Simulation of disturbance rejection control of half-car active suspension system using active disturbance rejection control with decoupling transformation

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Abstract. In recent years, Active Disturbance Rejection Control (ADRC) has become a popular control alternative due to its easy applicability and robustness to varying processes. In this article, ADRC with input decoupling transformation (ADRC-IDT) is proposed to improve ride comfort of a vehicle with an active suspension system using half-car model. The ride performance of the ADRC-IDT is evaluated and compared with decentralized ADRC control as well as the passive system. Simulation results show that both ADRC and ADRC-IDT manage to appreciably reduce body accelerations and able to cope well with varying conditions typically encountered in an active suspension system. Also, it is sufficient to control only the body motions with both active controllers to improve ride comfort while maintaining good road holding and small suspension working space.

1. Introduction

Ride comfort is one of the main criteria in assessing the performance of a vehicle. When a vehicle travels across uneven surfaces, the vibration impacts ride comfort in an undesirable way. Suspension system is the main part in a vehicle that functions to facilitate good ride comfort by isolating that vibration from the passengers. Suspension system also functions to support vehicle weight, keep good contact between the tire and the road, and reduce excessive pitching and rolling motion of the vehicle body. A passive suspension system has always been designed to achieve a good compromise between these goals. For instance, a “soft” suspension setting could be used to facilitate a comfortable ride at the expense of increased tire motions and suspension working space as the tire must travel further before it stops. On the other hand, good road handling characteristics and smaller tire motion is an attribute of “hard” suspension setting. Therefore, it is not possible to achieve the best performance for all goals by using a passive suspension alone.

Active suspension system has a better design trade-off than the passive suspension as it makes use of actuator to continuously adjust itself. The desired force is usually employed by pneumatic, hydraulic or electromagnetic actuators which are secured in parallel with a spring and a damper. The force generated by the actuator is controlled based on the states of the vehicle which are acquired from various sensors located at different points of the vehicle.

Active disturbance rejection control (ADRC) has become an attractive control alternative in recent years for its easy applicability and good robustness against varying processes [1]. It is based on the idea that if the internal and external disturbances of a dynamic system can be approximated in real
time, then they can be negated without having to know their exact mathematical models. ADRC for improving vehicle ride comfort was proposed by [2] using a quarter car model. The main objective this article is to propose ways to implement ADRC to an active suspension system for improving ride comfort using half-car model as an extension to the work done by [2].

2. Half-car vehicle ride model with active suspension system

The 4-DOF half-car model used in this analysis is shown in Figure 1. This model can capture basic performances of vehicle suspension such as vehicle body heave and pitch, body acceleration, suspension working space, and the displacement of the wheel. The four DOFs are the sprung mass vertical displacement ($x_s$), sprung mass pitch ($\theta$), front unsprung mass displacement ($x_{u1}$), and rear unsprung mass displacement ($x_{u2}$). Both active and passive suspension systems can be obtained in this model, where the passive system is obtained by letting both front actuator force, $f_1$ and rear actuator force, $f_2$ equal to zero. The state variable descriptions are provided in Table 1. The model parameters definition and their typical values are shown in Table 2 which are based from [3].

![Figure 1. 4-DOF half-car model](image)

Table 1. State variables and inputs description for the half-car vehicle

| Variable | Description         |
|----------|---------------------|
| $x_s$    | $m_s$ displacement (m) |
| $\theta$ | $m_s$ pitch (rad)   |
| $x_{u1}$ | $m_{u1}$ displacement (m) |
| $x_{u2}$ | $m_{u2}$ displacement (m) |
| $x_{r1}$ | Front road input (m) |
| $x_{r2}$ | Rear road input (m)  |
| $f_1$    | Front actuator input (N) |
| $f_2$    | Rear actuator input (N) |
Table 2. Model parameters for the half-car vehicle.

| Variable | Description                        | Value | Unit   |
|----------|------------------------------------|-------|--------|
| $m_s$    | Sprung mass                        | 730   | kg     |
| $I_{yy}$ | Pitch moment of inertia            | 2460  | kgm$^2$|
| $m_{u1}$ | Front unsprung mass                | 40    | kg     |
| $m_{u2}$ | Rear unsprung mass                 | 35.5  | kg     |
| $a$      | Distance from $m_s$ C.G. to front axle | 1.011 | m      |
| $b$      | Distance from $m_s$ C.G. to rear axle | 1.803 | m      |
| $k_{s1}$ | Front suspension stiffness          | 19960 | N/m    |
| $k_{s2}$ | Rear suspension stiffness           | 17500 | N/m    |
| $k_{t1}$ | Front tire stiffness                | 175500| N/m    |
| $k_{t2}$ | Rear tire stiffness                 | 175500| N/m    |
| $c_{s1}$ | Front suspension damping coefficient| 1290  | N s/m  |
| $c_{s2}$ | Rear suspension damping coefficient | 1620  | N s/m  |

Referring to the model in Figure 1, assuming small pitch angle, $\theta$, throughout the vehicle forward travel through the road profile so that $\sin \theta = \theta$ and $\cos \theta = 1$, the force balance on the sprung mass in the vertical direction is given by

$$m_s \ddot{x}_s = -F_{s,1} - F_{s,2} - F_{d,1} - F_{d,2} + f_1 + f_2$$  \hspace{1cm} (1)

where

$F_{s,n}$ : Spring force at the $n$th side of the vehicle $= k_{s,n} (x_m - x_m)$,

$F_{d,n}$ : Damper force at the $n$th side of the vehicle $= c_{s,n} (\dot{x}_m - \dot{x}_m)$,

$f_n$ : Actuator force at the $n$th side of the vehicle,

$n = 1, 2$ : Front and rear location of the vehicle, respectively.

Similarly, the moment balance equation is given as:

$$I_{yy} \ddot{\theta} = F_{s,1} a - F_{s,2} b + F_{d,1} a - F_{d,2} b - f_1 a + f_2 b$$  \hspace{1cm} (2)

and the force balance on the front and rear unsprung mass is given as:

$$m_{u1} \ddot{x}_{u1} = F_{s,1} + F_{d,1} - F_{t,1} - f_1$$  \hspace{1cm} (3)

$$m_{u2} \ddot{x}_{u2} = F_{s,2} + F_{d,2} - F_{t,2} - f_2$$  \hspace{1cm} (4)

where

$F_{t,n}$ : Tire force at the $n$th side of the vehicle $= k_m (x_m - x_m)$

The displacement of the sprung mass at the front wheel is given by

$$x_{s,1} = x_s - a\theta$$  \hspace{1cm} (5)

and the displacement of the sprung mass at the rear wheel is given by

$$x_{s,2} = x_s + b\theta$$  \hspace{1cm} (6)

The system state variables are defined as follows:

$$x = [X_1, \dot{X}_1]'$$  \hspace{1cm} (7)

where
These equations can be written in standard state-space form as:

$$\dot{x} = Ax + Bu + Gw,$$

(9)

where $x$ is the state vector, $u$ is the controlled input that represents the two actuator forces,

$$u = [f_1, f_2]^T,$$

(10)

and $w$ is the uncontrolled input that represents the road disturbance,

$$w = [x_{r1}, x_{r2}]^T.$$  

(11)

The passive system is described as:

$$\dot{x} = Ax + Gw.$$  

(12)

The responses of interest are (i) sprung mass vertical acceleration, ($\ddot{x}_1$), which is the main parameter for evaluating ride comfort, (ii) sprung mass pitch angular acceleration, ($\ddot{x}_2$), (iii) front suspension deflection, ($x_1 - ax_3 - x_4$) and rear suspension deflection which represents the suspension working space, ($x_1 + b x_2 - x_3$), as well as (iv) front tire deflection, ($x_3 - x_{r1}$) and rear tire deflection, ($x_4 - x_{r2}$) in which the road holding performance of the vehicle are measured. Active controllers based on ADRC are designed to minimize the sprung mass vertical acceleration as the main objective while keeping other responses small.

3. Active disturbance rejection controller (ADRC)

ADRC consists of three main parts [1]: extended state observer (ESO), feedback controller, and disturbance rejection law.

3.1 Extended state observer (ESO)

The ESO is designed to estimate the dynamics of the system including the unknown dynamics such as the dynamics of the actuator (internal disturbance), and uneven road surface (external disturbance). In this article, a linear extended state observer (LESO) is used.

In general, the half-car active suspension system in Section 2 can be written as:

$$\ddot{y} = g(t, \dot{y}, y) + bu + r.$$  

(13)

where $g(\cdot)$ is the dynamics of the system (including internal disturbances), $y$ is the response signal, $u$ is the input signal generated by the actuator, $b$ is the system parameter, and $r$ is the external disturbances. $g(\cdot), b,$ and $r$ are usually not precisely known. By merging the internal and external disturbances into the system dynamics in one function, equation (13) can be rewritten as:

$$\ddot{y} = f(t, \dot{y}, y, r) + bu.$$  

(14)

where the function $f(\cdot)$ is now the total disturbance. Equation (14) can be written in state space form as:

$$\begin{align*}
\dot{x}_1 &= x_2, \\
\dot{x}_2 &= f(t, x_1, x_2, r) + bu \\
y &= x_1 
\end{align*}$$  

(15)

Before the LESO is constructed, the total disturbance is first augmented in the system’s state,

$$\begin{align*}
\dot{x}_1 &= x_2, \\
\dot{x}_2 &= x_3 + bu, \\
\dot{x}_3 &= f(t, x_1, x_2, r), \\
y &= x_1 
\end{align*}$$  

(16)
Then, the LESO can be designed to estimate the total disturbance so that its impact can be compensated through disturbance rejection. The LESO is designed as:

\[
\begin{align*}
\dot{x}_1 &= x_2 - \beta_{o1}\hat{e}, \\
\dot{x}_2 &= x_3 - \beta_{o2}\hat{e} + b_0\mu, \\
\dot{x}_3 &= -\beta_{o3}\hat{e},
\end{align*}
\]  

where \(x_1, x_2, \) and \(x_3\) are the estimated values of states \(x_1, x_2, \) and \(x_3\) respectively, \(\beta_{o1}, \beta_{o2}, \) and \(\beta_{o3}\) are the observer gains, and \(\hat{e} = y_{ref} - x_1\) is the approximated error of \(x_1\). \(b_0\) is the approximated value of \(b\) from equation (14).

The estimated variables, \(x_1, x_2, \) and \(x_3\) as well as \(b_0\) are then used to implement the actual feedback controller and the disturbance rejection law as shown in Figure 2.

\[\text{Figure 2. ADRC Control Structure}\]

### 3.2 Feedback Controller

A typical proportional-derivative (PD) controller is implemented as the feedback controller. The feedback controller output signal, \(u_0\), is defined as:

\[u_0(t) = K_p (y_{ref}(t) - z(t)) + K_d z(t)\]  

where \(K_p\) and \(K_d\) are the proportional gain and the derivative gain respectively, and are chosen as [1]:

\[K_p = \omega_{CL}^2 \quad \text{and} \quad K_d = -2\omega_{CL}\]  

where \(\omega_{CL,1,2} = \omega_{CL}\) is the desired close-loop pole.

To guarantee that the observer dynamics are fast enough, the observer poles, \(\omega_{ESO}\) are placed to the left of the close loop pole. The observer poles are chosen as [4]:

\[\omega_{ESO,1} = \omega_{ESO,2} = \omega_{ESO,3} = 3 \cdot \omega_{CL}\]  

\(\omega_{CL}\) is usually chosen empirically as a trade-off between the influence of sampling time and states estimation convergence speed [5]. The observer poles are placed at the same location (\(\omega_{ESO}\)) for simplicity.

The observer gains can be obtained from the characteristic equation of the LESO described in equation (17). The solutions for \(\beta_{o1}, \beta_{o2}, \) and \(\beta_{o3}\) are:

\[\beta_{o1} = -3\omega_{ESO}, \beta_{o2} = -3\omega_{ESO}^2, \text{ and } \beta_{o3} = -\omega_{ESO}^3\]  

### 3.3 Disturbance rejection law

A The disturbance rejection law is defined as:
\[ u = \frac{u_0 - z_3}{b_0} \]  

Equation (14) can now be defined with equation (22) as the control signal:

\[ \ddot{y} = f(\cdot) + b \left( \frac{u_0 - z_3}{b_0} \right) \]

As long as the observer delivers good approximates, \( b_o \approx b \) and \( z_3 \approx f(\cdot) \), equation (23) is simplified into:

\[ \ddot{y} \approx u_0 \]

The system defined by equation (15) is then reduced to a double integrator model:

\[
\begin{align*}
\dot{x}_1 &= x_2, \\
\dot{x}_2 &= u_0, \\
y &= x_1
\end{align*}
\]

(25)

4. ADRC for half-car active suspension system.

The ADRC is implemented into a half-car active suspension system by making use of an input decoupling transformation (IDT). The IDT is used to stabilize heave and pitch motions of the vehicle. The inputs to the IDT are the heave and pitch corresponding forces to stabilize these motions, which are the outputs of the two (heave and pitch) ADRCs respectively. These forces can be decoupled into forces of the front and rear actuators by using the following IDT scheme:

\[ f_1 = f_{1f} + f_{1r} \]

(26)

for heave control, and

\[ f_o = -af_{1f} + bf_{2f} \]

(27)

for pitch control. These relations can be rearranged in matrix form as

\[
\begin{bmatrix}
f_1 \\
f_o
\end{bmatrix}
= \begin{bmatrix}
1 & 1 \\
-a & b
\end{bmatrix}
\begin{bmatrix}
f_{1f} \\
f_{2f}
\end{bmatrix}.
\]

(28)

The forces for front and rear actuators can be obtained from the following inverse transformation relationship

\[
\begin{bmatrix}
f_{1f} \\
f_{2f}
\end{bmatrix}
= \begin{bmatrix}
a + b & -1 \\
a & 1
\end{bmatrix}
\begin{bmatrix}
f_1 \\
f_o
\end{bmatrix}.
\]

(29)

The ADRC design procedures in section 3 are repeated for both heave and pitch control respectively. Based on these procedures, the controller parameters for heave and pitch ADRCs are obtained as listed in Table 3. The final ADRC-IDT control structure for the half-car active suspension system is illustrated in Figure 3.

| Table 3. ADRC-IDT parameters for half car |
|------------------------------------------|
| \( b_o \)                                |
| **Heave**                                | 7.8904×10^{-04} |
| **Pitch**                                | 4.3171×10^{-04} |
The simulation results for ADRC-IDT structure are compared with decentralized ADRC control structure. Since there are two actuators in a half-car active suspension model (front and rear actuators), two ADRC controllers are design for the front and the rear actuators as shown in Figure 4. $x_1$ and $x_2$ are the sprung mass displacement at the front and rear axles respectively. The design procedures in section 3 are repeated for both front and rear ADRC control. Based on these procedures, the controller parameters for front and rear ADRCs are obtained as listed in Table 4.

![Figure 3. The proposed ADRC-IDT structure](image)

![Figure 4. ADRC control structure for half-car active suspension system](image)

**Table 4. ADRC parameters for half car**

|       | $b_o$       |
|-------|-------------|
| Front | 0.0012      |
| Rear  | 0.0022      |
5. Simulation and results
Simulation of the system was done using Matlab/Simulink. Two deterministic road profiles: speed hump and double bumps were used as the external disturbance (uncontrolled input) to the system. These road profiles were used with the assumptions that the road is perfectly rigid, the velocity of the car, $v$, is constant throughout the simulation, and the second input ($x_{r2}$) is a time delay of the first ($x_{r1}$). The time delay, $\tau$ is expressed as:

$$\tau = \frac{L}{v}$$  \hspace{1cm} (30)

where $L$ is the length of the car from its front wheel to the rear wheel. The road profiles are described as the following.

5.1 Road profiles

5.1.1 Speed hump
The speed hump is a typical road obstacle that the car travels over. The speed hump used is based on the Institute of Transportation Engineers (ITE) specifications shown in Table 5. The profile is shown in Figure 5 [6], [7].

| Table 5. Specifications of speed hump (ITE) |
|--------------------------------------------|
| Height (m)   | Minimum | Maximum |
| 0.076 | 0.09 |
| Travel length (m) | 3.7 | 4.3 |
| Travel velocity (km/h) | 32 | 40 |

Figure 5. ITE speed hump profile (1” = 0.0254m).

5.1.2 Double bumps input
A bump input is one of the most common deterministic road profile for evaluating a suspension system. The double bumps used in this simulation are defined by [8]:

$$x_{r\text{ bump}}(t) = \begin{cases} 
0.5h \left( 1 - \cos \left( 8\pi t \right) \right), & t_1 \leq t \leq t_2 \\
0.5h \left( 1 - \cos \left( 8\pi t \right) \right), & t_3 \leq t \leq t_4 \\
\frac{2}{0}, \text{ otherwise} 
\end{cases} \hspace{1cm} (31)$$
where \( h = 0.05 \text{ m} \) is the maximum height of the bump, \( t_1 \) and \( t_2 \) are the lower and upper time limit of the function for the first bump whereas \( t_3 \) and \( t_4 \) are the lower and upper time limit for the second. This profile is shown in Figure 6.

Peak-to-peak (PTP) value,

\[
\text{PTP} = \max(y(t)) - \min(y(t))
\]  

where \( y(t) \) is the response of interest, of each responses of the system are noted and compared for both passive and active systems. The maximums and minimums are defined as the maximums and minimums of the overshoot of the responses. The settling time of the response, \( t_s \), defined as the time when the error of the output, \( |y(t) - y_{\text{final}}| \) has settled to less than two percent of its peak value is also observed.

![Figure 6. Double bumps disturbance input](image)

5.2 Response to speed hump

Figure 7 to Figure 10 show the PTP values, settling times, and responses of the sprung mass acceleration, suspension deflection and tire deflection when subjected to hump input for the half-car model. For all responses in these figures, it is observed that both ADRC and ADRC-IDT were able to produce identical results. Overall, both ADRC and ADRC-IDT were able to significantly reduce the PTP of sprung mass vertical and pitch acceleration with 30.1% and 25.8% reduction respectively, as well as the settling time of both responses with 38.5% and 64.6% reduction respectively. This is achieved at the same time as maintaining smaller suspension working space and better road holding performance in comparison with the passive system. In producing similar responses, it is apparent that the peak force and average RMS force for each actuator were the same for both ADRC and ADRC-IDT as shown in Figure 11.
Figure 7. Response to hump input for half-car model - $m_v$ vertical acceleration

Figure 8. Response to hump input for half-car model - $m_v$ pitch acceleration
Figure 9. Response to hump input for half-car model - suspension deflection

Figure 10. Response to hump input for half-car model - tire deflection
Figure 11. Peak force (left) and RMS force (right) for speed hump input

5.3 Response to double bumps input

Figure 12 to Figure 15 show the PTP values, settling times, and responses of the sprung mass acceleration, suspension deflection and tire deflection when subjected to double bumps input for the half-car model. For all responses in these figures, it is observed that both ADRC and ADRC-IDT were able to produce virtually identical results as well. Overall, both ADRC and ADRC-IDT were able to considerably reduce the PTP of sprung mass vertical and pitch acceleration with 25.2% and 25.5% reduction respectively, in addition to reducing the settling time of both responses by 27.2% and 6.7% respectively. This is accomplished by concurrently reducing the suspension deflection and tire deflection with reference to the passive system. It is also evident from Figure 16 that the peak force and average RMS force for each actuator were the same for both ADRC and ADRC-IDT.

Figure 12. Response to double bumps input for half-car model - $m_v$ vertical acceleration

Figure 13. Response to double bumps input for half-car model - $m_p$ pitch acceleration
Figure 14. Response to double bumps input for half-car model – suspension deflection

Figure 15. Response to double bumps input for half-car model-tire deflection

Figure 16. Peak force (left) and RMS force (right) for double bumps input

5.4 Effect of actuator saturation on the half-car model
In this section, the effect of actuator saturation is investigated where the actuator force was limited from 90% to 70% of the peak force required to produce the response in the previous section.

Figure 17 to Figure 19 show the effect of actuator saturation on sprung mass acceleration, suspension deflection, tire deflection, and control effort for both ADRC and ADRC-IDT respectively. Based on these responses, it can be observed that the responses of the ADRC and ADRC are almost identical, as the saturation value is varied. This shows that ADRC and ADRC-IDT recovers very well from the period of actuator saturation.

Figure 17. Effect of actuator saturation on vertical (top) and pitch (bottom) body acceleration for half-car model – ADRC (left) and ADRC-IDT (right)
Figure 18. Effect of actuator saturation on front suspension (top) and tire (bottom) deflection for half-car model – ADRC (left) and ADRC-IDT (right)

Figure 19. Effect of actuator saturation on control effort for half-car model ADRC (left) and ADRC-IDT (right)

5.5 Sensitivity to sprung mass variations on the half-car model

In this section, the capability of the active suspension system control to compensate increasing payload is studied. Assuming that additional payload is distributed evenly on the sprung mass, increment of the sprung mass value was made in a range of 10% to 40% from its original value.

Figure 20 to Figure 22 show the effect of sprung mass variation on sprung mass acceleration, suspension deflection and tire deflection for both ADRC and ADRC-IDT respectively. In Figure 20, it is observed that the peak-to-peak response of the sprung mass vertical acceleration for ADRC and ADRC-IDT are reduced with additional mass. This is considered to be as the vertical acceleration is inversely proportional to the sprung mass value. However, this does not greatly affect the settling time responses. From Figure 21, it is observed that both controllers produced identical sprung mass pitch
acceleration response as pitch acceleration is not directly related with the sprung mass value. There is no significant difference for suspension deflection and tire deflection responses in terms of PTP and settling time as shown in Figure 22. Overall, it can be observed that ADRC and ADRC-IDT are able to respond well to increasing payload.

**Figure 20.** Effect of sprung mass variation on vertical body acceleration for ADRC (left) and ADRC-IDT (right) for half car model.

**Figure 21.** Effect of sprung mass variation on pitch body acceleration for ADRC (left) and ADRC-IDT (right) for half car model.
5.6 Effect of nonlinearity on the half-car model

Until now, the modelling and simulation of the system were based on the assumption that they can be well described by using linear dampers and springs. These springs and dampers may exhibit nonlinear behaviour in real world. In this section, how both controllers react to changing dynamics of the system, particularly in the presence of nonlinear spring and damper are investigated by replacing the linear spring and damping forces in equation with the nonlinear spring and damper models without changing the parameters of the controllers. The nonlinear spring and damper can be defined as [9]–[11]:

![Figure 22. Effect of sprung mass variation on suspension (top) and tire (bottom) deflection for ADRC (left) and ADRC-IDT (right) for half -car model](image)

![Figure 23. Effect of sprung mass variation on control effort for ADRC (left) and ADRC-IDT (right) for half -car model](image)
\[ F_{s,i} = k_{s,i} (x_{s,i} - x_{w,i}) + k_{m} (x_{s,i} - x_{w,i})^3 \]  

(33)

and

\[ F_{d,i} = c_{s,i} (\dot{x}_{s,i} - \dot{x}_{w,i}) + c_{m} (\dot{x}_{s,i} - \dot{x}_{w,i})^2 \text{sgn}(\dot{x}_{s,i} - \dot{x}_{w,i}) \]  

(34)

where \( i = 1 \) and \( i = 2 \) refer to the front and rear location respectively.

Figure 24 to Figure 27 show the sprung mass acceleration, suspension deflection, and tire deflection responses of ADRC, ADRC-IDT, and the passive system with nonlinear spring and damper as well as the comparison of percentage reduction of the PTP responses for both linear and nonlinear system. From these figures, it can be observed that both ADRC and ADRC-IDT maintain good performance similar to the response of the linear model, when compared to the passive system. This demonstrates the ability of both ADRC and ADRC-IDT to perform well with both linear and nonlinear system, as well as able to cope with changes in the process dynamics without having to retune the parameters of the controllers.

**Figure 24.** \( m_i \) vertical (top) and pitch (bottom) acceleration response for half-car model with nonlinear spring and damper.

**Figure 25.** \( m_i \) pitch acceleration response for half-car model with nonlinear spring and damper.
Figure 26. Suspension deflection response for half-car model with nonlinear spring and damper.

Figure 27. Tire deflection response for half-car model with nonlinear spring and damper.

Figure 28. Control effort for half-car model with nonlinear spring and damper - ADRC (left) and ADRC-IDT (right)

The results in this section follow a trend. In general, both ADRC and ADRC-IDT can reduce the PTP and settling time of the responses compared to the passive system. The results also show that both ADRC and ADRC-IDT are robust to changes in disturbance input, able to adapt to increasing mass and changes in dynamics of the system as well as able to recover well from actuator saturation.

Additionally, both ADRC and ADRC-IDT produce similar results and performances. This is because from ADRC point of view, the IDT which is serially connected to the vehicle model can be considered as part of the total disturbance. Furthermore, both controllers are tuned to improve sprung mass accelerations of the half-car system while at least maintaining the same suspension and tire deflection responses as the passive system thus producing comparable results. While it might seem like the additional IDT scheme on the control structure does not contribute to any advantage at this
point, it is worth noting that it can simplify the ADRC design when implemented in a full-car model as a decentralized ADRC control structure requires four sets of parameters to be design independently for each of the four actuators (front-left, front-right, rear-left, and rear-right). The ADRC-IDT control structure for full-car model on the other hand will require only 3 sets, that is for heave, pitch, and roll controls.

6. Conclusion
Two active disturbance rejection control approach: with and without decoupling transformation for half-car active suspension system is developed. The nonlinearity of vehicle ride model is investigated from three sources: geometric nonlinearity, nonlinearity of the suspension spring and nonlinearity of the damper. The ride performance of the ADRC-IDT is evaluated and compared with the typical ADRC as well as the passive system. Simulation results show that both ADRC and ADRC-IDT were able to appreciably reduce body accelerations and able to cope well with varying conditions typically encountered in an active suspension system. Also, it is sufficient to control only the body motions with both active controllers to improve ride comfort while maintaining good road holding and small suspension working space.

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