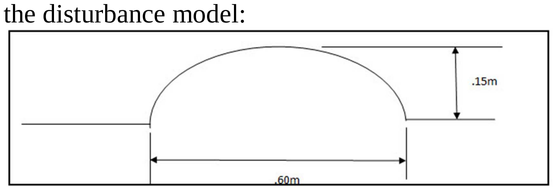


Figure 2.2. Disturbance model

### 2.3 Modeling Parameters

Modeling parameters are taken for a sedan vehicle. The parameters used in the study are taken from Reference [5]:

*Table 1. Vehicle Parameters*

| Parameter name | Parameter symbol | Value |
|---|---|---|
| Sprung mass | $m_s$ | 250 kg |
| Unsprung mass | $m_u$ | 50 kg |
| Tire stiffness constant | $k_t$ | 196000 N/m |

Suspension spring constant and damper co-efficient is changed for portray the effect of different combination on the driving comfort ability.

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*Table 2. Setup parameters*

| | | | |
|---|---|---|---|
| Setup 1: (Damper co-efficient, $C_s$= 2000 Ns/m ) | | | |
| Suspension spring constant, $k_s$ | 10000 N/m | 15000 N/m | 30000 N/m |
| Setup 2: (Spring constant, $k_s$ = 18600 N/m ) | | | |
| Suspension damper co-efficient, $C_s$ | 1000 Ns/m | 1500 Ns/m | 2000 Ns/m |
| | | | |

## 3. PID CONTROLLER DESIGN

For every system, PID controller should be set its value with respect to the system. For this reason, PID controller had to be tuned with system. In this research, Ziegler-Nicols Method is used to tune PID controller [6]. According to the continuous cycle method (Ziegler-Nichols method) first the PID block was attached to the system by setting $K_p$=1, $K_I$=0, $K_D$=0.

We adjusted the system and started to increase the proportional gain ($K'_p$) while forcing small disturbances to the set point until the system oscillated with a constant amplitude. We recorded the $K'_p$ value and the ultimate period ($P_u$).

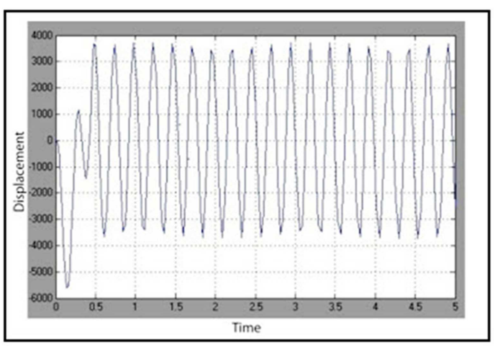


*Figure 3. PID Controller Tuning*

Here for $K'_p$=1635, $K_I$=0, $K_D$=0 we found the oscillation with constant amplitude. We found the ultimate period to be 0.25 seconds.

Using the values of $K'_p$ and $P_u$, we calculated the initial settings of $K_p$,$K_I$,$K_D$ :

$K_p$= 0.6 $K'_p$ = 0.6 x 1635 = 981

$K_I = \frac{Pu}{2} = \frac{.25}{2}$ = 0.125

$K_D = \frac{Pu}{8} = \frac{.25}{8}$ = 0.03125

After setting the value, we tuned the PID controller until the desired response was achieved. For the setup we tuned the $K_p$, $K_I$, $K_D$ to 1000, 0.125, 50.

## 4. MODEL SIMULATION

Mathematical modeling is transformed to computer simulation model and MATLAB/Simulink is used for the simulation. For the model variable-step continuous solver [ODE45 (Dormand-Prince)] is used which is based on an explicit Runge-Kutta formula. It is a *one-step* solver; that is, in computing $y(t_n)$, it needs only the solution at the immediately preceding time point, $y(t_{n-1})$. In general, ODE45 is the best solver to apply as a first try for most problems.

### 4.1 Passive Suspension System Simulation Model

The dynamical system is separated into two systems as the suspension system involves two degrees of freedoms (2 DOF). This passive suspension model was modeled in Simulink form as shown in Figure 4-1. This model was built based on the equation no. 1.1 and equation no. 1.2. There is an open loop system with no feedback element for appropriate adjustment of parameters.

### 4.2 Active Suspension System Simulation Model

Active suspension system requires an actuator force to provide a better ride and handling than the passive suspension system. The actuator force, $f_a$ is an additional input to the suspension system model. The model in Simulink was built based on the equation no. 2.1, equation no. 2.2 and shown in Figure 4.2. The actuator force is controlled by the PID which involves a feedback loop.

ISSN: 1992-8645 www.jatit.org E-ISSN: 1817-3195

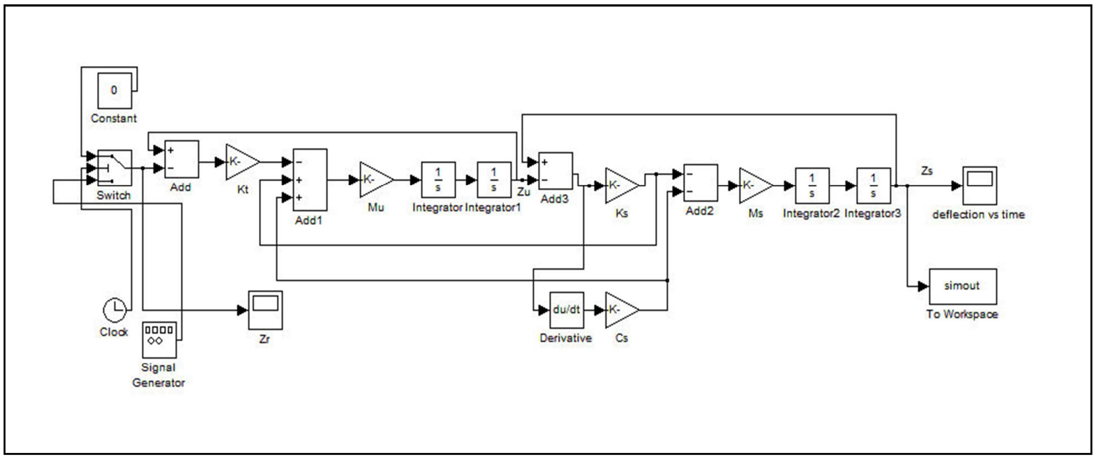


*Figure 4.1. Passive Suspension system simulation model*

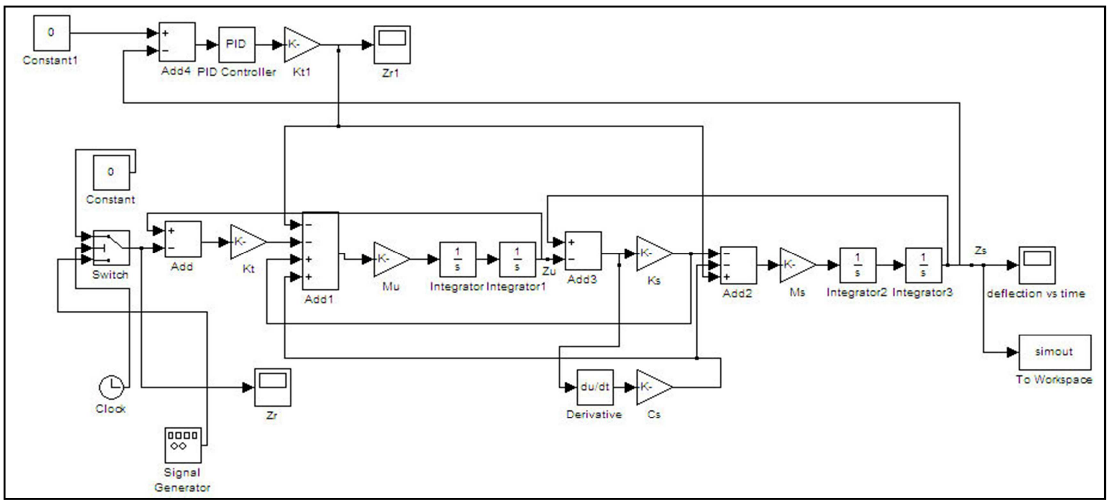


*Figure 4.2. Active Suspension system simulation model*

## 5. SIMULATION RESULTS

The main concern of the simulated suspension system responses results is the sprung mass displacement. Comparisons of the results for different combination of spring constant and damper co-efficient for passive and active suspension system are discussed. Best combinations for active and passive suspension system are also compared and discussed among themselves.

### 5.1 Passive Suspension System

Following figure 5-1 shows the displacement of sprung mass with the sinusoidal disturbance input for the passive suspension system by keeping damper co-efficient at constant value and changing spring stiffness constant.

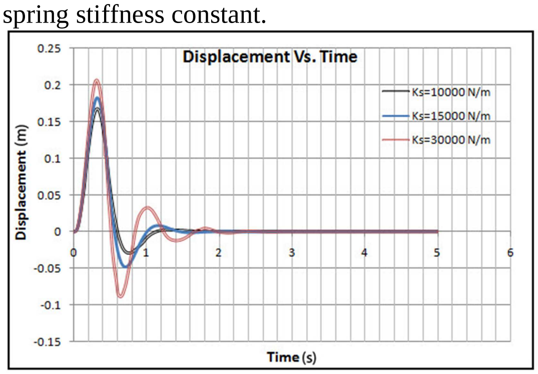


*Figure 5.1. Displacement vs. Time graph for passive system at various spring stiffness constants*

**Journal of Theoretical and Applied Information Technology**
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Figure shows that for higher spring constant sprung mass faces larger displacement and also it takes more time to stabilize the car body. Taking the maximum displacement and stabilizing time into consideration, the results are,

*Table 3. Results for Passive suspension simulation*

| Constant parameter (Damper co-efficient, $C_s$) | Variable parameter (Spring stiffness constant, $K_s$) | Value | |
|---|---|---|---|
| | | Maximum deflection (m) | Stabilizing time ( sec) |
| 1000 Ns/m | 10000 N/m | 0.1746 | 3.25 |
| | 15000 N/m | 0.1956 | 3.5 |
| | 30000 N/m | 0.2272 | 4.25 |
| 1500 Ns/m | 10000 N/m | 0.1700 | 2.25 |
| | 15000 N/m | 0.1872 | 2.6 |
| | 30000 N/m | 0.2150 | 2.75 |
| 2000 Ns/m | 10000 N/m | 0.1685 | 2.20 |
| | 15000 N/m | 0.1823 | 2.22 |
| | 30000 N/m | 0.2069 | 2.25 |

The values of $C_s$ and $K_s$ used here give under-damped vibrations.
From the result we can see that for $C_s$= 2000 Ns/m and $K_s$= 10000 N/m, maximum deflection of the sprung mass is lowest and it takes lowest time to stabilize. Here we see that we should take high value of damper co-efficient and low spring stiffness co-efficient for better performance and good ride comfort. This finding was verified by the finding in reference[7].

### 5.2 Active Suspension System

Figure 5-2 shows the response given by active suspension to the sinusoidal input disturbance. PID controller is used in this suspension and it is a close loop system. PID is tuned so that the response for the step input disturbance is good. PID gain used are as follows; $K_p$ = 1000, $K_i$ = 0.125 and $K_d$ = 50.

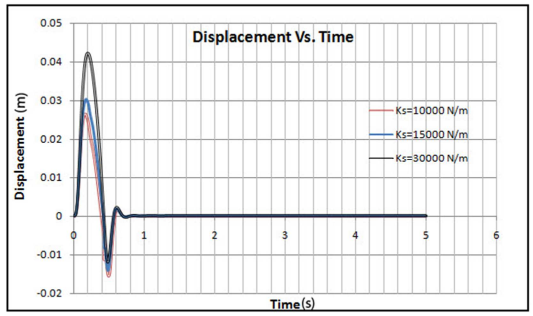

*Figure 5.2. Displacement vs. Time graph for Active system at various spring stiffness constants*

Figure shows that for higher spring constant, sprung mass faces larger displacement but for all the stiffness constant, stabilizing time is almost same. Taking the maximum displacement and stabilizing time into consideration, the results are:

*Table 4. Results for Passive suspension simulation*

| Constant parameter (Damper co-efficient, $C_s$) | Variable parameter (Spring stiffness constant, $K_s$) | Value | |
|---|---|---|---|
| | | Maximum deflection (m) | Stabilizing time (sec) |
| 1000 Ns/m | 10000 N/m | 0.0188 | 0.75 |
| | 15000 N/m | 0.0241 | 0.72 |
| | 30000 N/m | 0.0393 | 0.70 |
| 1500 Ns/m | 10000 N/m | 0.0224 | 0.65 |
| | 15000 N/m | 0.0267 | 0.62 |
| | 30000 N/m | 0.0403 | 0.60 |
| 2000 Ns/m | 10000 N/m | 0.0265 | 0.52 |
| | 15000 N/m | 0.0302 | 0.52 |
| | 30000 N/m | 0.0421 | 0.50 |

The values of $C_s$ and $K_s$ used here give under-damped vibrations.
From the result we can see that for $C_s$= 1000 Ns/m and $K_s$= 10000 N/m, maximum deflection of the sprung mass is lowest and but it takes larger time to stabilize. Here we see that we should take low value of damper co-efficient and low spring stiffness co-efficient for better performance.

### 5.3 Comparison Between Active And Passive Suspension System

For same modeling parameter, if we compare both passive and active suspension system we see that active suspension system works far better and shows better suspension characteristics. Following figure 5-3 shows the response of passive and active suspension system for damping co-efficient, Cs=

ISSN: 1992-8645 www.jatit.org E-ISSN: 1817-3195

2000 Ns/m, spring stiffness co-efficient, Ks= 10000 N/m.

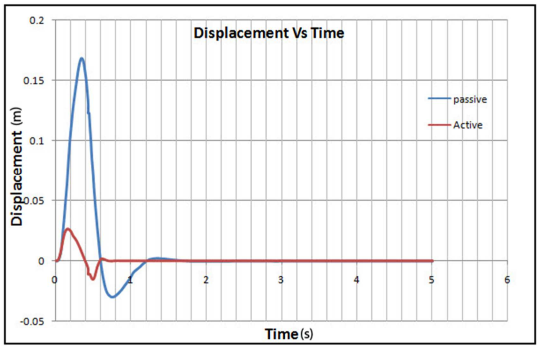


*Figure 5.3. Displacement vs. Time graph for comparison between Active and Passive Suspension*

Figure shows that active suspension system works better than passive suspension system for the same disturbance. Active suspension system absorbs the displacement quickly and stabilizes the sprung mass or car body. On the other hand passive suspension system transfer larger deflection to the car body and also takes time to attain stability.

## 6. CONCLUSIONS

This study intended to find a new approach on performance comparison between Active and passive suspension system. Performance of passive and active suspension system for road disturbance was studied using model simulation. Simulation of the system demonstrates that active suspension system performs much better and also shows better characteristics than passive suspension system. Actuator which is controlled by PID controller works effectively to minimize the sprung weight displacement. It actuates force opposite to the sprung mass displacement to stop the vibration. PID controller checks the error and sends signal to the actuator to execute force.

For passive suspension system for better suspension characteristics, we should use low stiffness spring and high damping co-efficient damper. On the other hand, for active suspension system, low stiffness and low damping co-efficient should be used. These findings are verified with the references.
The study has some limitations.

- Design of PID controller can also be improved by using other methods.
- In this study, single disturbance model has been used to verify the comparison approach. Different disturbance models which are more relevant to real life disturbance can be used to verify the method.
- Here 2 DOF is considered for the research. Higher degree of freedom by including parameters can be used to achieve more precise results.
- Quarter car model made the model simulation simple for calculation. This model does not count some parameters like full vehicle body center momentum.
- Experimental work is also needed for model evaluation and refinement.

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