Design of half-car active suspension system for passenger riding comfort

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Abstract. In this paper, the active suspension system is designed to reduce vibration on the passenger seat caused by change in road-surface shapes or disturbance. PID control is utilized in the design of this active suspension system, and Direct Synthesis method is proposed to tune PID parameters. Direct Synthesis method optimize PID tuning parameters to achieve desired output response in different road surface. In this study, three interference signals are used as disturbance from surface road i.e step, impulse, and sinus. The aim of the suspension control design is to obtain the control parameters that conform to ISO 2631. The simulation results show that value of \( \tau_c \) 0.0002 for step disturbance provide a \( \text{RMS} \) 0.086 m/s² and damping ratio 0.21. For impulse input the value of \( \tau_c \) 0.002 provide a \( \text{RMS} \) 0.112 m/s² and damping ratio of 0.22. For sinus input value of \( \tau_c \) 0.00005 provide a \( \text{RMS} \) 0.09 m/s².

1. Introduction
Nowadays, riding comfort becomes the main focus in the automotive industries. The passengers feel the vibration effects due to interaction of the vehicle and road surfaces. Many researchers said that vibration can degrade human fitness due to discomfort and fatigue [1]. There are several standards about whole body vibration that aim for restricting the allowable magnitude of vibration on human body such as ISO 2631 [2].

The vehicle suspension system is responsible for ride comfort and road holding as the suspension carries the vehicle body and transmit all forces between the body and the road. A classical car suspension or passive suspension system consists of a spring (coil spring, air spring or leaf spring) and a damping element. The spring and damping coefficients are chosen according to comfort, road holding, and handling specifications. However, conventional suspensions can achieve a trade-off between ride comfort and road holding since their spring and damping coefficients cannot be adaptively tuned according to driving efforts and road conditions. They can achieve good ride comfort and road holding only under the designed conditions. To avoid the tradeoff, intelligent suspension systems have been investigated since the 1980s with the development of microprocessor, sensor, and actuator technologies; their spring and damping coefficients can be controlled. The active suspension usually requires a substantial amount of external energy to generate the required control forces. As
intelligent suspension systems, active suspensions belong to the controlled suspension system, which consists of sensors, controllers, plants and actuators. Therefore the active suspension can improve riding comfort against vibration due to road surfaces change [3].

Recent years, a lot of studies about control method used in active suspension have been engaged. PID controller still widely used because of its simple structure and robust performance in linear model system. The application of PID on active suspension has been developed by many researchers and the result shows the response of active suspension of sprung mass decrease overshoot peak as compare with its passive suspension [4] [5]. However, it is necessary to build controller which will bear with any road changes. In this research Direct Synthesis method is used to optimize PID tuning parameters to achieve desired output response in different road surfaces. The presented controller is tested with three road surfaces i.e step, impulse and sinus. The robustness of control system will performed within simulation in MATLAB SIMULINK.

2. Mathematical Modelling

The 5DOF active suspension shown in Figure 1. The suspension, tire, passenger seat are modeled by linear springs in parallel with dampers. The $Y_s$ expresses seat displacement while $Y$ expresses sprung mass vertical displacement (bounce) at the center of gravity and $\Theta$ is the pitch displacement of sprung mass. Front and rear tire displacement are denoted by $Y_1$ and $Y_2$. The device $U$ set at the front suspension as the actuator which can be controlled. The $Z_1$ and $Z_2$ are the disturbance which hit the vehicle at front and rear tire.

![Figure 1. Half-car active suspension model](image)

The equation of motion of half-car active suspension system is being carried out by using Newton second law of motion and written bellow as

$$m_s\ddot{y}_s + c_s(\dot{y}_s + a\dot{\theta} - \dot{y}) + k_s(y_s + a\theta - y) = 0 \quad (1)$$

$$m_b\ddot{y} + (c_1 + c_2 + c_s)\dot{y} - k_2y_2 - k_sy_s + (c_1l_1 - c_2l_2 - c_s a)\dot{\theta} - c_1\dot{y}_1 - c_2\dot{y}_2 + (k_1 + k_2 + k_s)y + (k_1l_1 - k_2l_2 - k_s a)\theta - c_s\dot{\theta} - k_1y_1 = U \quad (2)$$

$$l\ddot{\theta} + (c_1l_1 - c_2l_2 + c_s a)\dot{\theta} - k_say_s - c_s a\dot{\theta} + (c_2l^2 + c_1l_1^2 + c_s a^2)\dot{\theta} + c_1l_1\dot{y}_1 + c_2l_2\dot{y}_2 + k_2l_2y_2 + (k_2l_2 - k_1l_1 + k_s a)\dot{\theta} + k_1(1y_1 + (k_2l_2^2 + k_1l_1^2 + k_s a^2)\dot{\theta} = Ul1 \quad (3)$$

$$m_{tz}\ddot{y}_2 - k_2y + k_{2l}\dot{\theta} - c_2\dot{y} + c_2l_2\dot{\theta} + c_2\dot{y}_2 + (k_2 + k_{tz})y_2 = k_{tz}z_2 \quad (4)$$

$$m_{t1}\ddot{y}_1 + c_1\dot{y} - k_1x_b - k_{sf}l\dot{\theta} + (k_{sf} + k_{tf})y - c_1l\dot{\theta} + c_1\dot{y}_1 = k_{t1}z_1 + U \quad (5)$$
In order to optimize the simulation and to analyze the half-car suspension behavior, the parameters are input as follow:

**Table 1. Suspension parameter of half-car**

| Description                        | Value  |
|------------------------------------|--------|
| Driver Seat Mass (ms)              | 75 Kg  |
| Sprung Mass (mb)                   | 505.1 Kg|
| Front unsprung mass (mt1)         | 28.58 Kg|
| Rare unsprung mass (mt2)          | 54.3 Kg |
| Moment of inertia of the vehicle (I) | 651 Kg.m |
| Rare body length from the CG (l2)  | 1.468 m |
| Front body length from the CG (l1) | 1.098 m |
| Driver seat length from the CG (a) | 0.7 m   |
| Spring stiffness of front and rare tire (kt1, kt2) | 155900 N/m |
| Spring stiffness of front and rare unsprung mass (k1, k2) | 15000 N/m |
| Damping coefficient of front and rare unsprung mass (c1, c2) | 1828 N.s/m |
| Spring stiffness of driver seat (ks) | 15000 N/m |
| Damping coefficient of driver seat (cs) | 150 N.s/m |

2.1. **Riding Comfort Standard Condition**

ISO 2631 has been recognized as minimum standard for whole body vibration on passenger as described below:

**Table 2. ISO 2631 standard for riding comfort**

| RMS (Root Mean Square) Acceleration | Ride comfort Criteria |
|------------------------------------|-----------------------|
| a<0.315 m/s²                       | Comfortable           |
| 0.315<a<0.63 m/s²                  | A little Uncomfortable|
| 0.5<a<1 m/s²                       | Fairy Uncomfortable   |
| 0.8<a<1.6 m/s²                     | Uncomfortable         |
| 1.25<a<2.5 m/s²                    | Very Uncomfortable    |
| a>2.5 m/s²                         | Extremely Uncomfortable|

Another riding comfort standard comes from Optimum G, an England automotive consultant, claims that passenger vehicles generally use a damping ratio (ξ) of approximately 0.2-0.25 for maximizing riding comfort [6]. Furthermore, the controlled responses should be conformed to those riding standards.

3. **PID Tuned by Direct Synthesis**

The first step to tune PID parameters using direct synthesis method is recognizing open loop behaviour of the system. Previous study has successfully arranged the equation to tune Kp, Ki, and Kd on the second order system shown in the equation (6)-(8) [7]:

\[
kc = (\tau_1 + \tau_2) / k_p \tau_c \quad (6)
\]

\[
\tau_i = (\tau_1 + \tau_2) \quad (7)
\]

\[
\tau_d = \tau_1 \tau_2 / (\tau_1 + \tau_2) \quad (8)
\]

\( \tau_c \) is the parameter in direct synthesis method that can be changed to reach the best response of the system. When the disturbance shape change, system may be unsteady and unrobust. Therefor in this
present attempt $\tau_c$ parameter used to build an adaptive control system to any different road surfaces. Table 3, table 4, and table 5 show PID gain parameters for step input, impulse input and sinus input with several varieties of $\tau_c$. The range of $\tau_c$ would be different for each input shapes since this value depend on the settling time and gain of each system.

| Table 3. Gain parameters for step input |
|----------------------------------------|
| Gain parameter | $\tau_c$ | 1 | 0.1 | 0.01 | 0.001 | 1E-04 | 2E-04 |
| Kp | 9 | 93 | 927 | 9268 | 92680 | 46340 |
| Ki | 3 | 34 | 336 | 3358 | 33580 | 16790 |
| Kd | 6 | 64 | 639 | 6395 | 63949 | 31974 |

| Table 4. Gain parameters for impulse input |
|-------------------------------------------|
| Gain parameters | $\tau_c$ | 1 | 0.1 | 0.01 | 0.001 | 0.002 |
| Kp | 39.5 | 395.3 | 3952.6 | 39526.4 | 19763.2 |
| Ki | 9.1 | 91.1 | 910.7 | 9107.5 | 4553.7 |
| Kd | 42.9 | 428.9 | 4288.6 | 42886.2 | 21443.1 |

| Table 5. Gain parameters for sinus input |
|-----------------------------------------|
| Gain parameters | $\tau_c$ | 0.001 | 0.0001 | 0.00005 |
| Kc | 10174.42 | 101744.2 | 203488.4 |
| Ki | 4069.767 | 40697.67 | 81395.35 |
| Kd | 6359.012 | 63590.12 | 127180.2 |

4. MATLAB simulation and result analysis
The MATLAB SIMULINK simulation is prepared for the passenger seat displacement and acceleration responses to analyse the achievement of riding comfort. The goals of this study will show the comparison between active suspension system response and its passive suspension for each road shapes.

4.1. Step road profile
The first road profile is step input with 0.08 m high. Figure 2 shows passenger seat displacement response while figure 3 shows passenger seat acceleration response.

![Figure 2. Seat displacement response due to step input with $\tau_c$ value 0.0002](image1.png)

![Figure 3. Seat acceleration response due to step input with $\tau_c$ value 0.0002](image2.png)
Figures show the maximum overshoot of the response from 0.0227 m of passive to 0.0021 m of active, and lessens the $a_{\text{RMS}}$ from 0.39 m/s$^2$ to 0.08 m/s$^2$ and gives the controlled response with 0.21 damping ratio. So that the active response meets all qualifications of riding comfort standard.

4.2. Impulse road profile
The second road profile formed as impulse with 0.08 m high and hit by the car for 0.06 seconds while it is traveling at 20 km/hour. Figure 4 shows passenger seat displacement response and figure 5 shows acceleration response.

![Figure 4. Seat displacement response due to impulse input with $\tau_c$ value 0.002](image)

![Figure 5. Seat acceleration response due to step impulse with $\tau_c$ value 0.002](image)

Figures observe that the result of active suspension system degrade maximum overshoot from 0.087 m of passive to 0.0012 m of active. Active suspension system with direct synthesis obtain $a_{\text{RMS}}$ 0.112 m/s$^2$ and damping ratio ($\xi$) 0.22, these two results absolutely meet riding comfort standard.

4.3 Sinus road profile
The third road profile formed as sinus wave. It assumed that the road is approximated as sinusoidal in cross section with amplitude of 0.035 m and the wavelength ($\lambda$) of 5 m. The car is traveling at 20 km/hour.

The period and cycle frequency for the harmonic input is as follow

$$Z_1 = 0.035 \sin 17.27t$$  \hspace{1cm} (9)

Figure 6 shows passenger seat displacement response and figure 7 shows passenger seat acceleration response.

![Figure 6. Seat displacement response due to sinus input with $\tau_c$ value 0.00005](image)

![Figure 7. Seat acceleration response due to sinus input with $\tau_c$ value 0.00005](image)
Figures observe that active suspension gives the better result. With maximum overshoot depression from 0.010 m of passive to 0.003 to active. And reach $a_{\text{RMS}}$ 0.099 m/s$^2$. This result successfully conform to ISO 2631 standard.

5. Conclusion
This research presents that direct synthesis is the appropriate method for active suspension system. $a_{\text{RMS}}$ and damping ratio obtained by active suspension system can adapted to three different road profiles and fulfill the requirements of passenger car riding comfort standard. It can be surely denoted that direct synthesis is the robust control strategy for automobile model.

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