# Finite Control Augmented with Fuzzy Logic for Automotive Air-spring Suspension System

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Abstract: This paper investigates the spring stiffness control of air suspension systems working under different operating conditions of road profile frequencies and amplitudes. Usually changing the stiffness of the air spring involves variations of the enclosed air pressure by pumping air into or out of the air chamber, or by changing its volume. Since, changing spring stiffness through controlling its pressure consumes power and is not instantaneous, controlling the stiffness through finite volume control is merged with a PI-like Fuzzy Logic Control (PI-FLC) in this paper. This is achieved by connecting the air spring volume to two additional unequal volumes. By controlling the total spring volumes through ON-OFF switching valves, four different stiffness settings are available, and one can achieve an improved performance of air suspension system. A nonlinear quarter-car model is used to evaluate the proposed approach while a Genetic Algorithm (GA) optimization is applied to estimate the PI-FLC optimal gains and the finite levels for switching the spring volumes. Numerical simulations results demonstrate the performance of the proposed control under different road profile. The vehicle body acceleration decreases by a value that reaches 4 cm/s2 which means improving the passenger ride comfort as well as maintaining the passenger safety. This in turns encourages the implementation of the proposed approach on an actual vehicle air suspension in the near future to further verify the system performance.

## 1 INTRODUCTION

Nowadays, many manufacturers are using pneumatic suspension for their vehicle as it is known for its low transmissibility coefficient. The load capacity for the air spring depends on the air pressure inside the spring and its effective area. The vehicle suspension system function is to carry the vehicle weight, isolate the vehicle body from road unevenness and to maintain the road-wheel contact. So, the design of the suspension system is responsible for ride quality and stability. A conflict between ride quality and handling is exist, where soft suspension is required for better ride quality, while a stiff suspension is required for good handling behavior (Wong, 2008).

Presthus (Presthus, 2002) proposed a new model for simulation of the air spring behavior of railway train. The model is three- dimensional and consists of two parts, describing vertical and horizontal behavior. The air spring model is implemented in the vehicle dynamic simulation program GENSYS and the results were consistent with the experimental data. Gavriloski et. al. (Gavriloski et al., 2014) proposed a frequency dependent mathematical model for air spring stiffness which enables the application in models with no need to experimental work.

Nieto et. al. (Nieto et al., 2011) developed an adaptive pneumatic suspension system based on excitation frequency. A control strategy is proposed to avoid undesirable resonant frequencies. The control procedure is based on the pre-knowledge of incoming vibration and an efficient prediction technique is used when the incoming frequency is unknown. An experimental estimation of air spring characteristics in



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active vibration control system is proposed in (Ballo, 2001) and then the results were compared with theoretical considerations estimation.

Bedarff and Pelz (Bedarff and Pelz, ) developed an active and integrated suspension system capable of changing its stiffness. This is done by changing the load carrying area. They choose double acting air spring bellow with adjustable piston segments.

Zepeng et. al. (Zepeng et al., 2017) tried to solve the overshoot phenomena found in electric vehicle. This is done by applying fuzzy control to electrically controlled air suspension (ECAS). This work is done theoretically and verified using AMESim. It was found that the fuzzy control can solve the overshoot problem. Miriji and Arockia (Mirji and Arockia, 2014) applied a fuzzy logic control to half-car suspension model using Matlab. The half car suspension (four-degree of freedom) model was established and the equations of motion were derived. Fuzzy logic and PID control for suspension model were achieved using Matlab. The results showed that the fuzzy control resulted in more improvement in stability than PID control. Omar and Ozkan (Omar and Özkan, 2015) applied Linear Quadratic Regulator (LQR) method to study the effect of in-wheel electric motors mass on the active suspension system performance. Their study showed that there was a bad effect of increasing the unsprung mass due to the addition of in-wheel motor to the system on the road holding and ride comfort. Another disadvantage was a higher actuator force was needed to work in the suspension system with in-wheel motor compared to the same system without IWM. Gohari and Tahmasebi (Gohari and Tahmasebi, 2015) designed neuro-active force control (AFC) technique and applied in active seat suspension system. The controller used is PID which controls the actuator and the force generated from the actuator was then measured. The estimated mass was estimated using Artificial Neural Network (ANN). The results of simulation demonstrated that neuro-AFC scheme improves the performance of control system compared to the uncontrolled and PID controller counterpart. Gokul and Malar presented a new design for air suspension using LQR control strategy which is applied on air suspension dynamic model, then the performance was checked on shaker table. A comparative study is done between the proposed control system and PID control system under different operating conditions and it was found the system performance is improved using LQR control strategy(S. and K., 2019). From the above paragraphs, it is divided into two main sections. The first is developing the air spring dynamic model to use it in a simulation for suspension system. The second is developing techniques to improve the performance of vehicle suspension such as variable area air spring, excitation frequency preknowledge, Apply fuzzy logic control or neuro active force control.

The motivation of this work is to develop an automotive suspension system with variable stiffness for the conflicting requirements of ride and handling. This paper is concerned with the control of air-spring suspension using PI-like Fuzzy Logic with control algorithm responsible for changing the air volume to vary the spring stiffness through using two unequal volumes.

This paper is organized as follows: the mathematical model of the air-spring suspension system is developed in Section II. The proposed finite control augmented with PI-like FLC is discussed in details in Section III, while the applied Genetic Algorithm optimization is highlighted in Section IV. The simulation results are presented and discussed in Section V. Finally, Section VI is the conclusion.

## 2 MATHEMATICAL MODEL

The stiffness, $K$ of the air spring is defined as the incremental force, $dF$ per incremental deflection, $dz$. The force can be calculated by gauge pressure $p_g$ inside the spring multiplied by the effective area, $A_e$.

$$F = p_g A_e \tag{1}$$

$$K = \frac{dF}{dz} = \frac{d(p_g A_e)}{dz} = A_e \frac{dp_g}{dz} + p_g \frac{dA_e}{dz} \tag{2}$$

Neglecting the change in the effective area, we obtain:

$$K = A_e \frac{dp_g}{dz} \tag{3}$$

To calculate the change of pressure inside the spring, polytrophic process is assumed (Gavriloski et al., 2014).

$$pV^n = constant \tag{4}$$

Where p is pressure, V is the volume and n is polytrophic coefficient. By differentiation:

$$\frac{d(p_g V^n)}{dz} = p_g n V^{n-1} \frac{dV}{dz} + V^n \frac{dp_g}{dz} = 0 \tag{5}$$

Since the cross-section area is considered constant, the rate of change of volume per unit deflection is the effective area but with negative sign. This is because decreasing the volume increases the deflection and vice versa.

$$\frac{dV}{dz} = -A_e \tag{6}$$

$$\frac{dp_g}{dz} = \frac{p_g n A_e}{V} \tag{7}$$

Hence,

$$K = \frac{p_g n A_e^2}{V} \tag{8}$$

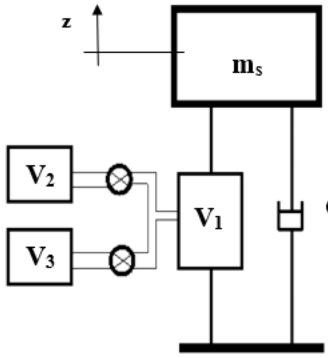


Figure 1: Proposed schematic diagram for the air volume connected to two additional volumes via ON/OFF valves.

The air spring stiffness is inversely proportional to the air volume. The stiffness value is controlled by connecting two unequal auxiliary volumes $V_2$ and $V_3 (V_3 > V_2)$ to the main air volume $V_1$ in successions through ON/OFF valves, as shown in Fig. 1. When the two valves are closed, the effective volume is $V_1$ (minimum volume), so that the effective stiffness will be maximum as given:

$$K_{max} = \frac{p_g n A_e^2}{V_1} \tag{9}$$

$$K_{min} = \frac{p_g n A_e^2}{V_1 + V_2 + V_3} \tag{10}$$

The dynamic model shown in Fig. 2 consists of two springs $K_1$ and $K_2$ in which they have four values $K_{11}$ and $K_{21}$, $K_{12}$ and $K_{22}$, $K_{13}$ and $K_{23}$ or $K_{14}$ and $K_{24}$ according to volume connection described in Table. 1. The mass $M$ represents the moving air inside the pipe and viscous damper $b_z$ which represents the viscous damping inside the pipe.

Table 1: Stiffness with volumes configuration.

| $K_2$ | $V$ |
|---|---|
| $K_{11}$ AND $K_{21}$ | $V_1$ |
| $K_{12}$ AND $K_{22}$ | $V_1 + V_2$ |
| $K_{13}$ AND $K_{23}$ | $V_1 + V_3$ |
| $K_{14}$ AND $K_{24}$ | $V_1 + V_2 + V_3$ |

The equation of motion (Gavriloski et al., 2014) for the model in Figure 2 is:

$$M\ddot{z}_1 = (z - z_1)K_2 - b_z \mid \dot{z}_1 \mid \dot{z}_1 \tag{11}$$

Where

$$K_2 = \frac{p_g n A_e^2}{V_1 + V_r} \frac{V_r}{V_1} \tag{12}$$

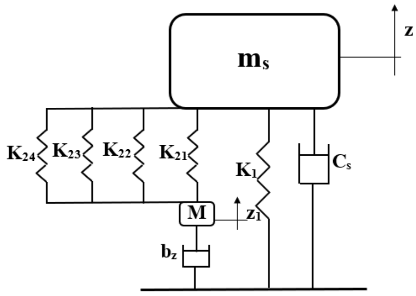


Figure 2: Dynamic model of the air spring.

$$b_z = 0.5\rho k_t A_p (\frac{A_e}{A_p} \frac{V_r}{V_r + V_1})^3 \tag{13}$$

$$M = \rho A_p l_p (\frac{A_e}{A_p} \frac{V_r}{V_r + V_1})^2 \tag{14}$$

Where:

$z_1$: air displacement inside the pipe
$V_1$: air bag volume
$V_r$: reservoir volume (0, V2, V3, V2+V3)
$\rho$: air density
$k_t$: total pressure drop in the pipeline
$A_p$: pipe area
$l_p$: pipe length

And the equivalent air suspension stiffness, $K_s$ equation will be:

$$K_s = \frac{RMS(F_z)}{RMS(z)} \tag{15}$$

Where, $F_z$‘ is the total force on the sprung mass:

$$F_z = p_g A_e + K_1 z + K_2 (z - z_1) \tag{16}$$

$$K_1 = \frac{p_g n A_e^2}{V_1 + V_r} \tag{17}$$

The system is merged in two degree of freedom model (DOF) as shown in Fig. 3. The excitation input for the system is $z_0$ where $K_s, C_s$ represent the variable stiffness of the air spring and the damping coefficient of the air suspension, respectively, while $K_{tr}$ and $C_{tr}$ represent the tire stiffness and damping coefficient.

The model parameters are:
$m_s$: Sprung mass, kg
$m_{us}$: Unsprung mass, kg
$K_s$: air spring stiffness, N/m
$K_{tr}$: Tire radial stiffness, N/m
$C_s$: Suspension damping coefficient, Ns/m
$C_{tr}$: Tire equivalent damping coefficient, Ns/m
$z$: Sprung mass vertical amplitude, mm
$z_2$: Unsprung mass vertical amplitude, mm
$z_0$: Road vertical excitation amplitude, mm

The system is modeled as 3-DOF with excitation input $z_0$. The equations of motion of the model shown

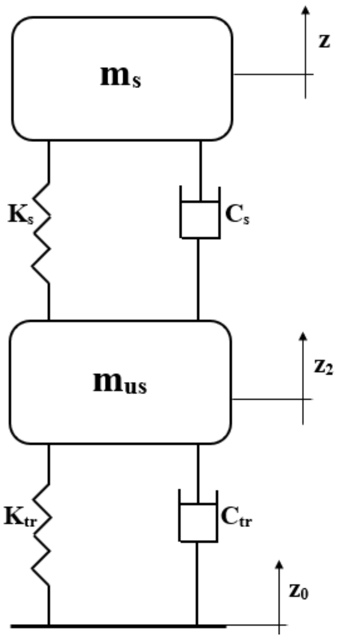

Figure 3: Quarter car air suspension model.

in Fig. 3 are:
For sprung mass

$$m_s\ddot{z}+C_s(\dot{z}-\dot{z}_2)+K_s(z-z_2)=0 \tag{18}$$

For unsprung mass

$$\begin{aligned} m_{us}\ddot{z}_2+C_s(\dot{z}_2-\dot{z})+K_s(z_2-z)\\ +C_{tr}(\dot{z}_2-\dot{z}_0)+K_{tr}(z_2-z_0)=0 \end{aligned} \tag{19}$$

## 3 FINITE CONTROL AUGMENTED WITH FUZZY LOGIC

The proposed control is applied where there are four different settings as described before. The default is all volumes $(V_1, V_2, V_3)$ are connected for minimum stiffness for better comfort. Until the unsprung mass displacement $z_2$ increases gradually to the excitation amplitude, the volume will decrease to $V_1$ by closing off the valves to increase the spring stiffness. The control algorithm is given in Algorithm 1.

A block diagram shows the schematic diagram for the control system shown in Figure 4. Since $K_s$ depends on the pressure $p_g$ of the air spring as well as the volume, a Mamdani PI like-fuzzy logic control (Figure 5) for incremental change $\Delta p_g$ is applied with three linguistic fuzzy input sets for acceleration $\ddot{z}$ and velocity $\dot{z}$ positive P, zero Z, negative N. The linguistic output fuzzy sets for $\Delta p_g$ are the same as the input. Figure 6 shows the membership of the fuzzy set of the input and outputs and Table. 2 shows the rule base.

Algorithm 1: Control Algorithm.

**if** $z_2 <=$ Threshold 1 $(l_1)$ **then**
  Connect $V_2+V_3$ to $V_1$
**else if** Threshold 1 $< z_2 <=$ Threshold 2 $(l_2)$ **then**
  Connect $V_3$ to $V_1$
**else**
  Disconnect $V_2$ and $V_3$ from $V_1$
**end if**

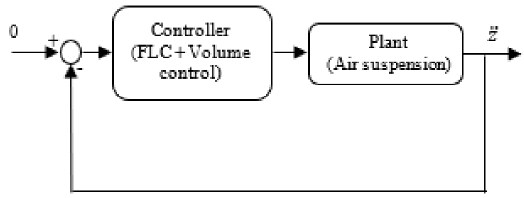

Figure 4: Block diagram for the control system.

Table 2: Fuzzy rule base.

| $\dot{z}$ \ $\ddot{z}$ | P | Z | N |
|---|---|---|---|
| P | P | P | Z |
| Z | P | Z | N |
| N | Z | N | N |

## 4 GENETIC OPTIMIZATION

As explained before, the fuzzy system has two input and one output on which each input or output has a gain. We need to know the optimum value for these gains $K_p, K_I$ and $K_u$. As well as, the thresholds of the volumes control algorithm $l_1, l_2$ and $l_3$. Genetic algorithm optimization technique is applied to find the set of optimal gains and threshold levels method. The objective function $J$ for the optimization is chosen as follow:

$$J = min\Sigma\ddot{z}^2 \tag{20}$$

The objective is minimizing the sum of squared sprung mass acceleration measured over the total time by minimizing the suspension stiffness, Ks as described before while limiting the unsprung mass displacement $z_2$ to a value not more than the road excitation amplitude to ensure tire road holding. Since the

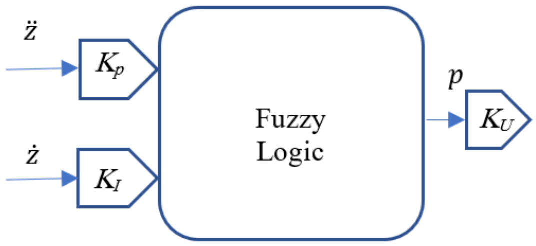

Figure 5: Fuzzy logic Schematic diagram.

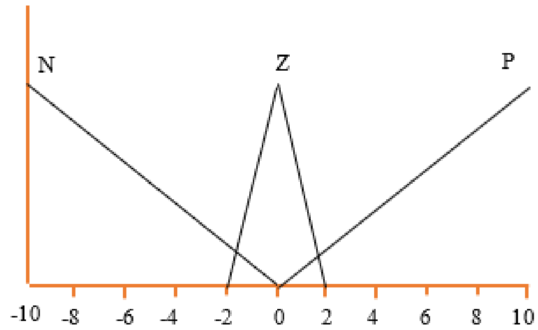


Figure 6: Member ship of the fuzzy sets for inputs and outputs.

threshold levels are arranged in sequence, i.e. $l_1 < l_2 < l_3$, the following inequality constraint is added to the GA problem.

$$\begin{bmatrix} 1 & -1 & 0 \\ 0 & 1 & -1 \end{bmatrix} \begin{bmatrix} l_1 \\ l_2 \\ l_3 \end{bmatrix} < \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

## 5 SIMULATION RESULTS

The two-DOF air suspension system is modeled and simulated using Matlab R2017a and a function stiff was created which depends on frequency, air volume and pressure to calculate the stiffness at different conditions which is then used to calculate the transmissibility ratio for the sprung and the unsprung mass. The system is modeled at different operating conditions of amplitudes and frequencies.

The values of the control gains and thresholds levels $K_e, K_{de}, K_u, l_1, l_2$ and $l_3$ are 13.866, 30.528, 540.552, 0.73, 0.782 and 0.783 respectively.

Figures 7 and 8 show the vertical amplitude of the unsprung mass and the vehicle body acceleration with the proposed finite control versus the one without finite control for an input excitation of 2 cm amplitude and 1 Hz frequency.

The figures show that implementing the proposed strategy reduces the sprung mass acceleration by a value that reaches 4 cm/$s^2$ while ensuring the vehicle safety as shown in Figure 7 which leads to improving the ride comfort.

Figures 9 and 10 show the unsprung mass vertical displacement and the vehicle body acceleration under excitation of 2 cm and 5 Hz frequency.

From the above figures, it can be found that the unsprung mass displacement curves coincide over each other. While for acceleration at 5 Hz, the proposed control reduces the transmissibility with a value reaches 3 cm/$s^2$. The system is not able to decrease

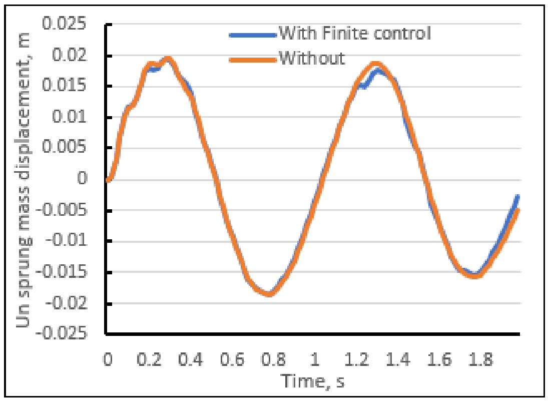


Figure 7: Unsprung mass vertical displacement with time.

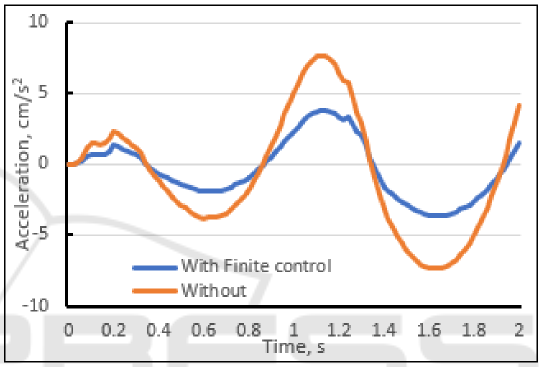


Figure 8: The vehicle body acceleration versus time.

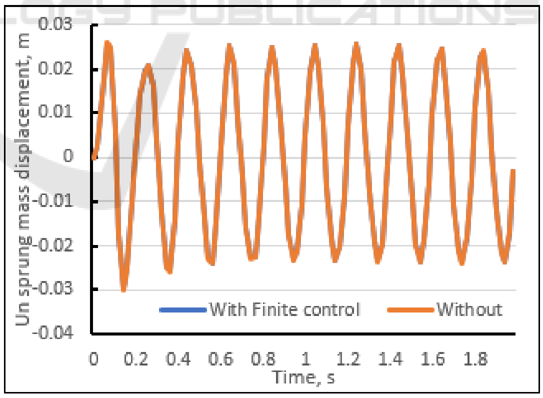


Figure 9: Unsprung mass vertical displacement with time under excitation of 2 cm amplitude and 5 Hz frequency.

the unsprung displacement at high frequencies but still have the benefit of improving comfort performance.

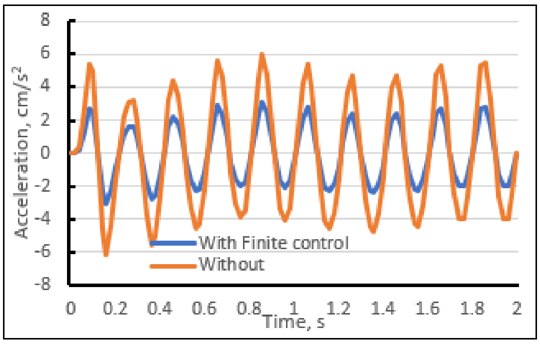


Figure 10: The vehicle body acceleration versus time under excitation of 2 cm amplitude and 5 Hz frequency.

# 6 CONCLUSION

A control strategy is proposed in this work in order to improve ride and safety by using additional volumes connected to the air spring and switching between them. The control strategy is merged with a PI like fuzzy control that control the pressure inside the air spring. The control parameters are estimated using Genetic algorithm. It was found that implementation of the proposed strategy improves the compromise between ride and handling behavior especially over a wide range of frequency.

The vehicle body acceleration decreases by a value that reaches 4 cm/$s^2$ which means improving the passenger ride comfort as well maintain the passenger safety. In future, we could compare our proposed approach with the Inverse Optimal Control (IOC) approach (El-Hussieny et al., 2015) found in the literature.

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