Design of independent full car suspension by using fuzzy control to improve ride comfort and steering ability of passenger car

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Abstract Un-sprung mass direction control, sprung mass rolling, pitching and ride comfort control are the essential integrated chassis control for vehicle. The first control, it utilizes direct yaw moment by independent wheel braking and drive and active steering. The second control, it uses varied independent damper constant and excitation force on suspension. This paper propose a design of fuzzy control to control independent full car suspension by considering brake and drive excitation and finally investigate the response of rolling, pitching of the sprung – mass. First step was developed mathematical model of full car suspension with considering the brake and drive excitation of each wheel. Second step designing the fuzzy control and investigate the response of rolling and pitching. Result of experiment shown, that the fuzzy intelligent control of suspension with considering the brake and drive force excitation has better rolling, pitching and ride comfort control in the form sixth time shorter to perform convergence time (1.5 seconds) compared with PID (7 seconds) in isolate the road, brake and drive disturbance.

1. Introduction
Suspension system is the essential aspect of vehicle dynamic component that can contribute for rolling, pitching body, improve the wheel ground grip for longitudinal and lateral friction. It can support the steering ability in high speed driving, and give a best ride for comfort convenience of the passenger. Many researcher exposed control method for anti-roll and anti-pitching to get vehicle stability during turning, varied slope and varied tilted road condition. Others have design a suspension control to improve damper ratio for convenience riding. Many researcher have done any significant research such as, Two controller type for an vehicle suspensions use Magneto-Rheological (MR) dampers are designed. The first is a model-based utility the Linear Parameter Varying (LPV), and the other is a controller use a Frequency Estimation Based (FEB)[1]. Considered to the relationship between the rear and front wheel axle wheelbase then an optimal control design method is introduced. It was exploited to the optimal design for a low frequency-active suspension system. A suspension constructs from a varied actuators range in series connection with a constantly spring and in parallel connection with a fixed damper [2]. A supervising fuzzy sliding mode with polynomial control model was proposed. This model was integrated with skyhook surface method. It was provided to develop the ride comfort of a vehicle semi-active suspension. A micro-genetic algorithm of multi-objective was used to investigate the PSF*SMC parameters guiding in a training process with 3 ride comfort output variable. It was applied to semi-active suspension as offline- step [3]. For a full car active suspension model an PI$^p$D$^p$ (FOPID) controller is designed. Fractional Order PI$^p$D$^p$ controller was producing an optimal values of parameters to minimize the cost function, which setup by an EA,
which evaluable to get an optimum result to a multivariable objective function. Under typical vehicle maneuvers, the control of suspension has been investigated: driving on bad road surface during deceleration and turning. The performance was showing the superior and robustness of the PI²D³ (FOPID) controller [4]. The acceleration input for 4 state variables use a study RMS.

The state variables were axial sprung mass acceleration, pitch angular acceleration, both front and rear sprung mass deflections. A semi-active control scheme under longitudinal were compared with passive in the form of frequency domain response and then it has implemented [5]. The memberships functions of a control fuzzy parameter are setup by applying GA. It are used to control force actuators. The control force and the vehicle body deflections have been obtained and compared with each other’s [6]. Implementing Matlab-Simulink, It was possible to validate control performance under randomly road excitation input within full car model. The result of simulation was proving that, the fuzzy increment controller performed better than RMSca in controlling pitching, and rolling angular acceleration and body’s vertical vibration acceleration, [7]. The parameters such as damping coefficient and spring stiffness are setup by using QGA (quantum genetic algorithm). Its control performance was proving, it produce superior than passive suspension model. the RMS of axial, pitch and roll sprung mass acceleration can significant be decreased [8]. The newly research was a STLQG control for compensating the time delay within controlling damping value a suspension. A result was showing that STLQG has a superior control performance [9]. Improving of weighting functions for fuzzy-H∞ control has been designed and applied to a novel of semi active control within a half car. This solution expose a good performance if it compared with a passive suspension [10][11].

Half car suspension dynamic, which constructed from 2 hydraulic cylinders, it was working on the anti-roll rods. The investigation of the modal and dynamic analysis was shown that the kinetic dynamic suspension system can eliminate the roll and articulation motions without adding bounce stiffness and improved its ride comfort [12]. A hydraulic electric inerter was integrated to an control force actuator to reduce damper ratio the suspension system for designing the predictive controller to develop the performance of vibration control. The result proved that suspension using ISD was better compared to passive suspension [13]. The optimal hybrid fuzzy PID-controller was used for an active suspension, implementing the GSA. In order to guarantee of the effectiveness, it was comprised with PSO or others control was also utilized with results of effectiveness of OHF-PID-controller on active suspension[14]. An design of the passive and active suspension of quarter-car model using a LQR results better in vehicle ride comfort [15]. Fuzzy logic and PID were used for the active control of car suspension systems. These fuzzy logic controller (FLC) method displayed the efficiency and convenience of vehicle suspension [16]. A half car model was using fuzzy control, bounded interval fuzzy, PID tuned fuzzy and ANFIS control using MATLAB/SIMULINK to reduce the uncertainties disturbance of road. The results proved that the sprung mass deflection and dynamics load acted on tire will be eliminated within the vehicle [17],[18],[19],[20],[21],[22],[23]. A quarter car models in the form of test rig were implemented realized with a Lab-VIEW to understand the real time behavior of three essential suspension parameters [24]. A control method of the LQR controller was designed. LQR matrix was estimated by using the simulated annealing through random searching characteristics of the. Finally, using this LQR compared to the chassis controlled by the normal LQR and the passive suspension, shows better control performance [25]. A research of stability control by using Hybrid Self Organizing Map And Locally Recurrent Neural Network Based Adaptive Backthrough. Result can improve the vehicle steer ability superior [26]. Design of fault for sensors operation by using Active fault tolerant control [27] and design of fault in actuators by using Active fault tolerant control [28].

2. Statement of contribution

2.1. Proposed Independent full car model with brake and drive excitation

We have developed the full car scale (figure 1&2) of suspension plant that is controlled by independent force control on each wheel. The excitation is defined the integration between axial road force and brake or drive force acted on each wheel. The force of brake and drive on the wheel-ground surface are randomly. The slope of road may even contribute to the variation, when its force are projected to axial direction and make a superposition within road amplitude force.
We develop 7 degree of freedom as follows:

1) Rolling degree of rigid sprung mass body as formula 2.
\[ I_r \ddot{\phi}_r = -b_f T_r (Z_{s1} - Z_{u1}) + b_f T_r (Z_{s2} - Z_{u2}) - b_r T_r (Z_{s3} - Z_{u3}) + b_r T_r (Z_{s4} - Z_{u4}) - k_f T_f (Z_{s1} - Z_{u1}) + k_f T_f (Z_{s2} - Z_{u2}) - k_r T_r (Z_{s3} - Z_{u3}) + k_r T_r (Z_{s4} - Z_{u4}) + T_r u_1 - T_f u_2 + T_r u_3 - T_r u_4 \] (2)

2) Pithing degree of rigid sprung mass body as formula 3.
\[ m_3 \ddot{Z}_3 = -b_f a (Z_{s1} - Z_{u1}) - b_f a (Z_{s2} - Z_{u2}) + b_r b (Z_{s3} - Z_{u3}) + b_r b (Z_{s4} - Z_{u4}) - d_f a (Z_{s1} - Z_{u1}) - k_f a (Z_{s2} - Z_{u2}) + k_r b (Z_{s3} - Z_{u3}) + k_r b (Z_{s4} - Z_{u4}) + a u_1 + a u_2 - b u_3 - b u_4 \] (3)

3) Vertical degree of COG rigid sprung mass body as formula 4.
\[ m_3 \ddot{Z}_u = b_f (Z_{s1} - Z_{u1}) - b_f (Z_{s2} - Z_{u2}) - b_r (Z_{s3} - Z_{u3}) - b_r (Z_{s4} - Z_{u4}) - k_f (Z_{s1} - Z_{u1}) - k_f (Z_{s2} - Z_{u2}) - k_r (Z_{s3} - Z_{u3}) - k_r (Z_{s4} - Z_{u4}) + u_1 + u_2 + u_3 + u_4 \] (4)

4) Vertical degree of front right wheel axle as formula 5&6.
\[ m_{u1} \ddot{Z}_{u1} = b_f (Z_{s1} - Z_{u1}) + k_f (Z_{s1} - Z_{u1}) - k_f (Z_{u1} - Z_{r1}) + F_{dfr} \sin(\beta_{fr}) - u_1 \] (5)
For acceleration during drive process and
\[ m_{u1} \ddot{Z}_{u1} = b_f (Z_{s1} - Z_{u1}) + k_f (Z_{s1} - Z_{u1}) - k_f (Z_{u1} - Z_{r1}) + F_{bfr} \sin(\beta_{fr}) - u_1 \] (6)
Is for deceleration during braking process

5) Vertical degree of front left wheel axle as formula 7&8.
\[ m_{u2} \ddot{Z}_{u2} = b_f (Z_{s2} - Z_{u2}) + k_f (Z_{s2} - Z_{u2}) - k_f (Z_{u2} - Z_{r2}) + F_{dfl} \sin(\beta_{fl}) - u_2 \] (7)
For acceleration during drive process and
\[ m_{ur} \ddot{Z}_{u2} = b_f(\dot{Z}_{s2} - \dot{Z}_{u2}) + k_f(\dot{Z}_{s2} - \dot{Z}_{u2}) - k_{tr}(Z_{u2} - Z_{r2}) + F_{bfr}. \sin(\beta_{fr}) - u_2 \]  
Is for deceleration during braking process

6) Vertical degree of rear right wheel axle as formula 9&10.
\[ m_{ur} \ddot{Z}_{u3} = b_f(\dot{Z}_{s3} - \dot{Z}_{u3}) + k_f(\dot{Z}_{s3} - \dot{Z}_{u3}) - k_{tr}(Z_{u3} - Z_{r3}) + F_{drr}. \sin(\beta_{rr}) - u_3 \]
For acceleration during drive process and
\[ m_{ur} \ddot{Z}_{u3} = b_f(\dot{Z}_{s3} - \dot{Z}_{u3}) + k_f(\dot{Z}_{s3} - \dot{Z}_{u3}) - k_{tr}(Z_{u3} - Z_{r3}) + F_{brr}. \sin(\beta_{rr}) - u_3 \]
Is for deceleration during braking process

7) Vertical degree of rear right wheel axle as formula 11&12.
\[ m_{ur} \ddot{Z}_{u4} = b_f(\dot{Z}_{s4} - \dot{Z}_{u4}) + k_f(\dot{Z}_{s4} - \dot{Z}_{u4}) - k_{tr}(Z_{u4} - Z_{r4}) + F_{drt}. \sin(\beta_{rt}) - u_4 \]
For acceleration during drive process and
\[ m_{ur} \ddot{Z}_{u4} = b_f(\dot{Z}_{s4} - \dot{Z}_{u4}) + k_f(\dot{Z}_{s4} - \dot{Z}_{u4}) - k_{tr}(Z_{u4} - Z_{r4}) + F_{brt}. \sin(\beta_{rt}) - u_4 \]
Is for deceleration during braking process

Where:
\[
\begin{align*}
Z_{s1} &= T_f \dot{\phi}_s + aT_s + Z_s \\
\dot{Z}_{s1} &= T_f \ddot{\phi}_s + aT_s + \dot{Z}_s \\
Z_{s2} &= -T_f \dot{\phi}_s + aT_s + Z_s \\
\dot{Z}_{s2} &= -T_f \ddot{\phi}_s + aT_s + \dot{Z}_s \\
Z_{s3} &= T_r \dot{\phi}_s - bT_s + Z_s \\
\dot{Z}_{s3} &= T_r \ddot{\phi}_s - bT_s + \dot{Z}_s \\
Z_{s4} &= -T_r \dot{\phi}_s - bT_s + Z_s \\
\dot{Z}_{s4} &= -T_r \ddot{\phi}_s - bT_s + \dot{Z}_s \\
\end{align*}
\]

If real variables are manipulated by state variable below
\[
\begin{align*}
\dot{\phi}_s &= X_1 \\
\theta_s &= X_2 \\
Z_s &= X_3 \\
\dot{Z}_s &= X_4 \\
Z_{u1} &= X_5 \\
\dot{Z}_{u1} &= X_6 \\
Z_{u2} &= X_7 \\
\dot{Z}_{u2} &= X_8 \\
Z_{u3} &= X_9 \\
\dot{Z}_{u3} &= X_{10} \\
Z_{u4} &= X_{11} \\
\dot{Z}_{u4} &= X_{12}
\end{align*}
\]

We will get state space equation, that need to be used for soft computing as follow
\[
\begin{align*}
\dot{X}_1 &= \phi_s = X_1 \\
\dot{X}_2 &= \theta_s = X_2 \\
\dot{X}_3 &= Z_s = X_3 \\
\dot{X}_4 &= \dot{Z}_s = X_4 \\
\dot{X}_5 &= Z_{u1} = X_5 \\
\dot{X}_6 &= \dot{Z}_{u1} = X_6 \\
\dot{X}_7 &= Z_{u2} = X_7 \\
\dot{X}_8 &= \dot{Z}_{u2} = X_8 \\
\dot{X}_9 &= \phi_s = \{-b_T T_f (Z_{s1} - Z_{u1}) + b_T T_r (Z_{s2} - Z_{u2}) - b_T T_r (Z_{s3} - Z_{u3}) + b_T T_r (Z_{s4} - Z_{u4}) - k_T T_f (Z_{s1} - Z_{u1}) + k_T T_r (Z_{s2} - Z_{u2}) - k_r T_f (Z_{s3} - Z_{u3}) + k_r T_r (Z_{s4} - Z_{u4}) + T_f u_1 - T_f u_2 + T_r u_3 - T_r u_4\}/I_T \\
\dot{X}_10 &= \theta_s = \{-b_T a (Z_{s1} - Z_{u1}) - b_T a (Z_{s2} - Z_{u2}) + b/r b (Z_{s3} - Z_{u3}) + b/r b (Z_{s4} - Z_{u4}) - k_T a (Z_{s1} - Z_{u1}) - k_T a (Z_{s2} - Z_{u2}) + k_T b (Z_{s3} - Z_{u3}) + k_T b (Z_{s4} - Z_{u4}) + a u_1 + a u_2 - b u_3 + b u_4\}/I_P \\
\dot{X}_11 &= Z_s = \{-b_T (Z_{s1} - Z_{u1}) - b_T (Z_{s2} - Z_{u2}) - b_T (Z_{s3} - Z_{u3}) - b_T (Z_{s4} - Z_{u4}) - k_T (Z_{s1} - Z_{u1}) - k_T (Z_{s2} - Z_{u2}) - k_T (Z_{s3} - Z_{u3}) - k_T (Z_{s4} - Z_{u4}) + u_1 + u_2 + u_3 + u_4\}/m_s
\end{align*}
\]
For wheel ground force dynamic by inserting the drive force of each wheel (four wheel drive type) to control force in each wheel.

\[
\begin{align*}
\dot{X}_{11} &= \ddot{Z}_{u1} = \{ b_f (Z_{s1} - Z_{u1}) + k_f (Z_{s1} - Z_{u1}) - k_{tf} (Z_{u1} - Z_{r1}) + F_{dfr} \sin(\beta_{fr}) - u_1 \}/m_{uf} \\
\dot{X}_{12} &= \ddot{Z}_{u2} = \{ b_f (Z_{s2} - Z_{u2}) + k_f (Z_{s2} - Z_{u2}) - k_{tf} (Z_{u2} - Z_{r2}) + F_{dfr} \sin(\beta_{fr}) - u_2 \}/m_{uf} \\
\dot{X}_{13} &= \ddot{Z}_{u3} = \{ b_f (Z_{s3} - Z_{u3}) + k_f (Z_{s3} - Z_{u3}) - k_{tr} (Z_{u3} - Z_{r3}) + F_{drr} \sin(\beta_{rr}) - u_3 \}/m_{ur} \\
\dot{X}_{14} &= \ddot{Z}_{u4} = \{ b_f (Z_{s4} - Z_{u4}) + k_f (Z_{s4} - Z_{u4}) - k_{tr} (Z_{u4} - Z_{r4}) + F_{drr} \sin(\beta_{rr}) - u_4 \}/m_{ur}
\end{align*}
\]

The same action during braking processes, by superpositioning brake force to control force in each wheel.

\[
\begin{align*}
\dot{X}_{11} &= \ddot{Z}_{u1} = \{ b_f (Z_{s1} - Z_{u1}) + k_f (Z_{s1} - Z_{u1}) - k_{tf} (Z_{u1} - Z_{r1}) + F_{bfr} \sin(\beta_{fr}) - u_1 \}/m_{uf} \\
\dot{X}_{12} &= \ddot{Z}_{u2} = \{ b_f (Z_{s2} - Z_{u2}) + k_f (Z_{s2} - Z_{u2}) - k_{tf} (Z_{u2} - Z_{r2}) + F_{bfr} \sin(\beta_{fr}) - u_2 \}/m_{uf} \\
\dot{X}_{13} &= \ddot{Z}_{u3} = \{ b_f (Z_{s3} - Z_{u3}) + k_f (Z_{s3} - Z_{u3}) - k_{tr} (Z_{u3} - Z_{r3}) + F_{btr} \sin(\beta_{rr}) - u_3 \}/m_{ur} \\
\dot{X}_{14} &= \ddot{Z}_{u4} = \{ b_f (Z_{s4} - Z_{u4}) + k_f (Z_{s4} - Z_{u4}) - k_{tr} (Z_{u4} - Z_{r4}) + F_{btr} \sin(\beta_{rr}) - u_4 \}/m_{ur}
\end{align*}
\]

Finally, state space can be simple formulated as follow

\[
\dot{X}_{(t)i} = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \begin{bmatrix} X_{(t)i} \\ 1 \end{bmatrix} + B \begin{bmatrix} U_{(t)j} \\ 1 \end{bmatrix} + C \begin{bmatrix} Z_{r(t)i} \\ 1 \end{bmatrix}
\]

where

- \( \Lambda = \) matrix of displacement
- \( B = \) matrix force control of active suspension
- \( C = \) matrix of road excitation
- \( \Theta_s = \) rolling angle of body – sprung mass (rad)
- \( \Theta_{s} = \) rolling angle rate of body – sprung mass (rad/s)
- \( \Theta_b = \) pitching angle of body – sprung mass (rad)
- \( \Theta_{b} = \) pitching angle rate of body – sprung mass (rad/s)
- \( m_s = \) mass of the car body or sprung mass (kg)
- \( m_{uf} \) and \( m_{ur} \) = front and rear mass of the wheel or unsprung mass (kg)
- \( I_p \) and \( I_r \) = pitch and roll of moment of inertia (kg.m²)
- \( Z_s = \) car body displacement (m)
- \( Z_{s1}, Z_{s2}, Z_{s3}, Z_{s4} = \) car body displacement for each corner (m)
- \( Z_{r1}, Z_{r2}, Z_{r3}, Z_{r4} = \) excitation displacement of road surface (m)
- \( Z_{u1}, Z_{u2}, Z_{u3}, Z_{u4} = \) wheel displacement (m)
- \( T_f \) and \( T_r = \) front and rear treat (m)
- \( a = \) distance from COG to front wheel (m)
- \( b = \) distance from COG to rear wheel (m)
- \( b_f \) and \( b_r = \) front and rear damping (Nm/s)
- \( k_f \) and \( k_r = \) stiffness of car body spring for front and rear (N/m)
- \( b_{tf} \) and \( b_{tr} = \) tire stiffness (N/m)
- \( u_1 \) and \( u_2 \) = front right and left actuators force
- \( u_3 \) and \( u_4 \) = rear right and left actuators force

To realize the soft computing using Simulink matlab, all formula of 7 degree of freedom dynamic are inserted to the three main block, it is already developed shown in figure 3. First block covers about dynamics model of 3 degree of freedom for rolling, pitching and axial central of gravity displacement of sprung mass. Second block consists of dynamics model of axial displacement of each sprung mass corner (namely four corners) and the last block is dynamics model of both four wheels and its excitation. The excitation is the superposition of axial road, brake and drive axial modulation.

5
2.2. Proposed Suspension Fuzzy Control

The control target of 7 degree of freedom full car suspension, already designed are the column of position of $\theta$, $\phi$, $Z_{s1}$, $Z_{s2}$, $Z_{s3}$, $Z_{s4}$ as the independent variable, that should be a constant value. The $U_1$, $U_2$, $U_3$, $U_4$ are as control force actuators and $Z_{r1}$, $Z_{r2}$, $Z_{r3}$, $Z_{r4}$ are as disturbance variable. Assume it, that vehicle drive and brake in a flat road in average, than the control target should a given value. Let are $\theta = a$, $\phi = 0$, $Z_{s1} = Z_{s2} = c$, $Z_{s3} = Z_{s4} = d$ and if it are applied a hybrid excitation of braking-drive and maneuver or turning let are $\theta = a$, $\phi = b$, $Z_{s1} = Z_{s2} = c$, $Z_{s3} = Z_{s4} = d$. Input variable of fuzzy are 7 degree of freedom and designed as follow:

| Table 1. |
|----------|
| Input variable |
| error $\theta$ error $\theta$ error $Z$ error $Z$ error $Z$ error $Z$ error $Z$ |
| -0+0+0+0+0+0+0+ |

Output variable are all control force ($U_1$, $U_2$, $U_3$, $U_4$) mounted on suspension unit between sprung mass to un-sprung mass to balance the extreme amplitude of sprung mass.

| Table 2. |
|----------|
| Output variable |
| $U_1$ | $U_2$ | $U_3$ | $U_4$ |
| d | h | C | d | h | C | d | h | C |
| e | o | o | o | o | e | o | o | o |
| cl | m | cl | m | cl | m | cl | m |

Dec : decompression
Hold : Hold
Compression

The rule was optimized by number of 32 and for this research to be exposed for apart as figure 4&5

Figure 4. Design of Fuzzy input – output
Where i = 1…4

3. Simulation result and discussion
The excitation of road is setup with kinds of step disturbances. For the investigation of control performance between PID control and fuzzy logic, some input signal sets are provided as follows:

| Acted on | Initial value | Final value | Time step |
|----------|---------------|-------------|-----------|
| Wheel FR | 0             | 5           | 3,0       |
| Wheel FL | 0             | 7           | 3,5       |
| Wheel RR | 0             | 3           | 5,5       |
| Wheel RL | 0             | 3           | 6,5       |

The result of axial control for any sprung mass corner in the form of displacement can be analyzed that PID control cannot ensure high quality of convergence shown in figure 6 with convergence time 8 seconds, while the fuzzy control have proved and performed a better convergence level with time need 1,5 seconds shown in figure 7

Figure 5. design of full car suspension model

Figure 6. axial response of sprung mass corner by PID
Figure 7. axial response of sprung mass corner by Fuzzy

When the road excitation acted on 4 wheels with deference time step as shown in table 1 produce 2 responses of roll position and its velocity of sprung mass as shown in figure 9 for PID control responses need 8 second to get its steady state and figure 9 for fuzzy logic response performs shorter time need to become stable.

Figure 8. roll response of sprung mass corner by PID

Figure 9. roll response of sprung mass corner by Fuzzy

The un-balance of roll and axial movement affect position and velocity of sprung mass as rigid body start to pitch suddenly until become its steady state. It needs kind of time long depend on the quality of control getting convergence factor. Fuzzy control performs a better convergence of pitch (figure 11 ), which can still maintain convergence time 1.5 seconds compared by using PID control, which longer time to approach steady state (figure 10).
4. Conclusion

Based on simulation can be defined that fuzzy control available to perform short convergence time compared by using PID controller to isolate vibration from road, brake and drive excitation. Range of control operation is also more widely robust. The time needed to become steady 5 time shorter in every randomly excitation.

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