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Analysis Of An Active Vehicle Suspension System With Controllers Using Simulation

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ABSTRACT: *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 insensitivity to applied load requires stiff suspension. Good handling requires a suspension setting somewhere between the two. The objective of the project is to develop an active control law for the suspension system , for this only a quarter model of the vehicle suspension system with two degree of freedom is conceived and then effect of various types of input (Step, impulse) on the passive quarter model is evaluated followed by application of various control algorithms like PID control, Active force control using Crude approximation and Iterative learning method, Fuzzy logic is applied on the quarter model and the results are tabulated. The effect of inputs on different control laws is found and the results are compared which gives the relative significance of various control algorithms. The software utilized is MATLAB version 7.6.*
Key Words*: Active Suspensions; Vehicle System; PID Control, Active force control*

I. 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 insensitivity 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 passive suspension system has the ability to store energy via a spring and to dissipate it via a damper. Its parameters are generally fixed, being chosen to achieve a certain level of compromise between road holding, load carrying and comfort. An active suspension system has the ability to store, dissipate and to introduce energy to the system. It may vary its parameters depending upon operating conditions and can have knowledge other than the strut deflection the passive system is limited to.This chapter deals with the background information of the various types of suspension system, their mathematical modeling, and various techniques available to achieve the required response. In this the major thrust is given on 3 types of control mechanisms for suspension systems namely passive, semi-active, active suspension systems are described in detail. PID control, fuzzy control and active force control (AFC) strategies applying crude approximation, iterative learning method (ILM) are also discussed.

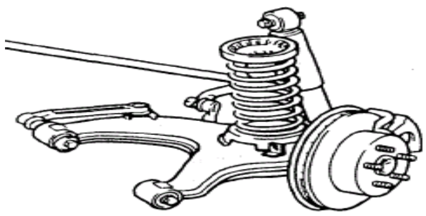

Fig.1. a: Schematic quarter car model.

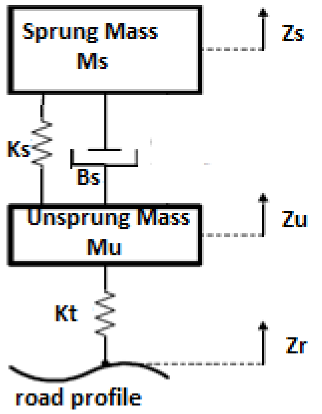

Figure1. b: Block Diagram of Passive Suspension System of quarter car model

II. Suspension System Model

Passive suspensions as shown in Fig.1. can only achieve good ride comfort or good road holding since these two criteria conflict each other and necessitate different spring and damper characteristics. While semi-active suspense with their variable damping characteristics and low power consumption, on systems offer a considerable improvement.A significant improvement can be achieved by using of an active suspension system, which supplied a higher power from an external source to generate suspension forces to achieve the desired performance. The force may be a function of several variables which can be measured or remotely sensed by various sensors, so the flexibility can be greatly increased.With rapid advances in electronic technologies, The development of design techniques for the synthesis of active vehicle suspension systems has been an active area of research over the last two decades to achieve a better compromise during various driving conditions.

Automotive companies are competing to make more developed cars, while comfort of passengers is an important demand and everyone expects from industries to improve it day by day. Therefore, in order to provide a smooth ride and satisfy passengers comfort, designing a modern suspension system is mandatory. A good and efficient suspension system must rapidly absorb road shocks and then return to its normal position, slowly. However, in a passive suspension system with a soft spring, movements will be high, while using hard springs causes hard moves due to road roughness Therefore, it's difficult to achieve good performance with a passive suspension system.As can be seen from Fig. 2, the fixed setting of a passive suspension system is always a compromise between comfort and safety for any given input set of road conditions on one hand and payload suspension parameters on the other. Semi-active/active suspension systems try to solve or at least reduce this conflict. In this regard, the mechanism of semi-active suspension systems is the adaptation of the damping and/or the stiffness of the spring to the actual demands. Active suspension systems in contrast provide an extra force input in addition to possible existing passive systems and therefore need much more energy. The illustration of Fig. 2 also clarifies the dependency of a vehicle suspension setup on parameter changes as a result of temperature, deflection, and wear and tear. These changes must be taken into account when designing a controller for an active or semi-active suspension to avoid unnecessary performance loss.

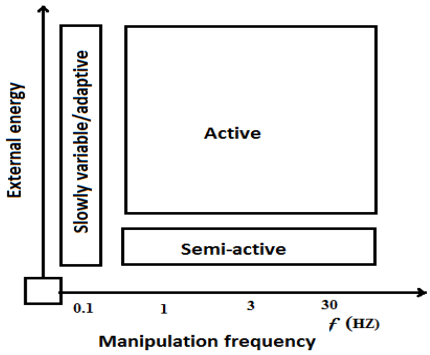

Fig. 2: Comparison between passive, adaptive, semi-active system.

III. Solution Methodology

The various steps involved in finding relevant solution are

1. To find the mathematical model of the quarter car
2. To develop passive (open loop) control model using simulink
3. To develop PID control model using Simulink
4. To develop AFM-CA control model using Simulink
5. To develop AFM-ILM control model using Simulink
6. To develop Fuzzy control model using Simulink
7. Comparison and discussion of results
8. Final conclusions

3.1 MATHEMATICAL MODEL FRO QUARTER VEHICLE PASSIVE SUSPENSION SYSTEM:

In this search, we are considering a quarter car model with two degrees of freedom. This model uses a unit to create the control force between body mass and wheel mass.
The motion equations of the car body and the wheel are
as follows:

$$m_s\ddot{z}_s = -k_s(z_s - z_u) + b_s(z_s - z_u)$$

$$m_s\ddot{z}_s = -k_s(z_s - z_u) + b_s(z_s - z_u)$$

The dynamic nature of the quarter model is given by the above differential equations
Where

m_s And m_u : Sprung mass and un-sprung mass respectively
b_s : Damping coefficient
k_s And k_t : Stiffness of spring and tyre respectively
z_s And z_u : Displacement of sprung mass and unsprung mass respectively
z_r : Displacement of road
$z_s - z_u$: Deflection of suspension
$z_u - z_r$: Deflection of tire
z_s And z_u : Velocity of sprung mass and un-sprung mass respectively
z_s And z_u : Acceleration of sprung mass and un-sprung mass respectively.

Parameters	Value
Sprung mass(m_s)	170kg
Un-Sprung mass(m_s)	25kg
Spring stiffness(k_s)	10520 N/M
Tire stiffness(k_t)	86240 N/M
Damper coefficient(b_s)	1130Ns/M

Table 1 : Table of parameters for quarter car suspension mode

The above equation is transfer function of a passive suspension system. It gives the relationship between sprung mass (Z_s) and road profile (Z_r).
Where

$Z_s(s)$ = Sprung mass displacement
$Z_r(s)$ = Road profile displacement

Therefore the equations for which control laws are developed are for equation

$$\frac{z_s(s)}{z_r(s)} = G(s)$$

3.2 MATHEMATICAL MODEL OF QUARTER CAR ACTIVE SUSPENSION SYSTEM:

$$m_s\ddot{z}_s = -k_s(z_s - z_u) + b_s(z_s - z_u) + fa$$

$$m_s\ddot{z}_s = k_s(z_s - z_u) + b_s(z_s - z_u) - k_t(z_u - z_r) - fa$$

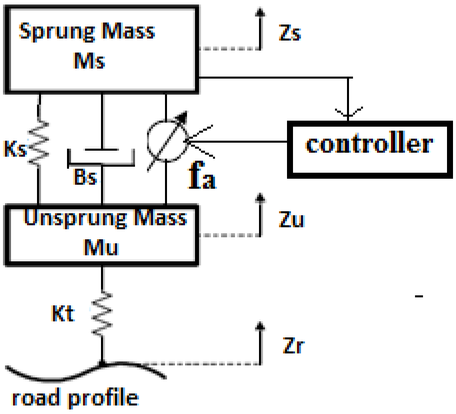

Fig.3: Suspension system block diagram

To model the road input let us assume that the vehicle is moving with a constant forward speed. Then the vertical velocity can be taken as a white noise process which is approximately true for most of real roadways.

To transform the motion equations of the quarter car model into a space state model, the following state variables are considered:

3.3. MATHEMATICAL MODEL OF PID CONTROLLER:

The PID control scheme is named after its three correcting terms, whose sum constitutes the manipulated variable (MV). The proportional, integral, and derivative terms are summed to calculate the output of the PID controller. Defining u (t) as the controller output, the final form of the PID algorithm is:

$$m(t) = k_p e(t) k_i \int e(t)\,dt + k_d \left(\frac{d}{dt}\right) e(t)$$

Where

k_p = Proportional gain, a tuning parameter

k_i = Integral gain, a tuning parameter

k_d = Derivative gain, a tuning parameter

$e(t)$ = error (output – input)

$e'(t)$ = Derivative error

t = Time or instantaneous time (the present)

To get transfer function of an active suspension system we are multiplying open loop and PID transfer functions (as both open loop and pid control transfer functions are in serial),

$$G1(s) = G(s) * \frac{6s^2 + 1000s + 8}{s}$$

3.4. MATHEMATICAL MODEL FOR ACTIVE SUSPENSION SYSTEM WITH AFC-CA STRATEGY:

To get transfer function of an active suspension system with (AFC-CA) we are multiplying active force (F= f_a - m*a) with active suspension system transfer functions,

Where

F = Active Force (0.9937)

F_a = Actuator Force 1m/sec

m = Estimated Mass of 200kg

a = Acceleration (0.001629m/sec^2)

$$G_2(s) = G_1(s)*F$$

3.5.MATHEMATICAL MODEL FOR ACTIVE SUSPENSION SYSTEM WITH AFC-ILM:

To estimate the estimated mass for AFC, systematic method such as intelligent method is appropriate to use rather than trial and error. One of the intelligent methods is iterative learning method (ILM).This type of method applied with AFC can be modeled as the transfer function of (AFC-ILM). The estimating mass in ILM method is U_{k+1} (200.9) and the initial mass (M_I) is (200

To get transfer function of an active suspension system with (AFC-ILM) we are multiplying estimated mass with active suspension system transfer functions

$$G_3(s) = G_2(s) * U_{k+1}$$

3.6.DESIGN OF A FUZZY CONTROLLER

It is necessary to formulate fuzzy rules, before simulating for comparison purpose. The input linguistic variables chosen for the fuzzy controller are sprung mass velocity and the suspension velocity (relative velocity of sprung mass to un-sprung mass).The output of the controller is the damping coefficient of the variable damper. The universe of discourse for both the input variables the sprung mass velocity and suspension velocity was divided in to three sections with the following linguistic variables. Positive (p), zero (z) and negative (n). The universe of discourse for the output variable, damping coefficient of the damper, was divided in to three sections with the following linguistic variables, small (s), medium (m) and large.

Trapezoidal membership functions are used for the linguistic variables because they produce smoother control action due to flatness at the top of the trapezoidal shape. The objective of control is contained in the fuzzy rule base in the form of the linguistic variables using the fuzzy conditional statement. It is composed of the antecedent (IF, AND clause) and the consequent (THEN-clause). For example, one of the control rules can be stated as “If the relative velocity is negative and the sprung mass velocity is positive THEN the damping coefficient is small”. Using these linguistic variables, a set of fuzzy rules was developed. The fuzzy rule base consisted of 9 rules. The fuzzy reasoning inference procedure used was max-min. The defuzzification procedure employed was bisector. Figure 4 shows the membership function with body velocity as input, ranging between [-1.5 1.5], the output is a value (degree of membership) which lies in the range [0, 1].

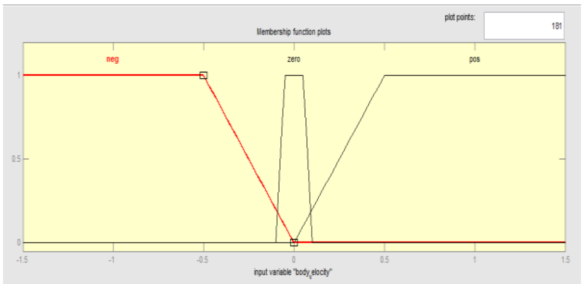

Figure 4: Membership function/input/body velocity

Figure 5 shows the membership function with relative velocity as input, ranging between [-1.5, 1.5], the output is a value (degree of membership) which lies in the range [0, 1].

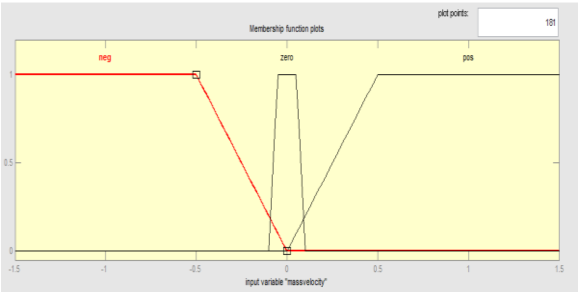

Figure 5: Membership function/input/mass velocity

Figure 6 shows the output membership function, damping coefficient with damper range [0, 2250].The output is a numerical value (degree of membership) between 0 and 1

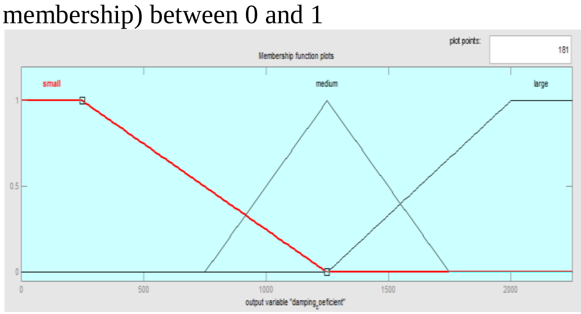

Figure 6: Membership function with damping coefficient

3.7.ATHEMATICAL MODEL FOR ACTIVE SUSPENSION SYSTEM WITH FUZZY LOGIC CONTROLLER USING AFC-CA

Instead of using only fuzzy controller, active suspension system in Simulink model was further develop by introduced active force control with crude approximation (AFC-CA) in the system. This model is shown in Figure. The AFC-CA control Simulink blocks includes the estimated mass gain. The input to the AFC control is the sprung mass acceleration and the output is summed with the fuzzy controller output before multiply with the actuator gain which finally results the generated actuator force. Crude approximation method is used to estimate the estimated mass in the AFC. Here estimated mass is E_m 200kg.

IV. Results And Discussions

This chapter presents the simulated results of a quarter car suspension responses that are described in the previous chapter. The response of sprung mass with respect to applied load is simulated. Comparative results of various control laws for quarter car model suspension system for two types of inputs (impulse and step) are given below.

Type of system	**Rise Time (t_r)in Sec**	**Peak Time (t_p)in Sec**	**% Peak over shoot (M_p)**	**Settling Time (t_s)in Sec**	**Steady state error (e_{ss})**
Passive	0.3100	0.521	44.6	3.685	0.0068
PID control	0.0116	**0.018**	22.3	**0.075**	**0.0017**
AFC-CA	0.1140	0.1602	23.1	0.582	0.0097
AFC-ILM	0.0586	0.0876	22.8	0.432	0.0103
Active (FLC)	0.285	0.4413	**18.76**	1.725	0.0045
FLC with AFC-CA	0.292	0.4717	21.6	1.925	0.0095

Table 2: Comparison Table of response of different type of Suspension Systems for step input:

Type of system	**Peak time (T_p)in Sec**	**Settling time (T_s)in Sec**
Passive	0.2781	2.895
Active(PID)	**0.0125**	**0.0724**
Active(AFC-CA)	0.1277	0.725
Active(AFC-ILM)	0.1761	0.865
Active(FLC)	0.4538	1.885
Active (AFC-CA)(FLC)	0.5123	3.665

Table 3: Comparison Table of response of different type of Suspension Systems for step input

V. Conclusions

- The implementation of an active suspension system, using different control (PID, AFC, FUZZY) laws to the vehicle suspension system has been successfully done in simulation.
- The most important thing in AFC is to estimate the initial mass. If the approximation is done accurately for initial mass, AFC will give a better performance.
- Crude approximation method is easier than iterative learning. In this the initial mass value can change directly.
- Iterative learning method is more intelligent to estimate the initial mass value as it will iterate and decrease the error until it get the right value. But the problem in this method is to tune the learning parameter.
- The % overshoot and settling time are best indicators of transient response (speed) of the system which show fuzzy controller is the best control law for step response.
- The best possible % overshoot of **18.7** is obtained by active fuzzy logic controller for step response which is about **58.07 %** better than the uncompensated system.
- The system to settle down fast is obtained by active PID controller for step response which is about 97.9% better than the uncompensated system.
- Steady state error is reduced from a value of 0.0068 to 0.0017 (PID control) with the implementation of active control laws, which is **75.6 %** improvement.
- The damping ratio(ζ) for uncompensated system is 0.547, which is reduced to a minimum value of 0.482 (Fuzzy control) with the use of active control laws.

- The best possible settling time (t_s) for impulse response is **0.0724sec.** It is attained in active fuzzy logic controller.
- The second best possible impulse response settling time (t_s) is obtained 0.725**sec** in iterative learning method.
- The system to settle down fast is obtained by active PID controller for impulse response which is about 97.4% better than the uncompensated system.

FUTURE SCOPE

- The above work is only simulation work, in order to check the validity of the results experimentation has to be done.
- The values for active force control law are randomly taken, the accurate approximate initial mass of AFC give better understanding performance values for various inputs.
- The analysis is done for quarter model; it can be extended to full car model and the degree of freedom for the system.

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